

Guide to
**STORAGE
TANKS &
EQUIPMENT**

**Bob Long
Bob Garner**



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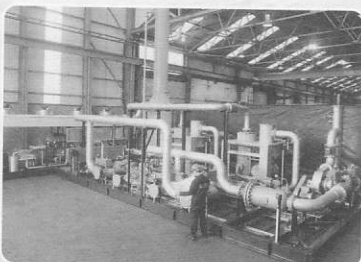
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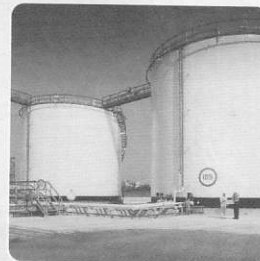
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Guide to STORAGE TANKS & EQUIPMENT

**The practical reference book and
guide to storage tanks and ancillary
equipment with a comprehensive
buyers' guide to worldwide
manufacturers and suppliers**

Bob Long

Bob Garner

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Foreword

Steel storage tanks are an important and costly part of oil refineries, terminals, chemical plants and power stations.

They should function efficiently and be trouble-free at their maximum storage capacity to ensure that these installations can have their planned maximum production capacity.

A sudden, unexpected loss of storage capacity due to accidents will cause a serious handicap for the production capacity of these installations and result in serious financial losses. It is therefore essential that accidents with storage tanks should be avoided as much as possible.

For this purpose it is not only essential that designers have adequate knowledge and experience of the design regulations and limits of storage tanks but also maintenance engineers and operation-personnel should be efficiently aware of important and crucial details of the storage tanks to avoid unexpected problems.

Thousands of steel storage tanks are operating at ambient temperature for oil and chemical products in almost every country in the world. The reported accidents with those tanks are in most cases caused by human errors or operational mistakes. Investigations demonstrate that in many cases they could have been avoided through adequate knowledge of the personnel involved.

Refrigerated steel storage tanks, for liquefied gases, eg. butane, propane and LNG are operating at storage temperatures of respectively - 6 °C, - 45 °C and - 165 °C. Their number is limited. The design and construction of such tanks is complicated and costly. Many special requirements are given, in addition to or deviating from the regulations of tanks operating at ambient temperatures.

For these tanks it is highly essential that designers, maintenance engineers and operation-personnel should have adequate and accurate knowledge of all requirements and crucial details. For such tanks, losses of capacity due to accidents would have very serious consequences.

This book will be most helpful in supplying the knowledge required and should therefore be available for designers, maintenance engineers and operation-personnel.

The guidance given is essential to ensure a trouble-free operation of the storage tanks. I therefore sincerely hope that this book will find its way worldwide.

John de Wit

Ex-tank specialist of Shell, The Hague

Previously chairman of the tank committees of:

The British Standards Institution, London

The Engineering Equipment and Materials Users Association, (EEMUA), London

The European Committee for Normalisation, Brussels.

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About the authors

Bob Long HND (Mechanical & Production Engineering), CEng, Eur Ing, FIMechE

Bob Long attended Woodbridge School in Woodbridge, Suffolk, before moving to the North East to take up a student apprenticeship with Whessoe Heavy Engineering Ltd in 1961. A four-year sandwich course provided an HND from Darlington Technical College and a sound background in both the white and blue-collar areas of the company's activities.

At that time Whessoe was a vigorous and broadly based engineering company working for and with the nuclear, petrochemical, power generation, chemical and sundry other industries, both at home and abroad. So there was plenty of scope for a young man, and a good place to start was in the development department. A thoroughly enjoyable five years was spent finding technical solutions to a variety of problems that emanated from the wide range of company activities.

A move to the storage tank department brought exposure, at first to tanks for the storage of ambient temperature products and then to the more exotic tanks for the storage of low temperature liquids. This was an interesting time in the evolution of low temperature tanks, as they moved from single containment through to double and finally to full containment systems. Many new problems had to be faced and overcome, in the design office, the fabrication shops and on sites in various countries.

The company's range of activities narrowed as time went on, but fortunately for Bob, the storage of liquid products and in particular of low temperature liquids became the main thrust of the business.

Bob became involved with the writing of British Standards, EEMUA guidelines and eventually European Standards in the field of liquid containment systems. He rose to become Engineering Manager and a Technical Director of Whessoe. He now works as a part time consultant for the same company.

A one-company man, a rare beast indeed these days!

Bob Garner HNC (Mechanical Engineering), CEng, MIMechE

Privately educated until the age of 15, Bob Garner left school and was taken on as office boy in an engineering department of Lever Bros. He attended day release and night school achieving a Pre National Certificate Diploma.

Bob was then apprenticed as a fitter/turner with C & H Crichton, maintaining the Ellerman City Line's shipping fleet. During this time he undertook day release gaining an ONC in Mechanical Engineering and subsequently a HNC. Vocational training covered operating lathes, boring machines and shaping machines, and the final year of the apprenticeship was spent in the drawing office. He was then asked to stay to assist with estimating for work required by local, land-based companies (as distinct from shipping).

At the age of 22, Bob was involved in the building of steel lock caissons for the new Langton/Canada Dock passage from the River Mersey. Spells as a draughtsman with the Mobil Oil Company followed, during which Bob was approached by a newly-formed storage tank company, McTay Engineering, and asked to prepare tankage calculations and drawings at home for £1/hr. Being a newly-married man with a mortgage, this was a golden opportunity to earn extra cash to enhance his life style, and his relationship with McTay flourished.

After a couple of years however, Bob joined a completely different engineering organisation that designed and built stone crushing machinery for the quarrying industry.

He continued with his moonlighting for McTay until 1969 when he joined the company full time, being involved in designing tanks, draughting, estimating for new work, visiting potential clients, purchasing steel and tank components and assisting with technical backup on overseas visits to clients.

Bob Garner was made Technical Director in 1972, responsible for estimating, design & drawing office and purchasing and inspection. After continuing with further studies, in 1974 Bob became an Associate Member of the Institution of Mechanical Engineers. (Associate Members later became known as Chartered Engineers, which is the recognised title today.)

By 1977, expanding business opportunities took Bob to East Africa, The Falklands and America as well as much of Europe. His responsibilities during this time were principally for the operation of the estimating and engineering departments. This work continued until 2000 when, now as a single man, he took early retirement.

He still works for McTay, on a consultancy basis — as long as it does not interfere too much with holidays at home and overseas, cruises or golf!

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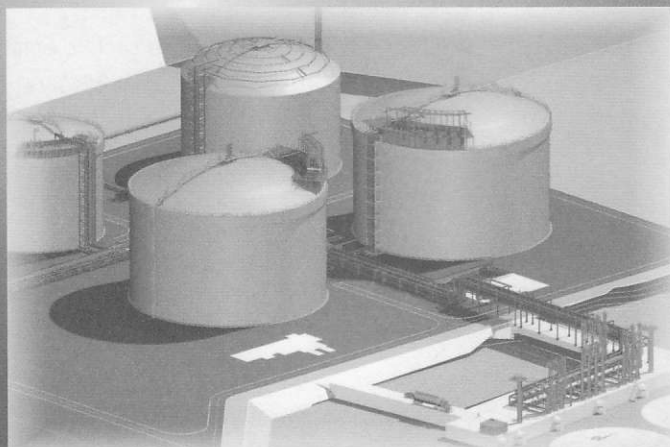
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STVZ

How to use this book

Storage Tanks & Equipment is a practical reference book written for specifiers, designers, constructors and users of ambient and low temperature storage tanks. It has been designed to provide practical information about all practical aspects of the design, selection and use of vertical cylindrical storage tanks. Other tank types are covered but in less detail. Although the emphasis is on practical information, basic theory is covered.

The book is aimed at everyone who has technical problems as well as those wanting to know more about all aspects of tank technology and also those who want to know who supplies what, and from where.

Storage Tanks & Equipment is not intended to be a comprehensive design manual, but sufficient information is included to enable the reader to understand the design process and to identify potential problem areas in tank type selection, fabrication and erection. The principal Standards are covered and detailed comparisons between the main ones are given. The main Codes* include: BS 2654, BS 7777, API 650, API 620, prEN 14015 and prEN 14620. Other Standards include those such as NFPA, DOT, IP, CEN, HSE etc.

Storage Tanks & Equipment can be used in a variety of ways depending on the information required. For specific problems it is probably best used as a reference book. The detailed contents section at the front of the book and in particular the Reference index, Chapter 29, at the end of the book, will simplify finding the appropriate topic. The introductions at the start of each chapter will also provide valuable guidance. Technical and other references are listed at the end of most chapters. Consulting these will lead to more references and hopefully sufficient information to satisfy those who need to know more on any particular subject.

As a practical textbook, though, *Storage Tanks & Equipment* may be read from cover to cover to obtain a comprehensive understanding of the subject. Of course, individual chapters may be studied separately. *Storage Tanks & Equipment* follows a logical sequence, starting with a general history of storage tanks, the design of tanks for the storage of products at ambient temperatures together with sections covering material selection, fabrication, erection, foundations, layout, venting, seismic design and operation of these tanks. There then follows a parallel series of chapters which concern themselves with tanks for the storage of products at low temperatures.

The various formulae used in *Storage Tanks & Equipment* have come from a large number of sources and many of the formulae are well known, as is their use of the variables contained within them. Rather than use a single system of variables in the book, which could give rise to confusion, it was decided in all cases to define the variables local to the equations themselves. Please note also that all pressures referred to throughout *Storage Tanks & Equipment* are gauge pressures unless otherwise stated.

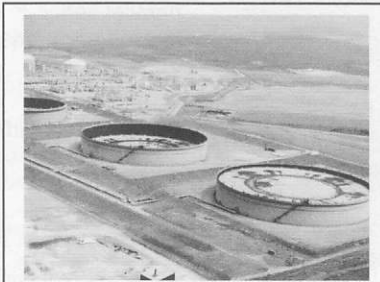
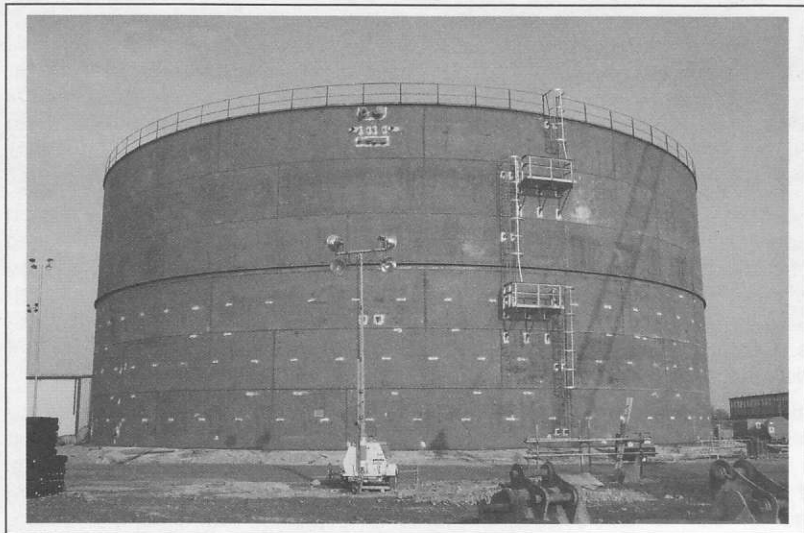
The Classification guide in Chapter 28 is an invaluable and important part of *Storage Tanks & Equipment*. It summarises ambient and low temperature liquid storage tanks, classifying them according to tank type, size or capacity, materials of construction, products stored, materials of construction etc. Companies are listed alphabetically here and in the other sections including ancillary products and services, by their country of origin. The information and data is for guidance only. It is strongly recommended that direct contact with all companies be made to ensure their details are clarified wherever necessary.

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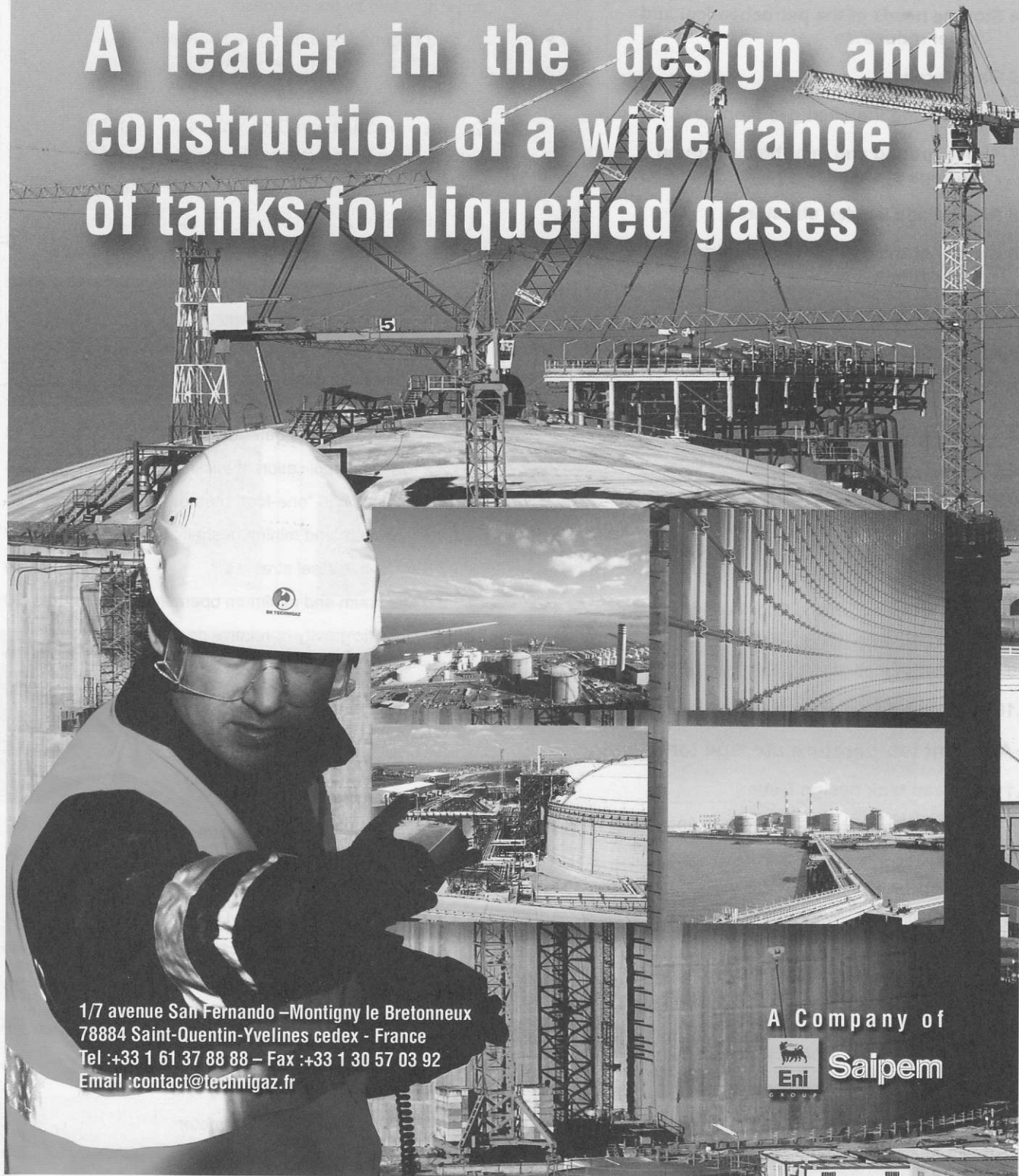
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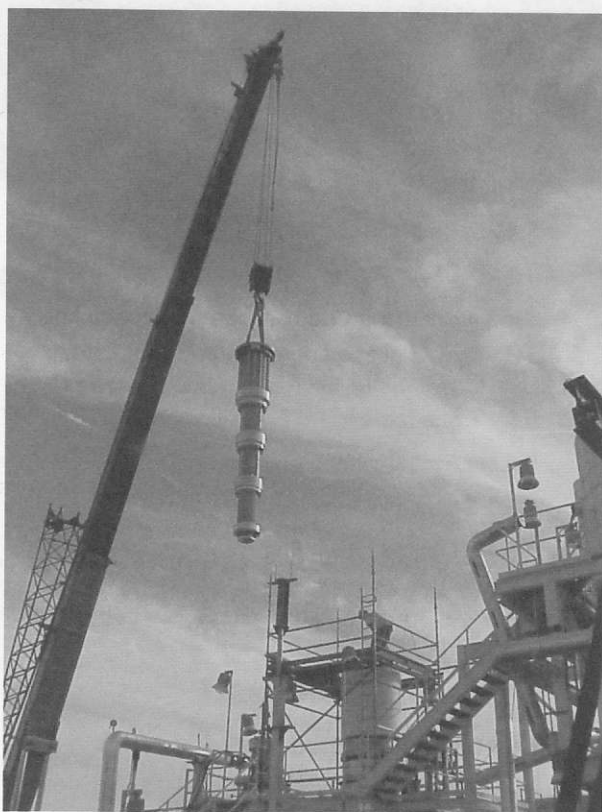
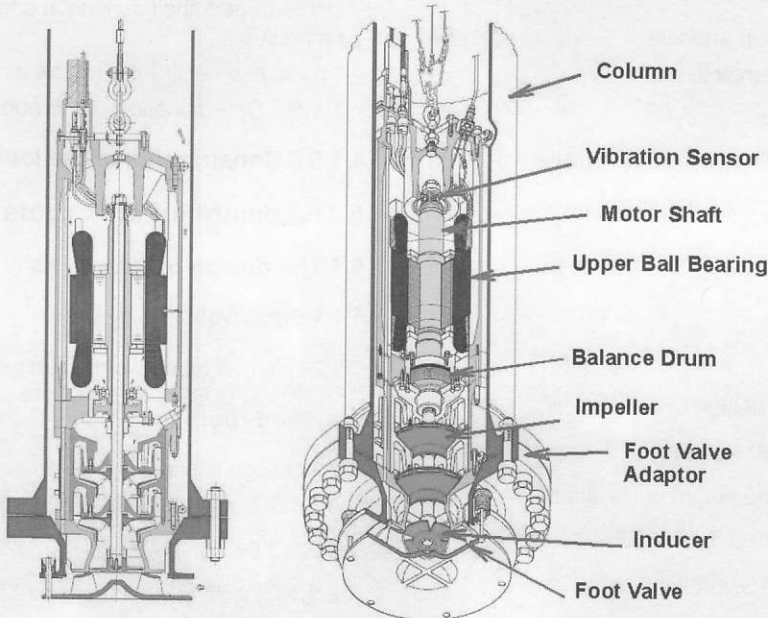
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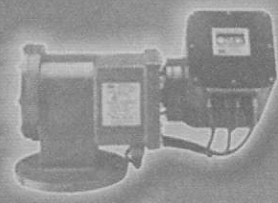
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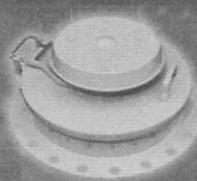
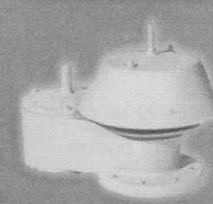


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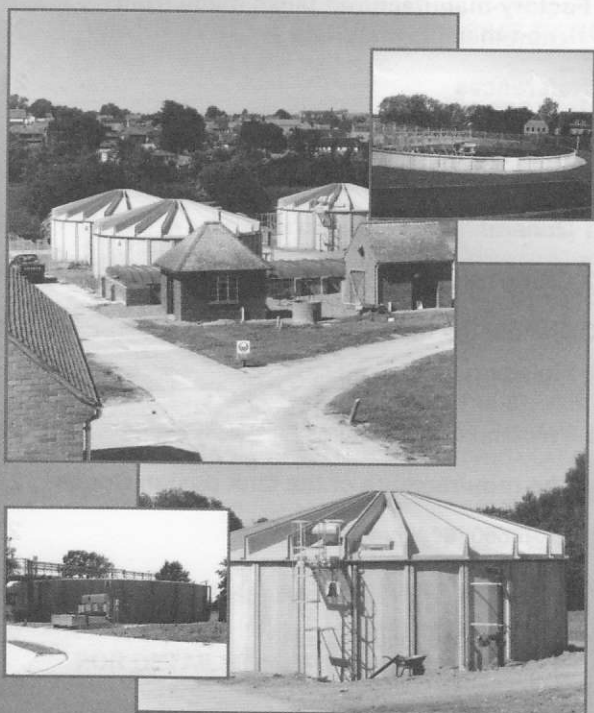


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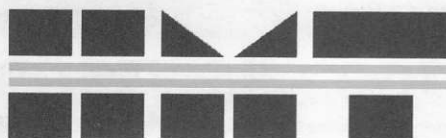
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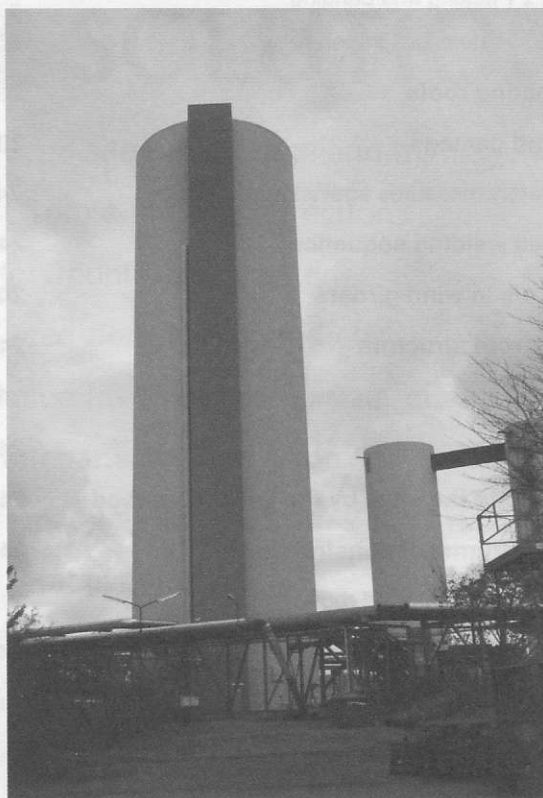
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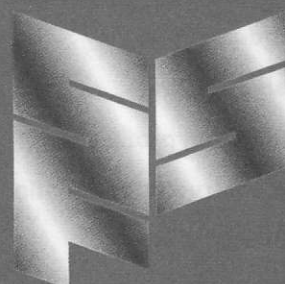
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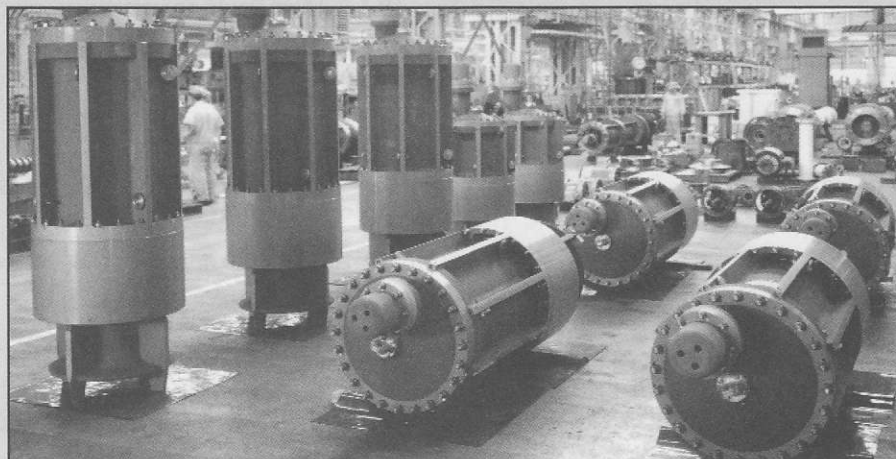


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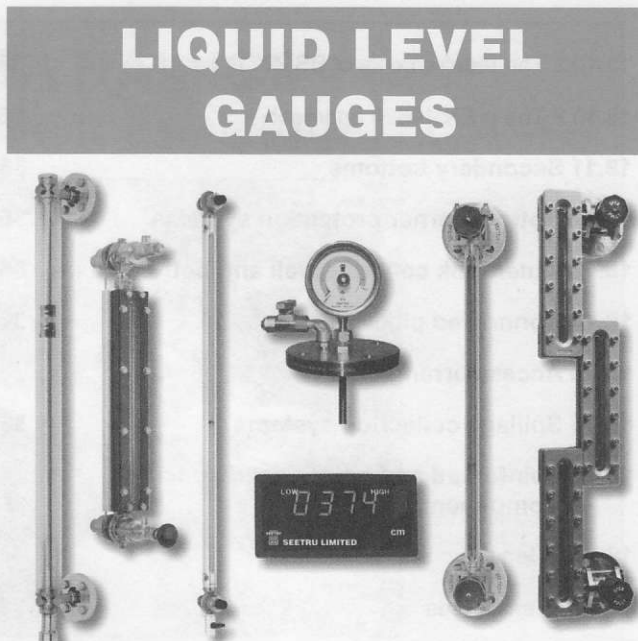
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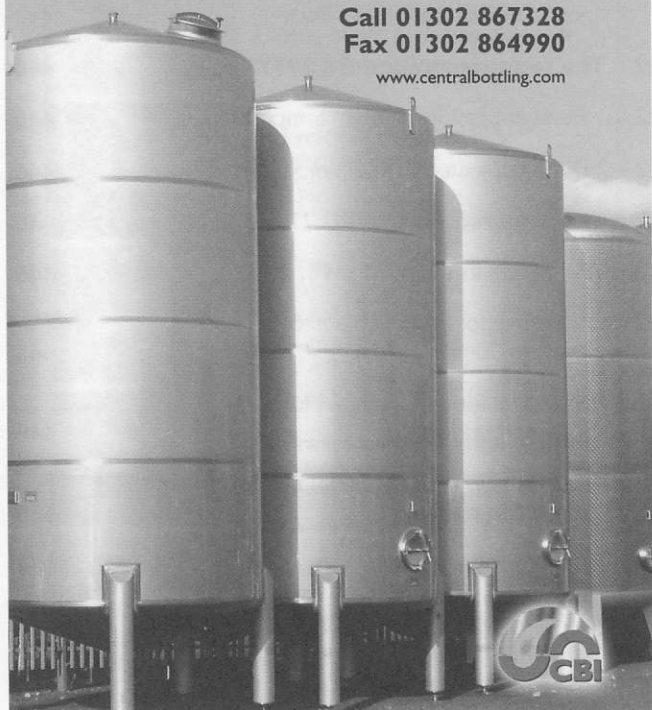
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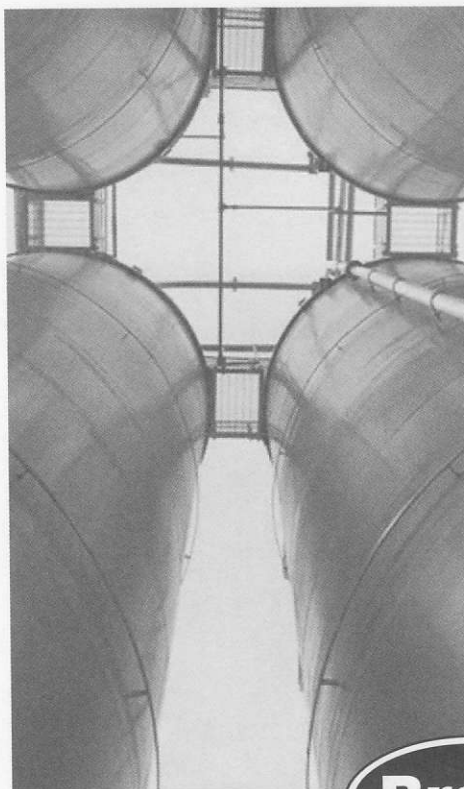
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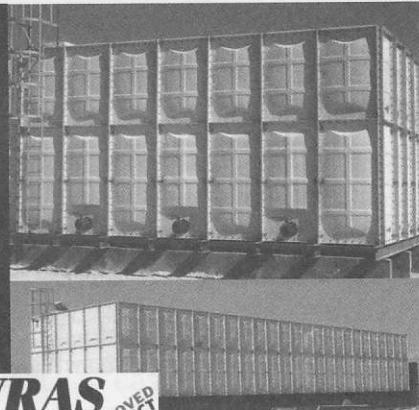
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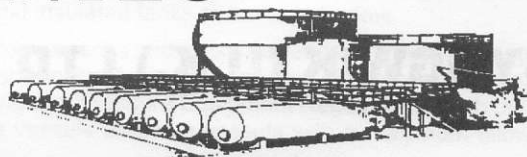
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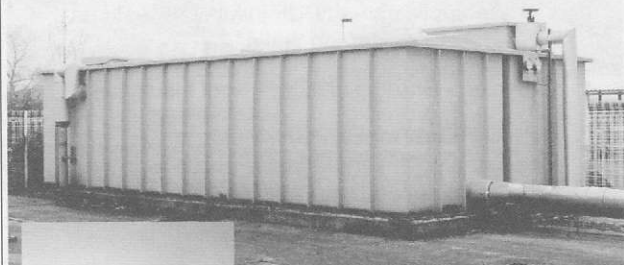
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1 Introduction

Storage tanks are a familiar part of our industrial landscape. They are used to store a multitude of different products and come in a range of sizes, from small to truly gigantic.

The transport of fluids such as oil, gas and water from their places of production or collection to the end users is rarely a continuous process. Even in cases where there seem to be direct links between the point of production and the point of use, such as gas from the United Kingdom's suppliers in the North Sea where there is a direct pipeline from the offshore rig to the consumer, the inability to match exactly production to consumption means that a pause in the overall scheme must be introduced. For water the rate of collection is a weather dependent matter and a pause is clearly a matter of necessity.

The ability to store large quantities of liquid and gaseous products was an essential element in the development of a number of industries. The petrochemical industry and locally-based town gas (i.e. gas made from coal) manufacturing facilities are those which most immediately come to mind. The movement of crude and refined oil products from their places of origin to the various markets would not be possible without the existence of economic and safe storage facilities. Similarly from the mid 19th century onwards, the ability to store large quantities of town gas in gasholders was an essential link in the industrial chain. More recently the liquid natural gas (LNG) trade, accounting for the bringing to markets of some 20% of the world's natural gas, would not be possible without the development of large scale cryogenic storage units at both export and import terminals.

In a processing plant such as an oil refinery, a chemical works or a food processing factory, production pauses are often necessary at stages in the process, perhaps to allow reactions to occur at different rates, or because products from differing intermediate processes must be brought together for a finishing process. At the end of the production process, the product cannot be immediately delivered to the customer and a further pause may be necessary to allow a suitable batch of material to be accumulated for transport. All of these pauses create the need for bulk storage.

Storage tanks are to be found constructed above ground, in ground and below ground. In shape they are most usually of vertical cylindrical form, but also come in horizontal cylindrical, spherical and rectangular forms. Products range from gases, liquids, solids and mixtures thereof. Tanks for the storage of particulate solids are more usually known as silos. Temperatures range from high temperature heated storage tanks (for products such as bitumen) through to -163°C for the storage of LNG and -196°C for liquid nitrogen.

A wide variety of storage tank types exist, including those with fixed roofs, floating roofs, internal roofs, with single walls, double walls and insulated tanks to name but a few.

It is important to distinguish between storage tanks and pressure vessels. This at first appears to be a difficult task, but help is at hand in the form of the European Pressure Equipment Directive (97/23/EC) and the United Kingdom Pressure Equipment Regulations. Both of these regulatory documents define pressure vessels as those vessels with a maximum allowable pressure greater than 0.5 bar.

Note: All pressures in this book are gauge pressures unless stated otherwise.

Thus it is convenient to define storage tanks as vessels with a maximum allowable pressure (which has been loosely taken by the industry to mean a maximum design pressure) less than 0.5 bar. The majority of storage tanks have design pressures much lower than this. For various reasons which will be discussed later, low temperature tanks have increasingly tended to have higher design pressures, but 500 mbar is still a sensible maximum. Various UK and European design Codes share this view. The USA view is somewhat different and API 620 allows a maximum design pressure of 15 psi (approximately 1000 mbar).

Pressure vessels are the subject of a companion volume in this series of publications entitled *European Pressure Equipment* written by Simon Earland, ISBN 1 86058 345 8. Pressure vessels will not be discussed in this book.

The companion books in the *European Series* confine themselves to European practices and design Codes. In the case of storage tanks, this approach does not make sense. As will become apparent, many of the major customers for the storage tank industry come from the petrochemical industry which is very much American dominated. The majority of storage tanks, including those constructed within the European Community, are specified and built to American Codes. *Storage Tanks & Equipment* therefore will seek to cover the practices and Codes of the UK, Europe and the USA.

As mentioned above the majority of storage tanks are of the vertical cylindrical type, constructed of steel or of steel alloys and fitted with fixed or floating roofs for the storage of liquids at ambient or low temperatures. It is to these tanks that this book will direct its main effort. Other tank types will be discussed, but in less detail.

2 History of storage tanks

Storage tanks in one form or another have been around for a long time. This Chapter includes a brief historical background describing how and why the current types of tanks have evolved.

A few words are devoted to in-ground tanks and to the transition from rivetted to welded tanks.

The historical development of the relevant American, British, European and some company specific design and construction Codes are reviewed.

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2.10 References

2.1 Introduction

This Chapter provides a brief resume as to why the need for liquid storage has come about and the driving forces which have caused the storage systems to increase in size and change in form with the passage of time.

2.2 Water storage

The need for the storage of water for domestic and other reasons has played a relatively minor part in the development of modern storage tanks.

Water is easily stored in reservoirs making the best use of local geographic features, clay-lined excavations or indeed in underground features accessed by wells.

Water storage tanks designed to provide a suitable pressure for local distribution systems are not uncommon. In the UK these frequently take the form of concrete tanks on elevated supporting structures located at the highest point that the local landscape will allow. These are usually of relatively modest capacity.

Elevated rectangular steel tanks of the Braithwaite type are also a common sight in industrial settings and airfields, again with the purpose of providing a suitable head of water.

In the USA and in particular in the flat landscapes of the mid-west, water towers have been used to advertise the products for which the particular town is best known. Hence water towers in the form of beer cans, pineapples and other unlikely items can often be seen. Figure 2.1 shows a typical example of such a water tower.

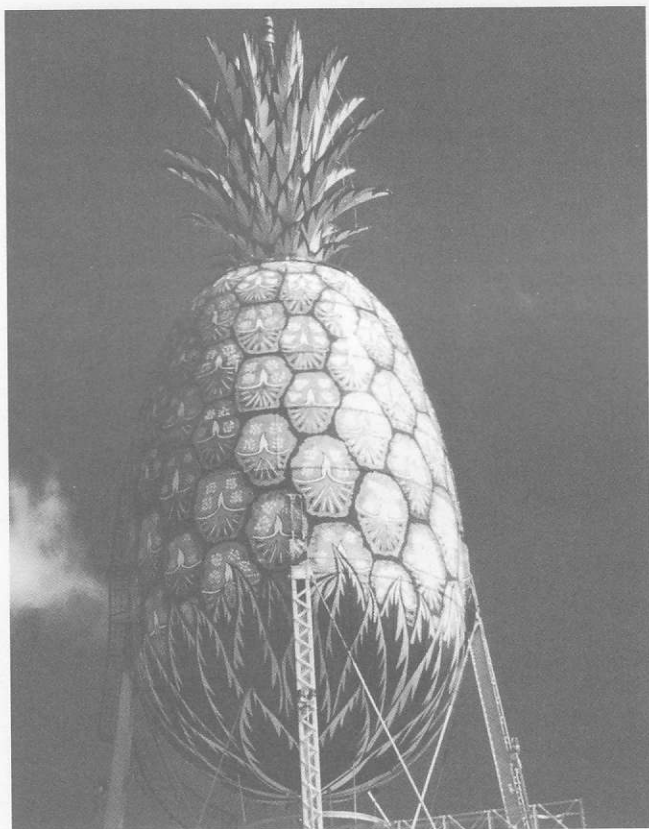


Figure 2.1 An unusual water tower
Courtesy of Chicago Bridge & Iron Company (CB & I)

The USA is also the main home of the prestressed concrete water tank. Usually these are of the Preload wire wound type. Figure 2.2 shows such a tank.

Water storage for industrial use is common, especially at power stations but despite this the real reasons for the rapid increase



Figure 2.2 Wire-wound concrete water tank
Courtesy of Preload Inc

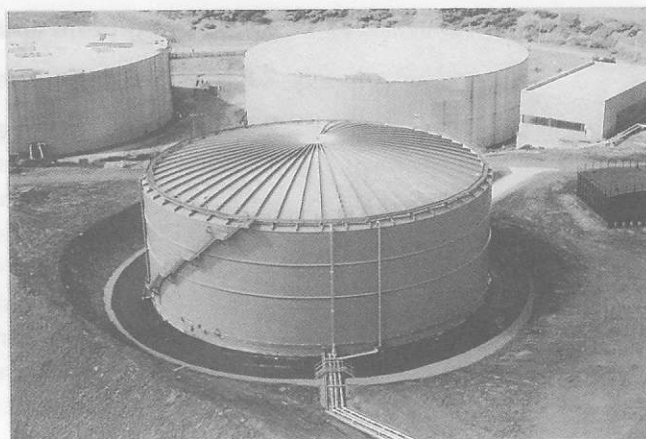


Figure 2.3 A 45 m diameter water tank
Courtesy of Whessoe

in the number and size of storage tanks lies elsewhere. Figure 2.3 shows a water tank of 45 m in diameter at the Peterhead power station in Scotland.

2.3 Oil storage

The first successful oil wells in the USA were generally agreed to have been drilled in Titusville, Pennsylvania in 1859. In Russia and Romania the first wells were drilled in 1860 and in the Dutch East Indies in 1865.

Oil-based products prior to the drilling of wells came from a variety of sources and were used in modest quantities. In addition to animal and vegetable sources, the distillation of naturally occurring mineral oil, often in the form of oil bearing shales, and the residual tars from gasworks, were the starting off point for the lighter oil products required for domestic lighting amongst other uses.

The drilling of the first wells in the USA were driven by the needs for cheaper sources of oil-based products, in particular kerosene, or paraffin as it is known in the UK. The dramatic expansion of the oil industry in the USA following the drilling of the early wells is well recorded. The formation of Standard Oil by John Rockefeller in 1870, led to this company dominating the industry from wellhead, through the refining process to the distribution and marketing of the finished products. Standard Oil not surprisingly eventually fell foul of the US antitrust laws and was broken up in 1911 into 34 separate and independent com-

panies. Many of these companies continue to exist to this day as household names such as Exxon, Mobil, Chevron, Texaco to name but a few.

Oil from the early wells in the US was placed in whisky barrels, these being a readily available receptacle at the time. The wooden barrels were not entirely suited to the storage of oil. They were originally designed for the storage of aqueous fluids which caused the wooden staves to swell and become progressively more leak tight. Oil did not have a similar effect and despite efforts to coat the insides of the barrels with glue, leakage caused by lack of tightness and mechanical damage was always a problem.

A report of the time records that at Vacuum Oil's Wandsworth works in the UK, barrels were stored in a field and during the summer they would dry out and leak. Eventually the ground became oil logged and pits had to be dug to recover the leaked oil. Figure 2.4 shows the piles of wooden barrels at Vacuum Oil's Millwall works.



Figure 2.4 Wooden barrels at Vacuum Oil's Millwall Works
Courtesy of Amadeus Press Ltd

Despite the drawbacks, wooden barrels were popular with customers providing a convenient means of storage; the general rule being that the barrel could be kept for one week before charges were imposed. They were also of appropriate size and weight for the transportation systems of the time.

Large depots included cooperages, barrelling sheds and stacking grounds where wooden barrels could be steam-cleaned, re-glued and stacked prior to being returned to service.

The wooden barrels were eventually replaced by steel barrels of 42 US gallon capacity. The barrel is to this day the most widely used measure of volume for oil based products. One US barrel = 0.159 cubic metres.

As late as 1921 it was reported that "...the barrel remains the one means of transporting and keeping oil in small volumes, although they are far from satisfactory as regards leakage. Anglo-American alone have half a million barrels in circulation.....".

The inconvenient fact that in general oil is found where there is no call for its immediate use, inevitably gave rise to the need to process, store and transport the various oil based products. Refineries were originally located close to the producing fields and the refined products transported to their markets.

Originally the bulk of the demand was for "illuminating oil" (Kerosene). As gas and electricity took the place of this oil derivative, the demand turned to lubricating oil, fuel oil and motor spirit. The spectacular increase in demand for the latter product led to refineries being gradually moved to the market end of the supply chain, where the various oil based products were produced and distributed, largely by rail in the first instance. An in-

Year	Site	Owner	Number off	Diameter (feet)	Height (feet)
1896	Hull	Crichton's Oil	1	78	30
1898	South Shields	Anglo Caucasian Oil	1	80	30
1898	Sunderland	Anglo American Oil	1	70	30
1897	Manchester	Lane and Macaudrew	1	80	38
1897	Manchester	Lane and Macaudrew	1	78	30
1897	Eccles	Lane and Macaudrew	2	80	38
1898	Eccles	Anglo Caucasian Oil	1	86	38
1899	Hull	Anglo Caucasian Oil	1	70	35
1901	Thameshaven	LATHOL	3	77	39
1901	Thameshaven	LATHOL	1	77	29
1901	Thameshaven	LATHOL	1	68	30
1901	Thameshaven	Caucasian Petroleum	1	95	39
1902	Purfleet	Anglo American Oil	1	70	30
1902	Manchester	Anglo American Oil	1	70	30
1902	London	A.Stuart	1	110	33
1902	Silvertown	J.Wilson	1	90	40
1903	Foynes	Consolidated Petroleum	1	90	30
1903	London	Caucasian Petroleum	1	70	30
1904	Manchester	General Petroleum	2	77	39
1907	Grangemouth	Anglo American Oil	1	70	30
1908	Portishead	British Petroleum	1	85	30
1908	Portishead	British Petroleum	2	60	30
1908	Thameshaven	LATHOL	2	77	39
1910	Felixtowe	Jacobs and Barringer	2	73	28

Figure e 2.5 A list of early storage tanks supplied by Whessoe
Courtesy of Whessoe

Year	Site	Number	Diameter (feet)	Height (feet)
1904	Portland	2	90	37
1905	Portland	1	90	37
1907	Gosport	4	90	37
1907	Portland	2	90	37
1908	Plymouth	4	90	37
1910	Portland	2	90	37
1911	Portsmouth	4	90	37
1913	Pembroke	2	90	37
1913	Haulbowline	2	90	37
1913	Invergordon	17	90	37
1914	Invergordon	3	90	37
1916	Perth	1	70	30
1916	Immingham	1	82	30
1916	Immingham	2	93	30
1919	Portland	1	90	37

Figure e 2.6 A list of early tank suppliers to the Admiralty
Courtesy of Whessoe

teresting book on this subject is entitled *Oil on the rails* (Reference 2.1). Storage tanks of ever increasing capacity were an essential element of this business and the listing of early tanks supplied by Whessoe (Figure 2.5) bears witness to this.

Up to the turn of the 19th century, most non sailing ships were fuelled by coal. Apart from the fact that "coaling" was hard and filthy work detested by all involved, it also ensured that around a quarter of any fleet was in port coaling up at any one time. In military terms this was a matter of serious inconvenience. The British Royal Navy prompted initially by Lord Fisher, the First Sea Lord, and later by Winston Churchill as First Lord of the Admiralty, changed the fuel of its major ships to oil prior to the start of the First World War. Oil fuelling gave the added bonus of ships being able to refuel at sea. The appearance of this new practice gave rise to the naval fuelling depots around the coast of the UK and the need for substantial reserves of storage capacity. This is reflected again in the early list of storage tanks supplied by Whessoe to the Admiralty, (Figure 2.6). Some of these tanks are still in service.

Increasing use of and trade in oil products gave rise to ever increasing requirements for transport and storage facilities. The early trade in oil and refined products was shipped in loads of around 5000 tons, carried in wooden barrels on tramp steamers or sailing ships. The earliest bespoke ships were barges used on the Caspian Sea to transport oil which was poured into the hold. These leaked so badly that ballast was placed on the decks to force the boat down and increase the water pressure to limit or reverse the leakage. Marcus Samuel of Shell ordered eight bulk oil carrying vessels of between 5000 and 6000 tons capacity each, the first one in 1892. The subsequent burgeoning in the number and size of oil tankers brought in turn corre-

sponding changes in the number and size of shore-based storage facilities.

As refining activities moved from the producing end of the chain to the supply end, refineries grew up. In the UK the first was in 1916 at Shell Haven, producing bunker fuel oil for the British Admiralty. Llandarcy followed in 1921 and in 1924 Shell opened refineries at Stanlow, Grangemouth and Adrossan, all refining imported crude oil.

The trend of increasing shipping capacity was for a while matched by the capacities of land-based storage tanks, providing the convenience of one ship filling one storage tank. The arrival on the scene of the Very Large Crude Carriers (VLCCs) of up to 500,000 dwt brought this situation to an end.

The speed at which storage facilities were being required around the world, particularly from the late 1950s up to the late 1970s gave rise the development of a standard range of tank designs, an initiative by Shell. These pre-designed tanks speeded up the ordering, fabricating and erection timescale for the refinery builders and will be discussed later in *Storage Tanks & Equipment*.

2.4 Storage needs of the petrochemical and other industries

The gradual appearance of the petrochemical industry around the world gave rise to the needs for storage of a much wider range of, mainly, liquid together with some solid products. Most were stored above ground in vertical cylindrical tanks. The properties of the different products caused the types of tanks to vary widely. Hence the development of heated tanks for bitumen storage, low temperature tanks for refrigerated liquid gases, corrosion resistant tanks for aggressive products, clean tanks for water, food and pharmaceutical materials, silos for solids and special measures for toxic materials.

2.5 Gas storage

The early gas industry in the UK was based on the production of coal gas in gasworks. Rather than transport the gas for large distances from producer to user, it was more convenient to transport the raw material (coal) and manufacture the gas close to the user. Hence the growth of the gasworks in most towns of any size in the UK.

As the production of gas was at best a batch process and as demand was on an uneven daily, and indeed often a longer term cycle, there arose a need to provide for buffer storage of gas. There was also a need to maintain the gas in the distribution system at a small positive pressure and it would be clearly be convenient to the user if this pressure could be relatively constant.

These two needs were admirably achieved by the evolution of the gasholder, once a familiar landmark of most UK towns, but perhaps less so these days. Incidentally, the gasholder seems to have become one of the very few forms of storage tank to have achieved a measure of affection in the eyes of the public, several indeed to the point where they have become listed buildings. The best known in the UK are perhaps the group which could be seen on leaving King's Cross Station in London, although sadly only one seems to have survived the current building developments in the area.

The gasholders seem to have increased in capacity earlier and faster than their liquid storage cousins and would have encountered and solved the various structural problems associated with size at an earlier date.

The list in Figure 2.7 of early gasholders designed and constructed by Whessoe shows this, indeed the 180 ft diameter tank at Newcastle, designed and constructed around the turn of

Year	Site	Owner	Number off	Diameter (feet)
1891	Redheugh	Newcastle and Gateshead Gas Co.	1	180
1892	Willington	J. Coates	1	42
1893	Stanley	West Stanley Colliery	1	44
1893	Durham	Durham County Assylum	1	60
1894	Easington	Gas Light and Coke Co.	1	38
1895	Blyth	Blyth Gas Co.	1	96
1896	Appelby	Appelby Gas Co.	1	42
1896	Huntley	Huntley Gas Co.	1	45
1896	Kilmarnock	Kilmarnock Gas Co.	1	100
1896	Malton	Malton Gas Co.	1	84
1896	Market Weighton	Market Weighton Gas Co.	1	42
1895	Pickering	Pickering Gas Co.	1	45
1897	Guisborough	Guisborough Gas Co.	1	45
1898	Filey	Filey Gas Co.	1	60
1903	Shildon	N.E. Railway Co.	1	60
1905	Tynemouth	Tynemouth Gas Co.	1	1.5 x 10 ⁶ ft ³
1914	Stanley	West Stanley Colliery	1	119

Figure 2.7 A list of early gasholders
Courtesy of Whessoe

the last century, would even have been considered a big tank some 50 years later. A 12 million cubic feet gasholder built in Sydney, Australia, during the First World War was considerable bigger with a diameter of 300 feet. Wet and dry seal gasholders are discussed briefly in Chapter 27 of *Storage Tanks & Equipment*.

2.6 Refrigerated liquefied gas storage

Products such as propane and butane were originally stored in small quantities in pressure vessels or spheres. As the requirement came to store ever larger quantities of these products, the pressure storage option became increasingly expensive and unattractive from a practical and safety point of view. Low pressure storage in refrigerated liquid form became the norm and the development of these tanks in terms of their increasing size and sophistication from a safety point of view will be covered in detail in later Chapters.

Natural gas is a methane-dominated mixture of gases which is often found with oil and used to be considered an inconvenience to the oil industry. Consequently the gas was often flared at the discovery site. Apart from being an economic nonsense to waste such a useful and valuable raw material, it is now sensibly considered environmentally unacceptable to burn large quantities of gas. The growth of the world's LNG trading from its early days between Arzew in Algeria, Canvey Island in the UK and Fos sur Mer in France, will be considered in the low temperature section of this book.

As with the oil trading, the scale of activities has changed here too. The first LNG carrier was *Methane Pioneer* which was a converted liberty ship with a liquid capacity of 5000 m³. This was followed by *Methane Princess* and *Methane Progress* each of 27,400 m³ capacity. The latest carriers are of up to 140,000 m³ in capacity. Similarly the first LNG tank at Canvey Island was of 2000 m³ capacity whilst in Japan an above ground tank of 180,000 m³ has been constructed and even larger tanks are being discussed.

2.7 Above ground and in or below ground storage systems

The bulk of the world's storage capacity for liquids is in the form of above ground tanks of the vertical cylindrical type. It is to this type of tank that the majority of *Storage Tanks & Equipment* will be devoted.

There are a number of areas where in ground storage is commonly adopted. One of these is petrol station forecourt tanks storing petrol and diesel fuels for sale to motorists. These tanks together with the smaller above ground tanks for the same purpose are described in considerable detail in Wayne Geyer's book (*Reference 2.2*). There seems little point in revisiting this type of tank in this book.

Another use for such tanks is for the storage of aviation fuel, particularly at military air bases, where the above ground storage of such flammable products would represent unacceptable risks.

Various products including LPG are stored in below-ground caverns. These caverns are conventionally mined in suitable rock and usually consist of interlinked horizontal tunnels of constant cross-section. These can have storage capacities of up to 250,000m³.

In Germany, a substantial part of the Federal Fuel Reserve is stored in caverns in salt domes. Salt domes are natural geological phenomena and can be mined by a technique known as "solution mining". These can be gigantic as illustrated by Figure 2.8.

All of these in and below ground storage solutions are briefly described in *Storage Tanks & Equipment*.

2.8 Riveted and welded structures

Most of the early liquid storage tanks were constructed from steel with riveted joints. API Standard 12A was the specification for "Oil Storage Tanks with Riveted Shells" (it allowed either riveted or welded bottoms) for tanks with capacities of between 240 bbl (38 m³) and 255,000 bbl (40,545 m³). The maximum end of the capacity range represents quite a big tank even by today's standards. Allowing for dead space at the bottom and top, this is a tank of 55 m in diameter and some 18 m in shell height. Much of the technology came from the shipbuilding industry.

Welding progressively took over from riveted construction from the late 1920s and riveted tanks became unusual from the late 1930s. The foreword to API 12A stated "at the November 1941 meeting the tank committee agreed that all committee activity

involving modifications and revisions of Standard 12A be suspended". This was clearly the end of the line for riveted tanks.

The Standard was last issued in 1951 and any copy currently provided bears the legend "copy provided for historical purposes only". The lengthy transition between the two metal joining techniques owed much to a suspicion within the more conservative operators of storage tanks that the newfangled welding was an unsuitable technique. This was based on a number of sudden failures of early welded tanks. Electric arc welding was not the closely controlled and well understood technique that it is today and the importance of toughness in preventing brittle fracture, particularly in the weld metal and the heat affected zone, (HAZ), was not appreciated.

It is interesting that welded bottoms with riveted shells were allowed. This is perhaps a tacit appreciation that the tank bottom, with its very low operating stresses, is not susceptible to brittle failure in the same way as is the more highly-stressed tank shell.

API 12C, first issued in 1935, covered welded tanks. This Standard imposed a "nick break test". This was a welded specimen which had a notch or nick made in it and was then subjected to an unquantified beating with a hammer. Brutal though this sounds, it was an attempt to ensure some measure of toughness in the welded joint, something that would be done by Charpy V-notch testing today.

Although riveted tanks are now only of historical interest, the reader of API 12A cannot fail to be impressed by the skills which must have been required at the design, fabrication and erection stages by the personnel involved with this type of tank. Even the simple shell joints appear complex and fittings must have been a nightmare to produce. Caulking of the shell (outside) and the bottom (inside) is a requirement. Bottoms, as a matter of necessity, had to be constructed at a height, and had the lower shell course added and the whole assembly water-tested whilst still supported.

2.9 History of the design and construction regulations

The storage of large volumes of products which were in the main highly flammable is a subject which was bound to attract regulation and standardisation from a number of interested parties.

2.9.1 American Standards

Tank owners, tank makers, fire officials and insurers in the USA were the first to address this subject and an association of tank manufacturers, later to become the Steel Tank Institute (STI) was formed in 1916. At or around the same time Underwriters Laboratories Inc (UL) was developing its safety standards for atmospheric storage tanks.

The first Standard for above ground steel storage tanks was produced by UL in 1922. UL 142 was entitled *Steel Above-ground Tanks for Flammable and Combustible Liquids*. The same organisation published the first edition of IL 58 entitled *Standard for Steel Underground Tanks for Flammable and Combustible Liquids* in 1925, a reaction to the increasing number of urban petrol stations in the USA.

The National Board of Fire Underwriters (NFBU) published NFBU 30 around 1904 with the unwieldy title *Rules and Requirements for the Construction and Installation of Systems for Storing 250 Gallons or Less of Fluids Which at Ordinary Temperatures Give Off Inflammable Vapors, as Recommended by its Committee of Consulting Engineers*.

Over a period of time the NFBU became the National Fire Protection Association (NFPA), an organisation which is familiar to

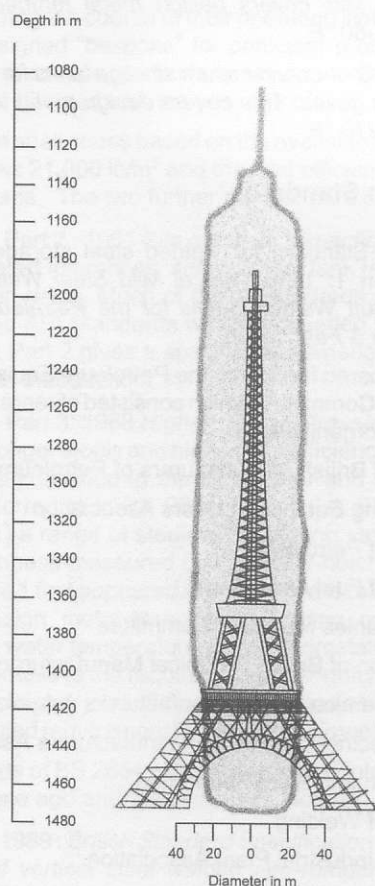


Figure 2.8 Salt domes are natural geological phenomena; these can be gigantic

us today. NFBU 30 became NFPA 30L published in 1913, and today this document has become NFPA 30 (Flammable and Combustible Liquids Code) first published in 1957. NFPA Codes are influential worldwide in both the ambient and the low temperature storage industries.

The American Petroleum Institute (API) was formed in 1919 and went on to produce two of the most influential Codes in the areas of ambient tankage (API 650, formerly API 12C) and low temperature tankage (API 620). These documents and their influence will be discussed in later Chapters at some length. API 12C is one of a family of Codes covering liquid storage tanks. The full set contains the following:

- **12A : Specification for oil-storage tanks with riveted shells.** This covers material selection, design, fabrication and erection requirements for vertical, cylindrical, above ground steel tanks with riveted shells in nominal capacities of 240 bbl (38 m³) to 255,000 bbl (40,545 m³) (in standard sizes) for oil storage.
- **12B : Specification for bolted production tanks.** This covers the material selection, design and erection requirements of vertical, cylindrical, above ground, bolted steel production tanks in nominal capacities of 100 bbl (16 m³) to 10,000 bbl (1590 m³) (in standard sizes) for oilfield service. It also includes appurtenance requirements.
- **12C : Specification for welded oil storage tanks.** This covers the material selection, design, fabrication and erection requirements for vertical, cylindrical, above ground, closed and open top, welded steel tanks in various sizes and capacities, for oil storage. It also includes appurtenance requirements and recommendations for the use of low alloy high strength steels, and aluminium alloys, in tank construction. The second edition of this part was published in 1936, so it must have its origins at an earlier date.
- **12D : Large welded production tanks.** This covers the material selection, design, fabrication and erection requirements for vertical, cylindrical, above ground, welded steel, production tanks in nominal capacities of 500 bbl (80 m³) to 3,000 bbl (477 m³) (in standard sizes) for oilfield service.
- **12E : Specification for wooden production tanks.** This covers the material selection, design, fabrication and erection requirements for vertical, cylindrical, above ground, closed top, wooden production tanks in nominal capacities of 130 bbl (21 m³) to 1,500 bbl (239 m³) (in standard sizes) for oil field service.
- **12F : Specification for small welded production tanks.** This covers the material selection, design and construction requirements for vertical, cylindrical, above ground, shop-welded, steel, production tanks in nominal capacities of 90 bbl (14 m³) to 400 bbl (63 m³) (in standard sizes up to a maximum diameter of 12 feet) for oilfield service.
- **12G : Specification for aluminium alloy welded storage tanks.** This covers the material selection, design, fabrication, erection and testing requirements for vertical, cylindrical, above ground, closed and open top, welded aluminium alloy storage tanks in various sizes and capacities.

The latest editions of the American Standards which interest tank designers and builders are:

- API 650 — *Welded Steel Tanks for Oil Storage: Tenth Edition, November 1998*
- API 620 — *Design and Construction of Large, Welded, Low-Pressure Storage Tanks: Tenth Edition, February 2002*

API 620 provides rules for ambient tanks for pressures up to 15 psig and is not restricted to vertical cylindrical forms. It has been used to produce designs for such interesting vessels as

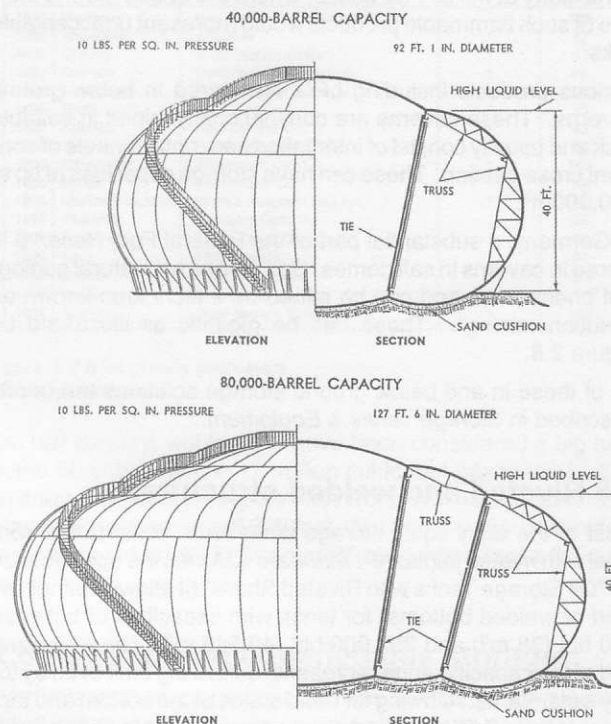


Figure 2.9 Noded hemispheroids

the noded hemispheroids shown in Figure 2.9. It also contains two Appendices for low temperature tank design. These are:

- **Appendix R — Low pressure storage tanks for refrigerated products.** This covers design metal temperatures from +40 °F to -60 °F.
- **Appendix Q — Low pressure storage tanks for liquefied hydrocarbon gases.** This covers design metal temperatures down to -270 °F.

2.9.2 British Standards

The first UK Standard for welded steel storage tanks was BS 2454: Part 1: 1956 *Vertical Mild Steel Welded Storage Tanks with Butt Welded Shells for the Petroleum Industry: Part 1 Design & Fabrication*.

This was prepared for BSI by the Petroleum Equipment Industry Standards Committee, which consisted of representatives of the following organisations:

Council of British Manufacturers of Petroleum Equipment
Engineering Equipment Users Association
Institute of Petroleum
Ministry of Fuel and Power
Oil Companies Materials Committee
Association of British Chemical Manufacturers
British Chemical Plant Manufacturers Association
British Electrical and Allied Manufacturers Association
British Iron and Steel Federation
Institute of Welding
Tank and Industrial Plant Association

It seems perhaps a little unnecessary to list all of the participating organisations in the preparation of this national Standard, but it serves to illustrate the width of industrial knowledge can-

vassed at that time and the size of the committee involved in the production of the document. This is something which contrasts with the present day where it is often difficult to assemble a viable committee to write or edit a Standard.

This Standard classified tanks into a number of categories:

- Non-pressure fixed roof tanks
- Pressure fixed roof tanks (limited to 128 ft diameter)
- Open-top tanks

It also proposed standard shell plate sizes and tank diameters giving effectively a standard range of tanks. This followed the Shell approach, which will be discussed later. This standardisation was a reaction to the level of tank building activity within the petroleum industry at that time. A range of standard tank sizes which had in effect been pre-designed was clearly in the interests of the industry in speeding up the fabrication and erection process and opening up the business to companies who perhaps did not have the facilities to carry out the detailed design aspects of this work.

The tanks were referred to by a coding system, which contained information on the tank diameter, shell height, pressure category and plate width. Hence the customer needed only to order a BNPB 1608, for the tank manufacturer to know that a "non-pressure fixed roof tank of 160 ft in diameter with eight shell courses each 7.25 ft wide" was required. Extracts from this Code are shown in Figure 2.10, explaining the coding system and show a few of the standard capacity/shell plate thickness tables.

Unlike the API Standard of the same period, the British Standard required a design product specific gravity of 1.00 in all cases. This was quite deliberate and allowed for the tank to be used for any product commonly encountered in the petrochemical industry without fear of over-stressing the tank shell. It is not uncommon for tanks to change their service from one product to another during the course of their operating lifetime and having tanks designed "bespoke" for particular product gravities runs the risk of misuse, particularly when records are not well maintained or dimmed with the passage of time.

The allowable shell stress based on the available carbon steels of the time was 21,000 lb/in² and the joint efficiency factor was 0.85 in all cases. The two further parts of BS 2654 followed:

- BS 2654: Part 2: 1961 *Site erection, inspection and testing* This covered tolerances, site welding, tank testing and inspection in detail. Much of these Standards owed a great deal to the API Standards which preceded them, indeed BS 2654: Part 2 gives a specific acknowledgement to this effect in its introduction.
- BS 2654: Part 3: 1968 *Higher Design Stresses* allowed the use of stronger steels and higher joint efficiencies. BS 4360: 1968 was published in the same year and added to the steels referred to in BS 2654: Part 1 (i.e. BS 13 and BS 1501-101) a range of steels with differing strength grades and toughness measured by Charpy V-notch impact testing. Figure 1 first appeared in this Standard relating the minimum design metal temperature during operation, the minimum water temperature during hydrostatic testing and plate thickness to the required Charpy V-notch test temperature. The higher joint efficiency of 1.0 was accompanied by an enhanced requirement for radiographic weld inspection.

The three parts of BS 2654 were consolidated into a single volume some time ago and the current version is:

- BS 2654:1989: *British Standard Specification for the Manufacture of vertical steel welded non-refrigerated storage tanks with butt-welded shells for the petroleum industry*. This Standard has not been updated since 1989 as may have been expected because of the "standstill" imposed

whilst the European Standard covering the same subject area was being prepared.

For the storage of low temperature products, the British Standards followed the practice adopted by API in providing separate rules for temperatures down to -50 °C and for temperatures from -50 °C down to -196 °C. Rather than using the API method of having two appendices covering the specific requirements of the two temperature ranges with the main body of the code addressing more general issues, it was decided to produce two separate codes. These were:

- BS 4741: 1971 *Vertical Cylindrical Welded Steel Tanks for low temperature service*. Single wall tanks for temperatures down to -50 °C. BSI London (now superseded by BS 7777: 1993).
- BS 5387: 1976 *Vertical Cylindrical Welded Storage Tanks for low temperature service*. Double Wall Tanks for Temperatures down to -196 °C. BSI London (now superseded by BS 7777: 1993).

These Standards only considered single containment storage systems. As will be described, various events created the need for a Standard which provided a framework for double and full containment systems for low temperature products. Following the work of the EEMUA storage tank committee described in Section 2.9.6, a new British Standard was issued in 1993 which addressed all of the low temperature products and all forms of containment. This was:

- BS 7777:1993 *Flat-bottomed, vertical cylindrical storage tanks for low temperature service: Parts 1 to 4*.

2.9.3 The European Standards

Around 1993 the European Standard Committee TC 265 was formed. The secretariat of this committee was given to the British Standards Institution (BSI) and most of the meetings were held at BSI headquarters in London. The work of the committee was divided into:

- A Standard for ambient temperature tanks entitled:
Specification for the design and manufacture of site built, vertical, cylindrical, flat-bottomed, above ground, welded, metallic tanks for the storage of liquids at ambient temperature and above – Part 1 – Steel Tanks (prEN 14015-1).

Note: Part 2 is intended to cover aluminium alloy tanks and will possibly follow later. It is currently suffering from limited industrial interest.

- A Standard for low temperature tanks entitled:
Specification for the design, construction and installation of site built, vertical, cylindrical, flat-bottomed steel tanks for the storage of refrigerated, liquefied gases with operating temperatures between -5 °C and -165 °C (prEN 14620 - Parts 1/2/3/4/5)

Note: The pr prefix indicates a provisional Euronorm, i.e. one where the committee responsible has finished its complete draft which is then issued for public comment. The comments received are reviewed by the committee and the draft edited prior to the Standard being issued as a full Euronorm without the prefix.

The work proceeded slowly, not least because of difficulties in resolving strongly held views from the various national delegations regarding differing practices in the countries which they represented. Indicative of the rate of progress was the comment by John de Wit, then chairman of CEN TC 265, that a final draft of the low temperature document would not be ready until the end of 1995.

The group working on the ambient tank Code issued a draft for public comment in 2000. Comments have been received and

B.S. 2654 : Part 1 : 1956

BRITISH STANDARD SPECIFICATION FOR VERTICAL MILD STEEL WELDED STORAGE TANKS, WITH BUTT-WELDED SHELLS, FOR THE PETROLEUM INDUSTRY

PART 1. DESIGN AND FABRICATION

FOREWORD

This British Standard, prepared under the authority of the Petroleum Equipment Industry Standards Committee, is designed to provide the petroleum industry with tanks of adequate safety, reasonable economy and in a range of suitable capacities.

In the further interests of economy, supply and uniformity of practice it is strongly recommended that the sizes of plates used for tanks of all capacities shall be limited to three (Clause 4). The standard tank sizes which result from the adoption of this proposal are given in Tables 1 to 8.

This part of the standard deals with design and fabrication of tanks; Part 2 will deal with site erection, inspection and testing.

SPECIFICATION

SECTION ONE : GENERAL

SCOPE

1. This British Standard relates to the materials, design and fabrication of vertical mild steel cylindrical welded tanks for the petroleum industry, for erection above ground, of the following designs:—

- Non-pressure fixed roof tanks (all sizes).
- Pressure fixed roof tanks (up to 128 ft diameter only).
- Open-top tanks (all sizes).

This standard specifies the use only of butt-welded shells and includes reference to mountings, stairways and handrailings.

This standard does not include the design and construction of floating roofs.

NOTE. Attention is drawn to Appendix F which tabulates the information to be supplied by the purchaser on alternatives permitted by this British Standard.

DESIGN CONDITIONS

2. *a. Non-pressure tanks* shall be suitable for working at atmospheric pressure, but designed for an internal pressure of 3 in. water gauge and a vacuum as specified for shells in Clause 14 *f* and for roofs in Clause 26 (see also Clause 15).

b. Pressure tanks shall be designed for an internal pressure of 8 in. water gauge and 2½ in. water gauge vacuum (see Clauses 15 and 26).

c. Tanks may be designed in accordance with this specification to withstand higher pressure and/or vacuum conditions, provided the allowable stresses given in this standard are not exceeded.

STANDARD TANK SIZES

3. Standard ranges of tank sizes based on the plate sizes specified in Clause 4 are given in the following tables:—

Type A (Maximum plate width 6·00 ft)	Table 1 Capacity in cubic feet
	Table 2 Capacity in cubic metres
	Table 3 Shell plate thicknesses
	Table 4 Heights in feet.
Type B (Maximum plate width 7·25 ft)	Table 5 Capacity in cubic feet
	Table 6 Capacity in cubic metres
	Table 7 Shell plate thicknesses
	Table 8 Heights in feet.

NOTE. Tables of equivalent capacity in U.S. barrels and imperial gallons are given in Appendices A, B, C and D.

In Tables 1 to 8 a maximum diameter of 200 ft and a maximum height of nine courses are given. These values may be exceeded provided the maximum shell plate thickness does not exceed 1½ in.

STANDARD PLATE SIZES

4. *a. General.* The standard plate sizes, which form the basis of the standard tank sizes and heights in Tables 1 to 8, are given below:

Thickness	Length	Width	
		Type A	Type B
Inches	Feet	Feet	Feet
⅜ or ¼	15·7 (5 π ft)	5·00	5·00
¼ up to but excluding ⅜	25·13 (8 π ft)	6·00	6·00
⅜ and over	25·13 (8 π ft)	6·00	7·25

The above plate sizes may be modified by agreement between the purchaser and the manufacturer.

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b. *Rolling margins.* Unless otherwise agreed between purchaser and manufacturer, no plate shall be under the specified thickness at any part, nor shall it exceed the

calculated weight by more than the appropriate rolling weight tolerance as shown in the following table:—

SCHEDULE OF PERCENTAGE ROLLING WEIGHT TOLERANCES FOR SHELL PLATES

Ordered thickness	Widths								
	Under 48 in.	48 in. to under 60 in.	60 in. to under 72 in.	72 in. to under 84 in.	84 in. to under 96 in.	96 in. to under 108 in.	108 in. to under 120 in.	120 in. to under 132 in.	132 in. and over
	Per cent	Per cent	Per cent	Per cent	Per cent	Per cent	Per cent	Per cent	Per cent
$\frac{3}{16}$ in. to under $\frac{1}{4}$ in.	10	10	10	10	10	—	—	—	
$\frac{1}{4}$ in. to under $\frac{5}{16}$ in.	5	5	5	7	9	12	12	—	
$\frac{5}{16}$ in. to under $\frac{3}{8}$ in.	5	5	5	6	7.5	10	11	12	
$\frac{3}{8}$ in. to under $\frac{1}{2}$ in.	5	5	5	6	6	8	9	11	15
$\frac{1}{2}$ in. to under $\frac{5}{8}$ in.	5	5	5	5	6	7.5	8	9	12
$\frac{5}{8}$ in. to under $\frac{3}{4}$ in.	5	5	5	5	6	6	7.5	9	10
$\frac{3}{4}$ in. to under 1 in.	5	5	5	5	5	5	7	8	9
1 in. to 1½ in.	5	5	5	5	5	5	6	7	8
1½ in. to 2 in.	5	5	5	5	5	5	5	6	7

CODING

5. For easy reference to tank sizes and types in cables and correspondence, etc., a coding system for each size of tank is given below.

The code system consists of a letter prefix denoting the three designs of tanks as listed below:

a. *Prefix.*

Fixed roof tank, non-pressure = BNP
 Fixed roof tank, pressure = BLP
 Open-top tank = BOT

b. The above prefixes to be followed by a type symbol A or B denoting the maximum plate width adopted, see Clause 4 a, together with a number consisting of the diameter of the tank in feet and number of courses.

c. *Examples.*

Pressure roof, maximum plate width 6.00 ft 96 ft diameter four courses deep = BLPA 964.

Non-pressure roof, maximum plate width 7.25 ft 160 ft diameter eight courses deep = BNPB 1608.

Open-top, maximum plate width 6.00 ft 80 ft diameter six courses deep = BOTA 806.

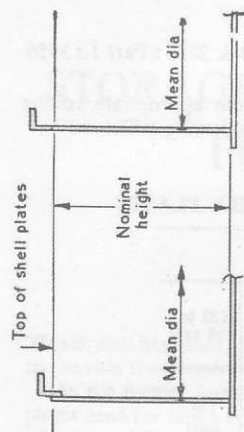


TABLE 4. STANDARD TANKS—TYPE A
NOMINAL TANK HEIGHTS IN FEET
BUTT-WELDED SHELLS
MAX. PLATE WIDTH 6·00 FEET

Number of courses	TANK DIAMETERS IN FEET															
	Type BNP—Design for non-pressure only Type BLP—Design for pressure								Type BNP Design for non-pressure only							
	10	15	20	25	30	35	40	48	56	64	80	96	112	120	128	136
1	5	5	5	5	5	5	5	6	6	6	6	6	6	6	6	6
2	10	10	10	10	10	10	10	12	12	12	12	12	12	12	12	12
3	15	15	15	15	15	15	15	18	18	18	18	18	18	18	18	18
4	20	20	20	20	20	20	20	24	24	24	24	24	24	24	24	24
5	—	25	25	25	25	25	25	30	30	30	30	30	30	30	30	30
6	—	30	30	30	30	30	30	36	36	36	36	36	36	36	36	36
7	—	—	—	35	35	35	35	42	42	42	42	42	42	42	42	42
8	—	—	—	—	—	—	40	48	48	48	48	48	48	48	48	48
9	—	—	—	—	—	—	45	54	54	54	54	54	54	54	54	54

I Denotes range of plates 5 ft 0 in. wide $\times$ 5 π feet (15·7 feet) long.
II Denotes range of plates 6 ft 0 in. wide $\times$ 8 π feet (25·13 feet) long.

are currently being reviewed and where appropriate edited into the final text. The document is hoped to be issued as a Euronorm (EN) shortly. As is the case with all new EN Standards, the national Standards in the areas covered by the new Standard are subject to standstill. This means that they are in effect frozen at the point when TC 265 began its work. In this particular case the standstill has been in force for much longer than was originally anticipated.

In terms of its contents the new ambient tank Standard will in the main follow the directions set by the earlier European national Standards, which in turn owe a great deal to the corresponding API Standards. The volume of fossilised experience in these earlier documents is both difficult and probably unwise to ignore.

The low temperature Euronorm is following close behind its ambient temperature counterpart and was issued for public comment in March 2003. It is hoped that the comments can be reviewed and consolidated into this Euronorm rather more quickly than has been the case with the ambient tank Standard.

Again, in terms of content it follows earlier European and API Standards as well as the EEUUMA Standard discussed in Section 2.9.6.

The differences will be described and discussed later in *Storage Tanks & Equipment*.

2.9.4 Other European national Standards

Most European countries have their own national Standards for ambient tanks (e.g. Germany has DIN 4119 Parts 1 and 2).

As these Standards are now about to be replaced by the two new Euronorms, there seems little point in discussing them further.

2.9.5 Related Standards

There are numerous Standards covering a whole variety of subjects such as materials, site layout and tank spacing requirements, safety issues, etc which are necessary for tank designers and manufacturers and which will be mentioned in this book. These come from organisations such as API, ASTM, The National Fire Protection Association (NFPA), European Standards, British Standards Institution (BSI) and bodies such as The Institute of Petroleum (IP).

These will be discussed as and when required.

2.9.6 The EEUUMA Standard

Up to about 1976 refrigerated gases were stored in single containment tanks surrounded by a low remote bund. An event in 1976 caused the industry to review liquid containment systems for these products from a safety point of view. The Standards in force at the time (API 620, BS 4741 and BS 5387) considered only single containment systems and there was clearly a need for a Standard which encompassed other forms of containment to avoid misunderstandings and misinterpretations.

The Engineering Equipment and Materials Users Association (EEMUA) is a UK-based equipment users association and was felt to be an appropriate body to propose and draft a set of rules to cover this regulatory shortfall. In 1987 EEMUA 147 was published, and after a period of time sufficient to allow for the industry's views of the document to be known, was given to the British Standards Committee PVE/15 to form the basis of BS 7777.

The subject of the various containment systems for the safe storage of refrigerated liquid gases will be discussed at greater length later in *Storage Tanks & Equipment*.

2.9.7 Company Standards

Over the years, and for a number of reasons, some of the major companies involved with the use of or the design and construction of storage tanks found the need to produce their own Standards. This could be because they thought that the national Standards available at the time did not reflect their requirements sufficiently, or for a need to standardise a range of tank types or sizes. Some of these have become influential within the industry and have attained the status of unofficial Standards.

2.9.7.1 The Shell Standards

The method of categorising and coding of vertical tanks used in BS 2654: Part 1: 1957, is almost identical to that used in the Shell publication *Standard Tanks*, also first published in 1957.

The closeness of the Shell and BS approaches in this matter is no real surprise. John de Wit, the Shell tank expert from SIPM in The Hague, was Chairman of the British Standards Committee CP12 (later PVE 15), which looked after ambient and low temperature storage tank codes. Shell always used BS Codes, unlike much of the petrochemical industry which was firmly wedded to Codes of US origin.

These Standards were updated and republished in three volumes in 1962/3. They included standard designs for a range of sizes of fixed roof and open top vertical tanks, together with a range of horizontal tanks. Not only did these designs cover the shell plating as the early BS, but they also included standard designs for roofs, bottoms and a range of standardised tank fittings as well. The roof types used were the folded plate cone, radial rafter cone, truss-supported cone and internally-framed dome. An example of a 96ft diameter trussed cone roof tank is shown in Figure 2.11.

Although these Standards were prepared for the exclusive use of the Shell Company to procure large numbers of tanks for the refinery expansions of the 1960s and 1970s. The need to issue the documents to tank building contractors ensured that they rapidly spread throughout the industry and were shamelessly copied and used by others. Consequently they became an "unofficial" Standard and are used as such to this day. Whilst this may have been annoying for the company, it is a tribute to the authors of these documents and to the sound and practical engineering that they contain.

2.9.7.2 The Chicago Bridge Engineering Standards

Chicago Bridge & Iron Company was responsible for numerous significant developments in the storage tank field and licensed its technology to a number of other companies over the years. Its floating roof designs were encapsulated in a series of particularly well-produced documents, which through the licensing process filtered out into the tank building industry and were again shamelessly plagiarised, becoming in effect the "unofficial" Standard.

2.9.7.3 The Exxon basic practices

The Exxon/Esso organisation published its own Standards covering a wide range of subjects including storage tanks for a number of products. These Standards were based on US Standards and practices adjusted to suit the perceived needs of the company.

2.9.8 Standards for other products

The foregoing has concentrated somewhat myopically on the storage of flammable products, mainly for the petrochemical industry. Indeed a number of the Standards discussed above have "petrochemical" or "oil industry" in their titles. There are other products and some of these have their own Standards.

The American Water Works Association (AWWA) has produced a number of Standards on its own and some of these are listed below:

ANSI/AWWA D100-96

Welded Steel Tanks for Water Storage

ANSI/AWWA D103-97

Factory-Coated Bolted Steel Tanks for Water Storage

ANSI/AWWA D110-95

Wire and Strand Wound Circular Prestressed Concrete Water Tanks

ANSI/AWWA D115-95

Circular Prestressed Concrete Water Tanks with Circumferential Tendons

These are all interesting documents and they will be discussed in later Chapters of *Storage Tanks & Equipment*.

AWWA D100 has a particularly good seismic design section. This is not surprising as the chairman of the D100 Revision Task Force is Bob Wozniak, a "world guru" in the area of seismic tank design and someone whose work will be discussed in detail in later Chapters.

2.10 References

- 2.1 *Oil on the rails*, Alan Coppin, The Historical Model Railway Society and Amadeus Press Ltd of Huddersfield. Published 1999, ISBN 0 902 835 17 3.
- 2.2 *Handbook of storage tank systems*, Edited by Wayne B. Geyer, Marcel Dekker Inc., ISBN 0 8247 8589 4.

3 Ambient temperature storage tank design

The design of vertical, cylindrical tanks for the storage of liquids at ambient temperatures can be divided into three basic areas:

- The shell
- The bottom
- The roof

The design of each of these is discussed in detail in this Chapter.

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3.1 European tank design Codes

The European Codes which will be discussed here are as follows:

- European Standard prEN 14015 -1 : 2000
- German Standard DIN 4119 Parts 1 & 2

3.1.1 European Standard prEN 14015-1 : 2000

This is a draft document which has been through the public comment procedure and will soon be issued as a full European Standard. The content of the final version is not expected to differ significantly from the draft. The full title of the English version is: "Specification for the design and manufacture of site built, vertical, cylindrical, flat bottomed, above ground, welded, metallic tanks for the storage of liquids at ambient temperature and above - Part 1: Steel tanks".

The Standard appears to be based on BS 2654 and API 650, together with some informative Annexes and all together is a comprehensive document. Some interesting aspects of certain parts of the Standard are outlined below:

3.1.1.1 Pressure rating

The Standard allows positive design pressures up to 500 mbar in four categories:

- Non-pressure, up to 10 mbar
- Low-pressure, up to 25 mbar
- High-pressure, up to 60 mbar
- Very high-pressure, up to 500 mbar

The maximum negative pressure which applies only to Very high-pressure tanks is -20 mbar. However the requirements given in the Standard for shell stability are only valid for negative pressures up to -8.5 mbar, beyond this value a suitable design methodology has to be agreed between the tank purchaser and the manufacturer.

3.1.1.2 Temperature rating

The temperature range is from 300°C down to -40°C. For temperatures above 100°C, the elevated temperature yield stress values of carbon and carbon manganese steels shall be certified by the steel supplier. The Standard gives a table of steels to Standard EN 10028 - 2 & 3 for use at elevated temperatures. For tanks constructed in stainless steel materials, the yield stress is taken as the 1 % proof stress for tanks subjected to ambient and elevated temperatures.

3.1.1.3 Materials

Carbon and carbon manganese steels for use in the manufacture of tanks are tabulated in the Standard. There is also a table of austenitic and austenitic-ferritic stainless steels to Standard EN 10088-1. Martensitic stainless steels cannot be used.

3.1.1.4 Floors

The requirements for tank floors is similar to BS 2654 and API 650. Minimum plate thickness for stainless floors is given as 5 mm for lap-welded floors and 3 mm for butt-welded floors. For carbon steel floors this are 6 mm and 5 mm respectively.

3.1.1.5 Shells

Minimum nominal shell thickness. The table of minimum nominal shell thickness for carbon steel tanks is similar to that of BS 2654 except that at the larger tank diameters, thinner plating than BS 2654 is allowed, although this is still thicker than that in API 650. A table of minimum nominal shell plate thickness is included for stainless steel shells.

Calculated shell plate thickness. Each shell course thickness is established from the greater value derived from two for-

mulae. This is similar to the API 650 "one-foot" method except that:

- In the first formula, the design stress is $\frac{2}{3}$ of the material minimum yield stress and the formula includes the design pressure (in the roof space) which can be neglected if < 10 mbar, and the corrosion allowance (if any).
- In the second formula, the test stress is $\frac{3}{4}$ of the material minimum yield stress and this formula includes only the test pressure (in the roof space), which is higher than the design pressure.

For both of these formulae, the maximum permitted design stress is 260 N/mm² (as is the case in BS 2654).

The API 650 "variable point" method of shell thickness calculation is not included in the Standard.

3.1.1.6 Yield stress

The yield stress shall be the minimum value specified for:

	Ambient temperature	Elevated temperature >100°C
Carbon and carbon manganese steels	Yield or 0.2 % proof stress	0.2 % proof stress
Stainless steels	1 % proof stress	>50°C 1 % proof stress

3.1.1.7 Primary and secondary wind girders

The requirements here are similar to that of BS 2654 and API 650 except that, for negative pressures more than -8.5 mbar, a design methodology has to be agreed between the tank purchaser and manufacturer.

3.1.1.8 Roof-to-shell compression zone

The requirements here are similar to that of BS 2654 and API 650.

3.1.1.9 Fixed and floating roof design

The requirements here are similar to that of BS 2654 and API 650.

3.1.1.10 Annexes to the Standard

The following annexes to the Standard are worthy of mention:

Annex B. Operational and safety considerations. Gives guidance on the selection of tank type, bunding requirements and fire protection.

Annex E. Requirements for floating roof seals. Gives details of the type of roof seals, which are available.

Annex F. Alternative steel specifications. Gives on the selection of other national standard steel specifications and the requirements, which govern their use within the parameters of the tank Standard.

Annex H. Recommendations for other types of floors. Gives recommendations for the thickness of floor plating, which is supported on a grillage. Also gives methods for constructing double containment floors.

Annex K. Design rules for frangible tanks. The rules here seem to apply principally to unanchored tanks and hence appear to be limiting in scope. Where frangibility cannot be achieved using the standard method given in the annex, then the "special arrangement" is recommended where a weak upper shell joint is proposed (as shown in Figure 3.71, Section 3.8.12).

Annex L. Requirements for venting systems. Gives detailed design parameters for venting under normal product import/export and climatic conditions, for tanks with and without thermal insulation. Emergency venting causing very high outbreathing capacities is considered, as in the case of a fire local to a tank, or due to operational malfunctions, which cause a rapid rise in

internal pressure. The possible requirement for emergency vacuum venting is also considered.

Annex P. Heating and/or cooling systems. Gives advice on heat transfer fluids and types of heat transfer devices, together with their installation.

Annex R. Surface finish. Gives general recommendations for the preparation of the internal and external surfaces of carbon and stainless steel tanks.

It must be remembered that the above information is based on the draft Standard and may be modified as and when the Standard is finalized and published as an adopted document.

3.1.2 The German storage tank Code DIN 4119

DIN 4119 is issued in two parts:

- Part 1 - Fundamentals, design and tests.
- Part 2 - Calculations.

The Codes does not take the same form as the BS, API or European prEN 14015 Codes, as it does not give specific formulae for designing the various elements of the tank.

3.1.2.1 Part 1

This advises on rules, which apply to: corrosion protection, material selection, fabrication, erection, welding and venting for fixed roof tanks. There are also directives for floating roofs. This part of the Code also lists many other related DIN Codes, which are referred to in the text of the Code, which are to be used for designing the tank.

3.1.2.2 Part 2

This is an elaboration of Part 1 and defines:

- 1) The mathematical symbols, which are to be used in the design process.
- 2) Design loads, including wind loads and test loads.
- 3) The principles for designing the shell, with minimum allowable thickness limitations but does not give a method for the design of the shell.
- 4) The principles governing shell stability under wind conditions, stating safety factors, which shall apply, but with no method for the calculation of shell stability.
- 5) The principles governing the design of the shell-to-bottom area, the shell-to-roof area and the requirements for frangibility.
- 6) Rules for the design of fixed and floating roofs.
- 7) Advice on the design of the tank foundations

Again, this part of the Code does not give any formulae for the design of the various areas of the tank but provides references to many related DIN Codes and learned papers on the subject. Also included in the list are the tank Codes API 650 and API 620.

The heading to both parts of the Code includes the following statement "The design, calculation and construction of the structural steel parts for tanks require a basic knowledge of steel construction and tank construction and the accepted codes of practice. Hence only companies employing experts having such knowledge and able to ensure proper construction may carry out such work."

This statement leads to the conclusion that any recognized tank design code methodology could be used in conjunction with the stipulations regarding: loadings, stress values safety factors etc., which are contained within DIN 4119.

However, as and when the draft European Code prEN 14015 becomes universally adopted (to which Germany is a signa-

tory) then, presumably DIN 4119, together with any other European national Codes, will become historical documents.

3.2 Design data

At the commencement of a project it is important that the tank purchaser clearly defines his exact requirements to the tank constructor, in order that there can be no misunderstandings between the two parties. To assist in this initial process, the design Codes each devote a section, which addresses this topic, and they are discussed in the following Sections.

Some of the terminology used in the following lists and data sheets may not be familiar to those who are not fluent in tank technology but such terms will become apparent on reading *Storage Tank & Equipment* and Codes to which it refers.

3.2.1 The BS Code 2654

Clause 3 of the Code lists the appropriate information together with references to other relevant clauses in the Code, to be exchanged prior to implementing the requirements of this Standard and inspections by the purchaser during erection, and is presented as follows:

3.2.1.1 Information to be specified by the purchaser

The following basic information to be specified by the purchaser shall be fully documented. Both the definitive requirements specified throughout the Standard and the documented items shall be satisfied before a claim of compliance with the Standard can be made and verified.

- (a) Geographical location of the tank.
- (b) Diameter and height or the capacity of the tank, including ullage. Where only the capacity of the tank is specified ground conditions shall be included.
- (c) Whether fixed or floating roof is to be supplied and the type of roof if the purchaser has specific preferences, i.e. for fixed roofs (cone, dome, membrane, etc.) or for floating roofs (pontoon, double deck, etc.).
- (d) All relevant properties of the contained fluid, including the relative density and corrosion allowance (if, how and where required).
- (e) The design vapour pressure and vacuum conditions inside the tank (see 2.1).
- (f) The minimum and maximum design metal temperatures (see 2.2).
- (g) The size, number and type of all mountings required showing locations. Maximum filling and emptying rates and any special venting arrangements (see 9.9).
- (h) The minimum depth of product which is always present in the tank (see 10.1(b)).
- (i) If the tank is to be thermally insulated (see 12).
- (j) Areas of responsibility between the designer, the manufacturer and the erector of the tank when these are not the same.
- (k) Quality of the water (particularly if inhibitors are to be present) to be used during tank water test (see 18.4.2).
- (l) Expected maximum differential settlements during water testing and service lifetime of the tank (see Appendix A).
- (m) Other specifications which are to be read in conjunction with this Standard.

3.2.1.2 Optional and/or alternative information to be supplied by the purchaser

The following optional and/or alternative information to be supplied by the purchaser shall be fully documented. Both the definitive requirements specified throughout this Standard and

the documented items shall be satisfied before a claim of compliance with the Standard can be made and verified.

- (a) Whether a check analysis is required (see 4.3.2).
- (b) Whether the weight of insulation is excluded from the minimum superimposed loadings (see 5.3.2).
- (c) Whether significant external loading from piping, etc. is present (see 5.5).
- (d) Whether seismic loading is present requiring specialist consideration including methods and criteria to be used in such analysis (see 5.7 and Appendix G).
- (e) Whether a fixed roof is required and if so:
 - (1) if cone roof slope is other than 1 in 5 (see 8.2.2);
 - (2) if radius of curvature of dome roof is other than 1.5 times tank diameter (see 8.2.2);
 - (3) whether made as a double-welded lap joint or a butt-joint (see 8.3.5);
 - (4) whether particular venting requirements are specified (see 8.6.1 and 8.6.2).
- or
- (f) Whether a floating roof is required and if so:
 - (1) whether floating roof is designed to land as part of the normal operating procedure (see 9.1.1);
 - (2) whether floating roof is designed for wind-excited fatigue loading (see 9.3);
 - (3) whether top edge of bulkhead is to be provided with continuous single fillet weld (see 9.5);
 - (4) floating roof ladder details (see 9.6.1, 9.6.2 and 9.6.4);
 - (5) type of primary roof drains (see 9.7.1);
 - (6) requirements for additional roof manholes (see 9.11);
 - (7) for selection of seal materials-whether maximum aromatic content of the product is greater than 40% (m/m) (see 9.13);
 - (8) requirements for the design of gauge hatch (see 9.14);
- (g) An alternative type of manhole cover (see 11.3).
- (h) Details of flange drilling if not in accordance with BS 1560 (see 11.7).
- (i) Details of painting requirements and whether pickling, grit or shot blasting is required (see 13.6.1, 13.6.3, and 14.12).
- (j) Details of erection marks for plates and sections (see 13.7.1).
- (k) Whether welding electrodes and/or key plating equipment are to be supplied by the tank manufacturer (see 14.1).
- (l) Alternative arrangements for provision of tank foundation (see 14.3).
- (m) Whether a welder making only fillet welds is required to be approved for such welding in accordance with BS EN 287-1 (see 16.3.2).
- (n) Whether tack welding of shell, roof and bottom is permitted to be carried out by non-approved operators (see 16.3.2).
- (o) Whether pneumatic testing of reinforcing plates is required (see 18.3.1).

3.2.1.3 Information to be agreed between the purchaser and the manufacturer

The following information to be agreed between the purchaser and manufacturer shall be fully documented. Both the definitive requirements specified throughout this Standard and the documented items shall be satisfied before a claim of compliance with the Standard can be made and verified.

- (a) Alternative materials selection other than those specified in the Code (see 3.1).
- (b) Precautions for avoiding brittle fracture during hydrostatic testing (see figure 1).
- (c) Alternative bottom plate layouts (see 6.1.2).
- (d) Spacing of the roof-plate-supporting members for dome roof (see 8.3.1).
- (e) Any increase in roof joint efficiency for tapped and welded roof plates (see 8.3.6).
- (f) Alternative loading conditions for floating roof design other than those specified in the Code (see 9.2.1.4).
- (g) The operating and cleaning position levels of the supporting legs (see 9.10.1).
- (h) Proposed method to hold the plates in position for welding (but see 14.5.1).
- (i) The location and number of checks on shell tolerances during erection (see 14.6.2).
- (j) Methods of protecting the shell during erection against wind damage, etc. (see 14.9).
- (k) If fixed roofs are to be erected in the tank bottom, and raised into position by an air pressure or suitable means (see 14.10).
- (l) Sequence in which joints are to be welded (see 15.2).
- (m) If previously approved appropriate welding procedures are acceptable (see 18.1.3).
- (n) Test procedures to be used during the tank water test (see 18.1.1).

3.2.2 The API Code 650

Appendix L of the Code gives four data sheets which should be completed, these are shown in Figure 3.1. On completion of tank erection, the purchaser shall receive from the manufacturer a copy of these sheets, filled in to show the "as built" details.

3.2.3 The draft European Code prEN 14015 -1:2000

Annex A of the Code lists the appropriate information together with references to the relevant clauses in the Code, and is presented as follows:

3.2.3.1 Annex A (normative) Technical agreements

A.1 Information to be supplied by the purchaser

The following information shall be fully documented:

- the design pressure and the design internal negative pressure (see 5.1 and Table 5.1);
- the stainless steel grade, and the risk of corrosion (see 6.2.1.2);
- the requirements for the surface finish of stainless steel (see 6.2.1.4);
- the value of the seismic load (see 7.2.11);
- the bottom type if not single (see 8.1.1);
- the bottom is to be butt-welded (see 8.4.1);
- the side of the roof that is welded and the size of the overlap (see 10.3.5);
- the venting requirements (see 10.6.1);
- that emergency pressure relief is not to be included (see 10.6.2);
- the provision of floating covers (see 10.7);
- the provision of floating roofs and floating roof seals (see 11);

GENERAL INFORMATION (T)	
1	NAME OF THE COMPANY
2	ADDRESS
3	CITY
4	STATE
5	ZIP
6	PHONE
7	FAX
8	E-MAIL
9	WEBSITE
10	PRODUCTS
11	MARKETING
12	SALES
13	FINANCE
14	OPERATIONS
15	RESEARCH & DEVELOPMENT
16	LEGAL
17	COMPLIANCE
18	ENVIRONMENTAL
19	SAFETY
20	QUALITY
21	LOGISTICS
22	IT
23	HR
24	TRAINING
25	WELFARE
26	COMMUNITY RELATIONS
27	GOVERNANCE
28	ETHICS
29	INTEGRITY
30	TRANSPARENCY
31	ACCOUNTABILITY
32	RESPONSIBILITY
33	STAKEHOLDERS
34	INTERESTS
35	CONCERNS
36	EXPECTATIONS
37	PERFORMANCE
38	IMPACT
39	CONTRIBUTION
40	VALUE
41	REPUTATION
42	BRAND
43	IDENTITY
44	IMAGE
45	PERCEPTION
46	RECOGNITION
47	RESPECT
48	INCLUSION
49	DIVERSITY
50	EQUALITY
51	JUSTICE
52	FAIRNESS
53	TRANSPARENCY
54	ACCOUNTABILITY
55	RESPONSIBILITY
56	STAKEHOLDERS
57	INTERESTS
58	CONCERNS
59	EXPECTATIONS
60	PERFORMANCE
61	IMPACT
62	CONTRIBUTION
63	VALUE
64	REPUTATION
65	BRAND
66	IDENTITY
67	IMAGE
68	PERCEPTION
69	RECOGNITION
70	RESPECT
71	INCLUSION
72	DIVERSITY
73	EQUALITY
74	JUSTICE
75	FAIRNESS
76	TRANSPARENCY
77	ACCOUNTABILITY
78	RESPONSIBILITY
79	STAKEHOLDERS
80	INTERESTS
81	CONCERNS
82	EXPECTATIONS
83	PERFORMANCE
84	IMPACT
85	CONTRIBUTION
86	VALUE
87	REPUTATION
88	BRAND
89	IDENTITY
90	IMAGE
91	PERCEPTION
92	RECOGNITION
93	RESPECT
94	INCLUSION
95	DIVERSITY
96	EQUALITY
97	JUSTICE
98	FAIRNESS
99	TRANSPARENCY
100	ACCOUNTABILITY

PAGE 1 OF 4

COMPLETED BY PURCHASER)

[illegible]

Figure 3.1 Storage tank data sheet - page 1
From API 650, Appendix L

API STANDARD 650 STORAGE TANK DATA SHEET

DATE _____
BY _____
FILE NO. _____
PAGE 2 OF 4

CONSTRUCTION DETAILS (TO BE COMPLETED BY MANUFACTURER AND/OR PURCHASER)			
1. MANUFACTURER _____			
ADDRESS _____			
CITY _____		STATE _____	ZIP CODE _____ PHONE _____
SERIAL NO. _____			
1. FABRICATOR _____			
ADDRESS _____			
CITY _____		STATE _____	ZIP CODE _____ PHONE _____
SERIAL NO. _____			
3. MATERIAL SPECIFICATIONS: SHELL _____			
ROOF _____			
BOTTOM _____			
STRUCTURALS _____			
4. NO. OF SHELL COURSES _____			
5. PLATE WIDTHS AND THICKNESSES (INCLUDING CORROSION ALLOWANCE), IN mm (in.)			
1 _____	4 _____	7 _____	
2 _____	5 _____	8 _____	
3 _____	6 _____	9 _____	
6. TANK BOTTOM: PLATE THICKNESS _____ mm (in.) ☐ LAP ☐ BUTT SEAMS			
SLOPE _____ mm/m (in/ft) ☐ TO ☐ FROM CENTER			
7. MINIMUM WIDTH AND THICKNESS OF BOTTOM ANNULAR PLATES (3.5), IN mm (in.): _____			
8. ROOF-TO-SHELL DETAIL (FIGURE F-1) _____			
9. INTERMEDIATE WIND GIRDER? ☐ YES ☐ NO		TOP WIND GIRDER FOR USE AS WALKWAY? ☐ YES ☐ NO	
10. ROOF TYPE: _____		☐ SUPPORTED ☐ SELF-SUPPORTED ☐ FLOATING	
SLOPE OR RADIUS _____ mm (in.)			
11. ROOF PLATE: _____		THICKNESS _____ mm (in.) ☐ LAP ☐ BUTT ☐ JOINT	
12. PAINT:			
SHELL—		EXTERIOR? ☐ YES ☐ NO INTERIOR? ☐ YES ☐ NO	
		SURFACE PREPARATION _____	
BOTTOM—		UNDERSIDE? ☐ YES ☐ NO INTERIOR? ☐ YES ☐ NO	
		SURFACE PREPARATION _____	
STRUCTURAL STEEL—		EXTERIOR? ☐ YES ☐ NO INTERIOR? ☐ YES ☐ NO	
		SPECIFICATION _____	
13. TANK BOTTOM COATING:		INTERIOR? ☐ YES ☐ NO MATERIAL _____	
		APPLICATION SPECIFICATION _____	
14. INSPECTION BY:		SHOP _____ FIELD _____	
15. WELD EXAMINATION:		RADIOGRAPH _____	
		SUPPLEMENTARY LIQUID PENETRANT OR ULTRASONIC _____	
16. FILMS _____		PROPERTY OF _____	
17. LEAK TESTING:		BOTTOM _____	
		SHELL _____	
18. MILL TEST REPORTS:		REQUIRED? ☐ YES ☐ NO	
		PLATE _____	
		STRUCTURAL SHAPES _____	
19. PURCHASER'S REFERENCE DRAWING _____			
20. TANK SIZE:		DIAMETER _____ m (ft) HEIGHT _____ m (ft)	
21. DATE OF STANDARD 650 EDITION/REVISION _____			
REMARKS _____			

Figure 3.1 Storage tank data sheet - page 2
From API 650, Appendix L

API STANDARD 650
STORAGE TANK
DATA SHEET

DATE _____
BY _____
FILE NO. _____
PAGE 4 OF 4

Top of shell height

Design liquid level*

Normal fill level

Overfill slot

Overfill protection level requirement:
(see Note 1)
_____ m³ (bbl) or _____ mm (in.)

Note 1

Maximum capacity:
_____ m³ (bbl)

Net working capacity:
_____ m³ (bbl)

Minimum fill level

Minimum operating volume remaining in the tank:
_____ m³ (bbl) or _____ mm (in.)

Top of bottom plate as shell

Notes: □
1. Overfill protection level (or Volume) requirement; see API 2350. □
* See 3.6.3.2.

Figure 3.1 Storage tank data sheet - page 4

From API 650, Appendix L

- the amount of product to be always present in the tank (see 12.1);
- the roof manhole cover (see 13.3.1);
- if the roof plates to be welded to the roof structure (see 15.8.4);
- the position of floating roof (see D.3.1)
- the floating roof design and type (see D.3.4);
- the additional roof manholes (see D.3.6);
- the support leg operating and cleaning positions (see D.3.13);
- the gauging device (see D.3.14);
- if a rolling ladder is not required (see D.3.15);
- the roof main drain if not a hose or articulated pipe type (see D.3.8.1);
- if a trial erection and inspection of a floating roof is required (see D.4);
- if floating roof rim seals are required (see E.1);
- the evaporation rate (see L.3.1.1 c);
- the maximum gas flow under malfunction conditions of the gas blanket (see L.4.3);
- the emergency flow capacity for other possible causes (see L.4.4);
- the emergency vacuum flow capacity (see L.5);
- the range of operating temperature (see Q.2.4);
- the procedure, qualification and acceptance tests for adhesive (see Q.3.3.1);
- the insulation thickness or heat loss requirements (see Q.6.1);
- the tank's external appearance and finish (see R.2.1).
- the painting system used (see R.2.2).

A.2 Information agreed between the purchaser and the contractor

- the additional requirements for roof plating and nozzle reinforcement (see Table 5.1)
- the design methodology and fabrication tolerances for design internal negative pressures above 8.5 mbar (see Table 5.1);
- the steel to be used if not from Tables 6.1.1-1 to 6.1.1-3 (see 6.1.1.1);
- the mounting materials, when different to the shell plates (see 6.1.7.1);
- the live loads (see 7.2.6);
- the concentrated live load (see 7.2.7);
- the value of the wind load if the wind speed is more than 45 m/s (see 7.2.10);
- the anticipated settlement loads (see 7.2.13);

- the emergency loads (see 7.2.14);
- the bottom gradient if more than 1:100 (see 8.1.1);
- the guaranteed residual liquid level to resist uplift (see 8.2.3);
- the incorporation of annular plates (see 8.3.1);
- the option to be used if the SG exceeds 1.0 kg/f (see 9.1.3);
- the shell thickness for stainless steel tanks of diameters greater than 45 mm (see Table 9.1.5 NOTE 3);
- whether the underside welds of stiffening rings shall be continuous or intermittent (see 9.3.1.11);
- the design methodology and load combinations (see 9.3.3.9);
- the span of roof supporting structure for dome roofs (see 10.3.1);
- the joint efficiency if different to the standard values (see 10.3.6);
- the minimum size of manholes (see 13.1.1);
- the details of non-standard nozzles (see 13.3.2);
- the method of heating or cooling the fluid (see 13.10);
- the non-standard distances between an opening and a plate edge (see 15.5);
- non-standard types of floating roofs (see D.2)
- non-standard floating roofs (see D.3.1);
- the specific requirement for a floating roof (see D.3.2.4);
- the alternative values for live load when resting on its support legs (see D.3.3);
- the method of assessing frangibility (see K.2);
- the safety coefficient for frangible roofs (see K.4);
- the design of flush-type clean-out doors (see 0.1.1);
- the proprietary system of insulation (see Q.1);
- the insulation system to be used (see Q.2.1);
- the basis for the wind load calculations (see Q.2.3);
- the type of foam insulation (see Q.8.2);
- the sequence of foaming and cladding (see Q.8.2);
- the means of checking the quality of foam (see Q.8.2);
- the type of foam and its physical and thermal properties (see Q.8.3).

3.3 The shell

3.3.1 The design of the tank shell

Storage tanks are often disparagingly referred to by constructors and users as "tin cans" and to some degree this is true in as much as there are similarities in the ratios of the shell thickness to diameter of both items.

For example a typical soup can is 75 mm diameter x 105 mm high ($d/h = 1/1.4$) and has a wall thickness of 0.15 mm. A storage tank of 10 m diameter x 14 m high has a wall thickness of 5 mm. It can be seen that the thickness-to-diameter ratio for the soup can is 0.002 and for the tank is 0.0005. The tank ratio is four times less than that of the soup can, which demonstrates how relatively flimsy the shell of a tank really is particularly if it is subjected to a partial vacuum condition as is demonstrated in Figure 3.2.

The scaffolding around the tank in Figure 3.2 was erected to allow the shell to be painted. Immediately after the painting was completed, the tank was put back into service but a plastic bag, which had been put over the roof vent valve to protect it during

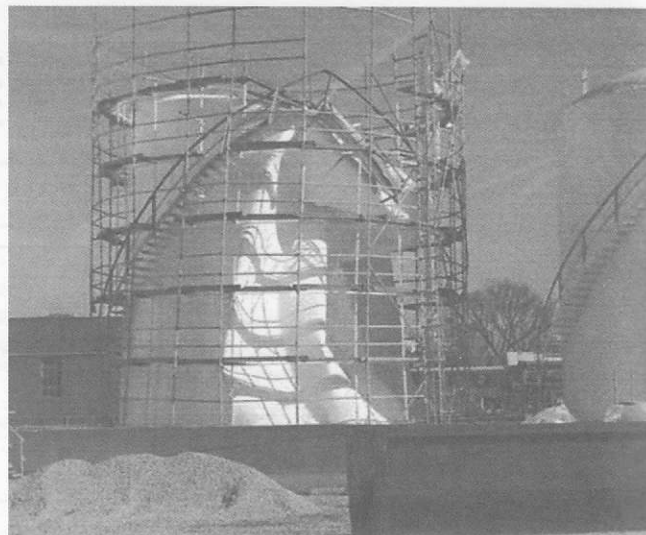


Figure 3.2 Example of a tank imploding

painting, had not been removed and the tank imploded when product was being drawn from it.

The various stresses to which the shell of a tank is subjected are as follows:

Hoop tension

The major stress in the shell is hoop tension which is caused by the head of product in the tank, together with any overpressure in the roof space of a fixed roof tank.

Axial compression

This stress is made up of the following components:

- The self-weight of the tank, comprising the shell, the roof, the superimposed load on the roof and any attachments to the tank.
- The compressive load due to any internal vacuum in the tank.
- Wind load acting on the shell of the tank causes an overturning effect and hence induces a compressive load on the leeward side of the shell.
- Where a tank is located in a geographical area which is subject to earthquakes, then compressive stresses due to the seismic action can be transmitted to the shell. This latter stress component is dealt with separately in Chapter 15 or 26 where seismic design is covered in detail.

Vertical bending

The natural elasticity in the shell material allows the shell to expand radially when under service loading, but this expansion is restrained at the shell-to-floor junction and therefore the shell suffers vertical bending stresses in this area.

3.3.1.1 Failure around the circumference of the cylinder

In order to demonstrate how tank shells are designed, some basic engineering design principles must be considered.

Figure 3.3 shows a cylindrical shell having a shell, which is comparatively thin, compared to its diameter, the ends are capped off and it is subjected to an internal pressure 'p'.

- D = diameter
- t = wall thickness
- L = length
- p = internal pressure
- P = horizontal load on the cylinder
- F = tangential load in the wall of the cylinder

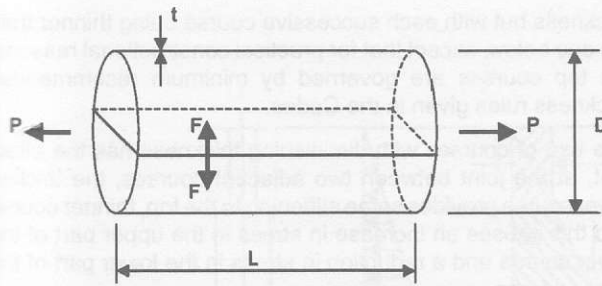


Figure 3.3 A cylindrical shell

Consider a failure around the circumference of the cylinder:

Load P = pressure $\times$ area

$$= p \times \pi/4 \times D^2 \quad \text{equ 3.1}$$

Resistance to a circumferential failure = stress $\times$ area of the cylindrical wall.

$$= f \times \pi \times D \times t \quad \text{equ 3.2}$$

Equating equations 3.1 and 3.2

$$p \times \pi/4 \times D^2 = f \times \pi \times D \times t$$

Then

$$f = \frac{P \times D}{4 \times t} \quad \text{equ 3.3}$$

3.3.1.2 Failure along the length of the cylinder

Consider a failure along the length of the cylinder:

Force F = pressure $\times$ area

$$= p \times D \times L \quad \text{equ 3.4}$$

Resistance to a longitudinal tear in the cylinder wall

= stress $\times$ area of the cylinder wall.

$$= f \times 2 \times L \times t \quad \text{equ 3.5}$$

Equating equations 3.4 and 3.5

$$p \times D \times L = f \times 2 \times L \times t$$

Then

$$f = \frac{p \times D}{2 \times t} \quad \text{equ 3.6}$$

By comparing equations 3.3 and 3.6 it can be seen that the highest stress is given by equation 3.6 and therefore a cylinder under pressure will fail by tearing along a line parallel to its axis rather than on a section perpendicular to its axis.

The basic equation 3.6 is used in the tank design Codes for determining the thickness for the tank shells.

The way the British, American and European tank design Codes apply the above basic principles differ in approach. Initially the British Standard 2654 will be considered, then later, the differing aspects of the other Codes will be discussed.

3.3.2 BS 2654

BS 2654 gives the shell thickness formula as:

$$t = \frac{D}{20 \cdot S} \{98 \cdot (H - 0.3) + p\} + \text{c.a.} \quad \text{equ 3.7}$$

where:

- t = shell thickness (mm)
- D = tank diameter (m)
- S = allowable design stress (N/mm^2)

- w = specific gravity of tank contents (non-dimensional) - but never taken as less than unity for design purposes
- p = design pressure in the vapour space above the product level (mbar)
- c.a. = corrosion allowance which, at the discretion of the tank customer, may be added to the design thickness (mm)
- H = distance from the bottom of the course under consideration to a predetermined height at the top of the tank, which is the limit of the fluid height (m)

The predetermined height at the top of the tank is either:

- The top of the shell.
- The level of an overflow designed to limit the fluid height in the shell.
- When the height of the shell includes a wind skirt with overflow openings and/or seismic freeboard, the maximum product height for calculation purposes shall be the overflow height, or the height less the seismic freeboard.
- $(H - 0.3)$ - The explanation of this term is given later in Section 3.3.2.2.

For the moment however, consider a tank having a shell of constant thickness over its full height, based on the full head of product in the tank represented by the simple term H (m).

Note: The tank diameter D is generally taken as the diameter measured to the centreline of the shell plating. However for floating roof tanks where it is preferable to have a smooth internal surface for the roof seal to act against, the diameter may be measured to the inside surface of each course of shell plates thus avoiding steps between adjacent courses.

Equation 3.6 is re-arranged for t as follows:

$$t = \frac{p \times D}{2 \times S} \quad \text{equ 3.8}$$

Where stress f is represented by S and p is the internal loading in the tank, which is made up of two components as shown in Figure 3.4.

The first component is due to the head of product in the tank H expressed as a height in metres.

The second component is the pressure in the vapour space ' p ' which is due to the natural gassing off of the stored product, or from the use of a positive pressure inert gas "blanket" over the product. This pressure is controlled by the use of pressure and vacuum relief valves fitted to the roof and these are covered later in Chapter 8, Section 8.2.4.2.

In order for the above formula to work, the input data has to be expressed in acceptable units as follows:

- p = N/mm^2
- D = mm
- S = N/mm^2

The first component of the pressure is converted from metres of product liquid head to mbar by multiplying by 98 and added to the second component, which is already expressed in mbar. This combination is then converted to N/mm^2 by multiplying by 0.0001.

D is converted to mm by multiplying by 1000 and S is already expressed in N/mm^2

Equation 3.8 is therefore transformed from $t = \frac{p \times D}{s \times S}$ to:

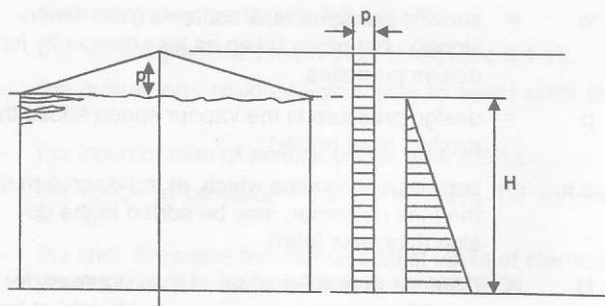


Figure 3.4 Loading on a tank shell

$$t = \frac{D \times 1000}{2 \times S} \left\{ [(H \times w \times 98) + p] 0.0001 \right\} + \text{c.a.}$$

$$t = \frac{D \times 1000}{2 \times S} \{ (0.0098 \times w \times H) + 0.0001p \} + c.a.$$

$$t = \frac{D}{2 \cdot S} \{ (9.8 \cdot w \cdot H) + 0.1p \} + c.a.$$

$$t = \frac{D}{20 \cdot S} \{ (98 \cdot w \cdot H) + p \} + \text{ca.} \quad \text{equ 3.9}$$

Earlier editions of BS 2654 limited the maximum allowable stress in the shell plating to 21,000 lbs/in² (145 N/mm²) and also included a welded joint efficiency of 85%.

The limitation on allowable stress has now been superseded, as shown later in Section 3.3.2.5. Also, due to improved modern welding technology and joint inspection techniques, as long as the welding and inspection procedures given in the Code are adhered to, the joint efficiency is deemed to be 100%. For example, the welded joints are considered to be at least as strong as the parent plate. Due to this increase in joint efficiency, tank shells are now 15% thinner than their earlier counterparts.

3.3.2.1 Principal factors determining shell thickness

It can be seen that the principal factors, which determine the thickness of the tank shell, are:

- the internal loadings due to the head of liquid and
- the pressure in the vapour space.

Adjustment may be required when axial, wind and seismic loads are considered but there is no allowance made for any other external loadings whatsoever. It is important to remember this, because on occasions, designers and constructors may be asked to impose additional external loads on the shell, or to allow for external piping loads to be transmitted to the shell nozzles, particularly those in the bottom course of the shell where more often than not the thickness of this course is a design thickness rather than a nominal thickness (the explanation of this difference is given later in Section 3.3.2.4).

Where additional loads are requested, separate consideration must be given to their effect on the stress in the shell. The American Code API 650 addresses the effect of nozzle loadings in Appendix P of the Code but its application is limited to tanks over 36 metres in diameter. This subject is dealt with in Chapter 4.

3.3.2.2 Practical application of thickness formula

Having established how the shell thickness formula was derived, the practical application of the formula to a storage tank can now be discussed.

From Figure 3.4 it can be seen that the pressure varies with the head of liquid and therefore the shell thickness varies from almost zero at the top, to a maximum at the bottom. As it is impractical to have a shell with a tapering thickness, it is instead, constructed of a number of plate courses each of a uniform

thickness but with each successive course being thinner than the one below, except that for practical constructional reasons, the top courses are governed by minimum recommended thickness rules given in the Codes.

The use of courses with diminishing thickness has the effect that, at the joint between two adjacent courses, the thicker, lower course provides some stiffening to the top, thinner course and this causes an increase in stress in the upper part of the lower course and a reduction in stress in the lower part of the upper course.

The design Codes assume, on an empirical basis, that the reduction in stress in the upper course reaches a maximum value at one foot (300 mm) above the joint and it is at this point, on each course from which the effective acting head is measured. This method of calculation is known as the "one foot" method or rule, (having evolved in an era when the Imperial measurement system was in vogue).

The above explanation can be shown diagrammatically as in Figure 3.5.

The displacement of the shell courses is shown diagrammatically in Figure 3.6.

The adoption of the “one-foot” method means that the shell thickness formula given in BS 2654 is written as set out in equation 3.7:

$$t = \frac{D}{20 \cdot S} \{98 \cdot (H - 0.3) + p\} + c.a.$$

3.3.2.3 Exception to “one-foot” method

There is an exception to the "one-foot" rule and this comes into use when steels of differing strengths are used in designing the shell courses. In such cases, when the ratio of:

height (H - 0.3), used for the computation of a given course, divided by the allowable stress for that course, is equal to or more than the $(H - 0.3) \div S$ ratio for the course beneath.

then the advantage of the "one-foot" method is deemed not to apply to the upper course and this course shall be designed using H instead of $(H - 0.3)$. The mathematical form of this is expressed as:

When:

$$\frac{H_u - 0.3}{S_u} \geq \frac{H_L - 0.3}{S_L}$$

$$\text{then } t = \frac{D}{20 \cdot S_u} \{ (98 \cdot w \cdot H_u) + p \} + \text{c.a.}$$

where:

H_u = distance from the bottom of the upper course to the maximum possible filling height (m)

S_u = allowable design stress for the upper course
(N/mm²)

H_L = distance from the bottom of the lower course to the maximum possible filling height (m)

S_L = allowable design stress for the lower course
(N/mm²)

There is a further very important stipulation, which must be remembered during the shell design, and this is that, no course shall be constructed at a thickness less than that of the course above, irrespective of the materials of construction.

There are other factors, which govern the use of the above formula, and these are now discussed.

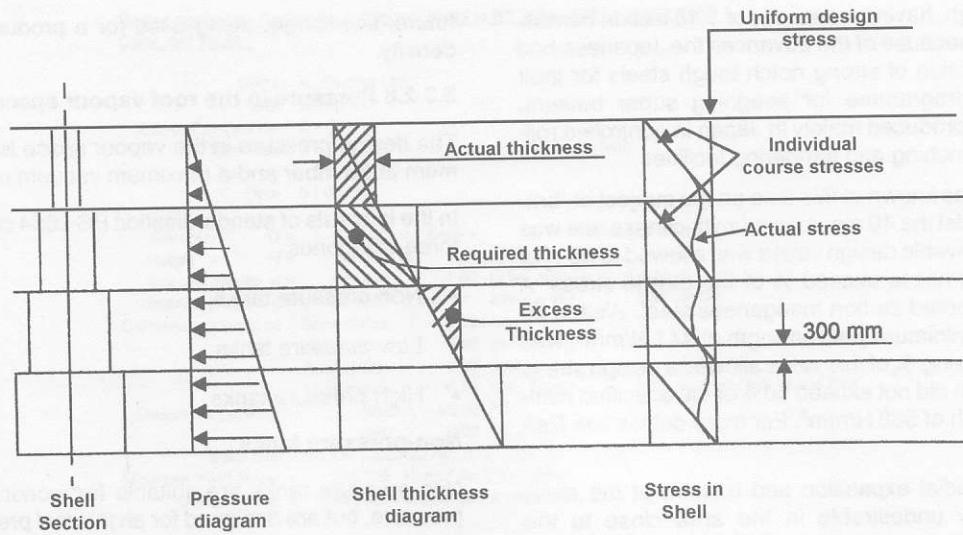


Figure 3.5 Diagrammatic explanation of the thickness formula or the "one-foot" method

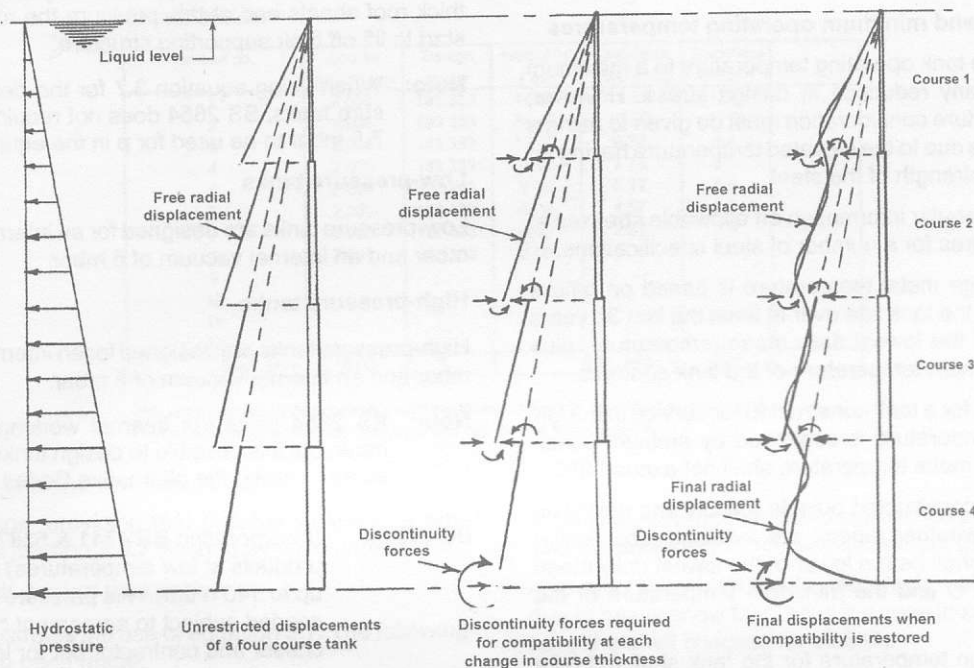


Figure 3.6 Displacement of the shell courses shown diagrammatically

3.3.2.4 Maximum and minimum shell thickness

Thick plates are known, under sub-zero temperature conditions, to be susceptible to brittle fracture. Tests made by the Wells Wide Plate test method in 1964 concluded that for operational safety, storage tank shell plates should be limited to a maximum thickness of 40 mm.

For the upper courses of shell plating the formula will give quite thin plate thickness which are impractical for constructional purposes. The Code therefore specifies minimum plate thickness, which must be used, and Table 2 in BS 2654 gives these and is shown in Figure 3.7. This minimum thickness may include any specified corrosion allowance, provided that the shell is shown by calculation to be safe in the corroded condition.

Nominal tank diameter D (m)	Minimum allowable shell plate thickness t (mm)
< 15	5
15 to < 30	6
30 to < 60	8
60 to < 75	10

Nominal tank diameter D (m)	Minimum allowable shell plate thickness t (mm)
75 to < 100	12
> 100	14

Figure 3.7 Minimum plate thicknesses according to Table 2, BS 2654

3.3.2.5 Allowable steel stresses

To keep the selection of shell plate material within the band of carbon and carbon manganese weldable steels, the maximum allowable design stress which may be used is 260 N/mm² or two thirds of the material, specified minimum yield strength at room temperature, whichever is the lower. This limit of 260 N/mm² discourages the use of steels with a minimum specified yield strength in excess of 390 N/mm², because of their increased hardness and reduced weldability.

However, steels with higher yield stresses than this have been used and this came about in the late 1960s and early 1970s, when the impetus in the petroleum industry gave rise to a demand for larger tanks with a capacity of 1 million barrels (159,000 m³) and greater. BP developed tankage on Das Island, offshore from Abu Dhabi, where the largest tank was 96 m

diameter x 25 m high, having a capacity of 1.18 million barrels. This was possible because of the advances the Japanese had made in the production of strong notch tough steels for their growing building programme for seagoing super tankers. These steels were produced mainly in Japan in controlled rolling and on-line quenching and tempering facilities.

Also, much more was known at this time on the subject of "brittle fracture" and whilst the 40 mm maximum thickness rule was maintained, the allowable design stress was allowed to be $\frac{2}{3}$ of the yield stress but not to exceed $\frac{1}{2}$ of the tensile stress. A quenched and tempered carbon manganese steel, Welton 60 having a specified minimum yield strength of 441 N/mm², was used for the shell. Using $\frac{2}{3}$ of this value allowed a design stress of 293 N/mm², which did not exceed 50% of the specified minimum tensile strength of 588 N/mm². For more details see *Reference 3.1*.

Also, it limits the radial expansion and rotation of the shell, which is especially undesirable in the area close to the shell-to-bottom junction where there is the added complication due to nozzle loadings. This aspect is developed further in Chapter 4.

3.3.2.6 Maximum and minimum operating temperatures

The Code limits the tank operating temperature to a maximum of 150°C without any reduction in design stress. However, above this temperature consideration must be given to using a lesser design stress due to the elevated temperature having an effect on the yield strength of the steel.

BS 5500 contains tabular information on allowable stresses at elevated temperatures for a number of steel specifications.

The minimum design metal temperature is based on official weather reports for the tank site over at least the last 30 years and is the lower of the lowest daily mean temperature, plus 10°C, and the minimum temperature of the tank contents.

BS 2654 states that for a tank constructed for service in the UK where the shell temperature is controlled by ambient conditions, the minimum metal temperature shall not exceed 0°C.

For a storage tank constructed outside the UK and where no long term data or weather reports are available, the design metal temperature shall be the lower of the lowest daily mean temperature plus 5°C and the minimum temperature of the contents.

The minimum design temperature for the tank shall not take into account the beneficial effect of heated or thermally insulated tanks.

It is interesting to note that the proposed European Standard prEN 14015 - 1, states a maximum design temperature of 100°C. Design temperatures above this value have to comply with clause 6 of the Standard which states that the steel supplier shall certify the yield stress values for steels used at elevated temperatures. Alternatively, a list of appropriate steels is given in the text. For design temperatures above 250°C, steels which are proven to be unaffected by ageing shall be used.

3.3.2.7 Specific gravity or relative density of the stored product

The specific gravity or relative density of the stored product for design purposes shall not be taken as less than unity (regardless that the actual specific gravity (SG) of the stored product may be less than unity). The basis of this requirement is the fact that the tank, on completion, is required to be hydrostatically tested with water prior to being put into service. Also, as many petroleum and chemical products have a SG less than unity, this gives an additional safety factor to the shell plating.

Also, experience has shown that designing to a SG of 1.0 gives flexibility of usage and guards against a tank, which may have been designed for a particular product density, some time in the

future, unwittingly, being used for a product having a higher density.

3.3.2.8 Pressure in the roof vapour space

The design pressure in the vapour space is limited to a maximum of 56 mbar and a maximum vacuum of 6 mbar.

In the interests of standardisation BS 2654 classifies tanks into three categories:

- Non-pressure tanks
- Low-pressure tanks
- High-pressure tanks

Non-pressure tanks

Non-pressure tanks are suitable for working at atmospheric pressure, but are designed for an internal pressure of 7.5 mbar and an internal vacuum of 2.5 mbar. However for tanks with column supported roofs an internal pressure of 4 millibars shall be assumed. 4 mbar equates approximately to the weight of 5 mm thick roof sheets and at this pressure the roof plates will just start to lift off their supporting structure.

Note: When using equation 3.7 for the design of non-pressure tanks, BS 2654 does not require the pressure of 7.5 mbar to be used for p in the equation.

Low-pressure tanks

Low-pressure tanks are designed for an internal pressure of 20 mbar and an internal vacuum of 6 mbar.

High-pressure tanks

High-pressure tanks are designed for an internal pressure of 56 mbar and an internal vacuum of 6 mbar.

Note: BS 2654 limits the internal working pressure to 56 mbar, but it is possible to design tanks for higher pressures by using the alternative Codes listed here:

BS 7777 (incorporating BS 4741 & 5387- Storage of products at low temperatures) and pressures up to 140 mbar. This pressure may be exceeded subject to agreement between the purchaser and contractor but for large diameter tanks the design of the roof-to-shell joint and anchorage might be limiting.

API 650 Pressures up to 2½ lbs/in² G (172 mbar)
Appendix F

API 620 Pressures up to 15 lbs/in² G (1034 mbar)

As is the case for BS 2654, these Codes also only allow for a small internal vacuum to be present in the tank.

prEN 14015 Pressures up to 500 mbar, and vacuum up to 20 mbar. Except that for a vacuum condition above 8.5 mbar, the design methodology is not given in the Code but it shall be agreed between the purchaser and the manufacturer.

A synopsis of the requirements of this Code were covered earlier in Section 3.1.1.

Note: Whilst BS 2654 gives maximum values for internal vacuum, these values are not actually incorporated into the design formula for the shell thickness, this is because it is assumed that the thickness derived from equation 3.7 will be adequate enough to withstand the low vac-

Design method for Carbon Steel Storage Tanks to BS 2654 : 1989 + amd.1:1997.
Cone roof Tanks

Client : A. Another Ltd.
 Site : Liverpool
 Est. or Contract No : C / 001
 Tank size : 30.00 m. dia. x 16.00 m. high
 Tank No : 001
 Date : 5 / 05 / 02

Diameter D = 30.000 m Shell = 12 Plates per Course
 Height H = 16.000 m
 Specific gravity w = 0.900 1.00 to be used for Shell design.
 Internal press. p = 7.50 m.bar Internal vac 2.50 m.bar
 Corrosion allowances :- Shell plates 0.00 mm
 Floor plates 0.00 mm
 Roof plates 0.00 mm
 Shell angles 0.00 mm, Total = 0.00 mm off each flange thks
 Design temperature :- Max. 90.00 °C
 Min. 0.00 °C
 Steel type :- BS EN 10025 S275
 Minimum Yield Stress = 275.000 N/mm² for 't' <= 16mm
 Design stress = 183.333 N/mm² (2/3 x min. Yield)

Shell thickness $D/20.S \{98.w (H - 0.3) + p\} + ca$ (ignore 'p' if <= 7.5 m.bar)

The Code requires a min. thickness 8.00 mm

Design of the Shell.

Course no.	Course height (m)	Design stress	Height 'H' (m)	Calculated thks. (mm)	Thickness chosen
1	2.000	183.333	16.00	12.59	12.6
2	2.000	183.333	14.00	10.98	11.0
3	2.000	183.333	12.00	9.38	9.4
4	2.000	183.333	10.00	7.78	8.0
5	2.000	183.333	8.00	6.17	8.0
6	2.000	183.333	6.00	4.57	8.0
7	2.000	183.333	4.00	2.97	8.0
8	2.000	183.333	2.00	1.36	8.0
9					
10					
11					
12					
Shell ht. = 16.00				Min. thks. = 8.00	

This shell calculation demonstrates how the formula produces very thin upper courses.
 The Code requires a minimum thickness of 8 mm for this tank diameter.

Figure 3.8 Tank shell design illustration using equation 3.7

uum ratings, providing that suitable stiffening is provided see Section 3.5.2 Secondary wind girders.

3.3.2.9 Tank shell design illustration

Figure 3.8 demonstrates the use of equation 3.7. The following assumptions have been made:

- A non-pressure cone roof tank
- Pressure rating +7.5 mbar and -2.5 mbar
- Dimensions: 30 m diameter x 16 m high
- Number of courses: 8
- Shell corrosion allowance - nil
- Design temperature: + 90°C and 0°C
- Steel specification: BS EN 10025 S275 having a minimum yield of 275 N/mm²

This shell calculation demonstrates how the formula produces very thin upper courses. The Code requires a minimum thickness of 8 mm for this tank diameter.

3.3.3 Axial stress in the shell

The design of the shell to cater for internal pressure loading which produces a tensile circumferential stress in the shell has been discussed. However, for large diameter tanks with low shell heights, the lowest shell courses may be rather thin and therefore the stability should be checked taking into account the vertical loads resulting from the roof weight, shell self-weight,

snow load, vacuum, wind and seismic loads, as applicable and also the possibility of uneven settlement of the foundation.

Also any tank which has to carry high roof loads for example due to heavy snow falls, as is the case in say, Canada, should have the shell checked for stability.

3.3.3.1 Derivation and assessment of axial stress in a cylindrical shell

The tank wall thickness has been determined using only the internal pressure to which it is subjected together with a limiting circumferential stress of 260 N/mm² or $\frac{2}{3}$ of the applicable material yield stress. The axial stress should now be calculated for each course because the existence of compressive membrane stresses in the shell could cause it to fail by buckling. The following theory is, in part, taken from work by the late Professor A. S. Tooth, Professor of Mechanical Engineering, University of Strathclyde, Glasgow.

The theory for the critical buckling stress in a thin walled circular shell subjected to longitudinal compression is given by Roark & Young (Reference 3.2) as:

$$Sc = \frac{1}{\sqrt{3}} \times \frac{E}{\sqrt{1-\nu^2}} \times \frac{t}{r} \quad \text{equ 3.10}$$

Taking $E = 207,000 \text{ N/mm}^2$ and $\nu = 0.3$ for carbon steel, then

$$Sc = 125.235 \times \frac{t}{r} \quad \text{equ 3.11}$$

where:

t is in (mm) and

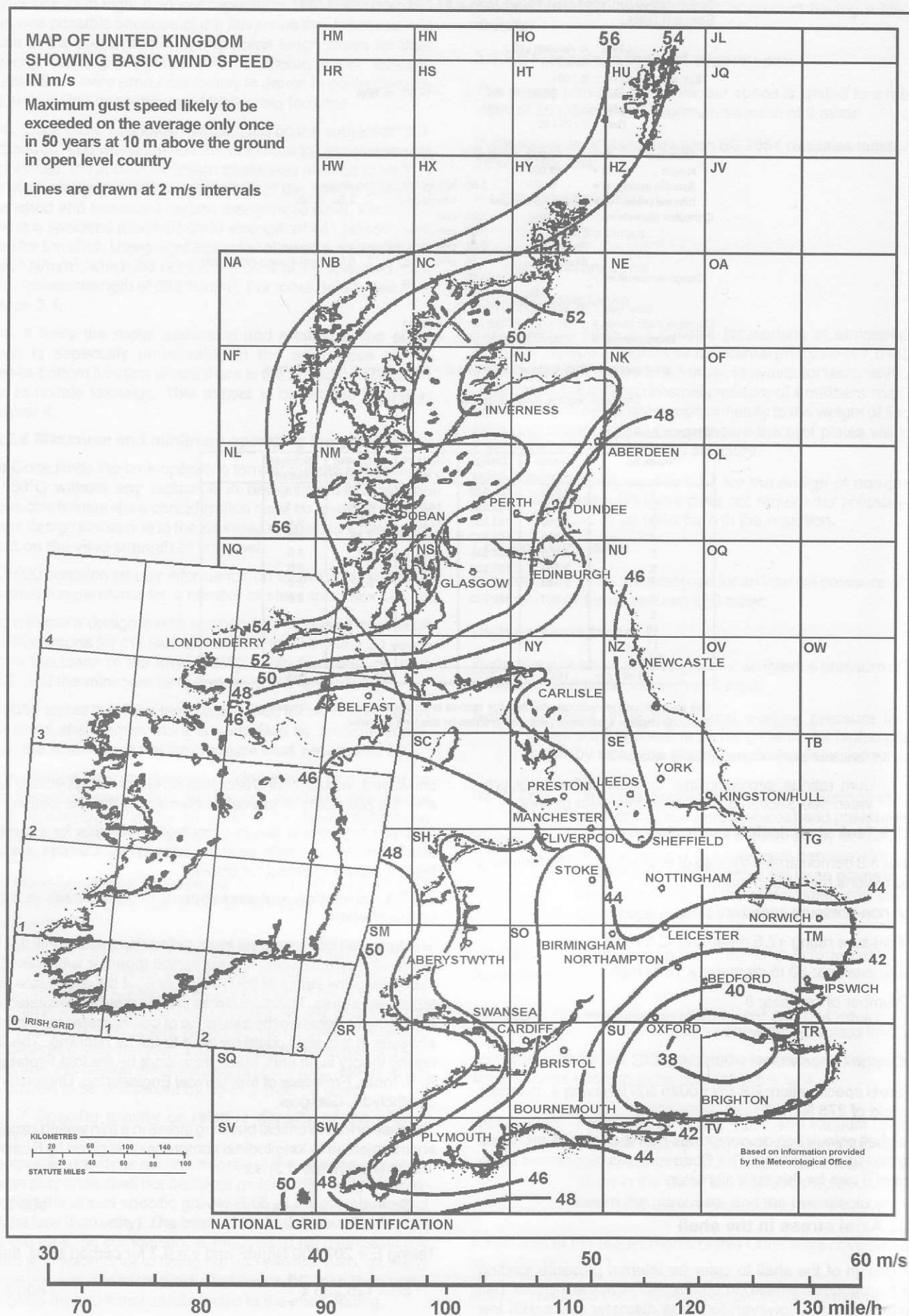


Figure 3.9 Basic wind speed for UK locations
From the Met Office, United Kingdom

Town or city	m/s	Town or city	m/s	Town or city	m/s	Town or city	m/s
Aberdeen	49	Carlisle	46	Leeds	46	Oxford	40
Bedford	40	Coventry	44	Leicester	44	Perth	51
Belfast	45	Dumfries	47	Liverpool	46	Plymouth	44
Birkenhead	46	Dundee	50	London	38	Portsmouth	40
Birmingham	43	Edinburgh	50	Londonderry	52	Preston	45
Blackpool	46	Exeter	43	Luton	40	Sheffield	46
Bournemouth	41	Glasgow	51	Manchester	45	Southampton	40
Bradford	47	Great Yarmouth	43	Middlesbrough	45	Stoke	43
Brighton	40	Huddersfield	46	Newcastle-upon-Tyne	45	Swansea	46
Bristol	43	Inverness	49	Northampton	42	Wolverhampton	43
Cambridge	41	Ipswich	41	Norwich	43	York	45
Cardiff	45	Kingston-upon-Hull	45	Nottingham	45		
Channel Islands							
Alderney	52	Guernsey	52	Jersey	52		

These values apply to cities and towns only and not necessarily to the surrounding areas

Figure 3.10 Basic wind speed in metres per second for some UK cities and towns

From British Standard CP3

Topography factor S_1

The basic wind speed, V , given in Figure 3.9 takes account of the general level of the site above sea level. This does not allow for local topographic features such as hills, valleys, cliffs, escarpments or ridges which can significantly affect the wind speed in their vicinity.

Near the summits of hills or the crests of cliffs escarpments or ridges the wind is accelerated. In valleys or near the foot of cliffs, steep escarpments or ridges, the wind may be decelerated. In all cases the variation of wind speed with height is modified from that appropriate to level terrain.

Where the average slope of the mound does not exceed 0.05 within a kilometer radius of the site, the terrain may be taken as level and the topography factor S_1 should be taken as 1.0.

In the vicinity of local topographic features the factor S_1 is a function of the upwind slope and the position of the site relative to the summit or crest, and will be within the range of $1.0 \leq S_1 \leq 1.36$. It should be noted that S_1 will vary with height above ground level, at a maximum near to the ground and reducing to 1.0 at higher levels.

In certain steep-sided enclosed valleys, wind speeds may be less than in level terrain. Caution is necessary in applying S_1 values less than 1.0 and specialist advice should be sought in such situations.

	(1) Open country with no obstructions			(2) Open country with scattered windbreaks			(3) Country with many windbreaks, small towns, outskirts of large cities			(4) Surfaces with large and frequent obstructions, e.g. city centres		
H	Class			Class			Class			Class		
m	A	B	C	A	B	C	A	B	C	A	B	C
3 or less	0.83	0.78	0.73	0.72	0.67	0.63	0.64	0.60	0.55	0.56	0.52	0.47
5	0.88	0.83	0.78	0.79	0.74	0.70	0.70	0.65	0.50	0.60	0.55	0.50
10	1.00	0.95	0.90	0.93	0.88	0.83	0.78	0.74	0.69	0.67	0.62	0.58
15	1.03	0.99	0.94	1.00	0.95	0.91	0.88	0.83	0.78	0.74	0.69	0.64
20	1.06	1.01	0.96	1.03	0.98	0.94	0.95	0.90	0.85	0.79	0.75	0.70
30	1.09	1.05	1.00	1.07	1.03	0.98	1.01	0.97	0.92	0.90	0.85	0.79
40	1.12	1.08	1.03	1.10	1.06	1.01	1.05	1.01	0.96	0.97	0.93	0.89
50	1.14	1.10	1.06	1.12	1.08	1.04	1.08	1.04	1.00	1.02	0.98	0.94
60	1.15	1.12	1.08	1.14	1.10	1.06	1.10	1.06	1.02	1.05	1.02	0.98
80	1.18	1.15	1.11	1.17	1.13	1.09	1.13	1.10	1.06	1.10	1.07	1.03
100	1.20	1.17	1.13	1.19	1.16	1.12	1.16	1.12	1.09	1.13	1.10	1.07
120	1.22	1.19	1.15	1.21	1.18	1.14	1.18	1.15	1.11	1.15	1.13	1.10
140	1.24	1.20	1.17	1.22	1.19	1.16	1.20	1.17	1.13	1.17	1.15	1.12
160	1.25	1.22	1.19	1.24	1.21	1.18	1.21	1.18	1.15	1.19	1.17	1.14
180	1.26	1.23	1.20	1.25	1.22	1.19	1.23	1.20	1.17	1.20	1.19	1.16
200	1.27	1.24	1.21	1.26	1.24	1.21	1.24	1.21	1.18	1.22	1.21	1.18

Class A = All units of cladding and roofing and their immediate fixings and individual members of unclad structures.

Class B = All buildings and structures where neither the greatest horizontal dimension nor the greatest vertical dimension exceeds 50 m (165 ft).

Class C = All buildings and structures whose greatest horizontal dimension or the greatest vertical dimension exceeds 50 m (165 ft).

Figure 3.10 Factors S_1 and S_2

From British Standard CP3

r is in (m)

Tests indicate that actual buckling occurs at between 40% and 60% of the value obtained using the above theory.

3.3.3.2 Allowable compressive stresses for shell courses

BS 2654 makes reference to BS 5387 "Specification for vertical cylindrical welded storage tanks for low-temperature service down to -196°C " and in particular to Clause 9-2-3 of that specification which gives a method for calculating the allowable compressive stresses for the shell courses, measured at each horizontal weld seam as:

$$S_c = 12.5 \frac{(t-c)}{R} \times G\eta \quad \text{equ 3.12}$$

where:

- S_c = the allowable compressive stress (N/mm^2)
- t = the shell plate thickness at the point under consideration (mm)
- c = the corrosion allowance, if applicable (mm)
- R = the radius of the tank (m)
- G = the factor for increase of the allowable stress for the loading combinations given below
- η = the joint efficiency factor which is 1.0 for butt-welded shells

The following loading combinations decide which value of 'G' is used in equation 3.12 as follows:

- (a) Dead weight above point under consideration + insulation + 50% pipe loads + superimposed load.
For this condition $G = 1.0$
- (b) Dead weight above point under consideration + insulation + pipe loads + wind load + 50% of superimposed load.
For this condition $G = 1.25$
- (c) Dead weight above point under consideration + insulation + pipe loads + earthquake load + 50% of superimposed load.
For this condition $G = 1.33$

Note: The superimposed load = 1.2 kN/m^2 of projected roof area which includes vacuum, snow and live loads.

There is apparent similarity between equation 3.11 and equation 3.12 but equation 3.12 recognises the limitations of the theoretical formula and also allows for the various loading possibilities given above and thus limits the allowable compressive stresses to well below the theoretical values which would be obtained from equation 3.11.

3.3.3.3 Actual compressive stress

Equation 3.12 gives the allowable compressive stress for each course and the actual compressive stress due to the various factors given in Sections 3.3.3.2 (a), (b) & (c) must be compared to this.

The actual stresses due to dead weight, insulation load, pipe loads, and superimposed load are fairly straightforward to calculate as:

$$\sigma_z = \frac{W}{\pi \cdot D \cdot t} \quad \text{equ 3.13}$$

where:

- σ_z = actual compressive stress
- W = summation of these loads
- D = tank diameter
- t = thickness of the course under consideration

The axial stresses due to the wind load and any seismic load are a little more complicated to calculate. Seismic analysis is dealt with later in Chapter 15 and the resulting axial stresses can be derived from there.

3.3.3.4 Axial stress due to wind loading on the shell

The axial stress due to wind load is now discussed and this is based on the "Engineering Bending Theory" where the circular shape is assumed to undergo small displacements. This is considered to be a reasonable assumption, in that the aim of the design approach is to maintain a circular cross section at all heights of the tank. This is certainly achieved at the base, where the axial stress has a maximum value.

The axial stress ' σ_z ' due to the wind load, causing a bending moment 'M' is therefore expressed as:

$$\sigma_z = \pm \frac{4 \cdot M}{\pi \cdot D^2 \cdot t} \quad \text{equ 3.14}$$

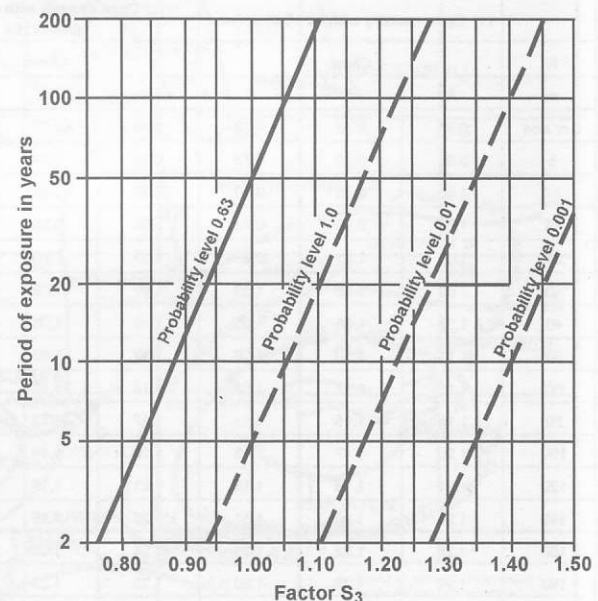
The moment M produces a stress σ_z which is approximately uniform across the wall thickness. On the windward side this axial stress is tensile and on the leeward side it is compressive.

The value of M in equation 3.14 is determined from the wind loading on the tank. In following the BS2654 approach, this is derived by determining:

- (a) The geographical location of the vessel and from this the basic wind speed, V which is the 3-s gust speed estimated to be exceeded on average once in 50 years.
- (b) Four wind speed factors, S_1 , S_2 , S_3 and S_4 defining the topography (S_1), ground roughness (S_2), a freak wind probability factor, (S_3) and a directional factor (S_4).

Values of basic wind speed for UK locations and values for the above factors are given in British Standard CP3, Chapter V, Part 2, 1972. They are reproduced in Figures 3.9 to 3.12.

For areas outside the UK, the wind speed information can be obtained from local meteorological stations.



Wind directions (degrees)	North			East			South			West		
	0	30	60	90	120	150	180	210	240	270	300	330
General S_4	0.78	0.73	0.73	0.74	0.73	0.80	0.85	0.93	1.00	0.99	0.91	0.82
Coastal S_4	0.84	0.78	0.78	0.79	0.78	0.86	0.91	1.00	1.00	1.00	1.00	0.88

(Coastal values of S_4 are applicable within 5 km of the coast for on-shore wind directions.)

Figure 3.12 Factors S_3 and S_4
From British Standard CP3

There is now an alternative Standard which is used for wind loadings and this is BS 6399 Part 2. But as CP3, Chapter V, Part 2 has been used successfully for many years and as BS 2654 still refers to it, its use will be continued here.

The design wind speed V_s is given by:

$$V_s = V S_1 S_2 S_3 S_4 \text{ (m/s)} \quad \text{equ 3.15}$$

This is converted to a dynamic pressure by using

$$q = \frac{1}{2} \rho V_s^2 \quad \text{equ 3.16}$$

where:

ρ is the density of air. The figure is the density of air at 15°C and under atmospheric pressure, viz., $\rho = 1.227 \text{ kg/m}^3$.

Thus:

$$q = 0.613 V_s^2 \text{ (N/m}^2\text{)} \quad \text{equ 3.17}$$

The pressure varies round the tank in such a way that on the windward side only $\pm 40^\circ$ the circumference of the tank is subject to a radial inward pressure. The rest of the tank is subject to suction i.e. an outward pressure. Details of this variation are given in British Standard CP3, Chapter V Part 2. In view of this variation the total horizontal wind load on the shell is given by:

$$F = C_f q A_e \quad \text{equ 3.18}$$

where:

C_f = the force coefficient for the tank and takes into consideration the pressure variation. It varies from 0.5 to 1.2 depending upon the height/diameter ratio, the velocity of the wind and the smoothness of the tank, i.e. pipe projections, etc. (see CP3 and Figure 3.13).

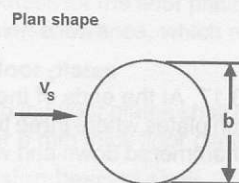
A_e = the effective frontal area, i.e. the area normal to the wind.

Other component parts attached to the shell may have a different factor, i.e. ladders, piping and equipment will have $C_f = 1.0$. If the cross-section changes, then the effective frontal area varies throughout the vessel length. Each section of the tank should therefore be considered and the wind load calculated.

It is generally assumed that the dynamic wind pressure is constant with the height of the tank so that the resultant wind force, F , acts at mid-height or alternatively it may be considered as a uniformly distributed force up the shell. Also it is general practice to allow for the effect of the horizontal wind force, which acts on the tank roof. Therefore the overall moment M on the tank can be shown with the help of Figure 3.14 as:

$$M = [F_s \cdot H/2] + [F_r(H + h/3)] \quad \text{equ 3.19}$$

where:



	$V_s b$	C_f for height / breadth ratio						
	m ² /s	Up to 1/2	1	2	5	10	20	x0
All surfaces	< 6	0.7	0.7	0.7	0.8	0.9	1.0	1.2
Rough or with projection	≥ 6	0.5	0.5	0.5	0.5	0.5	0.6	0.6
Smooth	≥ 6	0.5	0.5	0.5	0.5	0.5	0.6	0.6

Figure 3.13 Force coefficients C_f for clad buildings of uniform section (acting in the direction of the wind)

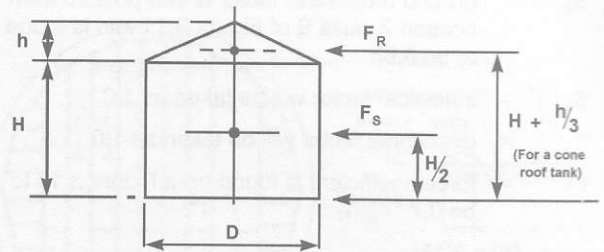


Figure 3.14 Effect of the horizontal wind force acting on the tank roof

$$F_s = C_f \cdot q \cdot D \cdot H \quad \text{equ 3.20}$$

and:

$$F_r = C_f \cdot q \cdot \frac{1}{2} D \cdot h \text{ (for a cone roof tank)} \quad \text{equ 3.21}$$

3.3.4 Allowable compressive stress

Using the data from the earlier tank design illustration in Figure 3.8, the axial stress in the shell bottom course, which is due to the vertical loadings, and the wind load can be analysed.

The allowable compressive stress from equation 3.12 is:

$$S_c = 12.5 \frac{(t-c)}{R} \times G \eta$$

Where in this case:

$$t = 12.6 \text{ mm}$$

$$c = 0 \text{ mm}$$

$$R = 15 \text{ m}$$

$$G = 1.25 \text{ (using the loading combination (b) in Section 3.3.3.2 for this example)}$$

$$\eta = 1.0$$

Then:

$$S_c = 13.125 \text{ N/mm}^2$$

The actual compressive axial load on the bottom course of the shell is made up of the following components:

The weight of the roof plating: = 29,000 kg or 284.40 kN
(assume to be 5 mm thick and the roof to have a 1:5 slope)

The weight of the roof supporting structure:
Assume to be 25,800 kg or 253.02 kN

50% of the superimposed roof load of 1.2 kN/m² 424.12 kN

The complete weight of the shell 1059.31 kN

Weight of thermal insulation Nil

Piping loads Say 20.00 kN

Total load = 981.54 kN

From equation 3.13:

$$\sigma_z = \frac{981.54}{\pi \cdot 30 \cdot 12.6} = 0.827 \text{ N/mm}^2$$

Referring to the design illustration in Figure 3.8, the compressive axial load due to the wind load on the tank can be found by using data from CP3, Chapter V Part 2,

where:

V = basic wind speed for the tank site in Liverpool
is taken from Figure 3.10 and is 46 m/s

S_1 = topography factor will be taken as 1.0

- S_2 = ground roughness factor is interpolated from column 2 class B of Figure 3.11 and is found to be 0.96
 S_3 = statistical factor will be taken as 1.0
 S_4 = directional factor will be taken as 1.0
 C_f = force coefficient is found from Figure 3.13 to be 0.7

From equation 3.15:

$$\text{The design wind speed } V_s = 46 \times 0.96 = 44.16 \text{ m/s}$$

From equation 3.17:

$$\text{The dynamic pressure } q = 0.613 \times 44.16^2 = 1195.40 \text{ N/m}^2$$

From equation 3.20:

$$\text{The load on the shell } F_s = 0.7 \times 1195.40 \times 30 \times 16 = 401,654.40 \text{ N}$$

From equation 3.21:

$$\text{The load on the roof } F_r = 0.7 \times 1195.4 \times 15 \times 3 = 37,655.10 \text{ N}$$

Using equation 3.19, the total wind moment on the tank is:

$$M = 401,654.40 \times 16/2 + 37,655.10 \times 16 \times 3/3$$

$$M = 3,853,371.90 \text{ Nm}$$

From equation 3.14:

$$\sigma_z = \frac{4 \times 3,853,371.90}{\pi \times 30^2 \times 0.0126} = 432,65128 \text{ N/m}^2 = 0.433 \text{ N/mm}^2$$

The actual axial compressive stress due to vertical loads and wind loading is:

$$0.827 + 0.433 = 1.26 \text{ N/mm}^2$$

which is well within the allowable stress of 13.125 N/mm^2 for this tank.

The effect of any seismic loading on the axial compressive stress is considered in Chapter 15.

3.4 Tank Floors

Section 3.3.2.1 explains how the shell is designed for a given set of conditions and therefore other conditions, which may impose additional stresses in the tank, must be avoided.

This being the case then the successful construction and operation of a storage tank relies on the tank being built on a firm foundation, which will not suffer undue differential settlement.

The foundation may take several forms and may be:

- Flat
- Rise to the centre, allowing drainage to the periphery of the tank
- Fall to the centre, allowing drainage to a centre sump
- Fall in one plane from one side of the tank to the other, allowing drainage to the low point at the periphery of the tank

The British and American tank Codes give recommendations for the construction of tank foundations in Appendix A and Appendix B of each Code respectively.

The tank floor is generally formed by a thin steel membrane, consisting of a number of plates welded together. This membrane has little inherent strength to resist distortion when the tank is loaded and will conform to the shape of the underlying foundation.

The floor plates, which are remote from the shell, will not be unduly stressed unless there is an abnormal amount of settlement in the foundation under them.

The area of the foundation immediately under where the shell meets the floor is particularly critical, because differential settlement here can cause the tank to try and "bridge" the area of settlement, thus inducing undesirable additional stresses in the shell-to-bottom area of the tank.

Out-of-plane, or differential settlement at the bottom edge of the tank can also cause flat areas to develop in the shell plating which in turn can affect the connecting nozzles and pipework, giving rise to additional stresses in these areas.

Floating roof tanks can also suffer a lack of circularity at the top of the tank, which can cause damage to the seal and in severe cases cause the floating roof to jam.

3.4.1 Floor plate arrangements

The floor plating may be one of two types:

- 1) A series of flat, generally rectangular, plates with lapped joints, fillet-welded on the top side only.

This type of floor is used for small tanks and in the areas where the tank shell passes over the outer lapped joints, the laps are joggled and any gap at the joggle is flushed off with weld metal to form a flat surface for the shell.

- 2) A ring of peripheral plates known as floor annular plates, which have a circular outside circumference and usually a regular polygonal shape inside the tank, are butt-welded together using backing strips. The inner floor plating is as described above, but in this case joggling is not necessary.

This type of floor is used for larger tanks where the annular plates allow the weight of the shell to be spread into the foundation and also to carry the radial bending stresses resulting from the discontinuity of the shell-to-floor joint. This is discussed in Section 3.3.6.

The requirements for floor plating, especially with regard to annular plates, differ between the British and American Codes, and these are explained as follows:

3.4.2 British Code requirements

3.4.2.1 Tanks up to and including 12.5 m diameter

The floors of tanks up to and including 12.5 m diameter, unless specified otherwise by the purchaser, shall be as 1) in Section 3.4.1.

The arrangement and details of the floor is as shown in Figures 3.15 to 3.17.

Floor plate joints

Referring to Figure 3.17. At the ends of the cross joints in the rectangular and sketch plates where three thickness occur, the upper plate shall be hammered down and welded as indicated in detail 'A' or 'B'.

The ends of the joints in the sketch plates under the bottom course of shell plating shall be joggled and welded for a minimum distance of 150 mm as shown in Figure 3.16 to ensure a flat surface on which to land the shell plating.

Welded joints

All lapped joints in the rectangular and sketch plates shall be full fillet-welded on the top side only. Care must be taken that the welds are continuous to ensure that there will be no leak paths through the joints particularly at the weld pick-up points.

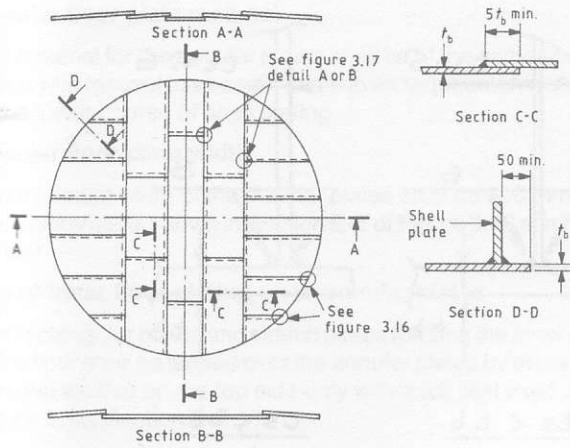


Figure 3.15 Typical floor arrangement for tanks up to and including 12.5 m diameter

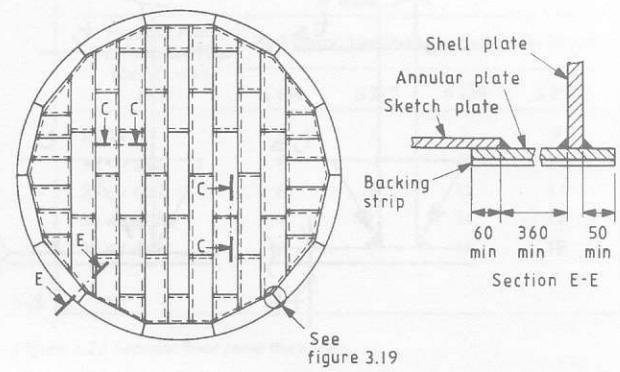


Figure 3.18 Typical floor arrangement for tanks over 12.5 m diameter

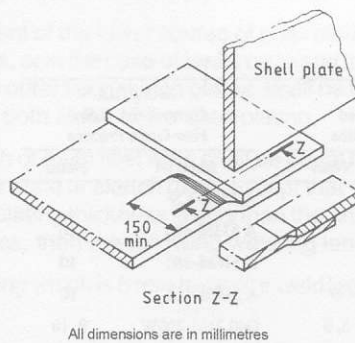


Figure 3.16 Joggled outer joints under shell plating

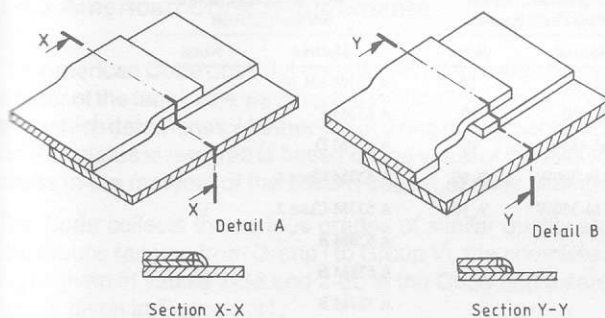


Figure 3.17 Joints in floor plates where three thicknesses occur

Floor plate minimum thickness

The minimum thickness for the floor plating shall be 6 mm, excluding any corrosion allowance, which may be required.

Minimum lap in floor plates

The minimum lap in the floor plates shall be 5 x the plate thickness i.e. 30 mm for 6 mm thick floor plates.

Floor plate extension beyond shell

The minimum extension of the floor plating beyond the shell plating shall be 50 mm. In practice designers usually allow between 60 and 80 mm to allow for possible shrinkage in the floor plating during welding and also for any irregularities in the circularity of the shell plating during erection and welding.

3.4.2.2 Tanks above 12.5 m diameter

Floor arrangement

The floors of tanks over 12.5 m diameter, unless specified otherwise by the purchaser, shall be as 2) in Section 3.4.1.

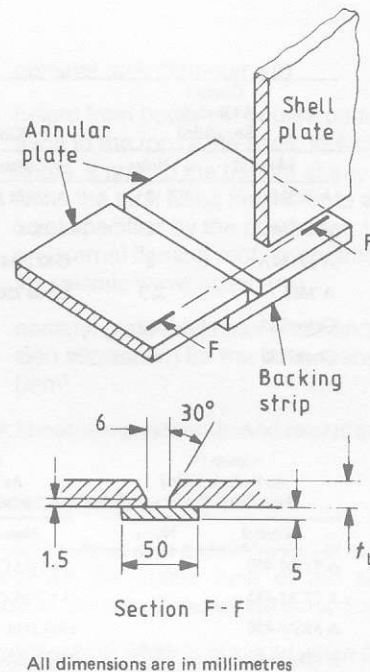


Figure 3.19 Joints between annular plates

The arrangement and details of the floor is as shown in Figures 3.18 and 3.19. The detail shown in Figure 3.17 also applies to this type of floor.

Minimum thickness of annular plates

The minimum thickness of the annular plates (excluding any corrosion allowance) shall be:

- 8 mm when the bottom course of shell plating is 19 mm thick or less.
- 10 mm when the bottom course of shell plating is over 19 mm and up to 32 mm thick.
- 12.5 mm when the bottom course of shell plating is over 32 mm thick.

Tanks up to and including 12.5 m diameter, if required by the purchaser, may be provided with a ring of annular plates, and in such cases the thickness of the annular plates shall not be less than 6 mm (excluding any corrosion allowance).

Annular floor plate welding

The radial seams connecting the ends of the annular plates shall be full penetration butt welds using backing strips as shown in Figure 3.19.

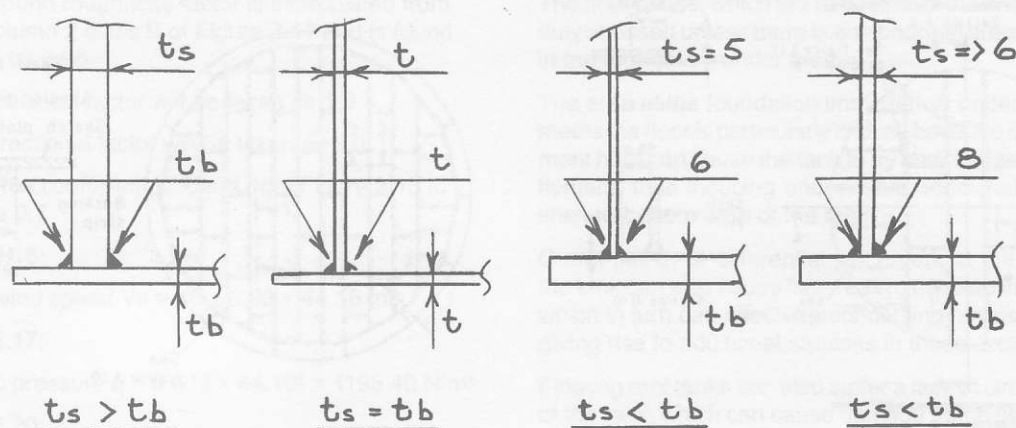


Figure 3.20 Leg lengths for shell-to-floor welds

Group I As Rolled, Semikilled		Group II As Rolled, Killed or Semikilled		Group III As Rolled, Killed Fine-Grain Practice		Group IIIA Normalized, Killed Fine-Grain Practice	
Material	Notes	Material	Notes	Material	Notes	Material	Notes
A 283M C	2	A 131M B	7	A 573M-400		A 131M CS	
A 285M C	2	A 36M	2, 6	A 516M-380		A 573M-400	10
A 131M A	2	G40.21M-260W		A 516M-415		A 516M-380	10
A 36M	2, 3	Grade 250	5, 8	G40.21M-260W	9	A 516M-415	10
Grade 235	3, 5			Grade 250	5, 9	G40.21M-260W	9, 10
Grade 250	6					Grade 250	5, 9, 10

Group IV As Rolled, Killed Fine-Grain Practice		Group IVA As Rolled, Killed Fine-Grain Practice		Group V Normalized, Killed Fine-Grain Practice		Group VI Normalized or Quenched and Tempered, Killed Fine-Grain Practice Reduced Carbon	
Material	Notes	Material	Notes	Material	Notes	Material	Notes
A 573M-450		A 662M C		A 573M-485	10	A 131M EH 36	
A 573M-485		A 573M-485	11	A 516M-450	10	A 633M C	
A 516M-450		G40.21M-300W	9, 11	A 516M-485	10	A 633M D	
A 516M-485		G40.21M-350W	9, 11	G40.21M-300W	9, 10	A 537M Class 1	
A 662M B				G40.21M-350W	9, 10	A 537M Class 2	13
G40.21M-300W	9					A 678M A	
G40.21M-350W	9					A 678M B	13
E 275	4, 9					A 737M B	
E 355	9					A 841	12, 13
Grade 275	5, 9						

Notes:

- Most of the listed material specification numbers refer to ASTM specifications (including Grade or Class); there are, however, some exceptions: G40.21M (including Grade) is a CSA specification; Grades E 275 and E 355 (including Quality) are contained in ISO 630; and Grade 37, Grade 41, and Grade 44 are related to national standards (see 2.2.5).
- Must be semikilled or killed.
- Thickness ≤ 20 mm.
- Maximum manganese content of 1.5%.
- Thickness 20 mm maximum when controlled-rolled steel is used in place of normalized steel.
- Manganese content shall be 0.80–1.2% by heat analysis for thicknesses greater than 20 mm, except that for each reduction of 0.01% below the specified carbon maximum, an increase of 0.06% manganese above the specified maximum will be permitted up to the maximum of 1.35%. Thicknesses ≤ 20 mm shall have a manganese content of 0.8–1.2% by heat analysis.
- Thickness ≤ 25 mm.
- Must be killed.
- Must be killed and made to fine-grain practice.
- Must be normalized.
- Must have chemistry (heat) modified to a maximum carbon content of 0.20% and a maximum manganese content of 1.60% (see 2.2.6.4).
- Produced by the thermo-mechanical control process (TMCP).
- See 3.7.4.6 for tests on simulated test coupons for material used in stress-relieved assemblies.

Figure 3.21 Sample from table 2-3a

From API 650

Annular floor plate material

The material for the annular plates shall be of the same specification with respect to strength and impact requirements as that of the lower course of shell plating.

Annular floor plate width

The minimum width of the annular plates shall be 500 mm and the requirements shown in Section E-E of Figure 3.18 shall also be met.

Lap of inner floor plating on to annular plates

The rectangular plates and sketch plates forming the inner area of the floor shall be lapped over the annular plates by at least 60 mm and welded on the top side only with a full fillet weld. (See Figure 3.18, Section E-E.)

Attachment of the lower course of shell plating to the floor plating

The following requirements apply to all sizes of tank.

The attachment of the lower course of shell plating to the annular floor plates, or in the case of tanks up to and including 12.5 m diameter, the outer floor sketch plates, shall be by a continuous fillet weld on both sides of the shell plating.

The leg length of each fillet weld shall be equal to the thickness of the annular plate or sketch plate, except that where the lower course shell plating thickness is less than the annular, or sketch plate thickness, then the following weld leg length shall apply.

For shell plating which is 5 mm thick, the weld leg length shall be 6 mm.

For shell plating which is 6 mm or thicker, the weld leg length shall be 8 mm.

These requirements are shown pictorially in Figure 3.20.

3.4.3 American Code requirements

The American Code does not classify the floor design by the diameter of the tank in the way that the British Code does. The criteria which determines whether or not a ring of segmental annular floor plates is required is based on the value of the allowable stress in the material of the bottom course of shell plating.

The Code collects the various grades of similar quality steels into groups ranging from Group I to Group VI, the complete listing is given in Tables 2-3a and 2-3b in the Code and a sample only is given in Figure 3.21.

3.4.3.1 Annular floor plates

When the bottom shell course is designed using the allowable stress for materials in Group IV, IVA, V, or VI, then butt-welded annular bottom plates shall be used.

When the bottom shell course is designed using the allowable stress for materials in Group IV, IVA, V, or VI and the maximum product stress 'Sd' (see equation 3.34) for the bottom course is less than or equal to 160 N/mm² (23,200 lbf/in²), or, the maximum hydrostatic test stress 'St' (see equation 3.35) for the bottom course is less than or equal to 172 N/mm² (24,900 lbf/in²), then lap-welded floor plates may be used instead of butt-welded annular plates.

Annular floor plate thickness

Where annular plates are used their thickness is determined using Table 3-1 of the Code and this is reproduced in Figure 3.22.

The hydrostatic test stress in the bottom course of the shell plating is found from:

Nominal Plate Thickness of First Shell Course (mm)	SI Units			
	Hydrostatic Test Stress in First Shell Course			
	≤ 190	≤ 210	≤ 230	≤ 250
$t \leq 19$	6	6	7	9
$19 < t \leq 25$	6	7	10	11
$25 < t \leq 32$	6	9	12	14
$32 < t \leq 38$	8	11	14	17
$38 < t \leq 45$	9	13	16	19

Figure 3.22 Annular floor plate thickness

From API 650, table 3-1

$$St = \frac{4.9 \cdot D \cdot (H - 0.3)}{t}$$

where

D = nominal tank diameter (m)

H = height from bottom of course under consideration to the top of the shell, including the top angle, if any; to the bottom of any overflow that limits the tank filling height; or to any other level specified by the purchaser, restricted by an internal floating roof, or controlled to allow for seismic wave action (m)

t = nominal plate thickness (including any corrosion allowance) for the bottom shell course (mm)

The above thickness are a minimum and exclude any corrosion allowance.

Annular floor plate width

Annular floor plates shall have a radial width of at least 600 mm measured between the inside face of the shell and any lap-welded joint in the remainder of the inner floor plating.

However a greater radial width is required when dictated by the following calculation:

$$\frac{215 \cdot tb}{\sqrt{H \cdot G}}$$

The detailed analysis of the width of annular plates is dealt with in Section 3.4.3.

The annular plate must also project at least 50 mm outside the outer face of the shell.

Annular floor plate welding

Floor annular plate radial joints shall be butt-welded by having their parallel edges prepared for butt-welding with either, square, or V grooves. If square grooves are used, the root opening shall not be less than 6 mm. The butt weld shall be made by tack welding a backing strip at least 3 mm thick to the underside of the annular plate such that it is centralised under the joint. A metal spacer shall be used to maintain the root gap between the adjoining plate edges to prevent shrinkage during welding, although other methods may be employed at the purchaser's approval.

Spacing of joints

Three plate lap joints in the inner floor plating must be at least 300 mm from each other, from the tank shell, from butt-welded annular plate joints and from joints between annular plates and the inner floor plating.

Inner floor plating

The inner floor plating, which is lapped on to the inner edge of the annular plates, shall conform to the requirements given below for "Floors formed from lap-welded plates only".

3.4.3.2 Floors formed from lap-welded plates only

Without annular plates

Where it is found that annular plates are not required, then all lap-welded floors can be employed.

Minimum thickness of lapped floor plates

The minimum thickness for all floor plates is 6 mm, excluding any corrosion allowance, which may be required.

Minimum width of floor plates

Unless otherwise agreed by the purchaser, all rectangular and sketch plates shall have a minimum width of 1800 mm and should be reasonably rectangular and square-edged.

Minimum lap

The overlap in lapped floor joints shall be a minimum of 5 x the floor plate thickness.

Three plate laps

Three plate laps in tank floors shall be at least 300 mm from each other, from the tank shell, from butt-welded annular plate joints and from joints between annular plates and the inner floor.

Note: The lapping of two inner floor plates on to the butt-welded annular ring does not constitute a three plate lap.

Floor projection

The lap-welded floor plates shall project at least 25 mm beyond the outside edge of the outer weld attaching the shell to the floor plating.

Welded joints

Lapped floor plates are to be welded on the top side only, with a continuous full fillet weld on all joints. Care must be taken, during welding, to ensure that no leak paths are left through the joints, particularly at the weld pick-up points.

Joints under the shell plating

The ends of the joints in the sketch plates under the bottom course of shell plating shall be joggled and welded for a minimum distance of 150 mm as shown in Figure 3.16, to ensure a flat surface on which to land the shell plating.

Attachment of the lower course of shell plating to the floor plating for all tanks

This attachment shall be by continuous fillet welds on each side of the shell plating.

The requirements of the American Code are more detailed than the British Code.

The American Code applies two sets of requirements, one for lapped floor plates or annular plates which are equal to or less than 12.5 mm thick, the other for annular plates which are more than 12.5 mm thick.

3.4.3.3 Lapped floor plates, or annular plates ≥ 12.5 mm thick

The following requirements shall be observed:

- 1) The size of the fillet welds shall not be less than the thinner of the two plates being joined (i.e. the floor or annular plate under the shell, and the shell plate).
- 2) The maximum size of the weld allowed is 12.5 mm.
- 3) The minimum size of weld shall not be less than that shown in the following table:

Nominal thickness of the shell plate (mm)	Minimum size of fillet weld (mm)
5	5
> 5 to 20	6
> 20 to 32	8
> 32 to 45	10

3.4.3.4 Annular plates >12.5 mm thick

The following requirements shall be observed:

The attachment welds shall be sized so that either the legs of the fillet welds, or the groove depth plus the leg of the fillet, for a combined weld, is of a size equal to the annular plate thickness, but shall not exceed the shell plate thickness. See Figure 3.23.

3.4.3.5 Shell-to-floor plate welds — consideration for specific materials

Shell-to-floor fillet welds for shell materials in Groups IV, IVA, V or VI shall be made with a minimum of two passes.

3.4.3.6 Tank floors which require special consideration

The floor arrangements shown in Figures 3.15 and 3.18 work well for the range of shapes listed above. They may be:

- Flat
- Rise to the centre, allowing drainage to the periphery of the tank
- Fall to the centre, allowing drainage to a centre sump
- Fall in one plane from one side of the tank to the other, allowing drainage to the low point at the periphery of the tank

The floor slope required to give a small fall or rise in the foundation to the centre of a tank can be accommodated by the lapped rectangular floor plates, as they will "scissor" at the edges to give a varying lap width down the length of the plate. However when the slope is more acute the "scissor" effect becomes more pronounced due to the conical form of the floor. In these cases the solution is to make the floor out of sector shaped petal plates.

Also, if annular plates are required, these will theoretically take on a conical form, but as these plates are relatively narrow, and if they are made in shorter than the normal length, then in most cases they will be found to accept the foundation shape and will not require to be developed, or rolled to a conical shape. For large diameter floors it may be found more economical, in terms of area of plate used, to make the floor petals in two pieces. If this is the case, then as an aid to erection and welding, the pieces forming one petal should be butt-welded together to form a flat plate thus avoiding another lap joint in the floor.

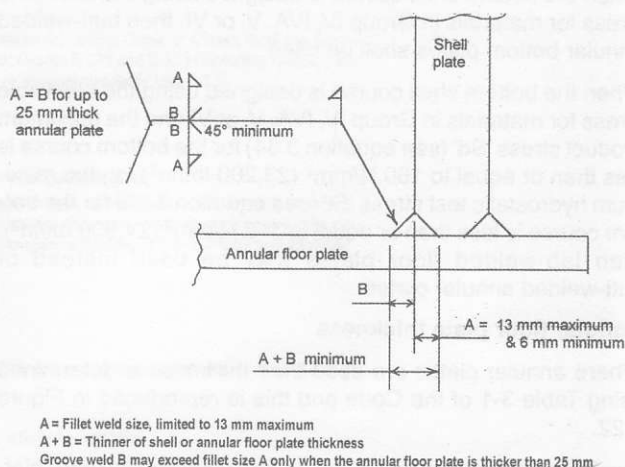


Figure 3.23 Detail of double fillet-groove weld for annular floor plates with a nominal thickness > 12.5 mm

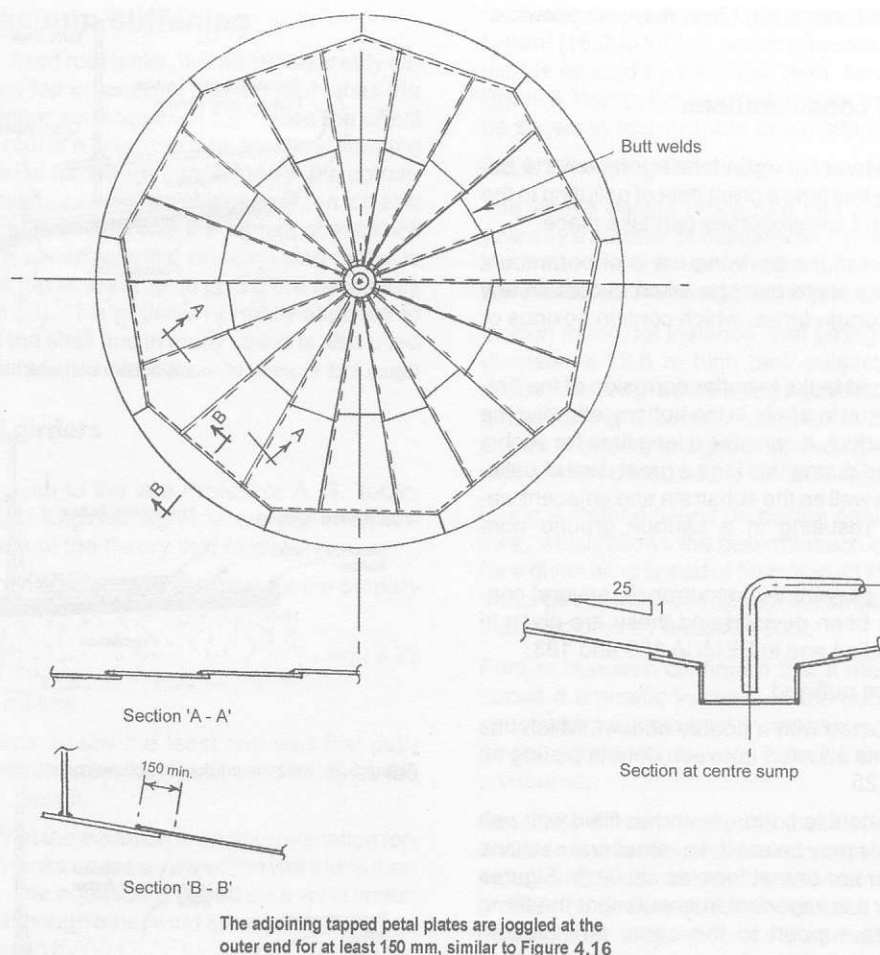


Figure 3.24 Floor plate arrangement for steeper sloping floors

The outer ends of the lap joints in the petal plates should be joggled to give a smooth transition on to the face of the annular plates.

Figure 3.24 shows the arrangement of such a floor.

3.4.3.7 Floor arrangement for tanks requiring optimum drainage

The presence of water in some stored products is highly undesirable. However as most petrochemical products are not miscible in water and the fact that they are generally lighter than water, any moisture in suspension in the liquid, tends to gravitate to the bottom of the tank.

One of the best ways to collect this water is to have a steeper sloping cone down floor, with a central collecting sump from which a suction drainpipe can be taken.

To ensure that the droplets of water drain to the sump it is important for the surface of the floor to be smooth, with no lap joints, discontinuities or pockets for the water to lodge in.

The arrangement of such a floor is similar to that shown in Figure 3.24 with certain alterations to the construction as follows:

The radial lap welds between the inner floor petals is acceptable but there must be no distortion due to welding which would allow the floor joint to lift in places thus forming pockets where water could lodge.

A means of preventing this, is to design the foundation as a solid concrete plinth into which are set radial steel members at the joint lines of the petal plates, the flanges of these members being flush with the conical surface of the foundation. The radial edges of the petal plates are welded to the flanges (either by lapping or by butt welding, using the flange as a backing strip) and hence the conical shape is maintained.

The lap at the outer end of the petal plates is reversed. That is to say the annular plates lie on top of the petal plates. This is to prevent the retention of water at the lap joint.

Care has to be taken to ensure that there is continuity of the backing strip for the butt joints between the annular plates, as this strip comes up against the outer edge of the petal plates.

This joint between the petal plates and the annular plates can be made as a butt-welded joint on to backing strips thus giving a smooth transition at the joint. The welding sequence and procedure for this approach needs careful consideration to avoid locked-in welding stresses, which can lead to distortion of the plates.

This latter type of floor construction is often favoured for tanks storing aviation fuel where it is of paramount importance to have "dry" fuel. Water in aircraft fuel lines at high altitude will freeze thus cutting off the supply to the engines with disastrous results, as airliners are not known to glide too well!

To keep the fuel clean, these tanks are very often internally lined with some form of epoxy coating. Also it is a common feature to make the relatively small-bore drain line from the sump out of a stainless steel material, because the successful internal coating a small-bore pipe is difficult. The problem with doing this is that if at some time the coating of the bottom of the sump is damaged or it perishes thus exposing the carbon steel plate, an electrolytic cell can be set up between the two dissimilar metals in the aqueous solution in the sump causing the carbon steel plate to erode and eventually perforate causing a leak.

This problem can be overcome by making the major part of the vertical section of the drainpipe in a fibreglass or composite pipe material, which is compatible with the fuel. The connection

between the stainless and composite pipes may be screwed or sleeved and clamped.

3.4.4 Environmental considerations

The effects of a leaking tank floor can take a long time to become evident and during this time a great deal of pollution to the surrounding substrata and watercourses can take place.

Nowadays the protection of the environment is of paramount importance, and therefore steps must be taken to contain any product leakage from storage tanks, which contain noxious or toxic products.

It is fairly common for aged tanks to suffer corrosion of the bottom plates, which can result in a hole in the bottom, allowing the release of the stored product. It can take a long time for such a leak to manifest itself and during this time a great deal of pollution of the foundation, as well as the substrata and adjacent watercourses can occur, resulting in a serious ground contamination problem.

In order to minimise, or prevent this occurrence, several construction methods have been devised and these are given in detail in API 650 Appendix I and in EEMUA 159 and 183.

A few of the methods are outlined:

- A) The tank is constructed with a double bottom, which has leak detection points situated between double plating as shown in Figure 3.25.

The space between the double bottom is shown filled with pea gravel but other materials may be used, i.e. structural sections or steel reinforcement in bar or mat form as shown in Figures 3.26 and 3.27. However it is important to ensure that the filling material gives adequate support to the upper tank bottom plates.

The drain pipes can be used as follows:

- As a visual indication of any leakage.
- For inserting a hydrocarbon sensor.
- For holding a vacuum in the interspace. The loss of vacuum indicates a leak.

In the event of a leakage, the disadvantage of the double bottom is twofold.

- 1) Dealing with the contaminated interspace in the confinements of the tank and without any hotwork being allowed.
 - 2) If the tank needs to be jacked vertically off its foundation at any time, then the additional weight of the double bottom construction makes this difficult.
- B) Two further examples of double bottoms (taken from the draft form of prEN 14015 -1: 2000) are shown in Figures 3.26 and 3.27.
- C) A membrane is introduced in the foundation between the tank bottom and the underlying substrate as shown in Figure 3.28.
- D) The tank is supported off a grillage on a concrete raft foundation as shown in Figure 3.29.

This arrangement is often used for acid storage tanks or tanks storing very toxic or noxious products where an early visual indication of a leaking bottom can be detected and dealt with without delay.

The spacing between the support beams, together with the height of the tank and the density of the stored product, will dictate the required thickness for the bottom plates.

This thickness is very often more than the minimum Code requirements and in many instances the thickness is such that lap-welded construction is impractical and the plates have to be butt-welded.

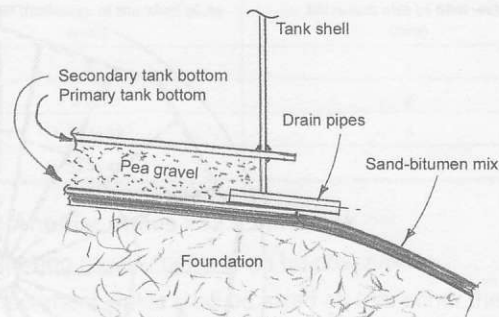


Figure 3.25 Example of double bottom with leak detection

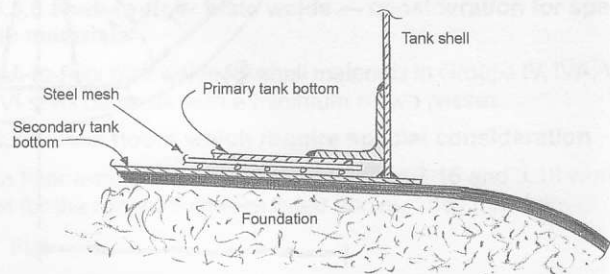


Figure 3.26 Variation on double bottom construction

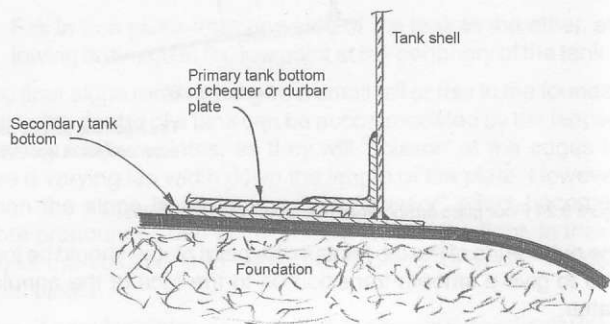


Figure 3.27 Further variation on double bottom construction

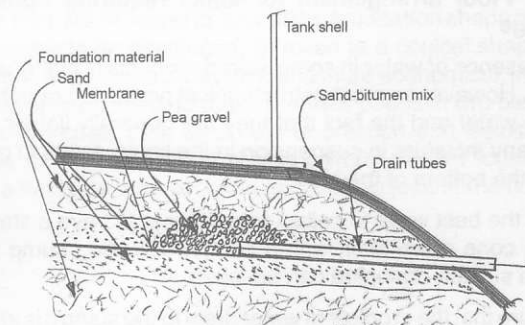


Figure 3.28 Use of membrane in foundation

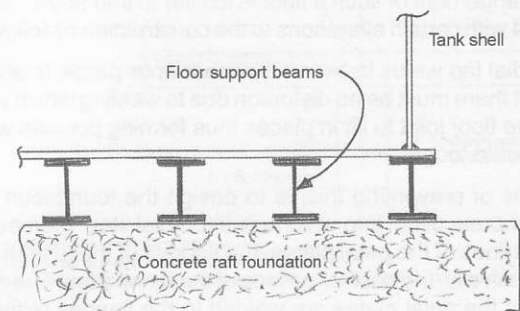


Figure 3.29 Concrete raft foundation

3.5 Wind and vacuum stiffening

For the case of closed, fixed roof tanks, the wind load is only external, whereas in open top or external floating roof tanks the wind also acts on the inner surface which can cause the effect of a vacuum load. The roof of a fixed roof tank assists in keeping the shell rigid and the wind forces are transmitted to the bottom of the tank as axial stresses as mentioned earlier. Open top and external floating roof tanks do not have the benefit of this shell rigidity and therefore a circumferential primary wind girder is provided at or near the top of the shell to give it the necessary stiffness (see Figure 3.30). This girder is normally attached to the external surface of the shell and in many cases is also used as an access and maintenance platform.

3.5.1 Primary wind girders

Acknowledgement is given to the late Professor A. S. Tooth, Professor of Mechanical Engineering at University of Strathclyde, Glasgow, for most of the theory that follows.

The equation to determine the section modulus for the primary wind girder is by:

$$Z = 0.58 \cdot D^2 \cdot H \text{ (cm}^3\text{)} \quad \text{equ 3.22}$$

where D and H are in metres.

The equation is simplistic to say the least and was first published in the early API tank Codes but is still used today as the basis of primary girder design.

Generally it is thought that the equation is an approximation formulated at a time when tanks under construction were less than 30 metres in diameter. The equation is based on a wind speed of 43.7 m/s (100 mph) although other wind speeds may be used by multiplying the equation by $(V/43.7)^2$ for SI units, or $(V/100)^2$ for Imperial units.

The equation may be derived, in SI units, using the above wind speed together with the dynamic wind pressure from equation 3.17. The horizontal wind load, using the terms D and H can be obtained from equation 3.18, using a C_f value of 0.6.

Assuming that the girder is loaded by a uniform pressure across the tank diameter and is supported by tangential shear, and that the pressure load on the top 25% of the shell has to be



Figure 3.30 Primary wind girder

carried by the girder, and the allowable design stress is 103.42 N/mm² (15,000 lbf/in²), which is increased by 25% because the load is caused by the wind, then, by referring to formulae by Roark & Young, the required section modulus for the girder can be shown to approximate to equation 3.22 above.

3.5.1.1 Refining the design technique

The above design procedure has been challenged over the years by a number of academics (e.g. Adams, Morton, Zick and McGrath) and the use of more analytical computer methods have enabled the design technique to be refined.

Morton found, for instance, that taking the example of an 84 m diameter x 12.5 m high tank subjected to a 100 mph wind speed, current practice using equation 3.22 suggests a primary girder having a section modulus of 2610 cm³ which can be shown to equate to a girder as shown in Figure 3.31, "Detail E", with a width dimension 'b' of 1050 mm.

Using a method based on design against plastic folding of the tank, which allows the determination of the girder dimensions for a given wind speed of 50 m/sec. (111.8 mph) it can be shown that a girder width of 432 mm is adequate, this is less than half that predicted by equation 3.22.

Further research confirmed that a modest girder section produced a dramatic increase in the buckling pressure and that subsequent incremental increases in the dimension 'b' of the girder produced a very small increase in the buckling pressures.

Generally it has been found that for large diameter open top and external floating roof tanks, say over 60 metres in diameter, equation 3.22 is over-conservative and that at, or over this diameter the girders calculate out to be unnecessarily wide. Accordingly, the present Code states that for tanks over 60 metres in diameter shall, for girder calculation purposes, be considered to be of this diameter when determining the section modulus of the primary girder.

However, as mentioned earlier, these primary girders are often used as access platforms and therefore, although a narrow girder may be found by design this may be increased in width to form a platform having a minimum width to Code of 600 mm.

For tanks where the primary girder is located 600 mm or more below the top of the shell the Code requires that the shell be provided with a top curb angle of the following dimensions:

For a top course thickness of 5 mm, the angle shall be 60 x 60 x 5 mm

For a top course thickness of 6 mm or more, the angle shall be 80 x 80 x 6 mm.

3.5.1.2 Design example

Using the principal dimensions for the tank in the earlier design illustration in Figure 3.8, but in this case assuming it is an external floating roof tank, and using a design wind speed of 46 m/sec, then:

$$D = 30 \text{ m diameter}$$

$$H = 16 \text{ m high}$$

$$V = 46 \text{ m/s}$$

From equation 3.22:

The section modulus for the primary girder is:

$$Z = 0.058 \cdot 30^2 \cdot 16 \cdot \left(\frac{46}{44.7} \right)^2 = 884.5 \text{ cm}^3$$

Referring to Figure 3.31 which is taken from BS 2654 it can be seen that a "Detail E" type girder will be sufficient and this has a horizontal web dimension 'b' of 500 mm when attached to a shell having a thickness of 8 mm.

Section through wind girder		Section modulus			
		Member size*	5 mm shell	6 mm shell	8 mm shell
			cm ³	cm ³	cm ³
Detail A top angle		60 X 60 X 5	4.64	4.77	—
		60 X 60 X 6	5.50	5.64	—
		60 X 60 X 8	7.13	7.29	—
		70 X 70 X 8	9.70	9.90	—
		80 X 80 X 8	12.68	12.92	—
Detail B curb angle		60 X 60 X 6	23.35	24.35	—
		70 X 70 X 6	31.88	33.24	—
		80 X 80 X 6	41.63	43.43	—
		90 X 90 X 6	52.69	54.98	—
		90 X 90 X 8	56.86	67.74	—
Detail C single angle		60 X 60 X 6	24.26	25.19	—
		60 X 60 X 8	29.70	31.06	—
		100 X 75 X 8	68.10	71.31	—
		100 X 75 X 10	79.26	83.43	—
		125 X 75 X 8	90.84	95.29	—
Detail D channel		178 X 76 (20.84 kg/m)	172.50	176.62	—
		203 X 76 (23.82 kg/m)	219.65	224.99	—
		229 X 76 (26.06 kg/m)	261.88	268.65	—
		254 X 76 (28.29 kg/m)	305.62	313.94	—
		254 X 89 (35.74 kg/m)	393.23	402.60	—
Detail E formed plate		305 X 89 (41.69 kg/m)	521.30	534.61	—
		305 X 102 (46.18 kg/m)	596.93	610.28	—
		b = 250	—	341.02	374.81
		b = 300	—	427.22	473.07
		b = 350	—	518.73	577.15
Detail E formed plate		b = 400	—	615.46	686.89
		b = 450	—	717.38	802.18
		b = 500	—	824.41	922.93
		b = 550	—	936.56	1049.09
		b = 600	—	1053.80	1180.59
		b = 650	—	1176.12	1317.40
		b = 700	—	1303.50	1459.48
		b = 750	—	1435.94	1606.81
		b = 800	—	1573.43	1759.34
		b = 850	—	1715.96	1917.07
		b = 900	—	1863.53	2079.98
		b = 950	—	2016.13	2248.04
		b = 1000	—	2173.58	2421.24

*Mass per unit length given in parentheses, where appropriate.
Dimensions are in millimetres, unless otherwise stated.

Figure 3.31 Wind girder sections
From BS 2654

This type of girder is normally shop-fabricated in several sections and is made of folded plate. In this case there would probably be 12 sections (the same number as the number of shell plates per course). The external flange of the girder sections would be polygonal with the inner edge of the web matching the radius of the tank shell.

This being the case, then to ensure the desired section modulus, the minimum width of the web will be 500 mm at the centre of the section, which will increase, reaching a maximum width at each end of the section, which by geometry will be found to be 1047 mm for this example. If the girder is to be used as a platform then the minimum width increases to 600 mm making the maximum width 1151 mm at the extremities of each section.

3.5.2 Secondary wind girders

3.5.2.1 Equivalent shell method

The shell of a storage tank is susceptible to buckling under the influence of wind pressure and internal vacuum, especially when in a near empty or empty condition. Accordingly the Design Code recognises this and requires an analysis of the shell to be made in order to ensure that it is stable under these conditions.

The fact that the shell is made up of courses of diminishing thickness, makes analysis difficult, so the method adopted in BS 2654 converts the multi-thickness shell into a equivalent shell having a thickness equal to that of the top course, with the height reduced in such a way that the stability of the actual shell is equal to that of the equivalent shell.

Work presented by Saunders and Windenberg (*Reference 3.3*) shows an approximate relationship for the uniform external pressure q' at which elastic buckling occurs in a short tube L , with ends held circular, or along tube held circular at intervals L . Their relationships have been simplified by Roark and may be written as:

$$q' = \frac{0.807 \cdot E}{L} \left(\frac{1}{1-\nu^2} \right)^{3/4} \frac{t^{5/2}}{R^{3/2}} \quad \text{equ 3.23}$$

where:

- E = modulus of elasticity for steel (N/mm^2)
- L = maximum length of shell (m)
- ν = poisson's ratio for steel
- t = constant shell thickness (m)
- R = radius of shell (m)

The individual shell course heights are derived using the dimensional analysis method and in conjunction with equation 3.23, with R constant in the equation, an equivalent buckling pressure q' is achieved when $L \propto t^{5/2}$.

Hence an equivalent height of each course can be found from the resulting equation:

$$H_e = h \left(\frac{t_{\min}}{t} \right)^{2.5} \quad \text{equ 3.24}$$

This equation is used in BS 2654 where:

- H_e = equivalent stable height of each course at thickness $t_{\min}$ (m)
- h = actual height of each course in turn below the primary ring (m)
- t = thickness of each course in turn (mm)
- $t_{\min}$ = thickness of the top course (mm)

Note: The course thicknesses are to be the corroded thicknesses if a corrosion allowance has been requested by the tank purchaser.

The total height of the equivalent shell, H_e , is found by adding together the equivalent heights of each course i.e.

$$H_e = \sum H_e \quad \text{equ 3.25}$$

3.5.2.2 Number of girders required

The dynamic wind pressure on the shell is obtained from the British Standard CP3, Chapter V, Part 2, Wind Loads, paragraph 6, and in SI units this is given as:

$$q = 0.613 \cdot V_s^2 \quad \text{equ 3.26}$$

where:

q = dynamic wind pressure (N/m^2)

V_s = design wind speed (m/sec)

The design vacuum in the tank V_a must be added to this, where V_a is in mbar and the equation becomes:

$$q = 0.613 \cdot V_s^2 + 100 \cdot V_a \quad \text{equ 3.27}$$

By equating the actual pressure q in equation 3.26 with the pressure q' to cause buckling in equation 3.23 it is possible to determine a value for the maximum permitted spacing L of the circumferential secondary wind girder(s) on the equivalent shell.

Note: L is given the notation H_p in BS 2654.

$$0.613 \cdot V_s^2 + 100 \cdot V_a = \frac{0.807 \cdot E}{L} \left(\frac{1}{1-\nu^2} \right)^{3/4} \frac{t^{5/2}}{R^{3/2}} \quad \text{equ 3.28}$$

Then

$$L = H_p = \frac{0.807 \cdot E}{(0.613 \cdot V_s^2 + 100 \cdot V_a)} \left(\frac{1}{1-\nu^2} \right)^{3/4} \frac{t^{5/2}}{R^{3/2}} \quad \text{equ 3.29}$$

Taking $E = 2.07 \times 10^{11} \text{ N/m}^2$, $\nu = 0.3$ and expressing t in mm then the equation becomes:

$$H_p = \frac{16,016}{(0.613 \cdot V_s^2 + 100 \cdot V_a)} \left(\frac{t^5}{D^3} \right)^{1/2} \quad \text{equ 3.30}$$

By multiplying the top and bottom of the equation by 5.8 the result approximates to the form given in BS 2654 as:

$$H_p = \frac{95,000}{(3.563 \cdot V_s^2 + 580 \cdot V_a)} \left(\frac{t_{\min}^5}{D^3} \right)^{1/2} \quad \text{equ 3.31}$$

BS 2654 further simplifies this equation into two equations. The first equation being given the constant value K thus:

$$K = \frac{95,000}{(3.563 \cdot V_s^2 + 580 \cdot V_a)} \quad \text{equ 3.32}$$

BS 2654 stipulates nominal values for V_a in equation 3.32 and these are as follows:

- 5 mbar for open top tanks irrespective of the design wind speed.
- 5 mbar for non-pressure, fixed roof tanks.
- 0.85 mbar for all other fixed roof tanks.

The second equation then becomes:

$$sH_p = K \left(\frac{t_{\min}^5}{D^3} \right)^{1/2} \quad \text{equ 3.33}$$

Which is the maximum permitted height of the unstiffened shell.

For any given tank, the results given by equation 3.25 and equation 3.31 are compared and if $H_p > H_E$ then the shell is sufficiently stable and does not require any secondary wind girders.

If $H_p < H_E$ then one or more secondary wind girders are required.

For instance if $H_p < H_E < 2H_p$ then one secondary wind girder is required.

This girder is positioned at $H_E/2$ down from the primary wind girder, or in the case of a fixed roof tank, down from the top of the shell.

If $2H_p < H_E < 3H_p$ then two secondary wind girders are required, and are positioned at $H_E/3$ and $H_E/2$ down from the primary girder, or top of the shell, as applicable.

The comparison between H_p and H_E is continued and hence the number of girders is established for each given tank.

In the event that multiple girders are found to be required, and this can happen on large tanks having a heavy shell corrosion allowance, then consideration can be given to increasing the upper course thickness in order to reduce the number of girders. This then becomes an exercise combining prudent design with construction costing to arrive at the most economic shell design.

For the method described above to be valid, the secondary wind girders must be located on shell courses having the same thickness as the top course. If this is not the case then adjustment to the position(s) has to be made by converting back the equivalent course heights to their actual values.

Also the Code requires that the girders must be at least 150 mm clear of the horizontal weld seams, but any adjustment for this must ensure that the maximum permitted height of the unstiffened shell, H_p is not exceeded.

Again, Morton found through his research, that secondary wind girders are required on the shell when under the influence of a uniform external pressure caused by sufficient wind pressure and internal vacuum. However, his research showed that the use of quite small ring sections produced a dramatic stiffening effect on a unreinforced shell. And that by increasing the size of the section did not significantly increase the buckling strength of the shell.

BS 2654 does not require the designer to calculate the section modulus for the secondary wind girders but instead tabulates the required angle ring girder section size against the tank diameter in question and these are given in Table 3 of the Code which is shown in Figure 3.32.

Tank diameter D (m)	Angle ring girder (other shapes may be provided having an equivalent section modulus) (mm)
$D \leq 20$	100 x 65 x 8
$20 < D \leq 36$	125 x 75 x 8
$36 < D \leq 48$	150 x 90 x 10
$48 < D$	200 x 100 x 12

Figure 3.32 Dimensions for shell circumferential secondary wind girders

It will be shown later in Section 3.6.7 that the American Code has a different approach to sizing secondary wind girder sections.

3.5.2.3 Worked example

An external floating roof tank 96 m diameter and 19 m high having eight, 2.375 m wide courses of thickness: 38.3, 33.4, 28.6, 23.7, 18.9, 14.0, 12.0 and 12.0 mm is to be designed for a wind speed of 60 m/sec. The primary girder is positioned at 1 m from the top of the shell.

Determine how many secondary wind girders are required, their size and their position on the shell.

$$V_s = 60 \text{ m/sec and}$$

$$V_a = 5 \text{ mbar}$$

Then from equation 3.32:

$$K = \frac{95,000}{3.563 \cdot 60^2 + 580 \cdot 5} = 6.041$$

and from equation 3.33:

$$H_p = 6.041 \cdot \left(\frac{12^5}{96^3} \right)^{1/2} = 3.203 \text{ m}$$

The total height of the equivalent shell H_E is found as follows:

H_E for each course is given by equation 3.24 and is tabulated as follows:

Course Number	h (m)	t (mm)	H_e (m)
1	1.375	12.0	1.375
2	2.375	12.0	2.375
3	2.375	14.0	1.615
4	2.375	18.9	0.763
5	2.375	23.7	0.433
6	2.375	28.6	0.271
7	2.375	33.4	0.184
8	2.375	38.3	0.131
			$H_E = 7.147$

As $2H_p < H_E < 3H_p$ ie. $6.406 < 7.147 < 9.609$

Then two secondary wind girders are required and these are located on the equivalent shell at $\frac{1}{3} H_E$ and $\frac{2}{3} H_E$ which is 2.382 m and 4.765 m down from the primary girder.

Both rings are more than 150 mm away from a horizontal weld seam and in this respect their position is acceptable. But it can be seen that when positioning the rings on to the actual shell, the top ring is on a course of minimum thickness but the lower ring is on the third course down which is 12.4 mm thick.

This lower ring will have to be repositioned on the 12.4 mm thick course by converting back the equivalent shell course height, H_e , to its actual value. This is accomplished by taking the section of the thicker course, measured from its top edge, down to the position of the girder and multiplying it by the reciprocal of the thickness as shown in equation 3.24 to the power 2.5.

This is performed as follows:

The section of the 14.0 mm course in this case =

$$4.765 - (1.375 + 2.375) = 1.015 \text{ m}$$

$$\text{This is adjusted to } 1.015 \times \left(\frac{14.0}{12.0} \right)^{2.5} = 1.492 \text{ m}$$

Then the new position for the girder measured down from the primary girder is:

$$1.375 + 2.375 + 1.492 = 5.242 \text{ m}$$

The complete mathematical equation can be shown as:

$$\{4.765 - (1.375 + 2.375)\} \times \left(\frac{14.0}{12.0} \right)^{2.5} + (1.375 + 2.375) = 5.242 \text{ m}$$

In this position the girder is also more than 150 mm clear of the adjacent horizontal weld seams.

The spacing between girders on the equivalent shell is, 2.382 m, 2.860 m and 1.905 m, which total 7.147 m (H_E). These spac-

ings are all less than the maximum permitted spacing of 3.203 m (H_p) and are therefore acceptable.

From Figure 3.32 it is seen that the size of the angle ring girders is to be 200 x 200 x 12.

The girders are located preferably on the outside of the tank shell but can be attached to the inside surface under certain circumstances, for example:

- To prevent a discontinuity in the insulation and cladding when the shell is to be thermally insulated.
- To prevent interference with a shell mounted spiral roof access staircase.

The disadvantages of internal girders are that:

- They hamper the internal cleaning of the tank shell.
- An internal floating cover cannot be installed in the tank.

3.5.3 Vertical bending of the shell

When a tank is being filled with product, the shell will expand radially due to the natural elasticity of shell plate material. This natural expansion is restrained at the point where the shell is welded to the bottom plating as shown in Figure 3.33 and this connection is therefore subjected to rotation.

Before analysing what occurs under this circumstance it is necessary initially to take the simplistic approach in order to establish what form the shell is trying to adopt under load. From basic engineering principles:

Young's modulus:

$$E = \frac{\text{Stress}}{\text{Strain}}$$

Then:

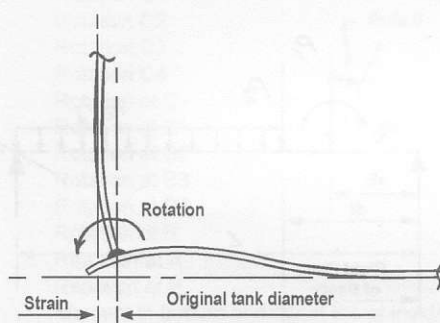


Figure 3.33 Shell-to-bottom connection under load

$$\text{Strain} = \frac{\text{Stress}}{E}$$

Then change in tank diameter = Original diameter x Strain.

3.5.3.1 Example

Consider the tank in the shell design illustration in Figure 3.8. The tank is 30 m diameter, with a bottom course thickness of 12.6 mm and a shell design stress of 183.333 N/mm² at a point 300 mm from the bottom of the course ($H - 0.3$). The tank is assumed to be full of product with a SG of 1.0.

Taking E to be 207,000 N/mm² for carbon steel, then the Strain is:

$$\frac{183.333}{207,000} = 0.000885666$$

The change in tank diameter is $0.000885666 \times 30,000 = 26.57$ mm or 13.29 mm on the radius.

3.5.3.2 Shell-to-bottom connection

The stresses in the tank shell have been dealt with earlier and further analysis is given later in Section 3.6, which deals with the "variable design point" method for shell design.

The amount of radial growth and the shape of the expanded shell can be best illustrated by modelling the area using a finite element analysis computer program and this can also include the effect of any external piping loads which are transmitted to the shell via the shell nozzles.

As mentioned above, the radial expansion of the shell is restrained at its junction with the bottom plating and it has been found in practice that the full theoretical hoop stress in the shell is not realised until a point which is about $\sqrt{D \cdot t}$ above the floor joint. This is illustrated later in Figure 3.40.

The rotation of the shell-to-bottom joint induces stresses in the bottom plating and the tank Codes give rules, which dictate the thickness and width requirements for the bottom plates, which are immediately under the shell. However, there are no specific design procedures given in the Codes for this critical area of bottom plating and whilst this Chapter is devoted to the design of the shell, it is difficult to divorce this area of bottom plating from the shell because the shell-to-bottom joint is very rigid and rotates as a unit when the tank is under hydrostatic load. This is demonstrated in Figure 3.34.

The section of the floor adjacent to the shell can be considered to be a horizontal projection of the shell itself and this section of the bottom therefore requires special consideration with regard to the stresses caused by the rotation and this analysis is included here.

Generally it is the larger diameter tanks which need detailed consideration in this area and it is found that the Codes require that these tanks are provided with a ring of annular floor plates which are butt welded together thus giving a smooth surface upon which the shell sits.

The expansion of the shell is restrained to practically zero at the welded joint between the shell-to-bottom plates and hence the shell tends to rotate in the outward direction about this joint. The welded connection of the shell to the bottom is very rigid and therefore as the shell rotates, the bottom plate also rotates which causes it to lift off the foundation for a distance inside the tank, until the pressure of the product acting on the floor, balances the lifting effect, this is depicted in Figure 3.34.

This action causes high bending stresses in the bottom plate and in the toe of the internal fillet weld, which are cyclic, due to the continual filling and emptying of the tank, and thus this area is subjected to low cycle fatigue.

The API 650 Code recognises this potential problem and specifies a design fatigue stress of 75,000 /in² (517 N/mm²) based

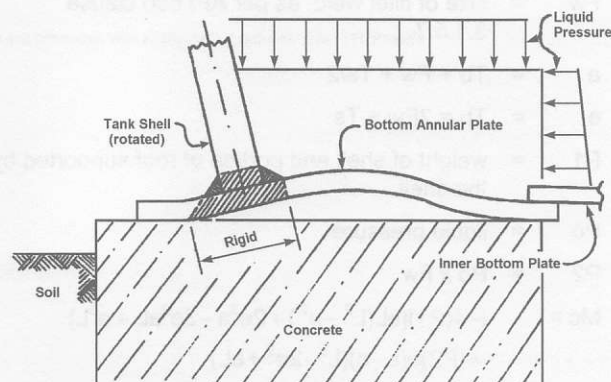


Figure 3.34 Rotation of the shell-to-bottom connection

upon 1300 cycles, which corresponds to one, filling/emptying cycle per week over 25 years.

3.5.3.3 Rotation and stress analysis

H. Kroon formulated a method for analysing the rotation and stresses at the joint (Reference 3.4) based on the following design conditions:

- The annular plate is considered to be a simply supported beam of unit width.
- The foundation is infinitely rigid (there is no vertical deflection).
- The length of the beam is the length required to reduce the rotation at the inside end to zero.
- The rotation of the shell is equal to the rotation of the bottom at the joint.
- Radial displacement is zero.
- The design fatigue stress is $75,000 \pm 100 \text{ lbs/in}^2$.
- The tank is at ambient temperature.
- The size of the fillet welds at the joint are as per the requirements of API 650 Clause 3.1.5.7.
- Elastic analysis. The use of elastic analysis for stresses beyond the yield strength assumes complete elastic action after a few repetitions of the stress cycle, which will increase the yield strength but leave a certain amount of permanent deformation.

3.5.3.4 Beam analysis

The beam is analysed by superposition of the rotation due to each load acting on the beam. The rotations are determined by the double integration method.

Referring to Figures 3.35 and 3.36. The unknowns M_c , R_a , R_b , L , and θ_c can be solved from the following equations:

- (1) M_c = moment in shell due to load and θ_c .
- (2) θ = 0
- (3) θ_c = θ_{shell}
- (4) $R_a + R_b$ = $P_1 + P_2 + p_o(L - e)$
- (5) $\sum M_b$ = 0

The example given later which demonstrates the use of H. Kroon's theory is given in Imperial units, the reason for this being that the theory is linked to the American API 650 Code, which at the time was exclusively expressed in Imperial units. However for the benefit of those not familiar with these units, the metric equivalents have been added. See Figure 3.37.

$$\begin{aligned}
 F_w &= \text{size of fillet weld, as per API 650 clause 3.1.5.7} \\
 a &= T_b + F_w + T_s/2 \\
 e &= T_b + 2F_w + T_s \\
 P_1 &= \text{weight of shell and portion of roof supported by the shell} \\
 P_o &= \text{liquid pressure} \\
 P_2 &= P_o \times F_w \\
 M_c &= \{-4(P_1)[aL(L^2 - a^2) + 2e^3a - 3e^2aL + a^3L] \\
 &\quad - 4(P_2)e(L - e)(L^2 - 2e^2 + eL) \\
 &\quad - (P_o)(L - e)^2(2eL^2 - 4e^3 + L^3 + e^2L)\} \\
 &\quad \div [4(-L^3 - 2e^3 + 3e^2L)]
 \end{aligned}$$

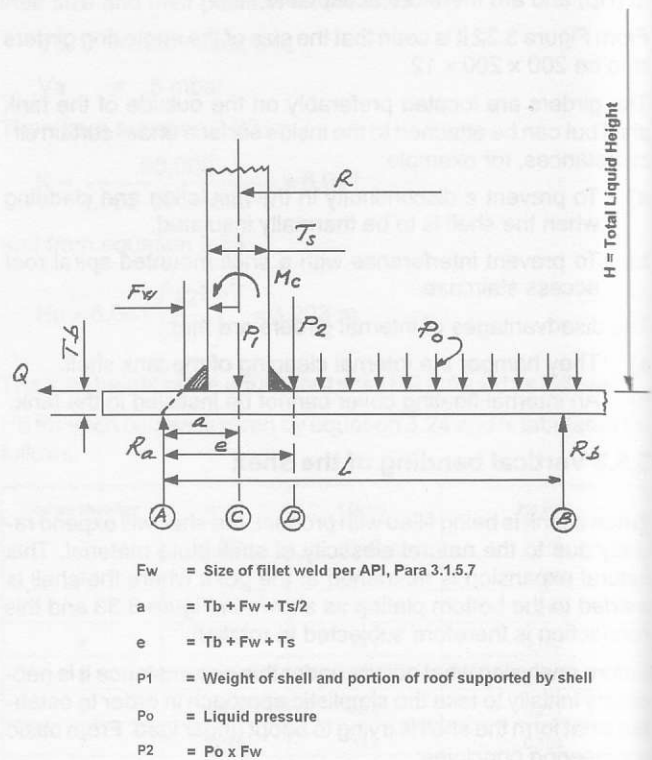


Figure 3.35 Annular plate loading diagram

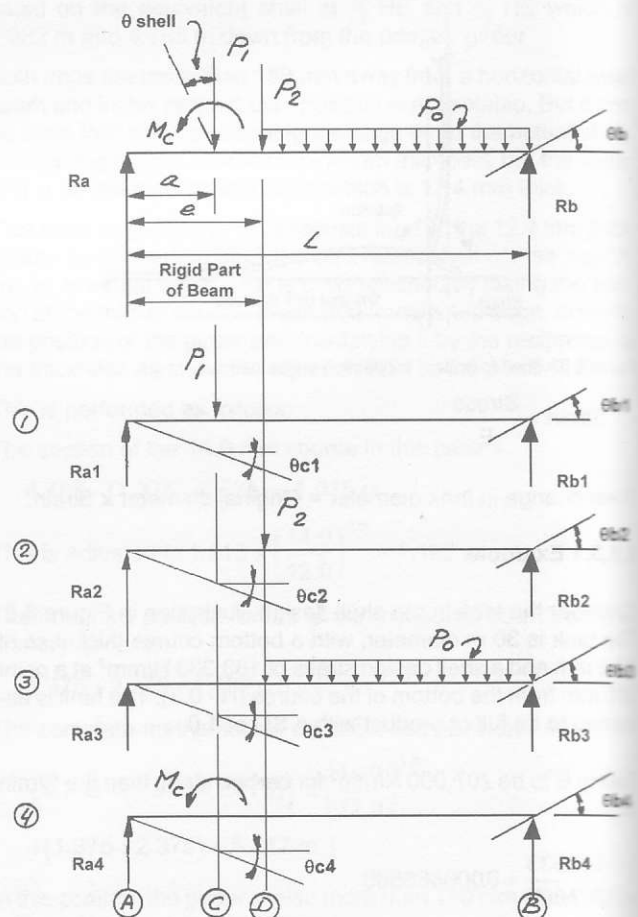


Figure 3.36 Superposition of loads

Example of a Tank bottom annular plate analysis using a "Excel" spreadsheet with the 'solver' method for evaluating the equations.

Tank diameter	* D =	98.43 feet =	30 m
Tank radius	R =	49.215 feet =	15 m
Design liquid level	* H =	629.952 inches =	16 m
Specific gravity of stored product	* SG =	0.9	
Thickness of bottom shell course	* Ts =	0.496 inches =	12.60 mm
Thickness of bottom annular plate	** Tb =	0.3150 inches =	8.00 mm
Leg length of shell-to-bottom fillet weld	* Fw =	0.3150 inches =	8.00 mm
Modulus of elasticity	* E =	29000000 lbs/inch ² =	200000 N / mm ²
Weight of shell + portion of roof supported by the shell	* P1 =	97.63 lbs/inch =	17.134 N / mm
Length of annular plate beam (found by iteration)	** L =	13.80185 inches =	350.567 mm
Design fatigue stress Sfat. <=75,000, o.k.	S.fat =	43893.79 lbs/inch ² =	302.716 N / mm ²
Characteristic length	λ =	0.075103 1/inch. =	0.002957 1 / mm
Moment of inertia of shell plate	Is =	0.011174 inch =	4651.117 mm
Moment of inertia of bottom annular plate	Ib =	0.002862 inch =	1191.362 mm
Unrestrained radial expansion at the bottom	y =	0.496442 inches. =	12.610 mm
Part length of bottom annular plate	a =	0.87796 inches. =	22.300 mm
Part length of bottom annular plate	e =	1.44092 inches. =	36.60 mm
Liquid pressure at the bottom	Po =	20.47342 lbs / inch ² =	0.1412 N / mm ²
Weight of shell + portion of roof supported by shell	P1 =	97.63 lbs / inch =	17.1342 N / mm
Liquid pressure on inside fillet weld	P2 =	6.44831 lbs / inch =	1.1317 N / mm
Moment in shell	Mc =	1075.127 in.-lbs/in. =	4784.442 mm.N/mm
Rotation of shell	θs =	-0.01438 radians	θs must = θc with opposite sign. OK
Rotation C1	θc1 =	-0.003413 radians	
Rotation C2	θc2 =	-0.00037 radians	
Rotation C3	θc3 =	-0.024643 radians	
Rotation C4	θc4 =	0.042807 radians	OK
Rotation at C	θc =	0.01438 radians	
Rotation at B1	θb1 =	0.002303 radians	θb = 0, OK
Rotation at B2	θb2 =	0.00025 radians	
Rotation at B3	θb3 =	0.026335 radians	
Rotation at B4	θb4 =	-0.028888 radians	
Rotation at B	θb =	0.00000 radians	θb = 0, OK
Reaction at A	Ra =	288.4169 lbs / inch =	50.61748
Reaction at B	Rb =	68.73185 lbs / inch =	12.06251
Moment in bottom annular at toe of inside fillet weld	Md =	714.5033 in.-lbs/in. =	3179.096 mm.N/mm
Horizontal force at bottom of shell	Q =	216.9575 lbs / inch =	38.07627 N / mm
Shear stress in fillet weld	σFw =	9745.074 lbs/inch ² =	67.20741 N / mm ²
Min. width of annular plate (inside shell to lap joint)	(L-e+Fw)=	12.67589 inches. =	321.9675 mm

The minimum width of the annular plate to API 650 cl. 3.5.2 is the greater of the length given by: $\frac{390.T_b}{(H.SG)^{1/2}}$, which is 17.87273 inches, or 24 inches

For this case the API 650 min. width is: 24 inches. = 600 mm

The API minimum requirement at 600 mm is very conservative in this case compared with all theoretical requirements to H. Kroon's theory of 322 mm.

* Manually inputted fixed data

** Manually inputted variable data

Figure 3.37 An example of H. Kroon's method for tank bottom annular plate analysis

Moment Mc the shell:

The equation for the rotation of the shell and moment Mc can be found in Hetényi's "Beams on Elastic Foundations", formula 22c, (Reference 3.5). The equation is as follows:

$$-Mc = 2(\lambda)(E)(I_s)(\lambda y_0 + \theta_0)$$

where:

$$\lambda = \sqrt[4]{\frac{3(1-\mu^2)}{R^2 T_s^2}}, \text{ for } \mu = 0.3 \rightarrow \lambda = \frac{1.2854}{\sqrt{R T_s}}$$

$$y_0 = \frac{Q \times L}{E \times T_b} - y$$

$$Y = \frac{P_o \times R^2}{E \times T_s} - \frac{(SG)(H-x)R^2}{E \times T_s}$$

$$I_s = \frac{T_s^3}{12(1-\mu^2)}, \text{ for } \mu = 0.3 \rightarrow I_s = \frac{T_s^3}{10.92}$$

$$\theta_0 = \frac{y}{H} - \theta_{\text{shell}}$$

$$\theta_{\text{shell}} = \frac{Mc}{2(\lambda)(E)(I_s)} - \lambda y + \frac{y}{H} + \frac{\lambda QL}{E \times T_b}$$

Note: The term (y/H) has been added to correct the equation for the triangular shape of the pressure diagram.

Rotation at point B:

(Found by the double integration method.)

$$\theta_{b1} = \frac{P_1}{6E(I_b)L^2} [aL(L^2 - a^2) + 2e^3a - 3e^2aL + a^3L]$$

$$\theta_{b2} = \frac{(P_2)e(L-e)}{6E(I_b)L^2} L^2 - 2e^2 + eL$$

$$\theta_{b3} = \frac{(P_o)(L-e)^2}{24E(I_b)L^2} (2eL^2 - 4e^3 + L^3 + e^2L)$$

$$\theta_{b4} = \frac{Mc}{6E(I_b)L^2} (-L^3 - 2e^3 + 3e^2L)$$

$$\theta_b = \theta_{b1} + \theta_{b2} + \theta_{b3} + \theta_{b4} = 0$$

The moment of inertia Ib for the annular plate is given as:

$$I_b = \frac{T_b^3}{12(1-\mu^2)}$$

$$\text{when } \mu = 0.3 \text{ then } I_b = \frac{T_b^3}{10.92}$$

The sum of the values $\theta_{b1} + \theta_{b2} + \theta_{b3} + \theta_{b4}$ is equated to zero, and by transposition of formulae the value Mc is found to be:

$$\begin{aligned} Mc = & \{-4(P_1)[aL(L^2 - a^2) + 2e^3a - 3e^2aL + a^3L] \\ & -4(P_2)e(L-e)(L^2 - 2e^2 + eL) \\ & -(P_o)(L-e)^2(2eL^2 - 4e^3 + L^3 + e^2L)\} \\ & \div [4(-L^3 - 2e^3 + 3e^2L)] \end{aligned}$$

Rotation at point C:

(Found by the double integration method.)

$$\begin{aligned} \theta_{c1} = & \frac{(P_1)}{24E(I_b)L^2} \{12Le^2(L-a) - 12L^2(e-a)^2 - 8e^3 \\ & (L-a) + 4L(e-a)^2(2e+a) - 4aL(L-a)(2L-a)\} \end{aligned}$$

$$\theta_{c2} = \frac{(P_2)}{24E(I_b)L^2} \{12Le^2(L-e) - 8e^3(L-e) - 4eL(L-e)(2L-e)\}$$

$$\theta_{c3} = \frac{(P_o)(L-e)^2}{24E(I_b)L^2} (7e^2L - 4e^3 - L^3 - 2eL^2)$$

$$\theta_{c4} = \frac{8Mc(L-e)^3}{24E(I_b)L^2}$$

$$\theta_c = \theta_{c1} + \theta_{c2} + \theta_{c3} + \theta_{c4} = -\theta_{\text{shell}}$$

where:

$$\theta_{\text{shell}} = \frac{Mc}{2(\lambda)(E)(I_s)} - \left(\lambda y - \frac{y}{H}\right) + \frac{\lambda QL}{E \times T_b}$$

Horizontal force at bottom of shell:

The horizontal force "Q" acting at the bottom of the shell is calculated by the substitution of values in the equation for " θ_{shell} " and the transposition of the equation which then gives:

$$Q = \frac{\left[\frac{2\lambda^2 R^2}{E \times T_s} Mc + y \right]}{\left[\frac{L}{E \times T_b} + \frac{2\lambda R^2}{E \times T_s} \right]}$$

Reaction force Ra:

The reaction force Ra acting at the outer end of the beam, can be calculated from:

$$Ra = \frac{\left\{ (P_1)(L-a) + (P_2)(L-e) + (P_o) \frac{(L-e)^2}{2} + Mc \right\}}{L}$$

Reaction force Rb:

The reaction force Rb acting at the inner end of the beam, can be calculated from:

$$Rb = (P_1) + (P_2) + (P_o)(L-e) - Ra$$

Moment Md:

The bending moment Md in the annular plate acting at the toe of the internal fillet weld, can be calculated from:

$$Md = Mc - (Ra)e + (P_1)(e-a)$$

Combined stress in annular plate:

Maximum combined stress due to moment Md and horizontal force Q is:

$$\sigma = \frac{Q}{(T_b)} + \frac{6(Md)}{(T_b)^2} \leq Sfat$$

Where Sfat is the design fatigue stress > 75,000 lbs/in².

Shear stress in fillet weld:

Maximum shear force acting on each fillet weld is:

$$w = \sqrt{\left(\frac{Mc}{T_s}\right)^2 + \left(\frac{Q}{2}\right)^2} \rightarrow \text{Shear stress } \tau = \frac{w}{0.7071 \times (FW)}$$

Solution of equations:

Figure 3.37, is an example of H. Kroon's theory where all the equations are solved using a "Excel" spreadsheet together with the "solver" function, which calculates the unknown variables for a given required target value, allowing also for any constraints which may apply.

In the following example, the thickness of the annular plate 'Tb' is targeted at 8 mm.

The variables are the fatigue stress 'Sfat' and the beam length 'L'.

The constraints are:

- 1) The rotation at the shell 'As' must be equal to, but opposite in sign to the rotation at point 'C' which is 'θc'.
- 2) The rotation at point 'B', which is 'θb', must be zero.

3.5.4 API 650

Up to now the British approach to tank shell design in accordance with BS 2654 has been discussed. As mentioned earlier the American Code API 650 differs from the British Code in certain aspects and these differences are now outlined.

3.5.4.1 General

The API 650 Code in its basic form is used for the design of tanks having (for fixed roof tanks) an internal pressure approximating to atmospheric pressure, or for a pressure not exceeding, that which equates to the weight of the roof plates. Unlike BS 2654 then, API 650 does not have the tank pressure categories (non-pressure, low-pressure and high-pressure). However, reference to Appendix F of the Code reveals that there are procedures for designing tanks with pressures up to $2\frac{1}{2}$ lbf/in² (172 mbar).

In Appendix F, the additional pressure in the space above the stored product is converted into an additional head of product and this is then added to the design head for use in computing the shell thickness.

The term 'H' in the following equations 3.34 and 3.35 then becomes

where:

- P = additional pressure (kPa) [1 kPa = 10 mbar]
G = design specific gravity

The effect of this additional pressure on the design of the roof-to-shell compression zone is dealt with later in Section 3.8.

As in BS 2654 there is no provision in API 650 for designing for an internal vacuum condition, but tanks which meet the minimum requirements of the Code are considered capable of withstanding a partial vacuum of one inch head of water gauge ($2\frac{1}{2}$ mbar).

With regard to temperature limitations, API 650 applies only to tanks in non-refrigerated service that have a maximum operating temperature of 90°C (200°F). However there is provision in Appendix M of the Code, which allows tanks to be designed up to a maximum temperature of 260°C (500°F).

This Appendix gives guidance on the design of fixed roof tanks for operating temperatures above 90°C (200°F) but not exceeding 260°C (500°F).

Through the use of a table of yield strength reduction factors for three bands of material yield strengths, against four temperature ranges, the Appendix shows how the allowable stress levels are reduced for the various parts of the tank. The Appendix also recognises the need to consider the effect of liquid head and temperature cycles on the shell-to-bottom joint and gives a procedure for dealing with these aspects.

3.5.4.2 Shell design stresses

API 650 has a different approach in setting allowable shell design stresses, in that, unlike BS 2654, which uses $2/3$ of the material minimum yield stress for the allowable design stress, API 650 considers both the yield and the ultimate tensile stress of the chosen shell material and uses two formulas for determining the final design shell thickness.

The allowable design stresses are defined as:

Sd, which is used in one shell thickness formula, based on the working parameters of the tank, including any corrosion allowance, which is required to be added to the computed thickness.

Unlike BS 2654, API 650 tanks are designed for a product specific gravity (SG), which is specified by the tank purchaser. The drawback to this philosophy is that the tank should not be used for storing products with higher SGs, unless a lower maximum filling height is first calculated. It is therefore very important for the tank owner to keep all tank design records on hand in order to obviate a tank being inadvertently over-stressed.

St, is used in the other shell thickness formula based on the hydrostatic testing of the tank and in this case the corrosion allowance is excluded from the formula.

For any chosen shell material:

Sd is found to be the lesser of $\frac{2}{3}$ of the minimum yield stress, or $\frac{2}{5}$ of the ultimate tensile stress.

St is found to be the lesser of $\frac{3}{4}$ of the minimum yield stress, or $\frac{3}{7}$ of the ultimate tensile stress.

For convenience the API 650 Code includes the stress values for a popular range of steels in Table 3-2 which is reproduced in Figure 3.38.

The two shell design formulas are derived using exactly the same principles as the BS 2654 formula but they are simplified because there is no internal pressure to consider in the tank vapour space.

Referring to equation 3.7 and ignoring the term p and combining the constants 98 and 20, the design shell course thickness in mm is given as:

$$td = \frac{4.9 \cdot D(H-0.3) \cdot G}{Sd} + CA \quad \text{equ 3.34}$$

And the hydrostatic test shell thickness in mm is given as:

$$tt = \frac{4.9 \cdot D(H-0.3)}{St} \quad \text{equ 3.35}$$

The above equations are given in API 650 together with their equivalents in US customary Imperial units (feet, inches and lbs/in²), as below:

$$td = \frac{2.6 \cdot D(H-1) \cdot G}{Sd} + CA$$

$$tt = \frac{2.6 \cdot D(H-1)}{St}$$

where:

- td = design shell thickness, in mm (inches)
tt = hydrostatic test shell thickness, in mm (inches)
D = nominal tank diameter, in m (feet)
H = height from bottom of course under consideration to the top of the shell, including the top angle, if any, to the bottom of any overflow that limits the tank filling height; or to any other level specified by the purchaser, restricted by an internal floating roof, or controlled to allow for seismic wave action, in m (feet)
G = design specific gravity of the liquid to be stored, as specified by the purchaser
CA = corrosion allowance, in mm (inches), as specified by the purchaser
Sd = allowable stress for the design condition, in N/mm² (lbs/in²)

Plate Specification	Grade	Minimum Yield Strength MPa (psi)	Minimum Tensile Strength MPa (psi)	Product Design Stress S_d MPa (psi)	Hydrostatic Test Stress S_t MPa (psi)
ASTM Specifications					
A 283M	C	205 (30,000)	380 (55,000)	137 (20,000)	154 (22,500)
A 285M	C	205 (30,000)	380 (55,000)	137 (20,000)	154 (22,500)
A 131M	A, B, CS	235 (34,000)	400 (58,000)	157 (22,700)	171 (24,900)
A 36M	—	250 (36,000)	400 (58,000)	160 (23,200)	171 (24,900)
A 131M	EH 36	360 (51,000)	490 ^a (71,000 ^a)	196 (28,400)	210 (30,400)
A 573M	400	220 (32,000)	400 (58,000)	147 (21,300)	165 (24,000)
A 573M	450	240 (35,000)	450 (65,000)	160 (23,300)	180 (26,300)
A 573M	485	290 (42,000)	485 ^a (70,000 ^a)	193 (28,000)	208 (30,000)
A 516M	380	205 (30,000)	380 (55,000)	137 (20,000)	154 (22,500)
A 516M	415	220 (32,000)	415 (60,000)	147 (21,300)	165 (24,000)
A 516M	450	240 (35,000)	450 (65,000)	160 (23,300)	180 (26,300)
A 516M	485	260 (38,000)	485 (70,000)	173 (25,300)	195 (28,500)
A 662M	B	275 (40,000)	450 (65,000)	180 (26,000)	193 (27,900)
A 662M	C	295 (43,000)	485 ^a (70,000 ^a)	194 (28,000)	208 (30,000)
A 537M	1	345 (50,000)	485 ^a (70,000 ^a)	194 (28,000)	208 (30,000)
A 537M	2	415 (60,000)	550 ^a (80,000 ^a)	220 (32,000)	236 (34,300)
A 633M	C, D	345 (50,000)	485 ^a (70,000 ^a)	194 (28,000)	208 (30,000)
A 678M	A	345 (50,000)	485 ^a (70,000 ^a)	194 (28,000)	208 (30,000)
A 678M	B	415 (60,000)	550 ^a (80,000 ^a)	220 (32,000)	236 (34,300)
A 737M	B	345 (50,000)	485 ^a (70,000 ^a)	194 (28,000)	208 (30,000)
A 841M	—	345 (50,000)	485 ^a (70,000 ^a)	194 (28,000)	208 (30,000)
CSA Specifications					
G40.21M	260W	260 (37,700)	410 (59,500)	164 (23,800)	176 (25,500)
G40.21M	300W	300 (43,500)	450 (65,300)	180 (26,100)	193 (28,000)
G40.21M	350WT	350 (50,800)	480 ^a (69,600 ^a)	192 (27,900)	206 (29,800)
G40.21M	350W	350 (50,800)	450 (65,300)	180 (26,100)	193 (28,000)
National Standards					
	235	235 (34,000)	365 (52,600)	137 (20,000)	154 (22,500)
	250	250 (36,000)	400 (58,300)	157 (22,700)	171 (25,000)
	275	275 (40,000)	430 (62,600)	172 (25,000)	184 (26,800)
ISO 630					
E 275	C, D	265 (38,400)	425 (61,900)	170 (24,700)	182 (26,500)
E 355	C, D	345 (50,000)	490 ^a (71,000 ^a)	196 (28,400)	210 (30,400)

^aBy agreement between the purchaser and the manufacturer the tensile strength of these materials may be increased to 515 MPa (75,000 psi) minimum and 620 MPa (90,000 psi) maximum [and to 585 MPa (85,000 psi) minimum and 690 MPa (100,000 psi) maximum for ASTM A 537M, Class 2, and A 678M, Grade B]. When this is done, the allowable stresses shall be determined as stated in 3.6.2.1 and 3.6.2.2.

Figure 3.38 Stress values for a popular range of steels
From API 650, table 3-2

St = allowable stress for the hydrostatic test condition, in N/mm² (lbs/in²)

As is the case in BS 2654, API 650 also stipulates that the nominal diameter shall be taken as the centreline diameter of the bottom shell course plates, unless otherwise specified by the purchaser.

An exception to this rule may be requested when ordering a tank, which is to have a floating roof, as it can be considered preferable to have a shell with a smooth internal surface for the roof seal to act against. For these tanks, the diameter may be measured to the inside surface of each course of shell plating, thus avoiding steps between adjacent courses.

However, the "one-foot" method in the API 650 Code can only be used for designing tank shells up to 60m in diameter. Larger tanks have to be designed using an alternative method known as the "variable design point" method, which is described in Section 3.6.

3.5.4.3 Use of shell design formulae

The use of the shell design formulae can be demonstrated as follows, using the fixed roof tank depicted earlier in the tank shell design illustration in Figure 3.8, constructed in steel specification BS EN 10025 S275.

From Figure 3.38, under the heading "National Standards", the Grade 275 Steel has a minimum yield strength of 275 N/mm² and a minimum tensile strength of 430 N/mm²

The product design stress is the lesser of $\frac{2}{3} \times 275 = 183.333$ N/mm² and $\frac{2}{5} \times 430 = 172$ N/mm², in this case 172 N/mm²

The hydrostatic test stress is the lesser of $\frac{3}{4} \times 275 = 206.25$ N/mm² and $\frac{3}{7} \times 430 = 184.29$ N/mm², in this case 184.29 N/mm²

The tank is 30 m diameter and 16 m high, in eight equal width courses.

The stored product has a specific gravity (SG) of 0.9.

The course thickness is determined using equations 3.34 and 3.35 as follows:

$$td = \frac{4.9 \cdot D(H-0.3) \cdot G}{Sd} + CA$$

$$tt = \frac{4.9 \cdot D(H-0.3)}{St}$$

For the bottom course:

$$td = \frac{4.9 \cdot 30(16-0.3) \cdot 0.9}{172} + 0 = 12.08 \text{ mm}$$

$$tt = \frac{4.9 \cdot 30(16-0.3)}{184} = 12.54 \text{ mm}$$

The greater of these two values is taken to be the thickness for the bottom course i.e. 12.54 mm.

The calculation can be tabulated as follows:

Course No	h (m)	(H-0.3) (M)	td (mm)	tt (mm)	Min t (mm)	Rounded t (mm)
1 (Btm)	2	15.7	12.08	12.54	12.54	12.6
2	2	13.7	10.54	10.95	10.95	11.0
3	2	11.7	9.00	9.35	9.35	9.4
4	2	9.7	7.46	7.75	7.75	7.8
5	2	7.7	5.92	6.15	6.15	6.2
6	2	5.7	4.38	4.55	4.55	4.6
7	2	3.7	2.85	2.96	2.96	3.0
8	2	1.7	1.31	1.36	1.36	1.4

3.5.4.4 Shell plate thicknesses

Similarly as for BS 2654, API 650 also specifies minimum allowable shell plate thickness for the "as constructed" tank and these are given in the table below.

The API 650 Code quotes Imperial and metric equivalents throughout its text but only the metric terms are given here.

Nominal tank diameter (m)	Minimum allowable shell plate thickness (mm)
< 15	5
15 to < 36	6
36 to 60	8
> 60	10

Then for the shell design above the minimum course thickness for the 30 m diameter tank is 6 mm and therefore the minimum final course thickness will be:

12.6, 11.0, 9.4, 7.8, 6.2, 6.0, 6.0 and 6.0 mm.

The comparable shell thicknesses for the tank designed to BS 2654 (Tank shell design illustration in Figure 3.8) were found to be:

12.6, 11.0, 9.4, 8.0, 8.0, 8.0, 8.0 and 8.0 mm.

For this particular tank, the only significant difference being in the minimum allowable shell plate thicknesses, this being 6 mm for the API Code and 8 mm for the BS Code.

Comparison between the above table and Figure 3.7 for BS 2654 shows that the American Code is not quite so stringent as the British Code as is demonstrated below:

Nominal tank diameter D (m)		Minimum allowable shell plate thickness t (mm)
BS 26554	API 650	
< 15	< 15	5
15 to < 30	15 to < 36	6
30 to < 60	36 to < 60	8
60 to < 75	> 60	10
75 to < 100		12
> 100		14

Whereas the American Code allows a minimum shell plate thickness of 6 for tanks up to 36 m in diameter, the British Code limits the diameter for this thickness to under 30 m.

Also the American Code allows all tanks above 60 m in diameter to have a minimum thickness of 10 mm. The British Code specifies a further two size categories having minimum thicknesses of 12 mm and 14 mm.

The maximum shell thickness allowed in the American Code is 45 mm, which is more than the 40 mm maximum in the British Code.

3.5.4.5 Choosing BS or API shell thickness design methods

The logical question which comes to mind when considering the BS and API methods for shell thicknesses is - which one is most advantageous from a commercial point of view? i.e. which gives the thinner shell for a given material?

This question is not easily answered, because of the effect of the following variables in the equations;

- Specific Gravity (SG) of the stored product.
- Any corrosion allowance (CA) which might be required.
- The varying ratio of minimum yield strength to minimum tensile strength of the range of steels used for the design of shells.

For SG = 1.0 and CA = 1mm.

Shell thicknesses in (mm) :-

Courses	A.P.I. Values. 'td' 'tt'		Final API thickness	Based on:	B.S. thickness	Thickest result.
Btm.	15.74	13.47	15.74	Sd	15.74	Same
2	13.86	11.75	13.86	Sd	13.86	Same
3	11.98	10.04	11.98	Sd	11.98	Same
4	10.11	8.32	10.11	Sd	10.11	Same
5	8.23	6.61	8.23	Sd	8.23	Same
6	6.35	4.89	6.35	Sd	6.35	Same
7	4.48	3.18	4.48	Sd	4.48	Same
8	2.6	1.46	2.6	Sd	2.6	Same

Min. thks. Allowed by code :-

6 mm

8 mm

For SG = 1.5 and CA = 1mm.

Shell thicknesses in (mm) :-

Courses	A.P.I. Values. 'td' 'tt'		Final API thickness	Based on:	B.S. thickness	Thickest result.
Btm.	23.1	13.47	23.1	Sd	23.1	Same
2	20.29	11.75	20.29	Sd	20.29	Same
3	17.47	10.04	17.47	Sd	17.47	Same
4	14.66	8.32	14.66	Sd	14.66	Same
5	11.84	6.61	11.84	Sd	11.84	Same
6	9.03	4.89	9.03	Sd	9.03	Same
7	6.21	3.18	6.21	Sd	6.21	Same
8	3.4	1.46	3.4	Sd	3.4	Same

Min. thks. Allowed by code :-

6 mm

8 mm

For SG = 0.8 and CA = 1mm.

Shell thicknesses in (mm) :-

Courses	A.P.I. Values. 'td' 'tt'		Final API thickness	Based on:	B.S. thickness	Thickest result.
Btm.	12.79	13.47	13.47	St	15.74	BS
2	11.29	11.75	11.75	St	13.86	BS
3	9.79	10.04	10.04	St	11.98	BS
4	8.29	8.32	8.32	St	10.11	BS
5	6.78	6.61	6.78	Sd	8.23	BS
6	5.28	4.89	5.28	Sd	6.35	BS
7	3.78	3.18	3.78	Sd	4.48	BS
8	2.28	1.46	2.28	Sd	2.6	BS

Min. thks. Allowed by code :-

6 mm

8 mm

Figure 3.39 Calculation of comparison of BS and API shells - page 1

For SG = 1.0 and CA = nil.

Shell thicknesses in (mm) :-

Courses	A.P.I. Values.		Final API thickness	Based on:	B.S. thickness	Thickest result.
	'td'	'tt'				
Btm.	14.74	13.47	14.74	Sd	14.74	Same
2	12.86	11.75	12.86	Sd	12.86	Same
3	10.98	10.04	10.98	Sd	10.98	Same
4	9.11	8.32	9.11	Sd	9.11	Same
5	7.23	6.61	7.23	Sd	7.23	Same
6	5.35	4.89	5.35	Sd	5.35	Same
7	3.48	3.18	3.48	Sd	3.48	Same
8	1.6	1.46	1.6	Sd	1.6	Same

Min. thks. Allowed by code :-

6 mm

8 mm

For SG = 1.5 and CA = nil.

Shell thicknesses in (mm) :-

Courses	A.P.I. Values.		Final API thickness	Based on:	B.S. thickness	Thickest result.
	'td'	'tt'				
Btm.	22.1	13.47	22.1	Sd	22.1	Same
2	19.29	11.75	19.29	Sd	19.29	Same
3	16.47	10.04	16.47	Sd	16.47	Same
4	13.66	8.32	13.66	Sd	13.66	Same
5	10.84	6.61	10.84	Sd	10.84	Same
6	8.03	4.89	8.03	Sd	8.03	Same
7	5.21	3.18	5.21	Sd	5.21	Same
8	2.4	1.46	2.4	Sd	2.4	Same

Min. thks. Allowed by code :-

6 mm

8 mm

For SG = 0.8 and CA = nil.

Shell thicknesses in (mm) :-

Courses	A.P.I. Values.		Final API thickness	Based on:	B.S. thickness	Thickest result.
	'td'	'tt'				
Btm.	11.79	13.47	13.47	St	14.74	BS
2	10.29	11.75	11.75	St	12.86	BS
3	8.79	10.04	10.04	St	10.98	BS
4	7.29	8.32	8.32	St	9.11	BS
5	5.78	6.61	6.61	St	7.23	BS
6	4.28	4.89	4.89	St	5.35	BS
7	2.78	3.18	3.18	St	3.48	BS
8	1.28	1.46	1.46	St	1.6	BS

Min. thks. Allowed by code :-

6 mm

8 mm

Figure 3.39 Calculation of comparison of BS and API shells - page 2

The many differing strength ratios which apply to the last variable factor, when taken in conjunction with varying SGs and CAs, make a generalised conclusion difficult.

However, it is found that comparisons can be made based on the premise that if the minimum tensile strength is taken hypothetically to be 166.66% or more, of the minimum yield strength, for a given material, then the allowable design stress 'Sd' for the API equation 3.34 and 'S' for the BS equation 3.7 will have the same value and these will determine the shell thicknesses as 'St', by definition will always be greater than 'Sd' or 'S'.

Then under these conditions the following is found for various combinations of SG and CA:

When SG = 1.0 and CA = 0
then BS & API thicknesses are equal.

When SG > 1.0 and CA = 0
then BS & API thicknesses are equal.

When SG < 1.0 and CA = 0
then the BS thickness is > than the API thickness.

When SG = 1.0 and CA > 0
then the BS & API thicknesses are equal.

When SG > 1.0 and CA > 0
then the BS & API thicknesses are equal.

When SG < 1.0 and CA > 0
then the BS thickness is > than the API thickness.

3.5.4.6 Worked examples

The following worked examples demonstrate the validity of the above statements:

Taking the 30 m diameter x 16 m high tank used in earlier examples, which has 8 x 2 m wide shell courses, and using the steel specification ASTM A131 Gr. B which has a minimum yield strength of 235 N/mm² and a minimum tensile strength of 400 N/mm².

The ratio of UTS/Yield = 170.213%.

This is more than 166.66% and therefore satisfies the requirements for this exercise.

Sd = 156.667 N/mm², St = 171.429 N/mm²

and for the BS Code, S = 156.667 N/mm².

Then taking each of the six above conditions in turn, a set of results are obtained which are presented in Figure 3.39.

3.6 The "variable design point" method

One very significant difference between the British and American Codes, is the alternative shell design method to the "one-foot" method which is included in the American Code. This method is called the "variable design point" method.

The American Code specifies that this method may only be used when the purchaser has not specified that the "one-foot" method be used and when the following is true:

$$\frac{L}{H} \leq \frac{1000}{6} \quad \text{equ 3.36}$$

where:

$$L = (500 \cdot D \cdot t)^{1/2} \text{ (mm)}$$

$$D = \text{tank diameter (mm)}$$

$$t = \text{bottom-course shell thickness (mm)}$$

$$H = \text{maximum design liquid level (m)}$$

The above condition is found to be satisfied for most tank sizes with the possible exception of certain tanks, which have large diameter to height ratios.

Also the Code specifies that this method must be used for tanks larger than 60 m in diameter.

3.6.1 "Variable design point" method development

The "variable design point" method normally provides a reduction in shell course thicknesses and total material weight, but more important is its potential to permit construction of larger diameter tanks within the maximum plate thickness limitation.

The following work, developed by the late Professor A.S. Tooth, Professor of Mechanical Engineering, University of Strathclyde, Glasgow, explains how the method evolved.

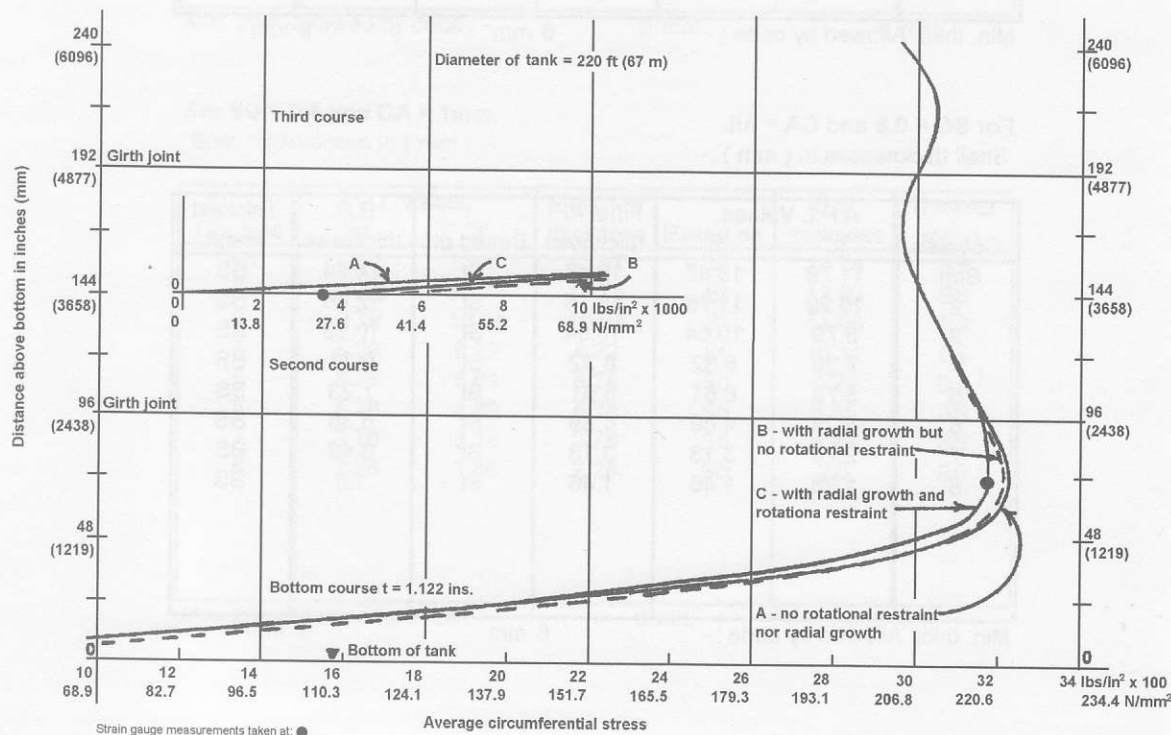


Figure 3.40 Distribution of circumferential stresses in a tank 220 ft (67 m) diameter and 56 ft (17.1 m) high with different base boundary conditions

Zick and McGrath analysed a number of large tanks, which were designed using the "one-foot" method. The analysis used is based on a computer program developed by Kalnins. In this, the basic shell equations are solved by a step-by-step integration method. A number of comparisons are made to examine the influence of different base restraints and of different allowable design stresses and tank size.

Because the theory was formulated some time ago when the American tank Code was written using Imperial units, the theory is similarly in the same units. However, the equations of the worked example at the end of the Section have been converted into the now more acceptable metric units.

Figure 3.40 provides a plot showing the distribution of the circumferential stress in a tank 220 ft (67 m) diameter and 56 ft (17.1 m) high, for three different restraints:

A = no rotational restraint and no radial growth i.e. a hinge.

B = allows radial growth but no rotational restraint.

C = allows radial growth but with rotational restraint.

It is noted that the differences in these three cases are small, away from the edge. The two strain gauge values presented give a measure of confidence in the analytical method, showing close agreement with curve C.

Figure 3.41 provides results of the analysis for the same tank as in Figure 3.40 but with three different values of allowable stress, namely: 17,850, 23,000, and 30,000 lbf/in². The tanks are designed using the API "one-foot" method.

The variation in the stress levels is noted. The maximum stress in the bottom courses is reasonably close to the design stress and in upper courses is less than the design stress. It is also noted that the location of the maximum stress at each course junction occurs at approximately one foot, or higher, above the junction.

The final comparison, shown in Figure 3.42 is for two different sizes of tanks:

1) 280 ft (85.3 m) diameter and 64 ft (19.5 m) high.

2) 120 ft (36.6 m) diameter and 48 ft (14.6 m) high.

The effects are similar to Figure 3.41, though the smaller tank is nearer to the design stress.

From the plots contained in Figures 3.40 and 3.41 it is clear that there is some variation in the magnitude of the actual circumferential stress in different courses of the tank. The bottom course and occasionally the second course are the most highly stressed. This is unfortunate since the bottom course is usually considered to be the most vulnerable course in the tank. In addition it may have piping attached, resulting in the possibility of a thrust and/or bending moment, being superimposed on the shell. It would therefore be desirable, if the design procedure used produced a shell which, when subject to the hydrostatic loading, had a stress in the bottom course which was lower than, or of similar magnitude to that of the upper courses.

Assuming that most designers would prefer the maximum stress in each shell course to be the same value, an alternative approach to calculate approximate plate thicknesses was proposed by Zick and McGrath in 1968. It was later incorporated into API 650, but not into BS 2654.

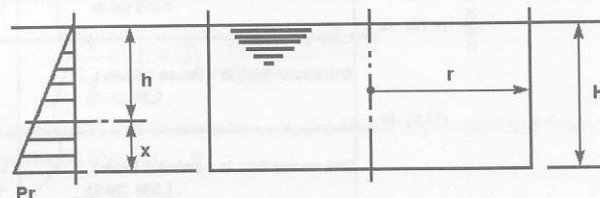
In this, the location of the "design point" on each shell course, at which the hydrostatic pressure is to be considered can be obtained from the radial and rotational movement of the plate edges at each joint. The movements are those caused by the self-equilibrating forces and moments and by the hydrostatic pressure. The aim is to find the point in the shell course called the "design point", where the stresses are close to the maximum for that course.

3.6.2 The bottom shell course

To explain the "variable design point" method, starting with the bottom shell course, it is assumed that the junction of the vertical shell and base connection is "pin-jointed" - that is, there is no rotational restraint and no radial growth allowed at the base junction. The fixing moment is thus zero and a horizontal force Q is required to sustain the no radial growth condition.

The value of this force can be obtained from shell analysis, the procedure being as follows:

The hydraulic head produces a linear variation of the radial pressure in the vessel. This is maximum at the base and zero at the liquid level. The value of this pressure is γh , where γ is the specific weight of the liquid in N/m³ and h is the height of fluid.



The circumferential stress σ_θ due to the hydraulic head is:

$$\sigma_\theta = \frac{\gamma r h}{t_1} = \frac{\gamma r}{t_1} (H - x) \quad \text{equ 3.37}$$

The free radial displacement of the cylinder at any height x , depends upon the values of the circumferential stress σ_θ and axial stress σ_x

$$w = \frac{r}{E} (\sigma_\theta - \nu \sigma_x)$$

where

ν = Poisson's ratio

For this treatment the axial stress is ignored. Thus the free base radial displacement from equation 3.37 is:

$$w = \frac{r}{E} \sigma_\theta = \frac{\gamma H r^2}{E t_1} \quad \text{equ 3.38}$$

To restrain this radial growth to zero, the bottom plate weld must exert a horizontal force Q per unit length of circumference in the inward direction. The deflection at the cylinder end due to Q is given by:

$$w = \frac{Q r^3}{2 K \beta^3} \quad \text{equ 3.39}$$

where:

$$K = E t_1^3 / [12(1 - \nu^2)]$$

and

$$\beta^4 = 3(1 - \nu^2) \left(\frac{r}{t_1} \right)^2$$

The membrane displacement (equation 3.38) and the edge bending displacement (equation 3.39) must be equal.

$$\therefore \frac{Q r^3}{2 K \beta^3} = \frac{\gamma H r^2}{E t_1}$$

$$\therefore Q = \frac{\gamma H r}{2 \beta} \quad \text{equ 3.40}$$

This force produces a mid-surface circumferential stress. At a location x from the cylinder end this is:

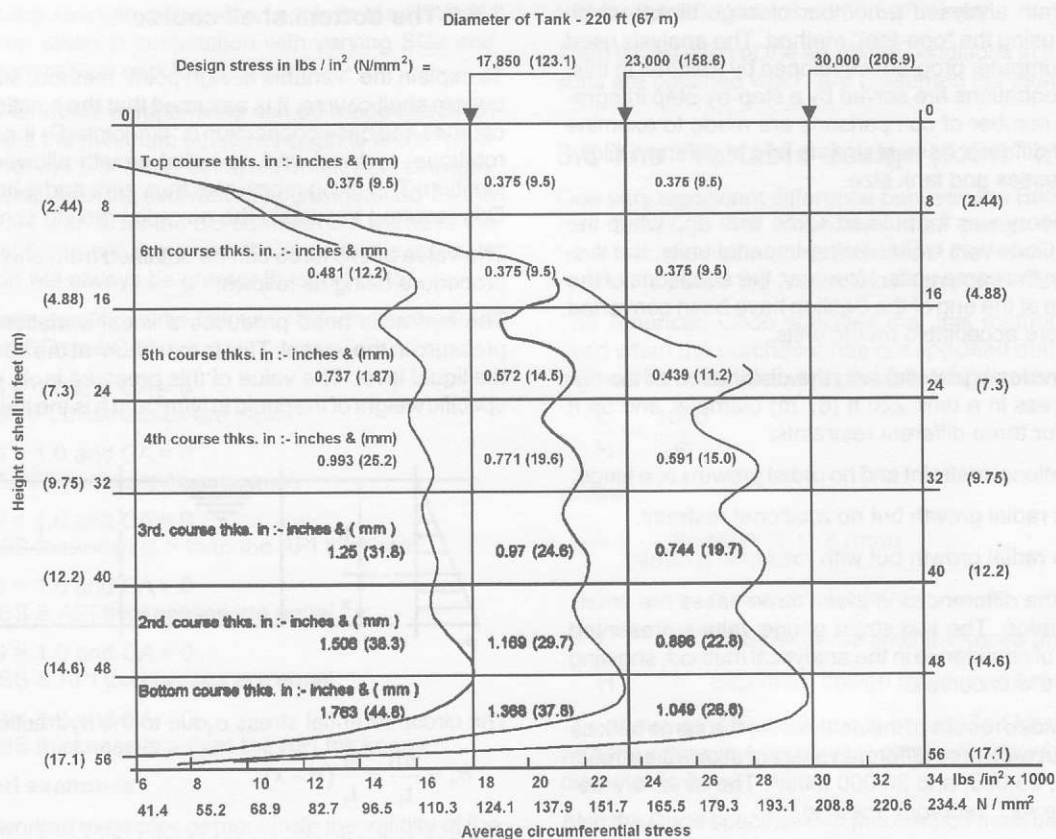


Figure 3.41 Actual stresses by analysis in a tank designed by the "one-foot" method, with API stress limits

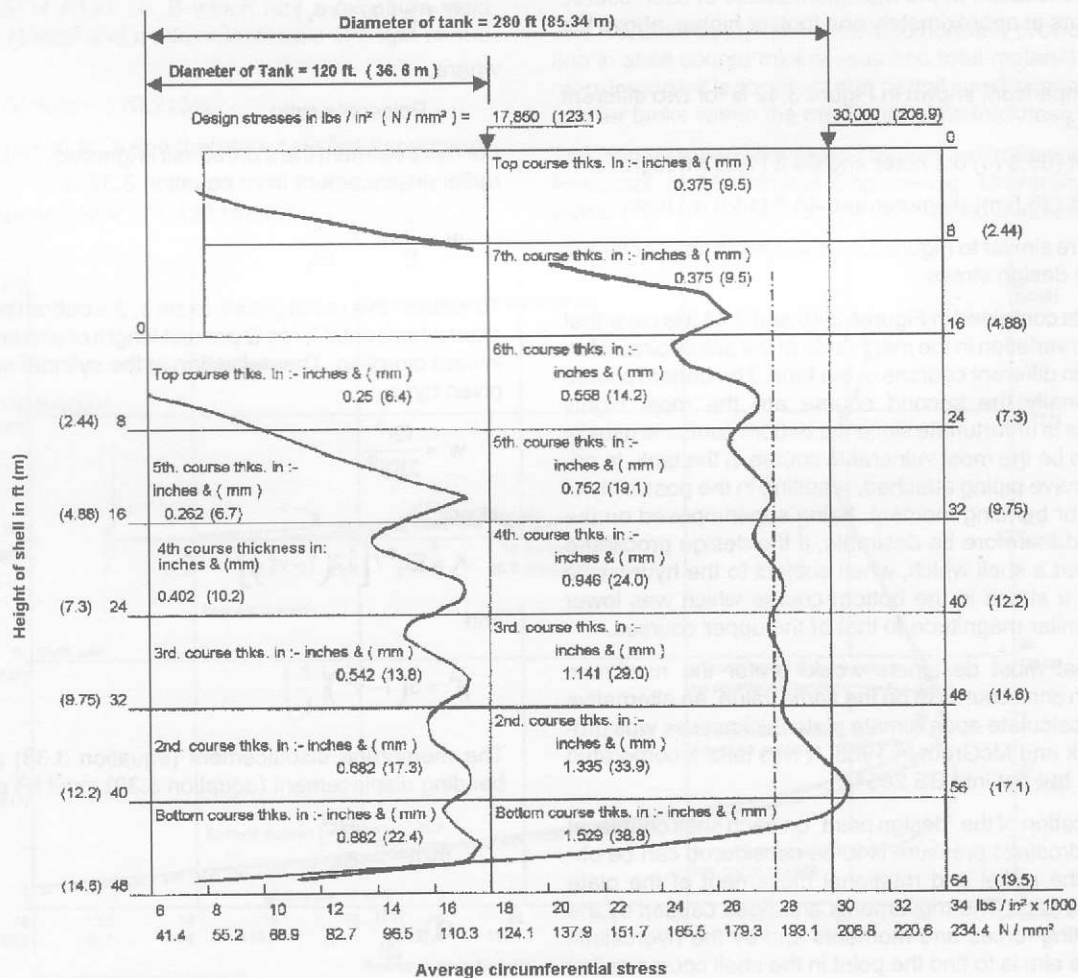


Figure 3.42 Actual stresses by analysis in smaller tanks designed by the "one-foot" method, with API stress limits

$$\sigma_\theta = -2 \frac{Q\beta}{t_1} e^{-\beta x/r} \cos \frac{\beta x}{r}$$

From equation 3.40

$$\sigma_\theta = \frac{\gamma H r}{t_1} e^{-\beta x/r} \cos \frac{\beta x}{r} \quad \text{equ 3.41}$$

The total mid-surface circumferential stress at any location x from the end is given by combining equations 3.37 and 3.41

$$\sigma_\theta = -\gamma \frac{Hr}{t_1} e^{-\beta x/r} \cos \frac{\beta x}{r} + \gamma \frac{r}{t_1} (H - x) \quad \text{equ 3.42}$$

To illustrate the behaviour of equations 3.37, 3.41 and 3.42 a specific example is considered:

In this the tank diameter $D = 76$ m, the height $H = 25$ m and the thickness of the bottom course $t_1 = 40$ mm.

The distribution of the circumferential mid-surface stress in a tank in this case full of water is shown in Figure 3.43 for the three equations. The following points are worthy of note.

1. The stress due to the edge bending (equation 3.41) is compressive at the base and dies away fairly rapidly reaching a turning value at a height of $1.83 \sqrt{rt_1} = 2260$ mm from the base.
2. The stress due to the hydraulic head (equation 3.37) is tensile and linear.
3. The combined stress (equation 3.42) is tensile and has a maximum at a height of 2040 mm. When the edge bending and hydraulic head stresses are combined the position of the maximum stress is always less than $1.83 \sqrt{rt_1}$ as shown by the plot of Figure 3.43 (in metric units).

The value of the height x at which the maximum occurs, depends on the geometry of the tank. The value used in equation 3.42 to derive the equation presented in API 650 (that is equation 3.47) is uncertain to the author.

One can but surmise that a number of actual tanks were analysed using the exact shell theory and an average value ob-

tained. The value which provides the numerical value given in API 650 is a height of x equal to $1.4949 \sqrt{rt_1}$, which gives the following equation:

$$\beta \frac{x}{r} = 4 \sqrt{3(1-\nu^2)} \sqrt{\frac{r}{t_1}} x \frac{1.4949 \sqrt{rt_1}}{r} = 1.9216$$

Substituting this into equation 3.42:

$$\sigma_\theta = \frac{\gamma H r}{t_1} e^{-1.9216} \cos(1.9216) + \frac{\gamma H r}{t_1} - \frac{\gamma r}{t_1} x 1.4949 \sqrt{rt_1}$$

$$\sigma_\theta = \frac{\gamma H r}{t_1} 1.0503 - \frac{\gamma H r}{t_1} \frac{1.4949 \sqrt{rt_1}}{H}$$

Putting $\sigma_\theta = S_d = \text{Allowable design stress}$ and rearranging:

$$t_1 = \frac{\gamma H r}{S_d} (1.0503) - \frac{\gamma H r}{S_d} \frac{1.4949 \sqrt{rt_1}}{H}$$

Noting that the thickness t_0 is the thickness obtained from the hydraulic loading, i.e.

$$t_0 = \frac{\gamma H D}{2 S_d} = \frac{\gamma H r}{S_d} \quad \text{equ 3.43}$$

$$\therefore \frac{t_1}{t_0} = 1.0503 - \frac{1.4949}{H} \sqrt{rt_0} \cdot \frac{t_1}{t_0}$$

From equation 3.43:

$$\frac{t_1}{t_0} = 1.0503 - 1.4949 \sqrt{\frac{\gamma r^2}{S_d H} \left(\frac{t_1}{t_0} \right)}$$

Substituting the nomenclature and dimensions of API 650:

$$\frac{t_1}{t_0} = 1.0503 - 1.4949 \sqrt{\frac{62.4 G (6D)^2}{12^3 S_d 12H} \sqrt{\frac{t_1}{t_0}}}$$

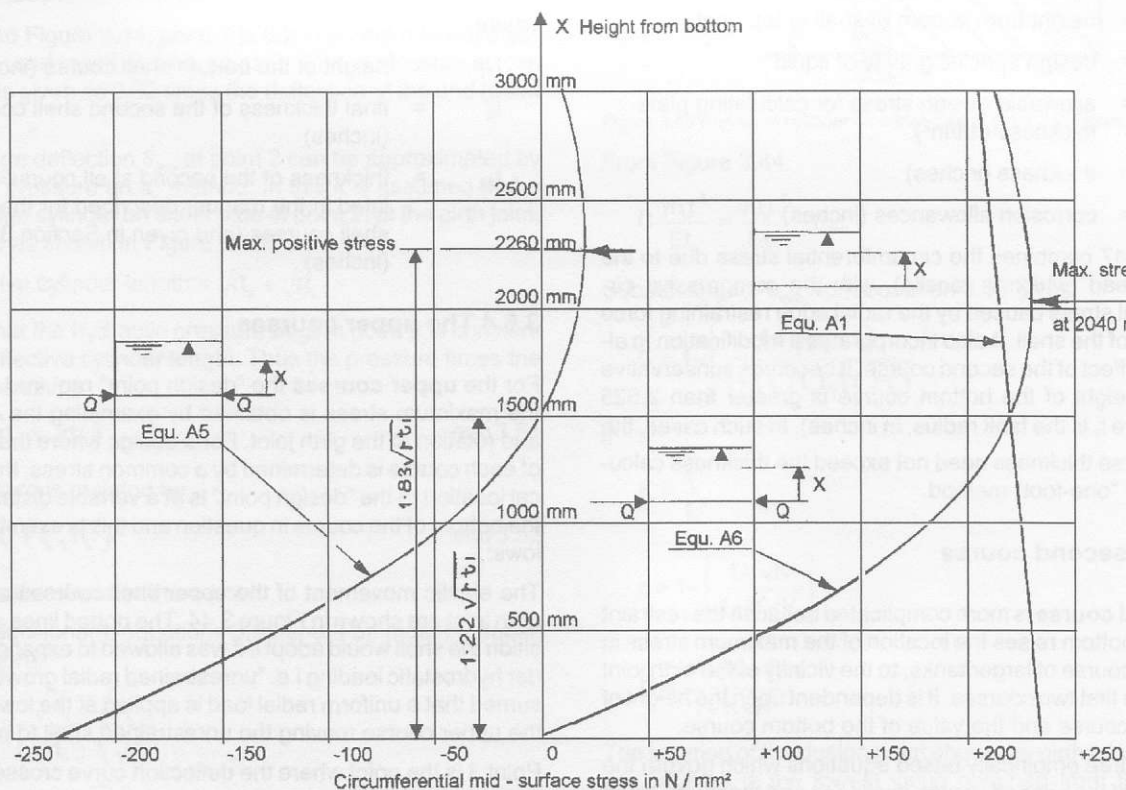


Figure 3.43 The variation of circumferential mid-surface stress in a tank, 76 m diameter and 25 m high, with a bottom course thickness of 40 mm

$$\frac{t_1}{t_0} = 1.0503 - 1.4949 \times 0.329 \sqrt{\frac{GD^2}{HS_d}} \sqrt{\frac{t_1}{t_0}}$$

$$\frac{t_1}{t_0} = 1.0503 - 0.4918 \frac{D}{H} \sqrt{\frac{HG}{S_d}} \sqrt{\frac{t_1}{t_0}} \quad \text{equ 3.44}$$

This is a quadratic equation in t_1/t_0 . It could have been used in this form in the Standard. However, it was simplified into a linear form. It would appear that this was done by examining a number of vessels of different diameters, heights and allowable stress design values and solving the quadratic equation 3.44 exactly. If this is done it is found that the (t_1/t_0) values are in the range of 1 to 0.87.

It would appear that the lowest value was taken, one presumes for conservatism.

Putting this in equation 3.44:

$$\frac{t_1}{t_0} = 1.0503 - 0.4918 \frac{D}{H} \sqrt{\frac{HG}{S_d}} \sqrt{0.87}$$

$$\frac{t_1}{t_0} = 1.0503 - 0.458 \frac{D}{H} \sqrt{\frac{HG}{S_d}} \quad \text{equ 3.45}$$

A further factor of 1.01 was introduced to equation 3.45 to compensate for a possible loss due to a thinner second course:

$$\frac{t_1}{t_0} = 1.061 - 0.463 \frac{D}{H} \sqrt{\frac{HG}{S_d}} \quad \text{equ 3.46}$$

Putting $t_0 = 2.6 \cdot H \cdot D \cdot \frac{G}{S_d}$ a modified form of the previous basic equation 3.46 is obtained as follows:

$$t_1 = \left[1.061 - 0.463 \frac{D}{H} \sqrt{\frac{HG}{S_d}} \right] \frac{2.6HDG}{S_d} + CA \quad \text{equ 3.47}$$

where:

- D = nominal tank diameter (ft)
- H = height from bottom of shell to top angle (ft)
- G = design specific gravity of liquid
- S_d = allowable design stress for calculating plate thickness (lb/in²)
- t_1 = thickness (inches)
- CA = corrosion allowances (inches)

Equation 3.47 combines the circumferential stress due to the hydraulic head (which is tensile), with the compressive circumferential stress caused by the radial edge restraining force at the base of the shell. It also incorporates a modification to allow for the effect of the second course. It becomes conservative when the height of the bottom course is greater than $2.625 \sqrt{r \cdot t_1}$, (where r , is the tank radius, in inches). In such cases, the bottom course thickness need not exceed the thickness calculated by the "one-foot" method.

3.6.3 The second course

The second course is more complicated because the restraint of the tank bottom raises the location of the maximum stress in the bottom course of larger tanks, to the vicinity of the girth joint between the first two courses. It is dependent upon the height of the bottom course and the value of the bottom course.

There are three empirically based equations which govern the calculation of the second course thickness and these are given as follows:

- 1) If the height of the bottom course is less than or equal to

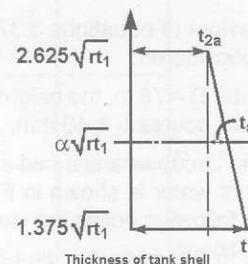
$$1.375 \sqrt{r \cdot t_1} \quad \text{equ 3.48}$$

then the second course t_2 should be the same thickness as the bottom course t_1 .

- 2) The influence of the second course is negligible when

$$h_1 = 2.625 \sqrt{r \cdot t_1} \quad \text{equ 3.49}$$

- 3) When $\frac{h_1}{\sqrt{r \cdot t_1}}$ lies between 1.375 and 2.625 a linear variation is introduced, and this is as follows:



$$\frac{t_1 - t_{2a}}{(2.625 - 1.375) \sqrt{r t_1}} = \frac{t_\alpha}{(2.625 - \alpha) \sqrt{r t_1}}$$

$$\therefore t_\alpha = (t_1 - t_{2a}) \frac{(2.625 - \alpha)}{1.25}$$

$$t_\alpha = (t_1 - t_{2a}) \left(2.1 - \frac{\alpha}{1.25} \right)$$

when the height of first course is equal to:

$$\alpha \sqrt{r t_1} \text{ i.e. } \alpha = h_1 / \sqrt{r t_1}$$

the thickness is:

$$t_2 = t_{2a} + t_\alpha = t_{2a} + (t_1 - t_{2a}) \left[2.1 - \frac{h_1}{1.25 \sqrt{r t_1}} \right] \quad \text{equ 3.50}$$

where:

- h_1 = height of the bottom shell course (inches)
- t_2 = final thickness of the second shell course (inches)
- t_{2a} = thickness of the second shell course calculated in the manner described for the upper shell courses (and given in Section 3.6.4) (inches)

3.6.4 The upper courses

For the **upper courses** the "design point" required to provide the maximum stress is obtained by examining the expansion and rotation of the girth joint. For a design where the thickness of each course is determined by a common stress, the theoretical location of the "design point" is at a variable distance above the bottom of the course in question and this is examined as follows:

The elastic movement of the upper shell courses at a typical girth joint are shown in Figure 3.44. The dotted lines are the position the shell would adopt if it was allowed to expand freely under hydrostatic loading i.e. "unrestrained radial growth". It is assumed that a uniform radial load is applied at the lower edge of the upper course moving the unrestrained shell to point 2.

Point 3 is the point where the deflection curve crosses the free deflection curve at a distance of $1.22 \sqrt{r \cdot t_u}$. This value is the same as given in Figure 3.43 since σ_0 and w decay in exactly the same way:

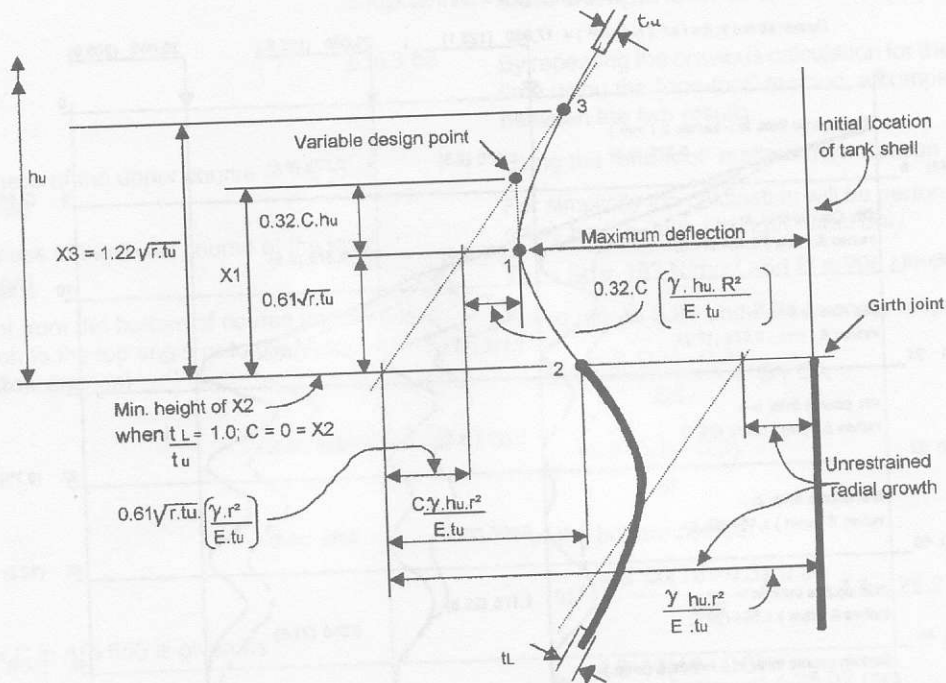


Figure 3.44 Elastic movement of upper shell courses at a typical girth joint

$$w = \frac{Qr^3}{2K\beta^3} e^{-\beta\chi/r} \cos \frac{\beta\chi}{r}$$

The deflection w is zero when $\cos \frac{\beta\chi}{r} = 0$, i.e.

$$\frac{\beta\chi}{r} = \frac{\pi}{2} \text{ and } \chi = \frac{\pi r}{2\beta} = \frac{\pi}{2} \frac{r}{\sqrt{3(1-\nu^2)}} \sqrt{\frac{t}{r}}$$

$$\chi = \frac{\pi}{2} \frac{\sqrt{rt}}{1.2854} = 1.22\sqrt{rt}$$

Referring to Figure 3.44, point 1 is taken as the mid-point between the end point 2 and point 3. The deflection at this mid-point is given as 0.32 times the deflection at the end (point 2).

The average deflection δ_{ave} at point 2 can be approximated by using the "pressure area" method. In this it is assumed that a portion of the cylinder on either side of point 2, at the girth joint, is involved as shown in Figure 3.45.

The effective cylinder length = $\sqrt{rt_u} + \sqrt{rt_L}$

Assume that the hydraulic pressure at girth point 2 is constant over the effective cylinder length. Thus the pressure times the projected area:

$$\gamma h_f (\sqrt{rt_u} + \sqrt{rt_L}) \quad \text{equ 3.51}$$

This pressure is resisted by:

$$S(t_u \sqrt{rt_u} + t_L \sqrt{rt_L}) \quad \text{equ 3.52}$$

where S is the stress in the vessel.

Applying equilibrium, equations 3.51 and 3.52 must be equal: and therefore:

$$S = \gamma h_f \frac{\sqrt{rt_u} + \sqrt{rt_L}}{t_u \sqrt{rt_u} + t_L \sqrt{rt_L}} \quad \text{equ 3.53}$$

$$\delta_{ave} = \frac{Sr}{E} = \frac{\gamma h_f^2}{E t_u} \left[\frac{\sqrt{rt_u} + \sqrt{rt_L}}{\sqrt{rt_u} + (t_L/t_u) \sqrt{rt_L}} \right]$$

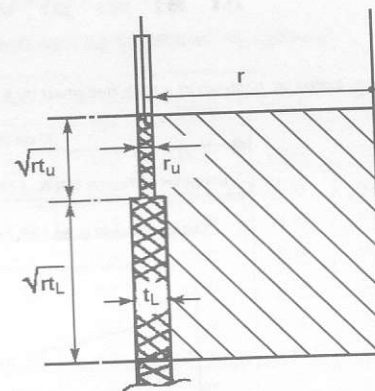


Figure 3.45 Portion of cylinder on either side of point 2 in Figure 3.44

From Figure 3.44

$$C \frac{\gamma h_f^2}{E t_u} = \frac{\gamma h_f^2}{E t_u} - \delta_{ave}$$

Substituting for δ_{ave} from equation 3.53 gives:

$$C = 1 - \left[\frac{1 + \sqrt{(t_L/t_u)}}{1 + (t_L/t_u) \sqrt{(t_L/t_u)}} \right] \quad \text{equ 3.54}$$

if

$$K = \frac{t_L}{t_u}$$

$$C = 1 - \left[\frac{1 + \sqrt{K}}{1 + K \sqrt{K}} \right]$$

$$C = \frac{\sqrt{K}(K-1)}{1 + K \sqrt{K}} \quad \text{equ 3.55}$$

The location of the design point above the girth joint, for the upper courses is the lowest value obtained from the resulting three expressions:

$$X_1 + 0.61\sqrt{rt_u} + 0.32Ch_u \quad \text{equ 3.56}$$

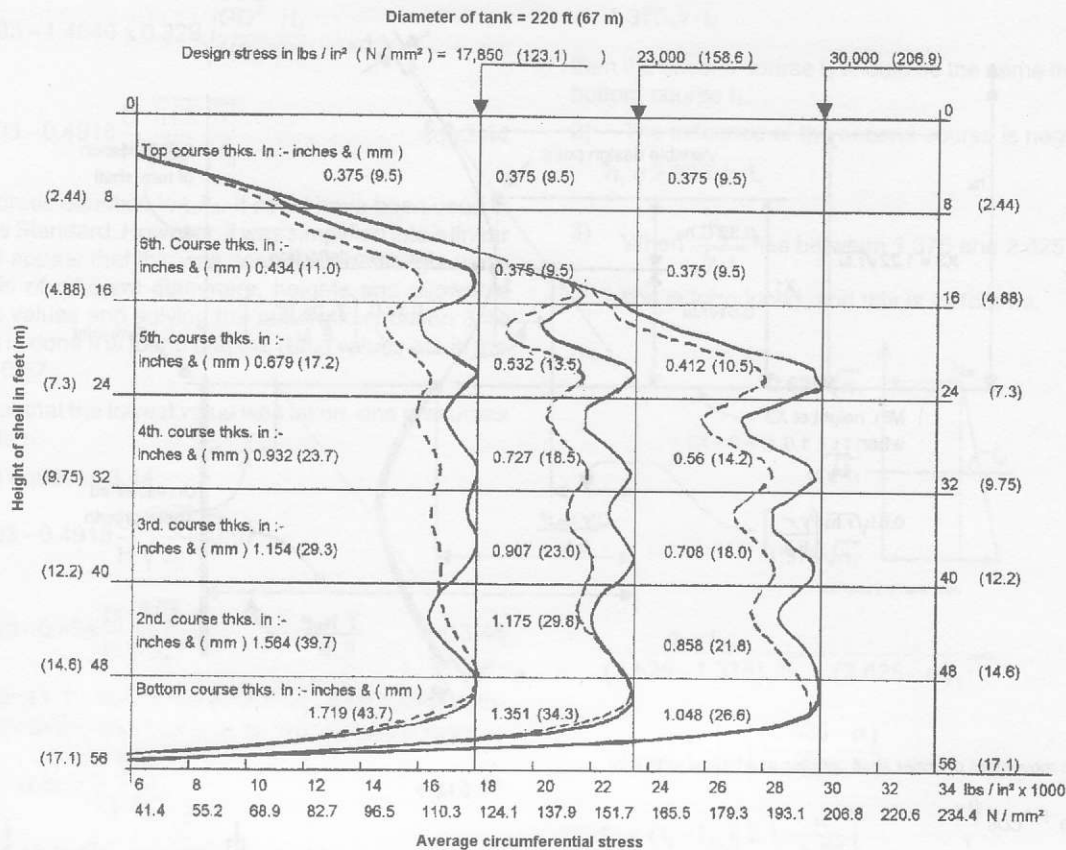


Figure 3.46 Actual stresses by analysis in a tank designed by the "variable design point" method (full line) and the "one-foot" method (chain dotted line)

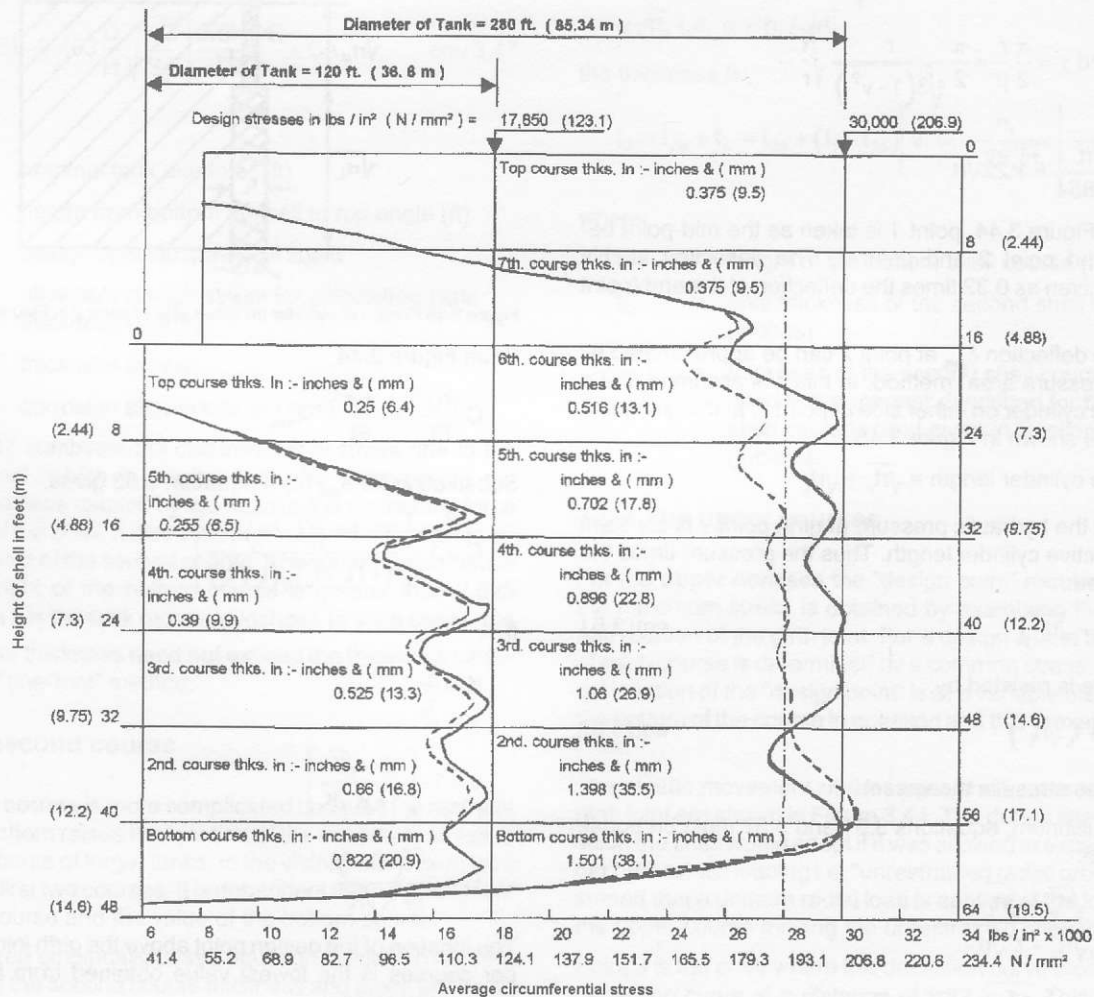


Figure 3.47 Actual stresses by analysis in smaller and in larger tanks designed by the "variable design point" method (full line) and the "one-foot" method (chain dotted line)

$$X_2 = Ch_u \quad \text{equ 3.57}$$

$$X_3 = 1.22\sqrt{rt_u} \quad \text{equ 3.58}$$

where:

- t_u = thickness of the upper course at the joint (inches)
 t_L = thickness of the lower course at the joint (inches)
 h_u = height from the bottom of course under consideration to the top angle or to the bottom of the overflow (inches)

$$C = \frac{\sqrt{K(K-1)}}{1+K\sqrt{K}} \quad \text{equ 3.59}$$

where:

$$K = \frac{t_L}{t_u}$$

The expression for C in API 650 is given as

$$C = \frac{K^{0.5}(k-1)}{(1+k^{1.5})}$$

which gives the same numerical value as equation 3.59.

3.6.5 Detailed "variable design point" method calculation

The preceding calculations require an estimated thickness for the upper course t_u .

This can be achieved by using the thickness obtained by the "one-foot" method equation 3.36. Using the thickness of the lower course, previously calculated, the value C can be obtained. From this X_1 , X_2 and X_3 can be calculated. The lowest of these X values, is then used to derive an improved value for t_u in a modified version of equation 3.36:

$$t_x = \frac{2.6 \cdot D(H - X/12)G}{S_d} + CA \quad (\text{Imperial units})$$

This first value of t_x is used to repeat the steps previously described, until there is only a small difference between the successive values of t_x . Invariably only three iterations are necessary to satisfy convergence.

The result of using the method is a tank where the upper courses are slightly thinner than those obtained with the "one-foot" method. When analysed using the Kalnins program, Zick and McGrath found that the maximum values of the actual stress compare well with the design stresses - see Figures 3.46 and 3.47. That is the maximum stresses in each course have a similar magnitude.

This reiterative method is somewhat labourious and was very time consuming for designers prior to the advent of modern computer software which is ideally suited for programming the above calculations.

By way of illustration Figure 3.48 shows a typical example of the use of this method of calculation and is reproduced in its entirety on pages 64-75.

The authors are grateful to the late Professor A. S. Tooth, Professor of Mechanical Engineering, Strathclyde University, to reproduce these calculations in full.

3.6.6 Comparison of the thickness results

By repeating the previous calculation for the same tank but this time using the "one-foot" method, a comparison can be made between the two results.

Using the "one-foot" method from Section 3.5.4.2:

For simplicity the calculation will be performed using the properties for the "high strength" steel only.

Then $S_d = 193 \text{ N/mm}^2$ and $S_t = 208 \text{ N/mm}^2$.

So equations 3.34 and 3.35 become respectively:

$$t_d = \frac{4.9 \cdot D(H-0.3)}{S_d} \cdot G + CA$$

$$t_t = \frac{4.9 \cdot D(H-0.3)}{S_t}$$

For the bottom course:

$$t_d = \frac{4.9 \cdot 60(18-0.3) \cdot 0.9}{193} + 1.0 = 25.27 \text{ mm}$$

$$t_t = \frac{4.9 \cdot 60(18-0.3)}{208} = 25.02 \text{ mm}$$

The greater of these two values is taken to be the thickness for the bottom course i.e. 25.27.

The calculation can be tabulated as follows:

Course No	H (m)	(H - 0.3) (m)	t_d (m)	t_t (m)	Min. t (m)	Rounded t (mm)
1(Btm)	2.25	17.7	25.27	25.02	25.27	25.3
2	2.25	15.45	22.18	21.84	22.18	22.2
3	2.25	13.2	19.10	18.66	19.10	19.1
4	2.25	10.95	16.01	15.48	16.01	16.1
5	2.25	8.7	12.93	12.30	12.93	13.0
6	2.25	6.45	9.84	9.12	9.84	9.9
7	2.25	4.2	6.76	5.94	6.76	6.8
8	2.25	1.95	3.67	2.76	3.67	3.7

The comparison between the thicknesses, in mm, is as follows:

Shell course	Btm	2	3	4	5	6	7	8	Weight of shell kg
"One-foot"	25.3	22.2	19.1	16.1	13.0	9.9	8.0	8.0	404,843
"Variable design point"	25.3	21.4	18.5	15.4	12.4	9.4	8.0	8.0	394,190
Difference	0	0.8	0.6	0.7	0.6	0.5	0	0	10,653

The saving in terms of weight of steel for the tank is 10,653 kg. in favour of the "variable design point" method.

Also the thinner plate gives savings in welding time, thus less manual labour, plant and welding consumables are utilised.

A further comparison is now made, with the shell designed to the BS 2654 method and the resulting thicknesses are found to be (in mm):

28.0, 24.5, 21.1, 17.7, 14.3, 10.9, 10.0 and 10.0

The weight of this shell is 454,450 kg which is 60,260 kg heavier than the shell designed to the API 650 "variable design point" method, and 49,607 kg heavier than the API 650 "one-foot" method. However, as the minimum allowable thickness for the top courses of the tank designed to BS 2654 is 10 mm instead of 8 mm, this accounts for 13,317 kg of the additional shell weight.

Design of Storage Tank Shell plating to A.P.I. 650, 10th. edition Nov 1998 + Add.1, Mar 2000.

Client : A. Another.
 Site : Europe.
 Contract No. C 001
 Calc. No. C 001 / 001
 Tank size : 60m dia. x 18m high.

Calculation in accordance with the "Variable - design - point" method (clause 3.6.4. of API 650)

Variables :		metric	imperial
D =		60 m	196.86 ft
H =		18 m	59.058 ft
G =		0.9	0.9
CA =		1 mm	0.0394 ins
Sd =		193 N/mm ²	27992 lb/in ²
St =		208 N/mm ²	30168 lb/in ²
No. of courses =		8	8
Height of each course =		2.25 m	7.38 ft

The first set of calculations will be made using a 'high' strength steel.

Material specification : - A.S.T.M. A 573M Gr.485

Check that $L/H \leq 1000/6$ where $L = (500.D.t)^{0.5}$
 D is the tank dia. in m.
 t is the bottom course shell thickness.

The bottom course shell thickness has not yet been established, but for the Variable point method not to be applicable for a tank of the above dimensions, it can be calculated that the bottom course would have to be
 > 300 mm thick and surely this will not be the case.

Calculations are worked simultaneously for both the 'design' & 'test' conditions.

For the Bottom course :

From Clause 3.6.3.2. Find values for "tpd" and "tpt".

$$tpd = \frac{4.9 \times D (H - 0.3) \times G}{Sd} + CA$$

$$tpt = \frac{4.9 \times D (H - 0.3)}{St}$$

$$tpd = 25.27 \text{ mm}$$

$$tpt = 25.02 \text{ mm}$$

From Clause 3.6.4.4.

$$tld = \left(1.06 - \frac{0.0696 D}{H} \sqrt{\frac{H G}{Sd}} \right) \left(\frac{4.9 H D G}{Sd} \right) + CA$$

$$tlt = \left(1.06 - \frac{0.0696 D}{H} \sqrt{\frac{H}{St}} \right) \left(\frac{4.9 H D}{St} \right)$$

$$tld = 25.50 \text{ mm}$$

$$tlt = 25.23 \text{ mm}$$

$$\text{Lesser of 'tpd' \& 'tld' = } 25.27 \text{ mm}$$

$$\text{Lesser of 'tpt' \& 'tlt' = } 25.02 \text{ mm}$$

The greater of these two latter figures is : - 25.27 ins.

Bottom course thickness = 25.3 mm.

The validity of using the Variable Point method can now be checked as required by Clause 3.6.4.1

Check that $L/H \leq 1000/6$ when $L = (500.D.t)^{0.5}$ = 871.21 and $H = 18$
 $L/H = 48.40$ As this is $\leq 1000/6$, the variable point method may be used

For the Second course :

$$\begin{aligned}
 h_1 &= 2250 \text{ mm. Width of bottom course.} \\
 r &= 30000 \text{ mm. Nominal Tank radius.} \\
 (t_{1d} - c.a.) &= 24.27 \text{ mm. Btm. course thks. less CA. Use for "t}_{2a}" (design). \quad \text{Used for ratio, } h_1 + \sqrt{r \cdot t_1} \\
 t_{1d} &= 25.27 \text{ mm. Total Btm. course thks.} \\
 t_{1t} &= 25.02 \text{ mm. Lesser of 't}_{1t}' \& 't_{1t}'. Used for ratio, } h_1 + \sqrt{r \cdot t_1} \\
 \text{Ratio for 't}_{1d}': \frac{h_1}{\sqrt{r \times (t_{1d} - c.a.)}} &= 2.637 \quad \text{Ratio for 't}_{1t}': \frac{h_1}{\sqrt{r \times t_{1t}}} = 2.597
 \end{aligned}$$

Ratio 't_{1d}' is ≥ 2.625 , then, $t_2 = t_{2a}$. This is found by trial for the 'Design' case as follows :-

Ratio 't_{1t}' is > 1.375 but < 2.625 , then, $t_2 = t_{2a} + (t_1 - t_{2a})[2.1 - \{h_1/1.25(r \cdot t_1)^{0.5}\}]$ and t_{2a} for the 'Test' condition is found as follows :-

Calculate the Second course 'Test' thickness by trial

$$t_{ud} = \frac{4.9 \times D (H - 0.3) G + CA}{S_d} = 22.18 \text{ mm}$$

$$t_{ut} = \frac{4.9 \times D (H - 0.3)}{S_t} = 21.84 \text{ mm}$$

1st. Trial

Find values of "x₁, x₂, & x₃" for both the Design & Test conditions. Course No. 2

K _d =	1.150	x _{1d} =	859.662	K _t =	1.1456	x _{1t} =	846.628
C _d =	0.072	x _{2d} =	1131.416	C _t =	0.0700	x _{2t} =	1102.782
H (m) =	15.75	x _{3d} =	995.217	H (m) =	15.75	x _{3t} =	987.476
		Use lowest value of 'x _d '	859.662 mm			Use lowest value of 'x _t ' =	846.628 mm
			0.860 m				0.847 m

$$t_{dx} = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 21.41 \text{ mm}$$

$$t_{tx} = \frac{4.9 \times D (H - x/1000)}{S_t} = 21.07 \text{ mm}$$

2nd. Trial

Repeat above calculation using new values for 't_u' & Btm. course thk's, for Design & Test.

$$t_{1d} = 25.27 \quad t_{1t} = 25.02 \quad t_{ud} = t_{dx} = 21.41 \quad t_{ut} = t_{tx} = 21.07$$

Find values of "x₁, x₂, & x₃" for both the Test & Design conditions.

K _d =	1.180	x _{1d} =	920.533	K _t =	1.188	x _{1t} =	934.163
C _d =	0.086	x _{2d} =	1348.770	C _t =	0.089	x _{2t} =	1403.867
H (m) =	15.75	x _{3d} =	977.853	H (m) =	15.75	x _{3t} =	969.850
		Use lowest value of 'x _d '	920.533 mm			se lowest value of 'x _t ' =	934.163 mm
			0.921 m				0.934 m

$$t_{dx} = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 21.33 \text{ mm}$$

$$t_{tx} = \frac{4.9 \times D (H - x/1000)}{S_t} = 20.94 \text{ mm}$$

3rd. Trial

Repeat above calculation using new values for 't_u' & Btm. course thk's, for Design & Test.

$$t_{1d} = 25.27 \quad t_{1t} = 25.02 \quad t_{ud} = t_{dx} = 21.33 \quad t_{ut} = t_{tx} = 20.94$$

Find values of "x₁, x₂, & x₃" for both the Test & Design conditions.

K _d =	1.184	x _{1d} =	930.062	K _t =	1.195	x _{1t} =	948.581
C _d =	0.088	x _{2d} =	1381.527	C _t =	0.092	x _{2t} =	1453.381
H (m) =	15.750	x _{3d} =	975.946	H (m) =	15.750	x _{3t} =	966.998
		Use lowest value of 'x _d '	930.062 mm			e lowest value of 'x _t ' =	948.581 mm
			0.930 m				0.949 m

$$t_{dx} = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 21.32 \text{ mm.} = t_{2a}$$

$$t_{tx} = \frac{4.9 \times D (H - x/1000)}{S_t} = 20.92 \text{ mm.} = t_{2a} \quad \text{Use to calc. value of } t_2 \text{ for the 'Test' case}$$

$$21.32 \text{ mm.} = 21.4 \text{ mm.}$$

$$\text{Test' } t_2 = t_{2a} + (t_1 - t_{2a}) [2.1 - h_1 / 1.25 (r \cdot t_1)^{0.5}] = 21.06 \text{ mm.}$$

Second course thickness =	21.4 mm.
---------------------------	----------

For the Third course.

1st. Trial

$$t_{Ld} = 21.32 \text{ mm.} \quad t_{Lt} = 20.92 \text{ mm.}$$

$$t_{ud} = \frac{4.9 \times D (H - 0.3) G + CA}{S_d} = 19.10 \text{ mm}$$

$$t_{ut} = \frac{4.9 \times D (H - 0.3)}{S_t} = 18.66 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.				Course No.	3		
Kd =	1.116	x1d =	705.270	Kt =	1.121	x1t =	710.091
Cd =	0.056	x2d =	761.111	Ct =	0.059	x2t =	792.872
H (m) =	13.5	xd3 =	923.428	H (m) =	13.5	xt3 =	912.745
Use lowest value of 'xd' =				Use lowest value of 'xt' =			

<u>3rd. Trial</u>	tLd =	18.45 mm.	tLt =	18.05 mm			
	tud =	15.41 mm.	tut =	14.92 mm			
<u>Find values of " x1, x2, & x3 " for both the Design & Test conditions.</u>							
Kd =	1.197	x1d =	751.227	Kt =	1.172	x1t =	703.433
Cd =	0.093	x2d =	1051.662	Ct =	0.082	x2t =	922.842
H (m) =	11.250	xd3 =	829.391	H (m) =	11.250	xt3 =	816.246
Use lowest value of 'xd' =			751.227 mm	Use lowest value of 'xt' =			703.433 mm
			= 0.751 m				= 0.703 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 15.39 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 14.91 \text{ mm}$$

$$\text{Fourth course thickness} = 15.4 \text{ mm}$$

For the Fifth course.

<u>1st. Trial</u>	tLd =	15.393654 mm.	tLt =	14.91 mm
	tud =	$\frac{4.9 \times D (H - 0.3) G + CA}{S_d}$		12.93 mm
	tut =	$\frac{4.9 \times D (H - 0.3)}{S_t}$		12.30 mm

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Course No. 5

Kd =	1.191	x1d =	640.609	Kt =	1.212	x1t =	658.774
Cd =	0.091	x2d =	814.771	Ct =	0.100	x2t =	900.847
H (m) =	9.00	xd3 =	759.764	H (m) =	9.00	xt3 =	741.007
Use lowest value of 'xd' =			640.609 mm	Use lowest value of 'xt' =			658.774 mm
			= 0.641 m				= 0.659 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 12.46 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 11.79 \text{ mm}$$

<u>2nd. Trial</u>	tLd =	15.39 mm.	tLt =	14.91 mm
	tud =	12.46 mm.	tut =	11.79 mm

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.235	x1d =	690.469	Kt =	1.196	x1t =	630.703
Cd =	0.110	x2d =	992.221	Ct =	0.093	x2t =	837.248
H (m) =	9.00	xd3 =	745.916	H (m) =	9.00	xt3 =	725.567
Use lowest value of 'xd' =			690.469 mm	Use lowest value of 'xt' =			630.703 mm
			= 0.690 m				= 0.631 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 12.39 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 11.83 \text{ mm}$$

<u>3rd. Trial</u>	tLd =	15.39 mm.	tLt =	14.91 mm
	tud =	12.39 mm.	tut =	11.83 mm

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.242	x1d =	697.973	Kt =	1.203	x1t =	639.782
Cd =	0.113	x2d =	1018.874	Ct =	0.096	x2t =	863.715
H (m) =	9.00	xd3 =	743.867	H (m) =	9.00	xt3 =	726.787
Use lowest value of 'xd' =			697.973 mm	Use lowest value of 'xt' =			639.782 mm
			= 0.698 m				= 0.640 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 12.38 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 11.82 \text{ mm}$$

$$\text{Fifth course thickness} = 12.4 \text{ mm}$$

For the Sixth course.

<u>1st. Trial</u>	tLd =	12.38 mm.	tLt =	11.82 mm
	tud =	$\frac{4.9 \times D (H - 0.3) G + CA}{S_d}$		9.84 mm
	tut =	$\frac{4.9 \times D (H - 0.3)}{S_t}$		9.12 mm

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 4

Find values of " x1, x2, & x3 " for both the Design & Test conditions.				Course No. 6					
Kd =	1.258	x1d =	590.692	Kt =	1.296	x1t =	613.197		
Cd =	0.120	x2d =	810.054	Ct =	0.136	x2t =	919.316		
H (m) =	6.75	xd3 =	662.950	H (m) =	6.75	xt3 =	638.032		
Use lowest value of 'xd' =			590.692 mm	Use lowest value of 'xt' =			613.197 mm		
			=	0.591 m				=	0.613 m
tdx = $\frac{4.9 \times D (H - x/1000) G + CA}{S_d}$				9.44 mm					
ttx = $\frac{4.9 \times D (H - x/1000)}{S_t}$				8.67 mm					
<u>2nd. Trial</u>									
tLd =	12.38 mm.	tLt =	11.82 mm						
tud =	9.44 mm.	tut =	8.67 mm						
Find values of " x1, x2, & x3 " for both the Design & Test conditions.									
Kd =	1.311	x1d =	632.268	Kt =	1.251	x1t =	564.119		
Cd =	0.142	x2d =	961.166	H (m) =	0.117	x2t =	790.453		
H (m) =	6.75	xd3 =	649.390	H (m) =	6.75	xt3 =	622.348		
Use lowest value of 'xd' =			632.268 mm	Use lowest value of 'xt' =			564.119 mm		
			=	0.632 m				=	0.564 m
tdx = $\frac{4.9 \times D (H - x/1000) G + CA}{S_d}$				9.39 mm					
ttx = $\frac{4.9 \times D (H - x/1000)}{S_t}$				8.74 mm					
<u>3rd. Trial</u>									
tLd =	12.3819497 mm.	tLt =	11.82 mm						
tud =	9.38731523 mm.	tut =	8.74 mm						
Find values of " x1, x2, & x3 " for both the Design & Test conditions.									
Kd =	1.319	x1d =	638.392	Kt =	1.259	x1t =	572.417		
Cd =	0.146	x2d =	983.370	Ct =	0.120	x2t =	812.505		
H (m) =	6.75	xd3 =	647.428	H (m) =	6.75	xt3 =	624.832		
Use lowest value of 'xd' =			638.392 mm	Use lowest value of 'xt' =			572.417 mm		
			=	0.638 m				=	0.572 m
tdx = $\frac{4.9 \times D (H - x/1000) G + CA}{S_d}$				9.38 mm					
ttx = $\frac{4.9 \times D (H - x/1000)}{S_t}$				8.73 mm					
				Sixth course thickness = 9.4 mm					
<u>For the Seventh course.</u>									
<u>1st. Trial</u>									
tLd =	9.38 mm.	tLt =	8.73 mm						
tud = $\frac{4.9 \times D (H - 0.3) G + CA}{S_d}$				6.76 mm					
tut = $\frac{4.9 \times D (H - 0.3)}{S_t}$				5.94 mm					
Find values of " x1, x2, & x3 " for both the Design & Test conditions.									
				Course No. 7					
Kd =	1.388	x1d =	524.336	Kt =	1.471	x1t =	552.814		
Cd =	0.173	x2d =	780.220	Ct =	0.205	x2t =	923.080		
H (m) =	4.50	xd3 =	549.331	H (m) =	4.50	xt3 =	514.858		
Use lowest value of 'xd' =			524.336 mm	Use lowest value of 'xt' =			514.858 mm		
			=	0.524 m				=	0.515 m
tdx = $\frac{4.9 \times D (H - x/1000) G + CA}{S_d}$				6.45 mm					
ttx = $\frac{4.9 \times D (H - x/1000)}{S_t}$				5.63 mm					
<u>2nd. Trial</u>									
tLd =	9.38 mm.	tLt =	8.73 mm						
tud =	6.45 mm.	tut =	5.63 mm						
Find values of " x1, x2, & x3 " for both the Design & Test conditions.									
Kd =	1.454	x1d =	554.646	Kt =	1.354	x1t =	480.858		
Cd =	0.199	x2d =	894.697	Ct =	0.160	x2t =	719.064		
H (m) =	4.50	xd3 =	536.685	H (m) =	4.50	xt3 =	501.516		
Use lowest value of 'xd' =			536.685 mm	Use lowest value of 'xt' =			480.858 mm		
			=	0.537 m				=	0.481 m
tdx = $\frac{4.9 \times D (H - x/1000) G + CA}{S_d}$				6.43 mm					
ttx = $\frac{4.9 \times D (H - x/1000)}{S_t}$				5.68 mm					
<u>3rd. Trial</u>									
tLd =	9.38 mm.	tLt =	8.73 mm						
tud =	6.43 mm.	tut =	5.68 mm						

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 5

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.458	x1d =	556.361	Kt =	1.357	x1t =	483.988
Cd =	0.200	x2d =	901.157	Ct =	0.161	x2t =	725.509
H (m) =	4.50	xd3 =	535.980	H (m) =	4.50	xt3 =	503.650
		Use lowest value of 'xd' =	535.980 mm			Use lowest value of 'xt' =	483.988 mm
			= 0.536 m				= 0.484 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 6.43 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 5.68 \text{ mm}$$

$$\text{Seventh course thickness} = 6.5 \text{ mm}$$

For the Eighth course.

1st. Trial $t_{Ld} = 6.43 \text{ mm.}$ $t_{Lt} = 5.68 \text{ mm}$

$$tud = \frac{4.9 \times D (H - 0.3) G + CA}{S_d} = 3.67 \text{ mm}$$

$$tut = \frac{4.9 \times D (H - 0.3)}{S_t} = 2.76 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Course No. 8

Kd =	1.752	x1d =	418.356	Kt =	2.059	x1t =	452.168
Cd =	0.300	x2d =	674.548	Ct =	0.384	x2t =	864.873
H (m) =	2.25	xd3 =	405.001	H (m) =	2.25	xt3 =	350.816
		Use lowest value of 'xd' =	405.001 mm			Use lowest value of 'xt' =	350.816 mm
			= 0.405 m				= 0.351 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 3.53 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 2.68 \text{ mm}$$

2nd. Trial

$$t_{Ld} = 6.43 \text{ mm.} \quad t_{Lt} = 5.68 \text{ mm}$$

$$tud = 3.53 \text{ mm.} \quad tut = 2.68 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.823	x1d =	429.656	Kt =	1.608	x1t =	355.842
Cd =	0.321	x2d =	722.386	Ct =	0.254	x2t =	571.045
H (m) =	2.25	xd3 =	396.986	H (m) =	2.25	xt3 =	346.215
		Use lowest value of 'xd' =	396.986 mm			Use lowest value of 'xt' =	346.215 mm
			= 0.397 m				= 0.346 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 3.54 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 2.69 \text{ mm}$$

3rd. Trial

$$t_{Ld} = 6.43 \text{ mm.} \quad t_{Lt} = 5.68 \text{ mm}$$

$$tud = 3.54 \text{ mm.} \quad tut = 2.69 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.817	x1d =	428.780	Kt =	1.603	x1t =	354.837
Cd =	0.319	x2d =	718.683	Ct =	0.252	x2t =	567.250
H (m) =	2.250	xd3 =	397.603	H (m) =	2.250	xt3 =	346.634
		Use lowest value of 'xd' =	397.603 mm			Use lowest value of 'xt' =	346.634 mm
			= 0.398 m				= 0.347 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 3.54 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 2.69 \text{ mm}$$

$$\text{Eighth course thickness} = 3.6 \text{ mm}$$

A summary of course thicknesses is given at the end of this set of calculations.

The upper courses often calculate to be thinner than the minimum allowable shell course thickness for the particular diameter of tank under consideration, therefore a second set of calculations is produced using a 'low strength' steel and this often results in a more financially economical design for one or more of the upper courses.

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 6

A second set of calculations is now made using a 'low' strength steel.

For the Bottom course :

First find "tpd" & "tpt".

Variables :		<u>metric</u>	<u>imperial</u>
D =		60 m	196.86 ft
H =		18 m	59.058 ft
G =		0.9	0.9
CA =		1 mm	0.0394 ins
Sd =		137 N/mm ²	19870 lb/in ²
St =		154 N/mm ²	22336 lb/in ²
No. of courses =		8	8
Height of each course =		2.25 m	7.38225 ft

Material Specification :- A.S.T.M. A 283 Gr.C

Calculations are worked simultaneously for both the 'design' & 'test' conditions.

From Clause 3.6.3.2.

$$tpd = \frac{4.9 \times D (H - 0.3) \times G}{Sd} + CA$$

$$tpt = \frac{4.9 \times D (H - 0.3)}{St}$$

$$tpd = 35.19 \text{ mm}$$

$$tpt = 33.79 \text{ mm}$$

From Clause 3.6.4.4.

$$tld = \left(1.06 - \frac{0.0696 D}{H} \sqrt{\frac{H G}{Sd}} \right) \left(\frac{4.9 H D G}{Sd} \right) + CA$$

$$tlr = \left(1.06 - \frac{0.0696 D}{H} \sqrt{\frac{H}{St}} \right) \left(\frac{4.9 H D}{St} \right)$$

$$tld = 35.08 \text{ mm}$$

$$tlr = 33.70 \text{ mm}$$

$$\text{Lesser of 'tpd' \& 'tld' = 35.08 mm}$$

$$\text{Lesser of 'tpt' \& 'tlr' = 33.70 mm}$$

The greater of these two latter figures is :- 35.0773689 ins.

$$\text{Bottom course thickness} = 35.1 \text{ mm.}$$

For the Second course :

$$h1 = 2250 \text{ mm. Width of bottom course.}$$

$$r = 30000 \text{ mm. Nominal Tank radius.}$$

$$(t1d - c.a.) = 34.08 \text{ mm. Btm. course thks. less CA. Use for "t2a" (design) Used for ratio } h1 \div \sqrt{r \cdot t1}$$

$$t1d = 35.08 \text{ mm. Total Btm. course thks.}$$

$$t1t = 33.70 \text{ mm. Lesser of 'tpt' \& 'tlr'. Used for ratio } h1 \div \sqrt{r \cdot t1}$$

$$\text{Ratio for 't1d': } \frac{h1}{\sqrt{r \times (t1d - c.a.)}} = 2.23$$

$$\text{Ratio for 't1t': } \frac{h1}{\sqrt{r \times t1t}} = 2.24$$

Ratio 't1d' is >1.375 but <2.625, then, $t2 = t2a + (t1 - t2a)[2.1 - \{h1/1.25(r \cdot t1)^{0.5}\}]$ and $t2a$ for the 'Design' condition is found as follows :-

Ratio 't1t' is >1.375 but <2.625, then, $t2 = t2a + (t1 - t2a)[2.1 - \{h1/1.25(r \cdot t1)^{0.5}\}]$ and $t2a$ for the 'Test' condition is found as follows :-

Calculate the Second course 'Test' thickness by trial

$$tud = \frac{4.9 \times D (H - 0.3)}{Sd} G + CA = 30.84 \text{ mm}$$

$$tut = \frac{4.9 \times D (H - 0.3)}{St} = 29.50 \text{ mm}$$

1st. Trial

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Course No. 2

$$Kd = 1.137 \quad x1d = 920.469 \quad Kt = 1.143 \quad x1t = 919.523$$

$$Cd = 0.066 \quad x2d = 1042.896 \quad Ct = 0.069 \quad x2t = 1080.352$$

$$H (m) = 15.75 \quad x3d = 1173.484 \quad H (m) = 15.75 \quad x3t = 1147.620$$

$$\text{Use lowest value of 'xd' = 920.469 mm} \quad \text{Use lowest value of 'xt' = 919.523 mm}$$

$$0.920 \text{ m} \quad 0.920 \text{ m}$$

$$tdx = \frac{4.9 \times D (H - x/1000)}{Sd} G + CA = 29.64 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{St} = 28.31 \text{ mm}$$

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 7

2nd. Trial.

Repeat above calculation using new values for 'tu' & Btm. course thk's, for Design & Test.

tld =	35.08	tlr =	33.70	tud = tdx =	29.64	tut = ttx =	28.31
Find values of "x1, x2, & x3" for both the Test & Design conditions.							
Kd =	1.183	x1d =	1014.799	Kt =	1.190	x1t =	1017.352
Cd =	0.087	x2d =	1373.657	Ct =	0.090	x2t =	1422.390
H (m) =	15.75	xd3 =	1150.459	H (m) =	15.75	xt3 =	1124.375
		Use lowest value of 'xd' =	1014.799 mm			Use lowest value of 'xt' =	1017.352 mm
			=				=
			1.015 m				1.017 m

$$tdx = \frac{4.9 \times D(H - x/1000)G + CA}{S_d} = 29.46 \text{ mm}$$

$$ttx = \frac{4.9 \times D(H - x/1000)}{S_t} = 28.13 \text{ mm}$$

3rd. Trial.

Repeat above calculation using new values for 'tu' & Btm. course thk's, for Design & Test.

tld =	35.08	tlr =	33.70	tud = tdx =	29.46	tut = ttx =	28.13
Find values of "x1, x2, & x3" for both the Test & Design conditions.							
Kd =	1.191	x1d =	1029.598	Kt =	1.198	x1t =	1033.308
Cd =	0.091	x2d =	1425.434	Ct =	0.094	x2t =	1478.055
H (m) =	15.75	xd3 =	1146.918	H (m) =	15.75	xt3 =	1120.661
		Use lowest value of 'xd' =	1029.598 mm			Use lowest value of 'xt' =	1033.308 mm
			=				=
			1.030 m				1.033 m

$$tdx = \frac{4.9 \times D(H - x/1000)G + CA}{S_d} = 29.43 \text{ mm.} = t_{2a} \text{ Use to calc. value of } t_2 \text{ for the 'Design' case}$$

$$ttx = \frac{4.9 \times D(H - x/1000)}{S_t} = 28.10 \text{ mm.} = t_{2a} \text{ Use to calc. value of } t_2 \text{ for the 'Test' case}$$

$$29.43 \text{ mm.} = 29.5 \text{ mm.}$$

$$\text{'Design' } t_2 = t_{2a} + (t_1 - t_{2a}) [2.1 - h_1 / 1.25(r \cdot t_1)^{0.5}] = 31.381 \text{ OK}$$

$$\text{'Test' } t_2 = t_{2a} + (t_1 - t_{2a}) [2.1 - h_1 / 1.25(r \cdot t_1)^{0.5}] = 29.832 \text{ mm.}$$

$$31.381$$

$$\text{Second course thickness} = 31.4 \text{ mm.}$$

For the Third course.1st. Trial tLd = 31.38 mm. tLt = 29.83 mm

$$tud = \frac{4.9 \times D(H - 0.3)G + CA}{S_d} = 26.49 \text{ mm}$$

$$tut = \frac{4.9 \times D(H - 0.3)}{S_t} = 25.2 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions. Course No. 3

Kd =	1.184	x1d =	922.642	Kt =	1.184	x1t =	907.971
Cd =	0.088	x2d =	1183.773	Ct =	0.087	x2t =	1179.957
H (m) =	13.50	xd3 =	1087.669	H (m) =	13.50	xt3 =	1060.769
		Use lowest value of 'xd' =	922.642 mm			Use lowest value of 'xt' =	907.971 mm
			=				=
			0.923 m				0.908 m

$$tdx = \frac{4.9 \times D(H - x/1000)G + CA}{S_d} = 25.29 \text{ mm}$$

$$ttx = \frac{4.9 \times D(H - x/1000)}{S_t} = 24.04 \text{ mm}$$

2nd. Trial

tLd = 31.38 mm. tLt = 29.83 mm

$$tud = 25.29 \text{ mm.} \quad tut = 24.04 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.241	x1d =	1017.683	Kt =	1.180	x1t =	887.251
Cd =	0.113	x2d =	1519.795	Ct =	0.085	x2t =	1153.829
H (m) =	13.50	xd3 =	1062.698	H (m) =	13.50	xt3 =	1036.052
		Use lowest value of 'xd' =	1017.683 mm			Use lowest value of 'xt' =	887.251 mm
			=				=
			1.018 m				0.887 m

$$tdx = \frac{4.9 \times D(H - x/1000)G + CA}{S_d} = 25.11 \text{ mm}$$

$$ttx = \frac{4.9 \times D(H - x/1000)}{S_t} = 24.08 \text{ mm}$$

3rd. Trial

tLd = 31.38 mm. tLt = 29.83 mm

$$tud = 25.11 \text{ mm.} \quad tut = 24.08 \text{ mm}$$

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 8

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.250	x1d =	1032.709	Kt =	1.188	x1t =	904.445
Cd =	0.117	x2d =	1572.787	Ct =	0.089	x2t =	1206.227
H (m) =	13.50	xd3 =	1058.835	H (m) =	13.50	xt3 =	1036.904
		Use lowest value of 'xd' =	1032.709 mm			Use lowest value of 'xt' =	904.445 mm
		=	1.033 m			=	0.904 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 25.08 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 24.05 \text{ mm}$$

$$\text{Third course thickness} = 25.1 \text{ mm}$$

For the Fourth course.

1st. Trial $tLd = 25.0791621 \text{ mm.} \quad tLt = 24.05 \text{ mm}$

$$tud = \frac{4.9 \times D (H - 0.3) G + CA}{S_d} = 22.15 \text{ mm}$$

$$tut = \frac{4.9 \times D (H - 0.3)}{S_t} = 20.90 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Course No. 4

Kd =	1.132	x1d =	727.111	Kt =	1.150	x1t =	742.836
Cd =	0.064	x2d =	718.354	Ct =	0.072	x2t =	811.765
H (m) =	11.25	xd3 =	994.476	H (m) =	11.25	xt3 =	966.142
		Use lowest value of 'xd' =	718.354 mm			Use lowest value of 'xt' =	742.836 mm
		=	0.718 m			=	0.743 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 21.34 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 20.06 \text{ mm}$$

2nd. Trial

$$tLd = 25.08 \text{ mm.} \quad tLt = 24.05 \text{ mm}$$

$$tud = 21.34 \text{ mm.} \quad tut = 20.06 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.175	x1d =	788.732	Kt =	1.127	x1t =	693.797
Cd =	0.084	x2d =	939.524	Ct =	0.061	x2t =	689.359
H (m) =	11.25	xd3 =	976.168	H (m) =	11.25	xt3 =	946.404
		Use lowest value of 'xd' =	788.732 mm			Use lowest value of 'xt' =	689.359 mm
		=	0.789 m			=	0.689 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 21.20 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 20.16 \text{ mm}$$

3rd. Trial

$$tLd = 25.08 \text{ mm.} \quad tLt = 24.05 \text{ mm}$$

$$tud = 21.20 \text{ mm.} \quad tut = 20.16 \text{ mm}$$

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.183	x1d =	799.419	Kt =	1.134	x1t =	707.094
Cd =	0.087	x2d =	977.786	Ct =	0.065	x2t =	727.153
H (m) =	11.250	xd3 =	973.054	H (m) =	11.250	xt3 =	948.809
		Use lowest value of 'xd' =	799.419 mm			Use lowest value of 'xt' =	707.094 mm
		=	0.799 m			=	0.707 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 21.18 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 20.13 \text{ mm}$$

$$\text{Fourth course thickness} = 21.2 \text{ mm}$$

For the Fifth course.

1st. Trial $tLd = 21.18 \text{ mm.} \quad tLt = 20.13 \text{ mm}$

$$tud = \frac{4.9 \times D (H - 0.3) G + CA}{S_d} = 17.80 \text{ mm}$$

$$tut = \frac{4.9 \times D (H - 0.3)}{S_t} = 16.61 \text{ mm}$$

Find values of " x1, x2, & x3 " for both the Design & Test conditions.				Course No. 5			
Kd =	1.190	x1d =	705.429	Kt =	1.212	x1t =	718.325
Cd =	0.090	x2d =	811.349	Ct =	0.100	x2t =	899.172
H (m) =	9.00	xd3 =	891.595	H (m) =	9.00	xt3 =	861.179
Use lowest value of 'xd' =				Use lowest value of 'xt' =			

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 17.02 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 15.81 \text{ mm}$$

<u>2nd. Trial</u>	tLd =	21.18 mm.	tLt =	20.13 mm
	tud =	17.02 mm.	tut =	15.81 mm

Find values of "x1, x2, & x3" for both the Design & Test conditions.								
Kd =	1.245	x1d =	764.989	Kt =	1.183	x1t =	670.236	
Cd =	0.114	x2d =	1028.455	Ct =	0.087	x2t =	781.644	
H (m) =	9.00	xd3 =	871.767	H (m) =	9.00	xt3 =	840.220	
value of 'xd' =		value of 'xd' =	value of 'xd' =	764.989 mm		Use lowest value of 'xt' =		670.236 mm
		=	0.765 m					0.670 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 16.90 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 15.90 \text{ mm}$$

<u>3rd. Trial</u>	tLd =	21.18 mm.	tLt =	20.13 mm
	tud =	16.90 mm.	tut =	15.90 mm

Find values of "x1, x2, & x3" for both the Design & Test conditions.							
Kd =	1.253	x1d =	774.044	Kt =	1.191	x1t =	681.867
Cd =	0.118	x2d =	1061.363	Ct =	0.090	x2t =	814.185
H (m) =	9.00	xd3 =	868.816	H (m) =	9.00	xt3 =	842.656
Use lowest value of 'xd' =		774.044 mm		Use lowest value of 'xt' =		681.867 mm	
		= 0.774 m				= 0.682 m	

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 16.89 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 15.88 \text{ mm}$$

Fifth course thickness = 16.9 mm

For the Sixth course.

<u>1st. Trial</u>	tLd =	16.89 ins.	tLt =	15.88 ins.
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$$tud = \frac{4.9 \times D (H - 0.3) G + CA}{S_d} = 13.46 \text{ mm}$$

$$tut = \frac{4.9 \times D (H - 0.3)}{S_t} = 12.31 \text{ mm}$$

Find values of " x1, x2, & x3 " for both the Design & Test conditions.				Course No. 6			
Kd =	1.255	x1d =	643.947	Kt =	1.290	x1t =	659.023
Cd =	0.119	x2d =	801.117	Ct =	0.133	x2t =	900.845
H (m) =	6.75	xd3 =	775.179	H (m) =	6.75	xt3 =	741.504
Use lowest value of 'xd' =				Use lowest value of 'xt' =			

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 12.79 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 11.63 \text{ mm}$$

<u>2nd. Trial</u>	tLd =	16.89 mm.	tLt =	15.88 mm
	tud =	12.79 mm.	tut =	11.63 mm

Find values of "x1, x2, & x3" for both the Design & Test conditions.							
Kd =	1.320	x1d =	693.500	Kt =	1.241	x1t =	603.966
Cd =	0.146	x2d =	986.244	Ct =	0.113	x2t =	761.502
H (m) =	6.750	xd3 =	755.804	H (m) =	6.750	xt3 =	720.572
Use lowest value of 'xd' =			693.500 mm	Use lowest value of 'xt' =			603.966 mm
			= 0.694 m				= 0.604 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 12.70 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 11.73 \text{ mm}$$

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 10

3rd. Trial	tLd =	16.89 mm.	tLt =	15.88 mm		
	tud =	12.70 mm.	tut =	11.73 mm		
Find values of "x1, x2, & x3" for both the Design & Test conditions.						
Kd =	1.330	x1d =	700.905	Kt =	1.251	x1t = 614.333
Cd =	0.150	x2d =	1013.811	Ct =	0.117	x2t = 788.819
H (m) =	6.75	xd3 =	752.972	H (m) =	6.75	xt3 = 723.821
		Use lowest value of 'xd' =	700.905 mm			Use lowest value of 'xt' = 614.333 mm
			= 0.701 m			= 0.614 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 12.68 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 11.71 \text{ mm}$$

$$\text{Sixth course thickness} = 12.7 \text{ mm}$$

For the Seventh course.

1st. Trial	tLd =	12.68 ins.	tLt =	11.71 ins.		
	tud =	$\frac{4.9 \times D (H - 0.3)}{S_d} G + CA$		9.11 mm		
	tut =	$\frac{4.9 \times D (H - 0.3)}{S_t}$		8.02 mm		
Find values of "x1, x2, & x3" for both the Design & Test conditions. Course No. 7						
Kd =	1.392	x1d =	570.944	Kt =	1.461	x1t = 589.208
Cd =	0.175	x2d =	787.548	Ct =	0.201	x2t = 906.346
H (m) =	4.50	xd3 =	637.857	H (m) =	4.50	xt3 = 598.354
		Use lowest value of 'xd' =	570.944 mm			Use lowest value of 'xt' = 589.208 mm
			= 0.571 m			= 0.589 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 8.59 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 7.47 \text{ mm}$$

2nd. Trial	tLd =	12.68 mm.	tLt =	11.71 mm		
	tud =	8.59 mm.	tut =	7.47 mm		

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.477	x1d =	608.169	Kt =	1.364	x1t = 524.695
Cd =	0.207	x2d =	932.921	Ct =	0.164	x2t = 737.507
H (m) =	4.50	xd3 =	619.270	H (m) =	4.50	xt3 = 577.386
		Use lowest value of 'xd' =	608.169 mm			Use lowest value of 'xt' = 524.695 mm
			= 0.608 m			= 0.525 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 8.52 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 7.59 \text{ mm}$$

3rd. Trial	tLd =	12.68 mm.	tLt =	11.71 mm		
	tud =	8.52 mm.	tut =	7.59 mm		

Find values of "x1, x2, & x3" for both the Design & Test conditions.

Kd =	1.489	x1d =	613.481	Kt =	1.375	x1t = 533.669
Cd =	0.212	x2d =	953.578	Ct =	0.168	x2t = 758.139
H (m) =	4.50	xd3 =	616.672	H (m) =	4.50	xt3 = 582.129
		Use lowest value of 'xd' =	613.481 mm			Use lowest value of 'xt' = 533.669 mm
			= 0.613 m			= 0.534 m

$$tdx = \frac{4.9 \times D (H - x/1000) G + CA}{S_d} = 8.51 \text{ mm}$$

$$ttx = \frac{4.9 \times D (H - x/1000)}{S_t} = 7.57 \text{ mm}$$

$$\text{Seventh course thickness} = 8.6 \text{ mm}$$

For the Eighth course.

1st. Trial	tLd =	8.51 mm.	tLt =	7.57 mm		
	tud =	$\frac{4.9 \times D (H - 0.3)}{S_d} G + CA$		4.77 mm		
	tut =	$\frac{4.9 \times D (H - 0.3)}{S_t}$		3.72 mm		

Figure 3.48 Illustration of the use of the "variable design point" method calculation - page 11

Find values of "x1, x2, & x3" for both the Design & Test conditions.				Course No.	8		
Kd =	1.785	x1d =	453.696	Kt =	2.034	x1t =	476.045
Cd =	0.310	x2d =	696.980	Ct =	0.378	x2t =	850.595
H (m) =	2.25	xd3 =	461.325	H (m) =	2.25	xt3 =	407.710
Use lowest value of 'xd' =				Use lowest value of 'xt' =			

3.6.7 Shell stiffening — wind girders

Having dealt with the differences in approach to designing shell thickness between the British and American Codes, the American approach to shell stiffening requirements is now considered.

3.6.7.1 Primary wind girders to API 650

The background for the requirements of primary wind girders to the API 650 Code are the same as for the BS Code and these have already been given in Section 3.5.1.

The API Code refers to top wind girders rather than primary wind girders and the formula for the required section modulus for the girder is the same as the BS formula except that it is presented in a slightly different format, as follows:

$$Z = \frac{D^2 H_2}{17} \quad \text{equ. 3.60}$$

where:

Z = required section modulus (cm^3)

D = nominal tank diameter (m)

H_2 = height of tank shell (m) including any freeboard provided above the maximum filling height as a guide for a floating roof

The constant $\frac{1}{17}$ equates to 0.058 used in the BS formula (see equation 3.22).

The formula is based on a wind speed of 100 mph and therefore must be modified for any other wind speed by multiplying the right hand side of the equation by $\left(\frac{V}{100}\right)^2$

where:

V = design wind speed (mph)

In SI units this becomes $\left(\frac{V}{44.7}\right)^2$ where V is in m/sec.

For tank diameters over 60 m, the section modulus required by equation 3.22 may be reduced by agreement between the purchaser and the manufacturer, but the modulus may not be less than that required for a tank diameter of 60 m.

As is the case for the BS Code, API requires that when the top wind girder is located more than 600 mm below the top of the shell, the tank shall be provided with a 60 x 60 x 5 mm top curb angle for shells with a top course thickness of 5 mm and a 80 x 80 x 6 mm angle for top courses more than 5 mm thick.

3.6.7.2 Secondary wind girders to API 650

Again, the theory behind the design of secondary wind girders (referred to as intermediate wind girders in the API Code) is the same as that given in Section 3.5.2 for the BS Code. However there are differences in the presentation of the formulae and the nomenclature used, as follows:

In the BS Code the maximum height of the unstiffened shell is given in equation 3.33 as:

$$H_p = K \left(\frac{t \min^5}{D^3} \right)^{1/2}$$

where:

K = $\frac{95,000}{3.563V_s^2 + 580 V_a}$

V_s = the design wind speed (m/sec)

V_a = the design vacuum (mbar)

The equivalent API formula is intended to apply to tanks with either open tops or closed tops and is based on the following factors taken from R.V. McGrath's *Stability of API 650 Standard Tank Shells*, (Reference 3.6).

- A design wind velocity (V) of 160 km/h (100 mph) which imposes a dynamic pressure of 1.23 kPa (25.6 lbf/ft²). The velocity is increased by 10% for either a height above ground or a gust factor; thus the pressure is increased to 1.48 kPa (31 lbf/ft²). An additional 0.24 kPa (5 lbf/ft²) is added to account for inward drag associated with open-top tanks or for internal vacuum associated with closed-top tanks. A total of 1.72 kPa (36 lbf/ft²) is obtained.

For the purposes of this Standard, this pressure is intended to be the result of a 160 km/h (100 mph) fastest mile velocity at approximately 9 m (30 ft) above ground. H_1 may be modified for other wind velocities, as specified by the purchaser, by multiplying the right side of the equation by $[(V_r / V)^2]$, where $V_r = 160$ km/h (100 mph). When a design wind pressure, rather than a wind velocity, is specified by the purchaser, the preceding increase factors should be added to the purchaser's specified wind pressure unless they are contained within the design wind pressure specified by the purchaser.

- The wind pressure being uniform over the theoretical buckling mode of the tank shell, which eliminates the need for a shape factor for the wind loading.
- The modified US Model Basin formula for the critical uniform external pressure on thin-wall tubes free from end loadings, subject to the total pressure specified in Item a.
- Other factors specified by the purchaser. When other factors are specified by the purchaser that are greater than the factors in Items a - c, the total load on the shell shall be modified accordingly and H_1 shall be increased by the ratio of 1.72 kPa (36 lbf/ft²) to the modified total pressure.

The resulting API formula is given as:

$$H_1 = 9.47 \sqrt[3]{\left(\frac{t}{D}\right)^3} \quad \text{equ. 3.61}$$

which is the same as:

$$H_1 = 9.47 \left(\frac{t \min^5}{D^3} \right)^{1/2} \quad \text{in the BS format.}$$

where:

H_1 = the vertical distance (m) between the intermediate wind girder and the top angle of the shell or top wind girder of an open top tank

t = the "as ordered" thickness (mm), unless otherwise specified, of the top shell course

D = nominal tank diameter (m)

Note: This implies that, unless directed otherwise by the purchaser, the tank designer can use the total, "as built" thickness of the top course in calculation without deducting from it any corrosion allowance which may have been included in the course thickness. The BS Code requires any corrosion allowance to be deducted from the top course thickness for this calculation.

For wind speeds other than 100 mph, H_1 is modified by multiplying the right hand side of equation 3.61 by $\left(\frac{100}{V}\right)^2$ where V is the design wind speed in mph.

For SI units this becomes $\left(\frac{44.7}{V}\right)^2$ where V is in m/sec.

To compare equations 3.33 and 3.61, consider a tank designed for a wind speed of 100 mph (44.7 m/s). The minimum

value for the partial internal vacuum used in the design of secondary wind girders to the BS Code is that quoted in the Code for open top, or non-pressure tanks, $V_a = 5$ mbar.

Then from equation 3.32

$$K = \frac{95,000}{3.563 \times 44.7^2 + 580 \times 5} = 9.482$$

This result is very similar to the constant of 9.47 derived for use in the API formula given in equation 3.61.

The principal difference between the Codes, is that the BS Code increases the value used for internal vacuum V_a for high-pressure tanks (56 mbar) to 8.5 mbar. Whereas no increase is required when designing for higher pressures when applying Appendix F of the API Code.

Applying the increased value of 8.5 mbar to equation 3.32 gives:

$$K = \frac{95,000}{3.563 \times 44.7^2 + 580 \times 8.5} = 7.884$$

This has the effect, for a given set of tank design parameters, to decrease the minimum allowable spacing of the girders on a high-pressure tank designed to the BS Code by 16.75% over the API requirements. Hence, depending upon the geometry of the tank, this could lead to an increase in the number of wind gliders required for the BS tank.

Section 3.5.2 showed how a tank shell of varying course thicknesses, designed to the BS Code, was transposed to a equivalent height shell having a constant thickness equal to the thickness of the top course.

The API Code follows exactly the same mathematical route in determining the equivalent, (or "transposed shell" as it is referred to in the API Code). Also the method for the determination of the number and positioning of the girders is the same as for the BS Code.

However, whereas the BS Code tabulates the required section for the secondary wind girders against ranges of tank diameters, the API Code requires the section modulus of the section to be calculated using the same equation as that used for the top girders (equation 3.60), except that the value for H is different. For Intermediate wind girders to the API Code:

$$Z = \frac{D^2 H_1}{17} \quad \text{equ 3.62}$$

where:

D = tank diameter (m)

H_1 = vertical distance (m) between the intermediate wind girder and the top angle of the shell, or the top wind girder of an open top tank (see equation 3.61)

Again, equation 3.62 is based on a wind speed of 100 mph.

For other wind speeds the right hand side of the equation is multiplied by $\left(\frac{V}{100}\right)^2$ where V is the required design wind speed.

For SI units this becomes $\left(\frac{V}{44.7}\right)^2$ where V is in m/sec.

The required section modulus for intermediate wind girders is based on the properties of chosen steel sections, which are attached to the shell. Normally rolled steel angles or channels are used but for larger girders, polygonal sections formed from folded plate are often used. (See Figures 3.30 and 3.31.)

When determining what steel section(s) is required to satisfy the section modulus given by equation 3.62, a portion of the

tank shell may be included in the calculation and the portion allowed is given by:

$$13.4\sqrt{D \times t} \quad \text{equ 3.63}$$

where

D = nominal tank diameter (m)

t = shell thickness (mm) at the point of attachment

The use in the API Code of equation 3.62 for determining the section size for intermediate wind girders usually results in larger section sizes than that required by Table 3 of the BS Code.

Comparisons between BS and API wind girder section requirements are given in Figure 3.49 for a range of tank diameters and minimum course thicknesses.

D (m)	T (mm)	equ 3.61 H_1 (m)	equ 3.62 Required Z (cm ³)	equ 3.63 $13.4\sqrt{D \cdot t}$	Selected section	Actual Z (cm ³)	Weight km/m circ.
15	6	14.374	190.250	127.124	200 x 100 x 10 angle	211.100	23.00
20	6	9.336	219.682	146.790	200 x 100 x 10 angle	229.800	23.00
25	6	6.681	245.612	164.116	200 x 100 x 12 angle	263.800	27.3
29	6	5.347	264.532	176.758	203 x 89 channel	273.180	29.78
36	8	7.936	605.031	227.406	Plate girder b = 370 wide	609.900	35.68
40	8	6.776	637.758	239.706	Plate girder b = 390 wide	657.390	37.43
45	8	5.679	676.445	254.247	Plate girder b = 400 wide	684.400	38.89
50	8	4.849	713.035	268.000	Plate girder b = 420 wide	734.510	38.32
55	8	4.203	747.838	281.081	Plate girder b = 430 wide	761.820	37.67
59	8	3.783	774.555	291.123	Plate girder b = 440 wide	788.660	37.19

D (m)	t (mm)	Selected section	Actual Z (cm ³)	Weight km/m circ.
15	6	100 x 65 x 8	64.7	9.94
20	6	125 x 75 x 8	95.3	12.20
25	6	125 x 75 x 8	95.3	12.20
29	6	125 x 75 x 8	95.3	12.20
36	8	150 x 90 x 10	173.9	18.20
40	8	150 x 90 x 10	173.9	18.20
45	8	150 x 90 x 10	173.9	18.20
50	8	200 x 100 x 12	314.4	27.30
55	8	200 x 100 x 12	314.4	27.3
59	8	200 x 100 x 12	314.4	27.3

Figure 3.49 Comparisons between BS and API wind girder section requirements

Note: Typical dimensions for plate girders made from formed plate are given in Figure 3.31.

The minimum thickness requirements for the top courses alter at differing tank diameters in each Code, so, in order to keep the comparisons on the same basis, tank diameters have been selected to fall into two of the top course minimum thickness categories, namely, 6 mm and 8 mm, for both Codes.

3.6.7.3 Comparison between British and American secondary wind girder requirements

The differing secondary wind girder requirements, between the British and American Codes, can be compared by designing a tank shell to both Codes using the same overall dimensions and design parameters.

Take the British tank design illustration in Section 3.5.2.3. Here it was demonstrated that the shell required two secondary wind girders, each being an angle section of 200 x 100 x 12 (27.3 kg/m).

Designing the shell to the American Code, and using the same design parameters (i.e. external floating roof tank 96 m diameter and 19 m high having eight 2.375 m wide courses), the shell is to be designed for a wind speed of 60 m/sec and the primary girder is 1 m down from the top of the shell.

Note: The shell, being over 60 m diameter, is designed to the "variable design point" method.

Also, due to the lower allowable stress for the American Code, which is based on the ultimate tensile stress of the shell material, rather than the minimum yield stress in the case of the British Code, the lower courses are thicker than those to the British Code, whereas the two upper courses are to the minimum allowable nominal thickness for construction purposes to the American Code. (i.e. 10 mm to API, and 12 mm to BS).

Course Number	h (m)	t (mm)	He (m)
1	1.375	10.0	1.375
2	2.375	10.0	2.375
3	2.375	14.1	1.006
4	2.375	19.2	0.465
5	2.375	24.7	0.248
6	2.375	28.8	0.169
7	2.375	39.2	0.078
8	2.375	40.7	.071
			5.787

However, as the stiffening requirements are being compared, rather than the differences in the shell thickness requirements, the upper two courses will be kept at the same thickness as that for the BS Code. The data used will therefore be as follows:

Course Number	h (m)	t (mm)	He (m)
1	1.375	12.0	1.375
2	2.375	12.0	2.375
3	2.375	14.1	1.006
4	2.375	19.2	0.465
5	2.375	24.7	0.248
6	2.375	28.8	0.169
7	2.375	39.2	0.078
8	2.375	40.7	.071
			5.787

The maximum spacing for stiffeners on the shell from equation 3.61 is:

$$H_1 = 9.47 \times 12 \sqrt{\left(\frac{12}{96}\right)^3} \times \left(\frac{44.7}{60}\right)^2 = 2.787 \text{ m}$$

$$\frac{H_e}{H_1} = \frac{5.787}{2.787} = 2.076$$

Therefore two secondary wind girders are required.

These girders are ideally spaced at $\frac{H_e}{3}$ apart = 1.929 m.

The first girder, when positioned 1.929 m down from the primary girder, is on a course of minimum thickness and is not within 150 mm of a horizontal girth weld. This position is acceptable.

The second girder is positioned 1.929 m below the first, i.e. at 3.858 m below the primary girder and in this position it is on the 14.1 mm thick course, which is not a course of minimum thickness and is also only 108 mm below a girth seam. On both counts its position must be adjusted.

Adjust the position for being on a course thicker than the minimum as follows:

$$\{3.858 - (1.375 + 2.375)\} \times \left(\frac{14.1}{12.0}\right)^{2.5}$$

$$+ (1.375 + 2.375) = 3.912 \text{ m below the primary girder}$$

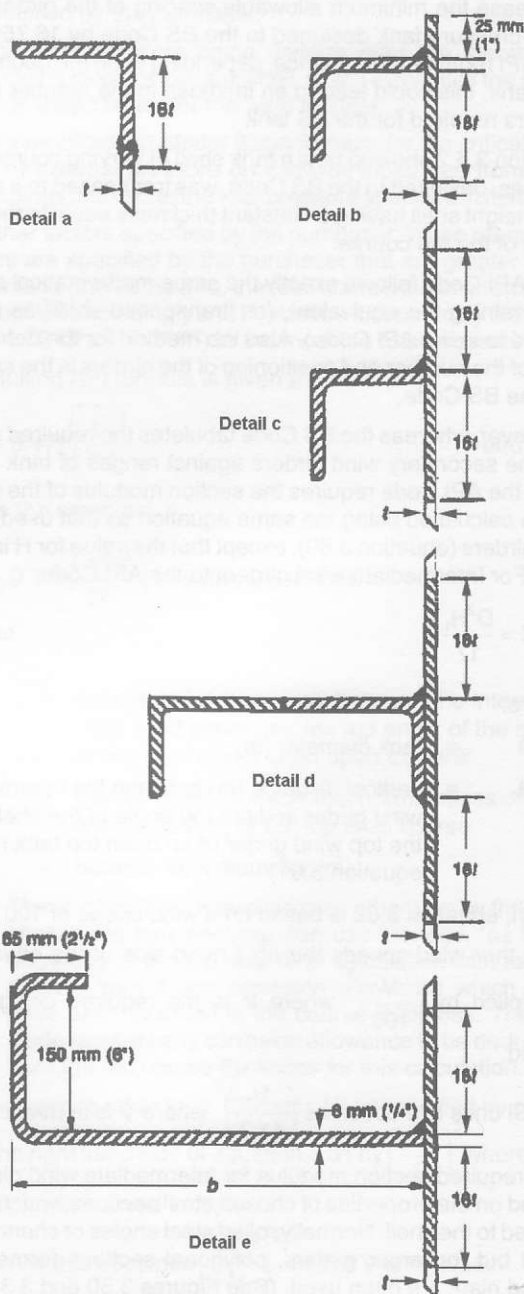


Figure 3.50 Typical stiffening ring sections for tank shells
From API 650, figure 3-20

Column 1		Column 2	Column 3	Column 4	Column 5	Column 6
Member Size		Shell Thickness [mm (in.)]				
mm	in.	5 (³ / ₁₆)	6 (¹ / ₄)	8 (⁵ / ₁₆)	10 (³ / ₈)	11 (⁷ / ₁₆)
Top Angle: Figure 3-20, Detail a						
64 × 64 × 6.4	2½ × 2½ × ¼	6.86 (0.41)	7.01 (0.42)	—	—	—
64 × 64 × 7.9	2½ × 2½ × ⅜	8.30 (0.51)	8.48 (0.52)	—	—	—
76 × 76 × 9.5	3 × 3 × ⅜	13.80 (0.89)	14.10 (0.91)	—	—	—
Curb Angle: Figure 3-20, Detail b						
64 × 64 × 6.4	2½ × 2½ × ¼	27.0 (1.61)	28.3 (1.72)	—	—	—
64 × 64 × 7.9	2½ × 2½ × ⅜	31.1 (1.89)	32.8 (2.04)	—	—	—
76 × 76 × 6.4	3 × 3 × ¼	38.1 (2.32)	39.9 (2.48)	—	—	—
76 × 76 × 9.5	3 × 3 × ⅜	43.0 (2.78)	52.6 (3.35)	—	—	—
102 × 102 × 6.4	4 × 4 × ¼	57.6 (3.64)	71.4 (4.41)	—	—	—
102 × 102 × 9.5	4 × 4 × ⅜	65.6 (4.17)	81.4 (5.82)	—	—	—
One Angle: Figure 3-20, Detail c (See Note)						
64 × 64 × 6.4	2½ × 2½ × ¼	28.5 (1.68)	29.6 (1.79)	31.3 (1.87)	32.7 (1.93)	33.4 (2.00)
64 × 64 × 7.9	2½ × 2½ × ⅜	33.1 (1.98)	34.6 (2.13)	36.9 (2.23)	38.7 (2.32)	39.5 (2.40)
102 × 76 × 6.4	4 × 3 × ¼	58.3 (3.50)	60.8 (3.73)	64.2 (3.89)	66.6 (4.00)	67.7 (4.10)
102 × 76 × 7.9	4 × 3 × ⅜	68.3 (4.14)	71.6 (4.45)	76.2 (4.66)	79.4 (4.82)	80.8 (4.95)
127 × 76 × 7.9	5 × 3 × ⅜	90.7 (5.53)	95.2 (5.96)	102.0 (6.25)	106.0 (6.47)	108.0 (6.64)
127 × 89 × 7.9	5 × 3½ × ⅜	101.0 (6.13)	106.0 (6.60)	113.0 (6.92)	118.0 (7.16)	120.0 (7.35)
127 × 89 × 9.5	5 × 3½ × ⅜	116.0 (7.02)	122.0 (7.61)	131.0 (8.03)	137.0 (8.33)	140.0 (8.58)
152 × 102 × 9.5	6 × 4 × ⅜	150.0 (9.02)	169.0 (10.56)	182.0 (11.15)	191.0 (11.59)	194.0 (11.93)
Two Angles: Figure 3-20, Detail d (See Note)						
102 × 76 × 7.9	4 × 3 × ⅜	186 (11.27)	191 (11.78)	200 (12.20)	207 (12.53)	210 (12.81)
102 × 76 × 9.5	4 × 3 × ⅜	216 (13.06)	222 (13.67)	233 (14.18)	242 (14.60)	245 (14.95)
127 × 76 × 7.9	5 × 3 × ⅜	254 (15.48)	262 (16.23)	275 (16.84)	285 (17.34)	289 (17.74)
127 × 76 × 9.5	5 × 3 × ⅜	296 (18.00)	305 (18.89)	321 (19.64)	333 (20.26)	338 (20.77)
127 × 89 × 7.9	5 × 3½ × ⅜	279 (16.95)	287 (17.70)	300 (18.31)	310 (18.82)	314 (19.23)
127 × 89 × 9.5	5 × 3½ × ⅜	325 (19.75)	334 (20.63)	350 (21.39)	363 (22.01)	368 (22.54)
152 × 102 × 9.5	6 × 4 × ⅜	456 (27.74)	468 (28.92)	489 (29.95)	507 (30.82)	514 (31.55)
Formed Plate: Figure 3-20, Detail e						
b = 250	b = 10	—	341 (23.29)	375 (24.63)	392 (25.61)	399 (26.34)
b = 300	b = 12	—	427 (29.27)	473 (31.07)	496 (32.36)	505 (33.33)
b = 350	b = 14	—	519 (35.49)	577 (37.88)	606 (39.53)	618 (40.78)
b = 400	b = 16	—	615 (42.06)	687 (45.07)	723 (47.10)	737 (48.67)
b = 450	b = 18	—	717 (48.97)	802 (52.62)	846 (55.07)	864 (56.99)
b = 500	b = 20	—	824 (56.21)	923 (60.52)	976 (63.43)	996 (65.73)
b = 550	b = 22	—	937 (63.80)	1049 (68.78)	1111 (72.18)	1135 (74.89)
b = 600	b = 24	—	1054 (71.72)	1181 (77.39)	1252 (81.30)	1280 (84.45)
b = 650	b = 26	—	1176 (79.99)	1317 (86.35)	1399 (90.79)	1432 (94.41)
b = 700	b = 28	—	1304 (88.58)	1459 (95.66)	1551 (100.65)	1589 (104.77)
b = 750	b = 30	—	1436 (97.52)	1607 (105.31)	1709 (110.88)	1752 (115.52)
b = 800	b = 32	—	1573 (106.78)	1759 (115.30)	1873 (121.47)	1921 (126.66)
b = 850	b = 34	—	1716 (116.39)	1917 (125.64)	2043 (132.42)	2096 (138.17)
b = 900	b = 36	—	1864 (126.33)	2080 (136.32)	2218 (143.73)	2276 (150.07)
b = 950	b = 38	—	2016 (136.60)	2248 (147.35)	2398 (155.40)	2463 (162.34)
b = 1000	b = 40	—	2174 (147.21)	2421 (158.71)	2584 (167.42)	2654 (174.99)

Note: The section moduli for Details c and d are based on the longer leg being located horizontally (perpendicular to the shell) when angles with uneven legs are used.

Figure 3.51 Section moduli of stiffening ring sections for tank shells (Values given in cm³ (in³))

From API 650, table 3-20

This position puts the girder 162 mm below the girth seam and therefore further adjustment is not required.

The spacing between the girders on the transposed shell is: 1.929 m, 1.983 m and 1.875 m = 5.787 m. These spacings are all less than H_1 at 2.787 m and therefore are acceptable.

The section sizes for the girders have now to be calculated.

From equation 3.62 the section modulus is calculated as follows:

For the upper secondary girder the value for H_1 is 1.929 m, and

$$Z = \frac{D^2 \cdot H_1}{17} \times \left(\frac{V}{44.7} \right)^2$$

$$= \frac{96^2 \times 1.929}{17} \times \left(\frac{60}{44.7} \right)^2 = 1884 \text{ cm}^3$$

Section type and size

Figure 3.50 shows typical stiffening ring sections and is taken from Table 3-30 of API 650 and typical values of section for various types of ring sections.

From equation 3.63 the participating portion of the shell plating which can be included in the calculation for the girder is:

$$13.4\sqrt{D \cdot t} = 13.4\sqrt{96 \times 12} = 455 \text{ mm}$$

Referring to Figure 3.50, a Detail 'e' type girder is required.

The table in Figure 3.51 does not have a shell thickness of 12 mm listed but at 11 mm the nearest Z value to 1884 cm³ is 1921 cm³ indicating that a minimum girder width of about 32 inches (813 mm) is required.

A detailed calculation gives an actual minimum width of 770 mm, giving a Z value of 1890 cm³.

For the lower secondary girder the value for H_1 is 1.983 m, and

$$Z = \frac{D^2 \cdot H_1}{17} \times \left(\frac{V}{44.7} \right)^2$$

$$= \frac{96^2 \times 1.983}{17} \times \left(\frac{60}{44.7} \right)^2 = 1937 \text{ cm}^3$$

The participating portion of shell is found to be 493 for the 14.1 mm plate, and the required Z value is 1937 cm³ indicating that a Detail 'e' type girder with a similar width to that for the upper girder is required.

A detailed calculation again shows that a minimum width of 770 mm, gives a Z value of 1940 cm³ for the 14.1 mm plate.

Both girders will have the same minimum cross section and it is found that if the girders are made in sections to match the number of shell plates there will be 32 polygonal sections per girder and these will each weigh an average of 50.64 kg/m of tank circumference.

Conclusion

The British design requires two girders each out of 200 x 100 x 12 x 27.3 kg/m angle, giving a total net weight of 16,467 kg.

The American design again requires two girders but of a much larger section made from 6 mm folded plate having an average fabricated weight of 50.64 kg/m giving a total net weight of 30,545 kg, which is 85% more than the British design.

Referring back to Morton's research in Section 3.5.2.2, it appears that the British Code has heeded his advice, which suggests that fairly small section girders give adequate stiffness to a shell, whereas the American Code seems not to have done so.

3.7 Compression area for fixed roof tanks

3.7.1 Effect of internal pressure

All closed tanks which are subjected to an internal pressure which is in excess of the weight of the roof plates, try to adopt a spherical form, whereby the meridional and latitudinal stresses at any given point in the containment parts would tend to equalise.

By way of illustration, the effect on a vertical cylindrical cone roof storage tank is shown in an exaggerated form in Figure 3.52.

Two critical areas of distortion become apparent:

- 1) The shell-to-bottom joint.
- 2) The shell-to-roof joint.

The distortion of the shell-to-bottom joint has already been discussed in Section 3.5.3 and the shell-to-roof joint is now considered.

The action of the pressure on the underside of the roof causes a compressive force to be induced in the shell-to-roof joint as shown in Figure 3.53.

The area in the vicinity of this connection needs to be strong enough to withstand the compressive force in order to prevent a buckling failure taking place as shown in Figure 3.54.

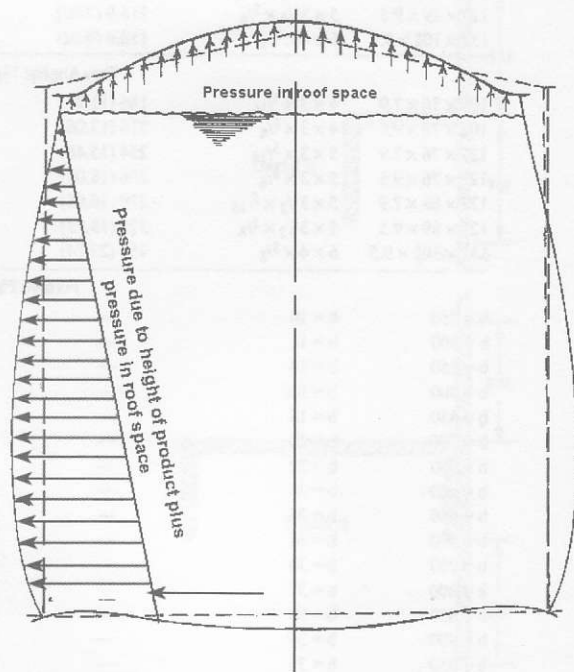


Figure 3.52 Diagrammatic illustration of a pressurised tank

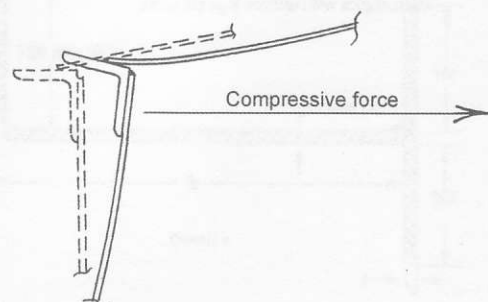


Figure 3.53 Compressive force at shell-to-roof joint

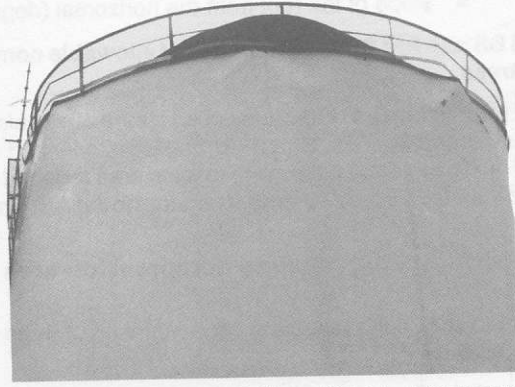


Figure 3.54 An example of a failed shell-to-roof joint due to internal pressure
Courtesy of EEMUA

3.7.2 Derivation of the required compression zone area

The compression area which is required is derived as follows:

The load acting normal to the underside of the roof

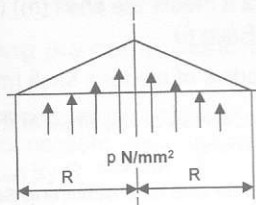
$$= p \cdot \pi \cdot R^2 \text{ (N)}$$

The circumference of the shell

$$= 2 \cdot \pi \cdot R \text{ (mm)}$$

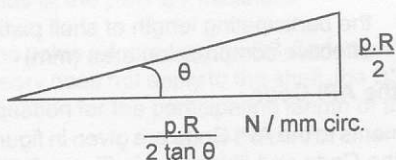
Then the vertical force in the shell

$$= \frac{p \cdot \pi \cdot R^2}{2 \cdot \pi \cdot R} = \frac{p \cdot R}{2} \text{ (N/mm circ.)}$$



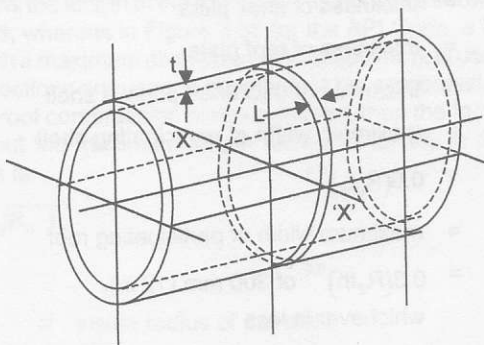
The horizontal component of this vertical force is found as:

Where θ is the angle between the roof and the horizontal, at the point where the roof meets the shell.



Consider an elemental ring of the tank shell having a thickness t of 1 mm and a length L of 1 mm and resolve the forces acting at axis XX .

Consider a unit cube of this ring, then the force F acting



$$= \frac{p \cdot R}{2 \tan \theta} \text{ N/mm circ.}$$

As this force is acting on area $t \times L$ (1 mm x 1 mm), it becomes a pressure

$$P = \frac{p \cdot R}{2 \tan \theta} \text{ N/mm} \quad \text{equ 3.64}$$

To find the circumferential (hoop) stress in the ring of diameter $2R$ and length L , proceed as follows:

The load on the elemental horizontal strip at axis

$$XX = \text{pressure} \times \text{area}$$

$$= P \times 2R \times L$$

$$\text{equ 3.65}$$

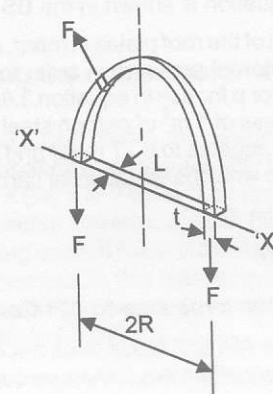
The force in the ring resisting this load at axis

$$XX = \text{stress} \times \text{area}$$

$$= S_c \times 2(t \times L)$$

$$\text{equ 3.66}$$

where S_c is the stress.



The load given by equation 3.65 must equate to the force given in equation 3.66 and therefore.

$$S_c \times 2 \times t \times L = P \times 2R \times L$$

Substituting equation 3.64 for P :

$$S_c \times 2 \times t \times L = \frac{p \cdot R}{2 \tan \theta} \times 2R \times L$$

Then:

$$t \times L = \frac{p \cdot R^2 \cdot L}{S_c \cdot \tan \theta}$$

The cross-sectional area A for the ring

$$= t \times L$$

but as both t and L are both 1 mm, then:

$$A = \frac{p \cdot R^2}{S_c \cdot \tan \theta} \quad \text{equ 3.67}$$

3.7.2.1 Effect of roof slope on cross-sectional area

It can be seen from equation 3.67 that for a given tank radius and pressure, the lower the slope of the roof, the lower the value for $\tan \theta$ and in consequence a higher value for the compression zone area is required. This is an important factor when designing "frangible" roof joints, which is discussed in Section 3.8.

3.7.3 Compression zones

3.7.3.1 Compression zone area to BS Code

In the BS Code the units which apply to equation 3.67 are:

$$A = \text{area to be provided within the compression zone (mm}^2\text{)}$$

- p = internal pressure in the roof space less the weight of the roof plates (mbar)
 R = radius of the tank shell (m)
 Sc = allowable compressive stress (N/mm²)
 θ = the angle between the roof and the horizontal, at the point where the roof meets the shell (degrees)

Note: The BS Code states that, unless otherwise specified, the value for Sc shall be taken as 120 N/mm².

p in mbar must be converted to N/mm² by multiplying by 0.0001 and R is converted from metres to millimetres. The equation then becomes:

$$A = \frac{p \times 0.0001 \times R^2 \times 1000^2}{2 \times Sc \times \tan \theta}$$

$$A = \frac{50 \cdot p \cdot R^2}{Sc \cdot \tan \theta} \quad \text{equ 3.68}$$

That is how the equation is shown in the BS Code.

Note: The weight of the roof plates in mbar, must be deducted from the internal pressure in order to arrive at the correct value for p for use in equation 3.68. The weight of 1 mm thickness of 1 m² of carbon steel plate is 7.85 kg, or 77N which equates to 0.77 mbar and so a more convenient way to write the equation for carbon steel tanks is:

$$A = \frac{50(p - 0.77tr) \cdot R^2}{Sc \cdot \tan \theta} \quad \text{equ 3.69}$$

3.7.3.2 Compression zone area to API Code

The basic American API 650 Code does not cater for pressurised tanks but merely stipulates minimum curb angle requirements for various sizes of tanks and these are given in Section 3.7.9.1, Figure 3.59.

However Appendix F of this Code caters for pressurised tanks and gives requirements for roof-to-shell compression zones.

Appendix F follows the same theory as that for the BS Code but in the API Code the tank diameter D in metres is used instead of the radius and the internal pressure p is expressed in kilopascals (kPa) instead of mbar, and as 1 kPa = 0.001 N/mm² the equation in the API Code becomes:

$$A = \frac{p \times 0.001 \times \left(\frac{D}{2} \times 1000\right)^2}{2 \times Sc \times \tan \theta} = \frac{125 \cdot p \cdot D^2}{Sc \cdot \tan \theta}$$

The API Code uses a value of 137.5 N/mm² (20,000 lbs/in²) for Sc and the equation reduces to:

$$A = \frac{p \cdot D^2}{1.1 \cdot \tan \theta} \quad \text{equ 3.70}$$

The value used for p is the internal pressure less the weight of the roof plates expressed in kPa and the API Code deems that 1 mm thickness of 1 m² of carbon steel plate weighs 0.08 kPa, then the formula becomes:

$$A = \frac{D^2(p - 0.08 th)}{1.1 \cdot \tan \theta} \quad \text{equ 3.71}$$

This is how the equation is shown in the API Code.

where:

- A = area to be provided within the compression zone (mm²)
 p = internal pressure in the roof space (kPa)
 D = diameter of the tank shell (m)
 th = thickness of the roof plates (mm)

- θ = slope of the roof from the horizontal (degrees)

3.7.3.3 BS and API Code differences of allowable compressive stress

Due to the difference in the values used for the allowable compressive stress S , (120 N/mm² in the BS Code and 137.5 N/mm² in the API Code), the compression area required to the BS Code is 14.6% greater than that required to the API Code.

3.7.4 Providing the required compression area

The roof-to-shell compression zone is made up of three basic components:

- 1) A participating area of the roof plating
- 2) A participating area of the shell plating
- 3) If required, the above areas can be augmented by adding steel sections at the roof-to-shell junction

In the case of 1) and 2) these areas may be increased by thickening up the plating in the area local to the joint. Additional steel sections, when added into the compression zone, must fall within the participating area of the shell plating. The areas which are considered to comprise the compression zone are illustrated in Figures 3.55, 3.56 and 3.57.

3.7.4.1 For the BS Code

The requirements to the BS Code are given in figure 7 of the Code and illustrated in Figure 3.55:

where:

- R_1 = the radius of curvature of the roof at the point where it meets the shell (m) (for conical roofs $R_1 = R/\sin \theta$)
 R = the radius of the tank shell (m)
 t = the thickness of the shell in the compression zone (mm)
 t_a = the thickness of a stiffening section (mm)
 t_r = the thickness of the roof plate in the compression zone (mm)
 W_h = the participating length of roof plating in the effective compression area (mm)
 W_c = the participating length of shell plating in the effective compression area (mm)

3.7.4.2 For the API Code

The requirements to the API Code are given in figure F-2 of Appendix F of the Code and illustrated in Figure 3.56:

where:

- t_a = thickness of angle leg
 t_b = thickness of bar
 t_c = thickness of shell plate
 t_h = thickness of roof plate
 t_s = thickness of thickened plate in shell
 W_c = maximum width of participating shell
 $\quad = 0.6(R_c t_s)^{0.5}$
 W_h = maximum width of participating roof
 $\quad = 0.3(R_2 th)^{0.5}$ of 300 mm (12 in), whichever is less
 R_c = inside radius of tank shell

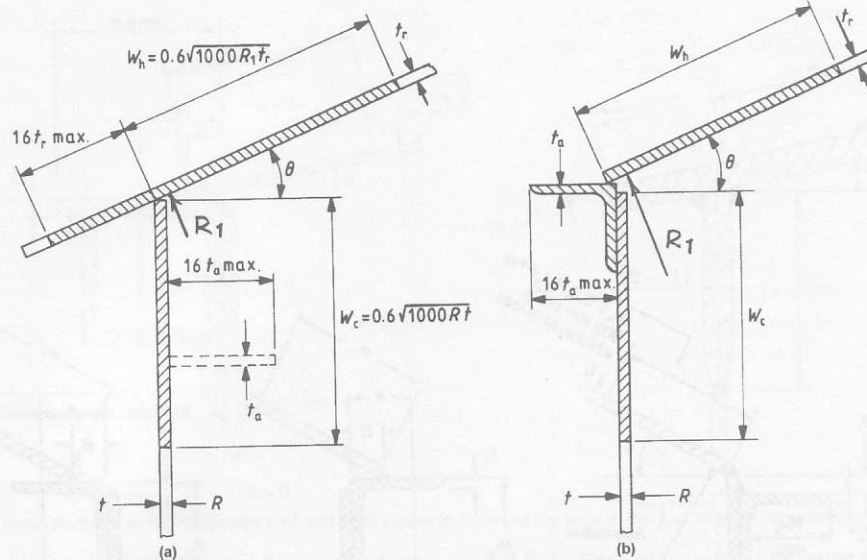


Figure 3.55 Shell-to-roof compression areas to BS 2654
From BS 2654, figure 7

$$R_2 = \text{length of the normal to the roof, measured from the vertical centreline of the tank}$$

$$= \frac{R_c}{(\sin \theta)}$$

Note: All dimensions and thicknesses are in millimetres and (inches).

Further examples for increasing the area in the roof-to-shell compression zone are given in Figure 3.57.

3.7.5 Establishing the compression area

The formulae for calculating the values W_h and W_c for the various roof-to-shell connections are arrived at empirically through research carried out by R. Perono, (Reference 3.7).

The increase in pressure in the roof space causes an upward deflection of the roof plating. Perono assumed the shape of this deflection to be parabolic in the region close to the shell and deduced that the length concerned was proportional to

$$0.6\sqrt{\text{Radius of the plating} \times \text{thickness}}$$

and this is the value adopted by the BS Code for W_h . Although the same theory does not apply to the shell, the BS Code uses the same equation for the participating length of the shell plating W_c .

3.7.6 API limitations for the length of the roof compression area

It is interesting to note that the BS Code uses a single factor of 0.6 for W_h the length of the roof compression area shown in Figure 3.55, whereas in Figure 3.56 for the API Code, a factor of 0.3, (with a maximum allowable value of 300 mm), is used when angle sections are used to supplement the compression area. Where roof compression plates are used, then the factor used is 0.6 but the maximum length allowable for W_h in these instances is:

$$0.9\sqrt{R_c \cdot t_b}$$

where:

- R_c = inside radius of the shell
 t_b = thickness of the roof compression plate

3.7.7 Calculating the compression zone area

When applying the above theory, the designer will calculate the W_h and W_c participating plate lengths and hence the available area as $(W_h \times t_r) + (W_c \times t)$. This is then compared with the required area from either equation 3.68 or 3.71 depending upon which Code is being used. If there is a deficiency, consideration may be given to redressing this deficiency by adding in one or more steel sections or thickened plates at tie joint as shown in Figures 3.55, 3.56 and 3.57.

Thickened plates may be used for either the roof or the shell section or for both together, depending upon the amount of additional area, which is required. When adopting this method it must be remembered that the participating length of the compression area W_h and/or W_c has to be recalculated using the new thicker plate chosen for the roof and/or shell section and this greater value is then multiplied by the thicker plate thus giving a larger compression area.

3.7.8 Practical considerations

The most suitable method for providing the required area for a particular application is found by trying various combinations of the available steel sections. For additional area requirements of up to say 9000 mm², angle sections can be used. Beyond this then horizontally disposed plate stiffeners and/or thickened shell and roof plate sections have to be considered.

If thickened sections of shell or roof plate are decided upon, then it should be borne in mind, that from a practical and commercial point of view, it is considered cheaper to produce a thickened shell plate section than roof section. This is because, unless flat bar can be sourced, the development of the cone frustum from rectangular plate is wasteful in terms of material. Also the labour involved in marking off, cutting and rolling the conical section, is more than that required for the cylindrical shell section. This is demonstrated later in Section 3.7.10.3.

3.7.9 Minimum curb angle requirements

For small diameter, or non-pressure tanks, (to the BS Code), the calculated compression area may be so small that it can be catered for by the allowable compression areas of the shell and roof plating alone. Therefore it can be argued that for these cases there is no need to introduce additional area at the joint in the form of a curb angle.

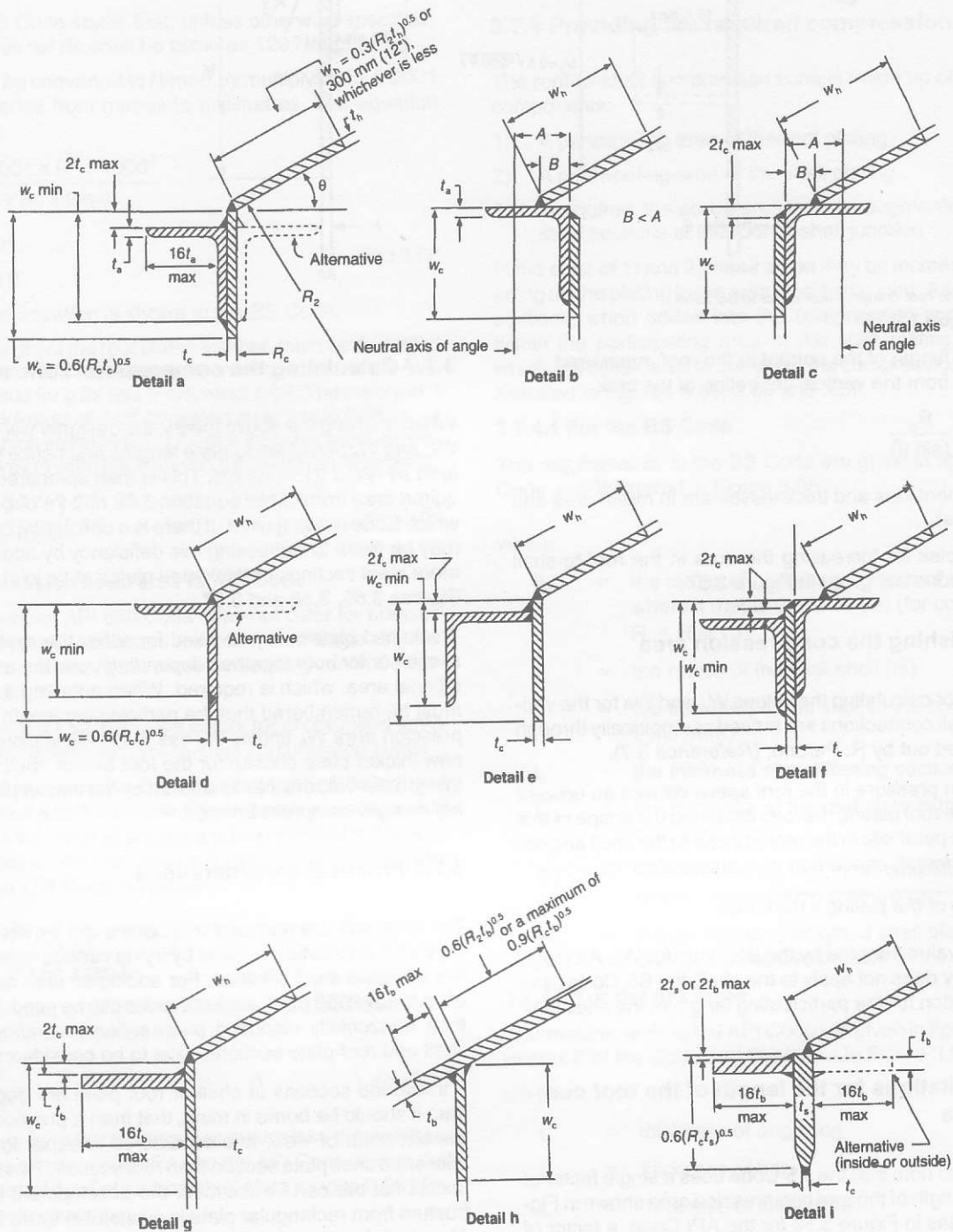


Figure 3.56 Roof-to-shell compression areas to API 650
From API 650, Appendix F

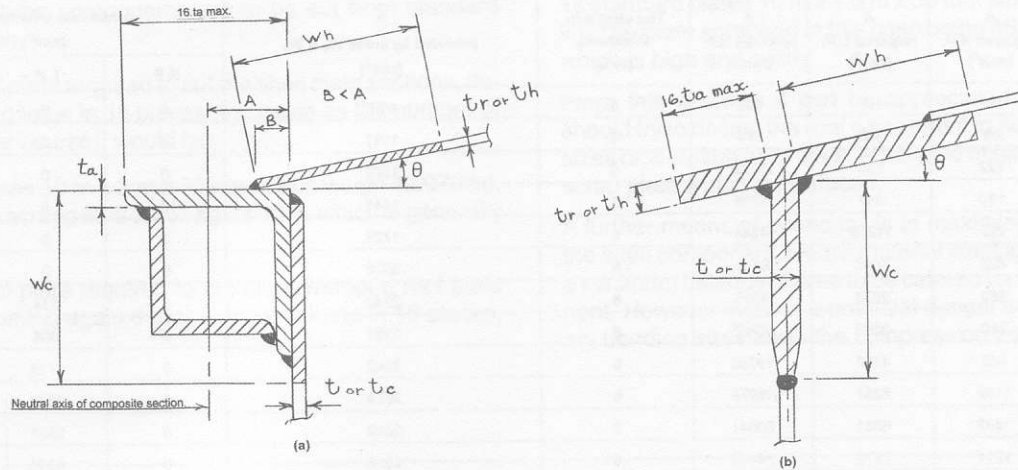


Figure 3.57 The use of two angle sections or two thickened roof and shell plates to increase the area in the roof-to-shell compression zone

From a practical point of view, both the BS and API Codes take the view that for construction purposes, (unless there are special circumstances which are given in Section 3.7.9.2), then tanks must be provided with a top curb angle of a certain minimum size.

The reason for this is to:

- Maintain shell circularity during construction
- Give a landing for the roof plating
- Give a landing for the roof handrail stanchions (where fitted)

3.7.9.1 Minimum curb angle sizes for fixed roof tanks

In the BS Code, the minimum size of curb angle which shall be fitted to the tank shall be that derived from equation 3.68 or as given in Table 4 of the Code (Figure 3.58) whichever is the greater.

Tank diameter (m)	Minimum size curb angle (mm)
≤ 10	60 x 60 x 6
$>10 \leq 20$	60 x 60 x 8
$>20 \leq 36$	80 x 80 x 10
$>36 \leq 48$	100 x 100 x 12
> 48	150 x 150 x 10

Figure 3.58 Minimum size of curb angle from BS 2654

The corresponding requirements to the API 650 Code are given in clause 3.1.5.9 of the Code and are shown in Figure 3.59.

Tank diameter (m)	Minimum size curb angle (mm)
≤ 11	50 x 50 x 5
$>11 \leq 18$	50 x 50 x 6
> 18	80 x 80 x 10

Figure 3.59 Corresponding requirements API 650 for minimum curb angle

3.7.9.2 Cases where minimum curb angle requirements do not apply

The stipulations given in Figures 3.58 and 3.59 do not apply to the following:

- Open top tanks.
- Tanks having self-supporting roofs to API 650 — these are governed by specific requirements given in clauses 3.10.5 and 3.10.6 of API 650 which can result in roof-to-shell connections as 'detail a' of Figure 3.55 or 'detail h' of Figure 3.56.
- For the API Code only. — Tanks ≤ 9 m diameter which have the top angle formed by flanging the top edge of the shell as shown in Figure 3.60.

3.7.9.3 Effect of internal pressure and tank diameter on required compression area

For the BS Code, the effect of the varying internal design pressure for a range of tank diameters is demonstrated in Figure 3.61, for the following tank design parameters.

Tank pressure type	Non	Low	High	
Internal pressure $p =$	7.5	20	56	mbar
Roof plate thickness $t_r =$	5	5	5	mm
P - wt. Roof plates $=$	3.65	16.15	52.15	mbar
Roof slope 1 in ?	5	5	5	
$\tan \theta =$	0.2	0.2	0.2	
$\sin \theta =$	0.1961	0.1961	0.1961	
$S_c =$	120	120	120	N/mm ²

From equation 3.68:

$$A_{\text{required}} = \frac{50 \cdot p \cdot R^2}{S_c \cdot \tan \theta}$$

From Figure 3.55, the available roof plate area

$$= W_h \cdot t_r = 0.6 \sqrt{1000 \cdot R \cdot t_r} \times t_r \quad \text{equ 3.72}$$

The available shell plate area

$$= W_c \cdot t = 0.6 \sqrt{1000 \cdot R \cdot t} \times t \quad \text{equ 3.73}$$

From Figure 3.65 it can be seen how the compression zone/requirements increase dramatically over the range of tank diameters, when moving from a non-pressure through to a high-pressure rating. This is because, in equation 3.68 the pressure increases by a factor of 14.29 i.e. $\frac{52.15}{3.65}$ linearly whilst the value for the tank radius is being squared.

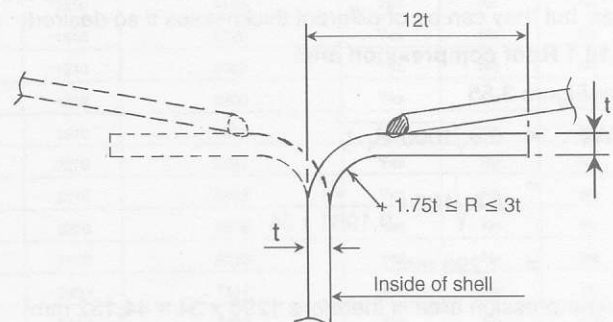


Figure 3.60 Top edge of shell flanged to form a landing for the roof plates

Tank diameter (m)	A required N.P. (mm ²)	A required L.P. (mm ²)	A required H.P. (mm ²)	Top shell min. thickness (mm)	A provided by areas W _h & W _c (mm ²)	Additional area required (mm ²)		
						N.P.	L.P.	H.P.
4	30	135	435	5	977	0	0	0
6	68	303	978	5	1197	0	0	0
8	122	538	1738	5	1382	0	0	356
10	190	841	2716	5	1545	0	0	1171
12.5	297	11314	4244	5	1728	0	0	2516
15	428	1893	6111	6	2076	0	0	4036
17.5	582	2576	8318	6	2242	0	334	6076
20	760	3365	10865	6	2397	0	968	8468
22.5	962	4258	13750	6	2542	0	1716	11209
25	1188	5257	16976	6	2679	0	2578	14296
27.5	1438	6361	20541	6	2810	0	3551	17731
30	1711	7570	24445	8	3518	0	4052	20927
33	2070	9160	29579	8	3690	0	5470	25889
36	2464	10901	35201	8	3854	0	7047	31347
39	2891	12794	41313	8	4011	0	8783	37301
42	3353	14838	47913	8	4163	0	10675	43750
45	3850	17033	55002	8	4309	0	12725	50693
48	4380	19380	62580	8	4450	0	14930	58130
51	4945	21878	70647	8	4587	358	17291	66060
54	5543	24528	79203	8	4720	824	19808	74483

Figure 3.61 Varying internal design pressure for a range of tank diameters

Hence, large diameter, high-pressure tanks require to be heavily stiffened at the roof-to-shell joint to prevent compressive failure in this area. Figure 3.62 shows the results from Figure 3.61 in graph form.

The effect of imposing a mandatory requirement for the provision of a minimum size of curb angle is shown in Figure 3.63.

Figure 3.78 shows that for the full range of non-pressure tanks selected, the minimum curb angle requirement satisfies the design area required for the compression zone for all the tanks. However this is not the case for all the low and high-pressure tanks and most of these will have to be provided with sections having larger cross-sectional areas.

3.7.10 Design example

Consider the 54 m diameter, high-pressure tank designed to BS 2654, details of which are shown in Figure 3.63.

The required roof-to-shell compression area is 79203 mm².

The range of angle sizes which are readily available are not large enough to satisfy the area which is required and so the use of thickened roof and shell plates will be employed.

By a trial and error method, a suitable arrangement can be found by using the maximum allowable roof and shell lengths together with a plate thickness of 34 mm, which will satisfy the total area requirement. For ease of calculation the same thickness plate has been used here for both the roof and shell plate areas, but they can be of different thicknesses if so desired.

3.7.10.1 Roof compression area

From Figure 3.55

$$\begin{aligned}
 W_h &= 0.6\sqrt{1000 \cdot R_1 \cdot t} \\
 &= 0.6\sqrt{100 \times \frac{27}{0.1961 \times 34}} \\
 &= 1,298 \text{ mm}
 \end{aligned}$$

The compression area is therefore $1298 \times 34 = 44,132 \text{ mm}^2$

The maximum allowable outstand of the roof plate beyond the shell is $16t$ which in this case is $16 \times 34 = 544 \text{ mm}$.

The area of this section is $544 \times 34 = 18,496 \text{ mm}^2$

Then the total roof compression area =
 $44,132 + 18,496 = 62,628 \text{ mm}^2$

3.7.10.2 Shell compression area

From Figure 3.55

$$\begin{aligned}
 W_h &= 0.6\sqrt{1000 \cdot R \cdot t} \\
 &= 0.6\sqrt{100 \times 27 \times 34} \\
 &= 575 \text{ mm}
 \end{aligned}$$

The shell compression area = $575 \times 34 = 19,550 \text{ mm}^2$.

The total of the roof and shell compression areas available

$$= 62,628 + 19,550$$

$$= 82,178 \text{ mm}^2$$

This is acceptable, although 2,975 mm² more than required. By reducing the roof plate outstand beyond the shell to 457 mm reduces the area by $(544 - 457) \times 34 = 2,958$.

This then gives a total compression area of 79,220 mm², which is acceptable.

3.7.10.3 Rationalising the calculation

The above example is based on using the maximum allowable participating lengths for W_h and W_c in the roof and shell area calculations. Using the maximum value for W_h resulted in a plate thickness of 34 mm being the ideal thickness to suit the calculated lengths. But 34 mm is not considered to be a "standard" thickness and 35 mm thick plate would be more appropriate. Repeating the above calculations for 35 mm plate and using appropriately chosen values for W_h and W_c , the result given in Figure 3.64 is obtained.

3.7.10.4 Economy of design

The net weight of the components is:

for the shell	29,703 kg
for the roof	77,281 kg
Total net weight	106,984 kg

Assuming that the components are to be cut from standard plate sizes then:

The amount of plate required to cut the shell plate sections, assuming the ring to be in 18 pieces (the same as the number of shell plates per course), would be:

6 standard plates 10 m x 2 m x 35 mm which weigh 32,970 kg. The plate thus scrapped is 3,267 kg. or 10%, which is generally acceptable.

The amount of plate required to cut the developed roof plate sections, assuming again that the ring would be in 18 pieces, would be:

18 standard plates 10 m x 2.5 m x 35 mm which weigh 123,638 kg. The plate scrapped in this case being 46,357 kg. or 37.5%, which is high and costly.

From this exercise it can be appreciated that the designer should try to design the roof component to suit standard flat bar sizes or, if cutting from plate, attempt to minimise the amount of scrap plate which is produced.

A further means of economy, is to maximise the area put into the shell component, where material wastage is lower, leaving a minimum balance of area to be catered for by the roof component. However there is a potential danger of inducing secondary bending stresses in the compression zone due to the cen-

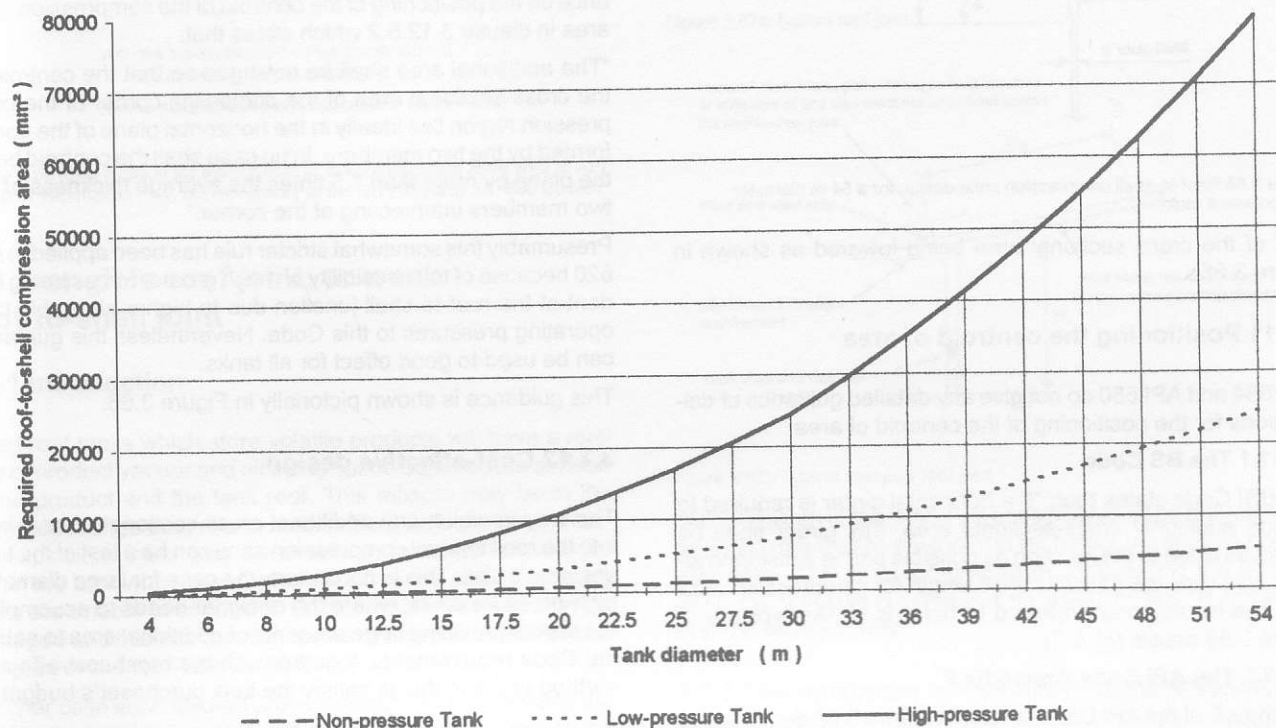


Figure 3.62 Comparison of roof-to-shell compression area requirements

Tank diameter (m)	A required N.P. (mm ²)	A required L.P. (mm ²)	A required H.P. (mm ²)	Top shell min. thickness (mm)	Min. curb size to Code (mm ²)	Area of min. size curb (mm ²)	A provided by min. size curb + areas W _h & W _c (mm ²)	Is the min. size curb sufficient? (mm ²)		
								N.P.	L.P.	H.P.
4	30	135	435	5	60 x 60 x 6	691	1668	Yes	Yes	Yes
6	68	303	978	5	60 x 60 x 6	691	1888	Yes	Yes	Yes
8	122	538	1738	5	60 x 60 x 6	691	2073	Yes	Yes	Yes
10	190	841	2716	5	60 x 60 x 6	691	2236	Yes	Yes	No
12.5	297	1314	4244	5	60 x 60 x 8	903	2631	Yes	Yes	No
15	428	1893	6111	6	60 x 60 x 8	903	2979	Yes	Yes	No
17.5	582	2576	8318	6	60 x 60 x 8	903	3145	Yes	Yes	No
20	760	3365	10865	6	60 x 60 x 8	903	3300	Yes	No	No
22.5	962	4258	13750	6	80 x 80 x 10	1510	4052	Yes	No	No
25	1188	5257	16976	6	80 x 80 x 10	1510	4189	Yes	No	No
27.5	1438	6361	20541	6	80 x 80 x 10	1510	4320	Yes	No	No
30	1711	7570	24445	8	80 x 80 x 10	1510	5028	Yes	No	No
33	2070	9160	29579	8	80 x 80 x 10	1510	5200	Yes	No	No
36	2464	10901	35201	8	80 x 80 x 10	1510	5364	Yes	No	No
39	2891	12794	41313	8	100 x 100 x 12	2270	6281	Yes	No	No
42	3353	14838	47913	8	100 x 100 x 12	2270	6433	Yes	No	No
45	3850	17033	55002	8	100 x 100 x 12	2270	6579	Yes	No	No
48	4380	19380	62580	8	100 x 100 x 12	2270	6720	Yes	No	No
51	4945	21878	70647	8	150 x 150 x 10	2930	7517	Yes	No	No
54	5543	24528	79203	8	150 x 150 x 10	2930	7650	Yes	No	No

Figure 3.63 Total compression zone areas, including minimum curb angle sizes

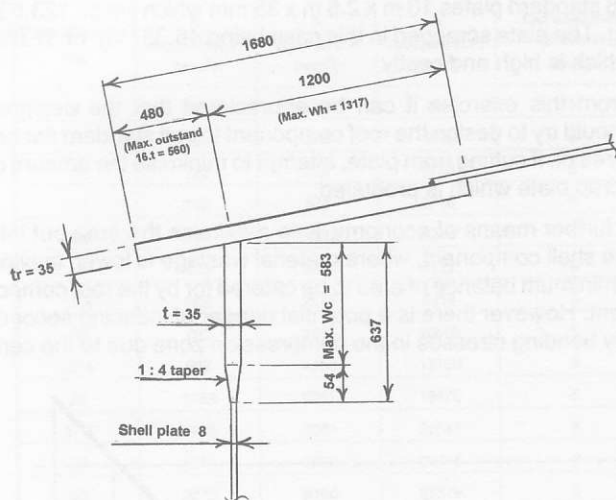


Figure 3.64 Roof-to-shell compression zone design for a 54 m diameter high-pressure tank

centroid of the cross sectional area being lowered as shown in Figure 3.65b.

3.7.11 Positioning the centroid of area

BS 2654 and API 650 do not give any detailed guidance or calculations for the positioning of the centroid of area.

3.7.11.1 The BS Code

The BSI Code states that: "If a horizontal girder is required to provide additional cross-sectional area, this girder shall be placed as close to the junction as possible and at a distance always less than the effective shell length for compression area W_c ". The arrangement referred to here, is shown typically in Figure 3.56 details (g) & (i)

3.7.11.2 The API Code Appendix F

Appendix F of the API Code, shows in Figure 3.56 detail 'b' and 'c' that the roof plate connection point on to the horizontal leg of the curb angle shall be between the position of the vertical neutral axis of the angle and the heel of the angle.

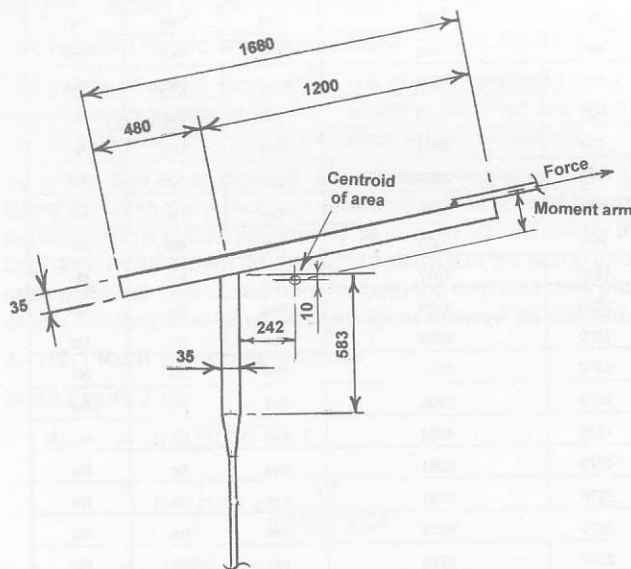


Figure 3.65a Compression zone having roof and shell plates of the same thickness

More specific guidance is given for tanks having dome roofs and self-supporting cone roofs, i.e. roofs without internal supporting structures. In these cases clause F7 states that the participating compression area shall be in accordance with clause 3.12.4 of the API Standard 620, "Design and Construction of Large, Welded, Low-Pressure Storage Tanks."

Except that the allowable compressive stress stated in API 620, shall be increased from 105 N/mm² (15,000 lbs/in²) to 140 N/mm² (20,000 lbs/in²)

3.7.11.3 Guidance on the positioning the centroid of area

Having mentioned API 620, which incidentally, allows design pressures up to 1035 mbar (15 lbs/in²). This Code gives guidance on the positioning of the centroid of the compression zone area in clause 3.12.5.2 which states that:

"The additional area shall be arranged so that the centroid of the cross-sectional area of the composite corner of the compression region lies ideally in the horizontal plane of the corner formed by the two members. In no case shall the centroid be off the plane by more than 1.5 times the average thickness of the two members intersecting at the corner."

Presumably this somewhat stricter rule has been applied in API 620 because of the possibility of much greater forces being evident at the roof-to-shell junction due to higher allowable tank operating pressures to this Code. Nevertheless this guidance can be used to good effect for all tanks.

This guidance is shown pictorially in Figure 3.66.

3.7.12 Cost-effective design

The way in which any additional cross-sectional area is built into the roof-to-shell compression zone can be a test of the tank designer's skill. This is particularly the case for large diameter, high-pressure tanks, where the designer needs to accomplish the task of providing large amounts of additional area to satisfy the Code requirements, together with the most cost-effective method of doing this to satisfy the tank purchaser's budget.

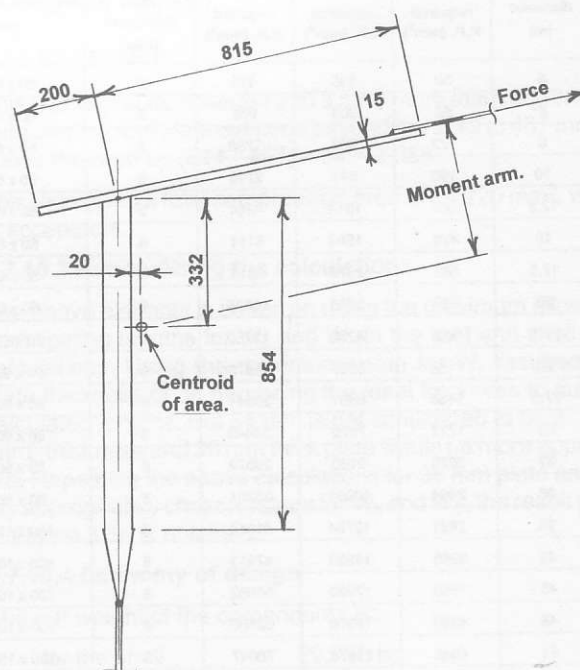


Figure 3.65b Compression zone with the shell thickness much greater than the roof

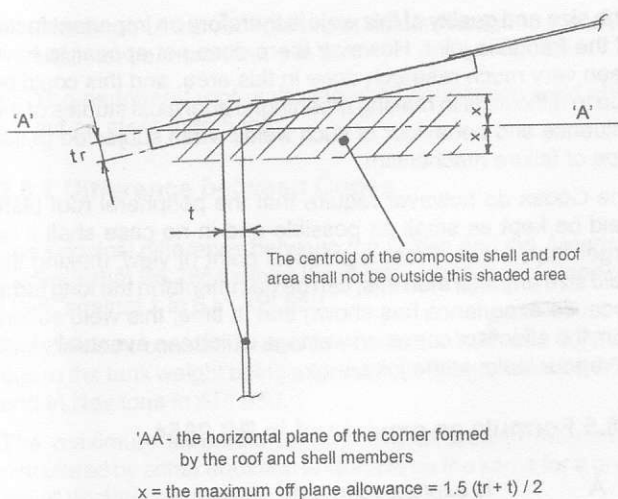


Figure 3.66 Ideal location for the centroid of the compression zone area to API 620. (For information only, not mandatory to the BS 2654 and API 690 Codes)

3.8 Frangible roof joint, or weak roof-to-shell joint

3.8.1 Introduction

Fixed roof tanks which store volatile products will have a mixture of product vapour and air in the space between the surface of the product and the tank roof. This mixture may be in the flammable range and, due to malfunction, external fire or internal explosion, there may be a sudden increase in pressure within the tank which the normal vent devices and emergency vents are unable to cope with. Consequently the tank may be damaged and this can result in failure of either the shell-to-bottom joint or the roof-to-shell joint.

In either case such failures are disastrous but the failure of the shell-to-bottom joint can be particularly horrendous due to the release of the stored product over the surrounding area causing the attendant ecological and environmental problems.

Of the two types of failure, the roof-to-shell failure is to be preferred, as this will normally create sufficient free-venting area to allow the release of the tank over-pressurisation without any loss of stored product. To increase the likelihood of a preferential roof-to-shell failure, some fixed roof tanks can be provided with a weak roof-to-shell connection, known as a "frangible roof joint". A typical arrangement of this type of joint is shown in Figure 3.67b.

3.8.2 Frangible roof joint theory

Assuming an empty cone roof tank, then, as the pressure in the tank increases above atmospheric pressure, a point will be reached when the upward force on the roof plating will equal the downward load due to the weight of the roof plating. As the pressure increases further, the roof plating will lift off its support structure and this further increase in pressure is withstood by tensile membrane forces 'T' in the roof plating (see Figure 3.68). These forces exert a pull at the shell-to-roof junction and so induce compressive forces in this area.

A point will be reached when the upward force due to further increase in pressure, will overcome the downward load due to the weight of the shell and support structure, and at this pressure, the floor plating at the tank periphery will start to lift off the tank foundation, as illustrated earlier in Figure 3.52.

The floor being allowed to lift off the foundation, can result in high stresses being set up in the shell-to-bottom joint which can

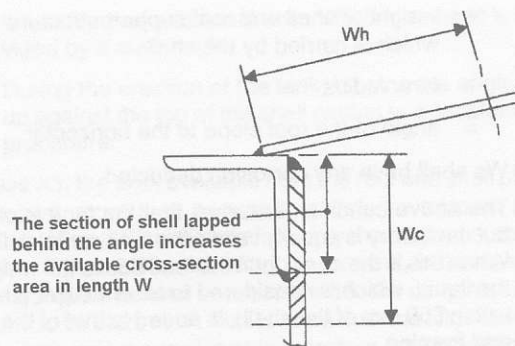


Figure 3.67a Typical roof joint

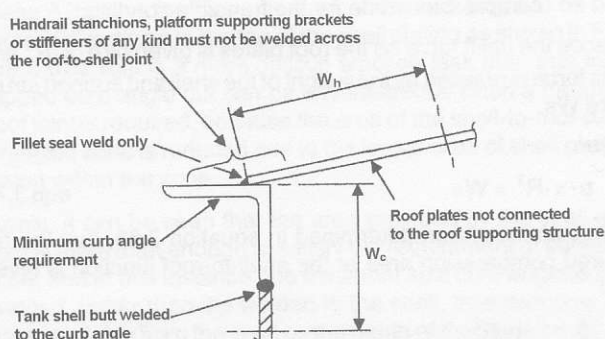


Figure 3.67b Typical frangible roof joint

result in failure of the joint. This possibility must be prevented by designing the roof-to-shell joint to fail before the shell-to-bottom joint does. This is accomplished by considering the point at which the pressure in the tank is such that the floor is just about to lift off its foundation.

3.8.3 The maximum compression zone area allowable

For a roof connection to be considered frangible, the maximum compression zone area allowable must be determined.

The roof plating is assumed to act as a membrane and any bending effects are ignored, as are any changes in geometry, also the angle between the slope of the roof and the horizontal θ , is assumed to remain at its design value.

Considering Figure 3.68.

p = internal pressure

T = membrane force in roof plating

W_r = weight of roof plating

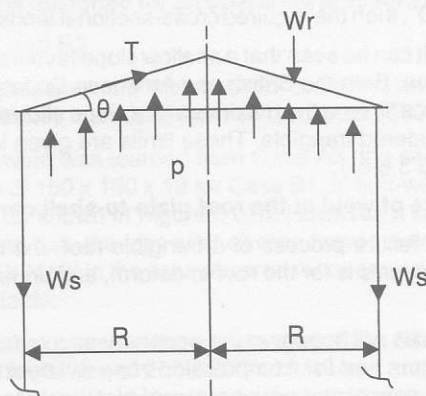


Figure 3.68 Tensile membrane forces

- Ws = weight of shell and roof support structure which is carried by the shell
- R = tank radius
- θ = angle of the roof slope to the horizontal

Wr and Ws shall have any corrosion deducted.

Note: The above condition assumes that the tank is empty, but the theory is equally valid if the tank contains liquid. When this is the case, then the load due to the weight of the liquid, which is considered to be effective, (i.e. say within 750 mm of the shell), is added to that of the shell and framing.

However, it is normal practice to design for the worst condition, which in this case, is when the tank is empty, thus giving a lesser value for the allowable area for the compression zone for the frangible condition.

Hence the uplift force on the roof plates is given by $p \cdot \pi \cdot R^2$ and this force is resisted by the weight of the shell and support structure Ws.

Then:

$$p \cdot \pi \cdot R^2 = Ws \quad \text{equ 3.74}$$

It has already been determined in equation 3.68, that the required compression area at the shell-to-roof junction is given by:

$$A = \frac{p \cdot R^2}{2 Sc \cdot \tan \theta}$$

And transposing for p:

$$p = \frac{2 \cdot A \cdot Sc \tan \theta}{R^2} \quad \text{equ 3.75}$$

Substituting for p in equation 3.74 then:

$$\frac{2 \cdot A \cdot Sc \tan \theta}{R} \times \pi \cdot R^2 = Ws$$

Hence:

$$A = \frac{Ws}{2 \cdot \pi \cdot Sc \cdot \tan \theta} \quad \text{equ 3.76}$$

The area A thus found, is the maximum that can be allowed for the shell-to-roof compression zone to be considered as a frangible joint.

3.8.4 Other factors affecting the frangible roof connection

3.8.4.1 Roof slope

In Section 3.7.2.1 it was demonstrated that as the roof slope becomes shallower, the value of θ decreases and hence the required cross sectional area increases. Taken to the extreme, as θ tends to 0° , then the required cross-sectional tends to infinity.

Therefore it can be seen that a shallow slope favours the frangible condition. Both the British and American Codes recognise this and put a limit on the maximum roof slope allowed for a roof to be considered frangible. These limits are given in Sections 3.8.5.1 and 3.8.6.1.

3.8.4.2 Size of weld at the roof plate-to-shell connection

During the failure process of a frangible roof, the normal sequence of events is for the roof to deform, and undergo elastic buckling.

Many creases will appear at the periphery as a reduction in diameter occurs and the compression zone will buckle and collapse. This causes the peripheral roof plate weld to tear away from its shell mounting and hence the excessive internal pressure is relieved.

The size and quality of this weld is therefore an important factor of the frangible joint. However there does not appear to have been very much research done in this area, and this could be due to difficulties in making meaningful analytical studies of the influence and behaviour of such welds when subjected to this type of failure mechanism.

The Codes do however require that the peripheral roof plate weld be kept as small as possible and in no case shall it be larger than 5 mm. From a practical point of view, making the weld size any less than this, can be detrimental in the long term, because experience has shown that in time, this weld suffers from the effects of corrosion wastage which can eventually lead to vapour leaks at the joint.

3.8.5 Formula as expressed in BS 2654

- A is expressed in mm^2
- Ws is given the notation 'T' and is the weight of the shell, shell stiffening and roof framework supported by the shell but excluding the roof plates, expressed in kilograms.
- Sc is expressed in N/mm^2 and curb failure is assumed to occur at 220 N/mm^2 , so this figure is built into the equation.
- θ is the slope of the roof at its point of connection to the shell in degrees.

The formula then becomes:

$$A = \frac{T \times 9.807}{2 \times \pi \times 220 \cdot \tan \theta} = \frac{T \times 7.07 \times 10^{-3}}{\tan \theta} \quad \text{equ 3.77}$$

Which is as it is shown in Appendix F of BS 2654.

3.8.5.1 Additional requirements to BS 2654

In addition to the restriction in cross-sectional area for the roof-to-shell zone for the frangible condition, the Code requires that the following conditions shall also be met, as described in Sections 3.8.4.1 and 3.8.4.2:

- The slope of the roof plating at its connection to the shell shall not be more than 1 in 5.
- The peripheral roof plating-to-shell connection weld shall not be more than 5 mm.

3.8.6 Formula as expressed in API 650

- A is expressed in mm^2
- Ws is given the notation W and is the weight of the shell, shell stiffening and roof framework supported by the shell but excluding the roof plates, expressed in Newtons
- Sc is expressed in N/mm^2 and curb failure is assumed to occur at 221 N/mm^2 , (32,000 lb/in^2) so this figure is built into the equation
- θ is the slope of the roof at its point of connection to the shell in degrees

The formula then becomes:

$$A = \frac{W}{2 \times \pi \times 221 \times \tan \theta} = \frac{W}{1390 \times \tan \theta} \quad \text{equ 3.78}$$

Which is as it is shown in clause 3.10.2.5.3 of API 650.

3.8.6.1 Additional requirements to API 650

In addition to the restriction in cross-sectional area for the roof-to-shell zone for the frangible condition, the Code requires that the following conditions shall also be met, as described above in Sections 3.8.4.1 and 3.8.4.2:

- The slope of the roof plating at its connection to the shell shall not be more than 1 in 6.
- The peripheral roof plating-to-shell connection weld shall not be more than 5 mm

3.8.7 Difference between Codes

The principal difference between the British and the American Codes is that BS 2654 allows the slightly steeper roof slope of 1 in 5, against 1 in 6 to API 650.

The different constants used in equations 3.77 and 3.78 are due to the tank weight being expressed in kilograms in BS 2654 and in Newtons in API 650.

The maximum allowable cross-sectional area in millimetres calculated by either equation is found to be the same for a given set of design parameters.

3.8.8 Conflict of design interests

During the initial tank design stage, the shell-to-roof joint will have been designed to suit the internal service pressure requirement, as detailed in Section 3.7. The most appropriate method of providing the required cross-sectional area in the roof-to-shell joint will have been established and hence the tank will be capable of withstanding the compressive forces which will develop in this area during normal operation of the tank.

However, it may be necessary to ensure, that in the event of an accidental over-pressurisation in the tank, it would be desirable for the shell-to-roof joint to fail. This may not always be possible because the compression area built into the tank to satisfy the operating pressure may be more than that allowed for a frangible roof joint, within the strictures of the Code.

The likelihood of this conflict occurring and the possible means by which it can be overcome, will become evident from the following Sections.

3.8.8.1 "Service" and "Emergency" design conditions

The maximum cross-sectional area at the compression zone which is allowable by equations 3.77 and 3.78 for the tank emergency condition, may be found to be less than that required to satisfy resistance of the internal pressure for the service condition calculated by equations 3.68 or 3.71.

When this occurs the tank is deemed not to have a frangible roof joint, but this situation may be overcome by providing the tank with anchor bolts or straps attached to the lower shell area of the tank and secured to a peripheral concrete foundation ring beam.

3.8.9 Examples of frangible and non-frangible roof joints

Using the tank shell design illustration given in Section 3.3.2.9, and assuming a roof slope of 1 in 5, and a roof plate to curb angle weld of 5 mm, then further calculations give the following information:

3.8.9.1 Tank designed for an operating pressure of 7.5 mbar

Case A1

Case A1 allows for the curb angle to be lapped on to the top of the shell, as shown in Figure 3.67a. This arrangement is generally adopted for two main reasons;

- 1) The available area of the compression zone which is required for the tank operating pressure is increased, because the top of the shell plating behind the angle is also included in the zone. This is advantageous as it minimises

the amount of additional area which may have to be provided by a curb angle.

- 2) During the erection of the tank, lapping the angle directly up against the top of the shell plating is a simpler erection procedure.

In Case A1, the area available from the roof and shell plating is, on its own, more than enough to satisfy the amount required from equation 3.67 and therefore only the minimum size of angle from Figure 3.58 will be fitted to the tank, in this case a 80 x 80 x 10 angle. The total area provided in the compression zone is found to be 5028 mm². This is more than the allowable area of 4811 mm², and the roof joint is therefore considered not to be frangible.

Case A2

Case A2 allows for the vertical leg of the curb angle to be butt welded directly on to the top of the shell plating as shown in Figure 3.67b. This is a more difficult erection task than that for a lapped curb angle but can be advantageous when a frangible roof joint is required, because the area of the shell-to-roof compression zone is reduced due to the lesser area of shell plating being within the zone.

Again, it can be seen that the area provided by the shell and roof is more than enough to satisfy the requirement of equation 3.64, and in this instance, the minimum size curb angle is butt welded, rather than lap welded to the shell, thus reducing the area available from the shell by the depth of the angle i.e. 80 x 8 = 640 mm².

This is enough to reduce the total available compression zone area to a figure which is less than the maximum allowed for a frangible joint and therefore the roof joint is frangible.

	Case A1	Case A2
Pressure	7.50 mbar	7.5 mbar
Compression zone area required for operation	1711 mm ²	1711 mm ²
Curb angle lapped or butted to shell?	Lap-welded to shell	Butt-welded to shell
Wh and Wc area	3518 mm ²	2878 mm ²
Additional area required	-1807	-1167
Selected curb angle size	80 x 80 x 10 RSA	80 x 80 x 10 RSA
Selected curb angle area	1510 mm ²	1510 mm ²
Total area provided	5028 mm ²	4388 mm ²
Is total area provided sufficient?	Yes	yes
Tank weight T	136089 kg	136089 kg
Maximum area allowed for frangible joint	4811 mm ²	4811 mm ²
Is the roof joint frangible?	No	Yes

3.8.9.2 Tank designed for an operating pressure of 20 mbar

Cases B1 and B2

At this higher pressure the required compression zone area has significantly increased from 1711 mm² to 7570 mm².

Following what was learned from Case A2, the selected curb angle size of 150 x 150 x 18 for Case B1, is butt-welded to the tank shell as shown in Figure 3.67b. However, it can be seen that in doing this, the loss of shell area leaves a deficit of 152 mm² (7570-7418) in the area required for operation, and this is not acceptable.

Case B2 is calculated in the same way as Case B1 except that the larger angle size of 200 x 200 x 16 is used and the consequent increase in the cross-sectional area of the angle gives an acceptable total area for the compression zone required for operational purposes.

For both Cases B1 and B2 however the area of the compression zone is far in excess of the maximum allowed for a frangible roof joint.

	Case B1	Case B2
Pressure	20.00 mbar	20.00 mbar
Compression zone area required for operation	7570 mm ²	7570 mm ²
Curb angle lapped or butted to shell?	Butt-welded to shell	Butt-welded to shell
Wh and Wc area	2318 mm ²	1918 mm ²
Additional area required	5252.32	5652
Selected curb angle size	150 x 150 x 18 RSA	200 x 200 x 16 RSA
Selected curb angle area	5100 mm ²	6180 mm ²
Total area provided	7418 mm ²	8098 mm ²
Is total area provide sufficient?	No	Yes
Tank weight T	139634 kg	140426 kg
Maximum area allowed for frangible joint	4936 mm ²	4964 mm ²
Is the roof joint frangible?	No	No

Case B3

From the previous Cases B1 and B2 it was found that for this particular tank size and its attendant design parameters, there was no advantage in butt-welding the curb angle to the shell.

Case B3 therefore is based on lap welding the curb angle as shown in Figure 3.67a. It can be seen from the results that in doing this the inclusion of the additional area of the shell plate behind the curb angle allows a smaller angle size of 150 x 150 x 15 to be used, and the combination gives an adequate overall total area in the compression zone.

	Case B3
Pressure	20.00 mbar
Compression zone area required for operation	7570 mm ²
Curb angle lapped or butted to shell?	Lap-welded to shell
Wh and Wc area	3518 mm ²
Additional area required	4052
Selected curb angle size	150 x 150 x 15 RSA
Selected curb angle area	4300 mm ²
Total area provided	7818 mm ²
Is total area provide sufficient?	Yes
Tank weight T	139041 kg
Maximum area allowed for frangible joint	4915 kg
Is the roof joint frangible?	No

However, as before in the previous cases, this area is well in excess of that allowable for a frangible roof joint.

3.8.10 Tank anchorage — a means to frangibility

The tank in Case B3 meets the Code requirement for having sufficient cross-sectional area in the roof-to-shell compression zone for operating conditions. But under an emergency over pressure condition, this area is too great to ensure that the roof-to-shell joint is frangible and therefore may not fail under this extreme condition. This could cause the shell-to-floor rim of the tank to lift off the foundation and the resulting distortion in this area could cause this joint to fail rather than the roof-to-shell joint.

This occurrence can be prevented by anchoring the tank to a suitably designed concrete ring beam which forms a part of the

tank foundation. Three methods of anchorage are illustrated in Figures 3.69 (a), (b) and (c).

3.8.10.1 Ensuring a frangible roof connection using anchorage

Apart from the frangibility consideration, anchorage may also be required due to the following conditions:

- The operating pressure causing uplift of the tank.
- The overturning effect on the tank of the prevailing wind.
- Instability of the tank caused by seismic action.

These instances are discussed in Section 3.9 and Chapter 15 or 26, but for now, the means of designing anchorage to ensure a frangible roof joint will be considered as follows:

3.8.10.2 Determining anchorage requirements

Where a roof is deemed not to be frangible, then the pressure at which it would fail has to be determined. This is done by transposing equation 3.69 or 3.71 depending upon which code is being used, and thus determining a failure pressure p .

Taking the case for the British Code then from equation 3.69:

$$p = \frac{A \cdot Sc \cdot \tan \theta}{50 \cdot R^2} + 0.77 \cdot tr \quad \text{equ 3.79}$$

Failure is considered to occur at a compressive stress Sc of 220 N/mm².

Hence failure pressure

$$p = \frac{4.4 \cdot A \cdot \tan \theta}{R^2} + 0.77 \cdot tr$$

Remember that in the British Code p is in mbar.

Similarly, for the American Code, from equation 3.71.

$$p = \frac{1.1 \cdot A \cdot \tan \theta}{D^2} + 0.08 \cdot th$$

For the American Code, failure is considered to occur at a compressive stress of 221 N/mm².

The constant 1.1 in equation 3.71 is calculated using a allowable stress of 137.5 N/mm² e.g. $\frac{137.5}{125} = 1.1$

This has to be recalculated using the failure compressive stress of 221 N/mm² and the new constant is $\frac{221}{125} = 1.77$

Failure pressure is therefore

$$p = \frac{1.77 \cdot A \cdot \tan \theta}{D^2} + 0.08 \cdot tr \quad \text{equ 3.80}$$

In the American Code p is in kilopascals — (1 kPa = 10 mbar)

3.8.10.3 Worked example

Consider the tank depicted in Section 3.3.2.9.

This tank is 30 m diameter, has a roof slope of 1: 5, a roof plate thickness of 5 mm and compression zone details as given in Section 3.8.9.2 for Case B3.

Anchorage is to be provided using bolts.

Using the BS Code for this example, then the failure pressure will be:

$$p = \frac{4.44 \times 7818 \times 0.2}{15^2} = 0.77 \times 5$$

$$= 34.43 \text{ mbar}$$

$$= 3.443 \text{ kN/m}^2$$

This pressure acting on the roof of the empty tank will produce a uplift of:

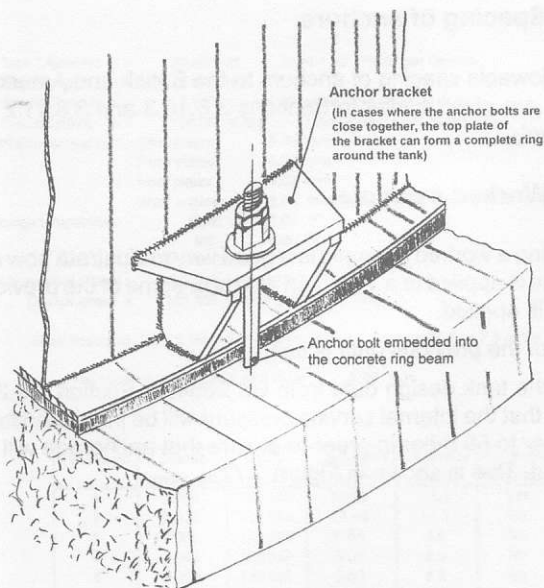


Figure 3.69a Anchorage using bolts

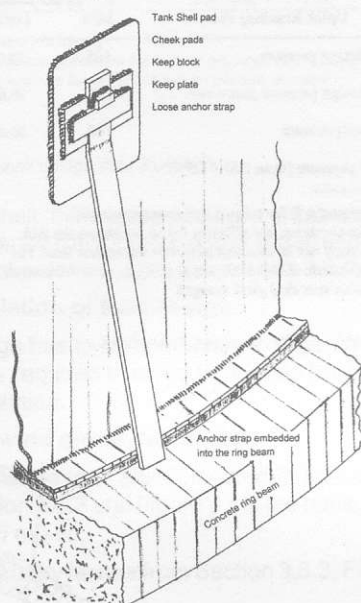


Figure 3.69b Anchorage using straps

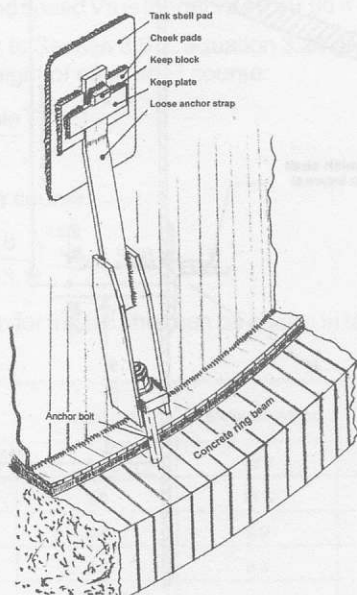


Figure 3.69c Combination using strap and bolt anchorage

$$U_p = \pi \cdot R^2 \cdot p$$

$$= \pi \times 15^2 \times 3.443$$

$$= 2433.71 \text{ kN}$$

The weight of the tank shell, stiffening and roof structure given in Case B3 is 139041 kg which equates to 1363.55 kN.

Then the net uplift = 2433.71 - 1363.55 = 1070.16 kN.

The BS Code requires anchors to be spaced around the tank circumference at a minimum of 1 m and a maximum of 3 m.

In this case a 3 m spacing will be used and hence the number of bolts required is:

$$\frac{30 \times \pi}{3} = 31.42$$

This is rounded up to 32.

However, as there are 12 plates per shell course, then 36 anchors will be selected, giving 3 per plate and thus clashes between anchor brackets and vertical shell course butt welds will be avoided.

The load per bolt due to the over-pressurisation uplift will be

$$\frac{1070.16}{36} = 29.73 \text{ kN}$$

The BS Code also requires anchors to have a minimum cross-sectional area of 500 mm². This equates to a bolt core diameter of 25.33 mm and hence a overall bolt diameter of 30 mm will be selected, which has an actual core stress area of 561 mm² (this excludes any corrosion which may be required).

The stress in each bolt due to the over-pressurisation uplift will be

$$\begin{aligned} &= \frac{29.73 \times 1000}{561} \\ &= 53.0 \text{ N/mm}^2 \end{aligned}$$

The BS Code states that the allowable tensile stress in the anchorage shall not exceed 50% of the specified yield strength, or 33.33% of the minimum tensile strength of the anchorage material, whichever is the lowest.

Taking medium strength steel having a minimum tensile strength of 430 N/mm² and yield of 255 N/mm² for this diameter of bolt, then the allowable tensile stress would be 127.5 N/mm². The selected bolt size is therefore acceptable.

3.8.10.4 Further design check

From above it can be seen that the tank can be subjected to a pressure greater than its design pressure i.e. 34.58 mbar instead of 20 mbar. The original tank design must therefore be checked to ensure that the allowable stress in the shell (equation 3.7) is not exceeded. This is accomplished by transposing S , the allowable stress and t in equation 3.7.

3.8.10.5 Other anchorage considerations

The anchorage design here is only catering for the uplift due to over-pressurisation and it must be borne in mind that this may have to be combined with any anchorage requirements which may be found to be necessary to stabilise an overturning moment on the tank due to wind loading which is dealt with in Section 3.9.

3.8.11 API 650 Code — anchor requirements

3.8.11.1 Minimum bolt diameter

The minimum anchor bolt diameter should not be less than 25 mm, plus a corrosion allowance of at least 6 mm, giving a minimum diameter of 31 mm. This is similar to that given in the BS

Code at 30 mm, except that in the case of the BS Code any corrosion allowance is added to 30 mm.

3.8.11.2 Spacing of anchors

The API Code does not specify a minimum spacing for anchors but states a maximum spacing of 3 m.

3.8.11.3 Allowable stresses in anchors

Table F-1 of Appendix F of API 650 gives the allowable stresses and this is reproduced in Figure 3.70.

3.8.12 Further guidance on frangible roofs

3.8.12.1 EEMUA

EEMUA (The Engineering Equipment and Materials Users Association) publication No. 180, gives very useful advice on the subject, (*Reference 3.8*).

One of the aspects covered, is an alternative method of ensuring a frangible joint in the tank shell near to the top of the tank and this is shown in Figure 3.71.

This method could also be used to convert an existing non-frangible roof tank, to have a frangible joint.

Note: Care must be exercised in using this method to ensure that the frangible shell-to-roof joint will fail before the shell-to-bottom joint, the shell joint or the anchorage. A thorough finite element analysis should be undertaken to make certain that the fillet weld between the angles fails before any other area of the tank.

3.9 Tank anchorage — further considerations

3.9.1 Wind loading and internal service pressure

The British, American and European Codes all address this subject. Fixed roof tanks shall be provided with anchorage if, due to one of the following conditions, there may be a tendency for the shell and the bottom plate, close to the shell, to lift off the foundation:

- Uplift on an empty tank due to internal design pressure, counteracted by the effective weight of the roof and shell.
- Uplift due to internal design pressure in combination with wind loading, counteracted by the effective weight of the roof and shell, plus the effective weight of product, considered by the tank operator, to be always present in the tank.

(This last condition is at the sole discretion of the tank operator.)

Note: The tank weights referred to are the weights after deducting any corrosion allowances.

3.9.2 Anchorage attachment

The principle point of attachment of the anchorage shall be on the tank shell plating and not the bottom plating and should be so designed to accommodate any tank movement due to thermal changes and hydrostatic pressure. Stresses induced into the shell due to the anchorage shall be kept to a minimum. Examples of tank anchoring methods are shown earlier in Figures 3.69 (a), (b) and (c).

The allowable stresses to the British, American and EN Codes are given earlier in Sections 3.8.10.3 and 3.8.11.3 and in Figure 3.70.

3.9.3 Spacing of anchors

The allowable spacing of anchors to the British and American Codes are given earlier in Sections 3.8.10.3 and 3.8.11.2 respectively.

3.9.4 Worked example

Following a worked example is a good way to illustrate how anchorage is applied to a tank, and also how some of the previous theory is applied.

Some of the previous data is used:

Using the tank design data from BS 2654, in Section 3.3.2.9, except that the internal service pressure will be increased from 7.5 mbar to 56 mbar in order to ensure that anchorage will be required. This is shown in Figure 3.72.

Uplift Resulting From	Allowable Stress at Root of Anchor Bolt Threads	
	MPa	(lbf/in. ²)
Tank design pressure	105	15,000
Tank design pressure plus wind ^a	140	20,000
Tank test pressure	140	20,000
Failure pressure (from F.6) $\times 1.5^b$	c	c

^aSee Appendix E for seismic design requirements.

^bFor this condition, the effective liquid weight on the tank bottom shall not be assumed to reduce the anchor load. The failure pressure shall be calculated using as-built thicknesses.

^cMinimum specified yield strength.

Figure 3.70 Allowable design stresses in anchors
From API 650, table F-1

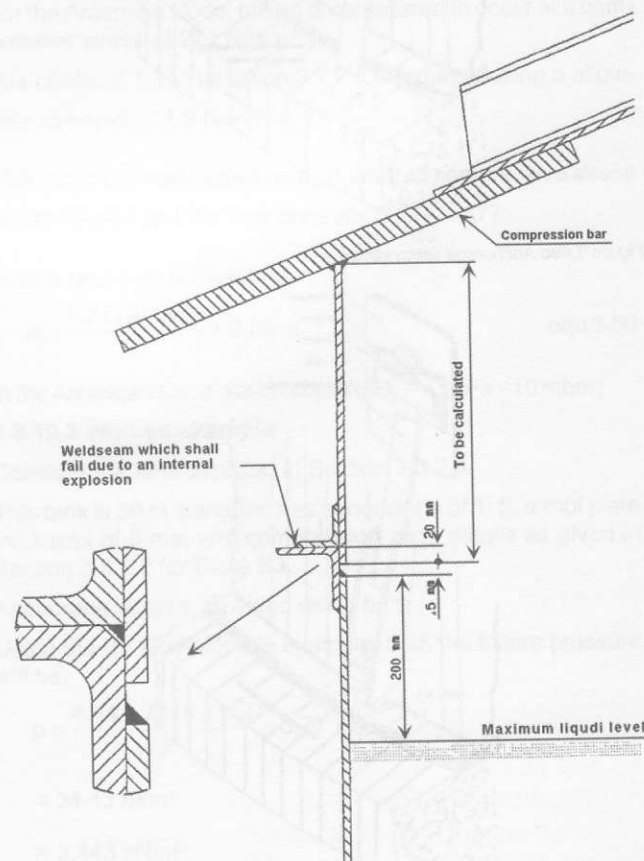


Figure 3.71 Frangible joint in a tank shell

Tank Diameter D =	30.000 m	Shell = 12 Plates per Course
Tank Height H =	16.000 m	
Specific gravity w =	0.900	1.00 to be used for Shell design.
Internal press. p =	56.00 m.bar	Internal vac 6.00 m.bar
Corrosion allowances :-	Shell plates 0.00 mm	
	Floor plates 0.00 mm	
	Roof plates 0.00 mm	
	Shell angles 0.00 mm, Total = 0.00 mm off each flange thks	
Design temperature :-	Max. 90.00 °C	
	Min. 0.00 °C	
Steel type :-	BS EN 10025 S275	
Minimum Yield Stress =	275.000 N/mm ² for 't' <= 16mm	
Design stress =	183.333 N/mm ² (2/3 x min. Yield)	

Shell thickness $D/20.S(98.w(H - 0.3) + p) + ca$ (ignore 'p' if ≤ 7.5 m.bar)

The Code requires a min. thickness 8.00 mm

Design of the Shell.

Course no.	Course height (m)	Design stress	Calculated thks. (mm)	Thickness chosen	Any impact Reqs'ts?
1	2.000	183.333	13.05	13.1	Yes
2	2.000	183.333	11.44	11.5	No
3	2.000	183.333	9.84	9.9	No
4	2.000	183.333	8.24	8.3	No
5	2.000	183.333	6.63	8.0	No
6	2.000	183.333	5.03	8.0	No
7	2.000	183.333	3.42	8.0	No
8	2.000	183.333	1.82	8.0	No
Shell ht. = 16.00			Min. thks. = 8.00		

The weight of the shell = 110,681 kg

This shell calculation demonstrates how the formula produces very thin upper courses.
The Code requires a minimum thickness of 8 mm for this tank diameter.

Figure 3.72 Tank shell design data illustration

Note: The shell thicknesses have increased slightly from those shown in Figure 3.8, this is due to the increase in internal pressure, from 7.5 mbar to 56 mbar.

3.9.4.1 Completion of tank design

The tank design has to be completed in order to obtain a tank weight. This is required in order to be able to perform the anchorage calculation.

3.9.4.2 Shell wind girder calculation

In this example the tank site is located in Liverpool, England and from Section 3.3.3 and Figure 3.10, the basic wind speed is found to be 46 m/sec.

Also the topography factors from Section 3.3.3, Figure 3.11 and 3.12 are:

$$S_1=1.0 \quad S_2=0.96 \quad S_3=1.0$$

The design wind speed V_s is therefore $46 \times 0.96 = 44.16$ m/sec

Referring back to Section 3.5.2, equation 3.24 gives the equivalent stable height of each shell course:

$$H_e = h \left(\frac{t \min}{t} \right)^{2.5}$$

For the bottom course:

$$H_e = 2.0 \left(\frac{8}{13.1} \right)^{2.5} = 0.538 \text{ m}$$

The calculation for the full shell can be shown in tabular form as follows:

Course No.	Height (m)	Thickness (mm)	Equivalent height (m)
1	2.0	13.1	0.853
2	2.0	11.5	0.807
3	2.0	9.9	1.174
4	2.0	8.3	1.824
5	2.0	8.0	2.000

Course No.	Height (m)	Thickness (mm)	Equivalent height (m)
6	2.0	8.0	2.000
7	2.0	8.0	2.000
8	2.0	8.0	2.000

The total equivalent stable height of the shell $H_e = 12.388$ m.

3.9.4.3 Maximum unstiffened height of the shell

This is obtained from Section 3.5.2, equation 3.33 which gives:

$$H_p = K \left(\frac{t \min^5}{D^3} \right)^{1/2}$$

But first a value for K must be obtained from equation 3.32

where

$$K = \frac{95,000}{(3.563 \cdot V_s^2 + 580 \cdot V_a)}$$

$$K = \frac{95,000}{(3.563 \times 44.16^2 + 580 \times 8.5)} = 7.998$$

Then:

$$H_p = 7.998 \left(\frac{8^5}{30^3} \right)^{0.5} = 8.811 \text{ m}$$

Comparing the maximum height of unstiffened shell allowable $H_p = 8.811$ m, to the equivalent stable height of the shell $H_e = 12.388$ m it can be seen that as $8.811 \text{ m} < 12.388 \text{ m} < 2 \times 8.811 \text{ m}$, then one secondary wind girder is required and the Code requires this to be positioned at $H_e/2 = 6.194$ m down from the top of the shell.

However, the girder may be positioned at a point $12.388 - 8.811 = 3.577$ m down from the top of the shell, as in this position the maximum permitted spacing of 8.811 m is still maintained.

There is an argument for placing the girder(s) as close to the top of the tank as possible because it has been found in practice that the upper courses tend to suffer more internal corrosion. This is due to the wetting and drying cycle in the upper area due to product movements in and out of the tank. Hence the girder(s) offer stiffness in the area where it is most needed.

In any event the girder(s) shall not be within 150 mm of a shell girth weld.

3.9.4.4 Section size for the secondary wind girder

From Figure 3.32, for a 30 m diameter tank the section size shall be a 125 x 75 x 8 mm angle.

The toe of the longer leg of the angle is welded to either the internal or external surface of the shell. The normal preference is to attach it to the external surface. This leaves a smooth internal surface, which makes for easier tank cleaning and also allows for the future fitting of an internal floating cover if, due to change of stored produce, this is found necessary.

The weight of this wind girder is 1,150 kg.

3.9.4.5 Shell-to-roof compression zone

From equation 3.69 the required area in this zone is:

$$A = \frac{50 \cdot (p - 0.77 \cdot tr) \cdot R^2}{Sc \cdot \tan \theta}$$

The minimum allowable roof plate thickness to the Code is 5 mm (to which any corrosion allowance has to be added).

The normal roof slope for a cone roof tank is 1 : 5 and this will be used here.

For this tank

$$A = \frac{50 \times \{56 - (0.77 \times 5)\} \times 15^2}{120 \times 0.2}$$

$$= 24,445 \text{ mm}^2$$

3.9.4.6 Participating roof and shell plate area

From Figure 3.55, in Section 3.7.4.1, the participating roof plate length:

$$Wh = 0.6 \sqrt{1000 \cdot R_t \cdot t_r}$$

R_t is the roof plate radius at the point where it meets the shell and is given by:

$$\frac{R}{\sin \theta} = \frac{15}{0.196} = 76.485 \text{ m}$$

Then:

$$Wh = 0.6 \sqrt{1000 \times 76.485 \times 5} = 371 \text{ mm}$$

The area of this length of roof plating:

$$= Wh \cdot t_r = 371 \times 5 = 1885 \text{ mm}^2$$

Similarly the participating shell plate length:

$$Wc = 0.6 \sqrt{1000 \cdot R \cdot t}$$

In this case the radius of the shell:

$$R = 15 \text{ m}$$

Then:

$$Wc = 0.6 \sqrt{1000 \times 15 \times 8} = 207.85 \text{ mm}$$

The area of this length of shell plating:

$$= Wc \cdot t = 207.85 \times 8 = 1662.8 \text{ mm}^2$$

The total participating area:

$$= Wh + Wc = 1885 + 1662.8 = 3547.8 \text{ mm}^2$$

The additional area required at the junction:

$$= A - (Wh + Wc)$$

$$= 24,445 - 3547.8 = 20,897.2 \text{ mm}^2$$

To comply with the Code, this additional area must lie within the participating roof and shell lengths of:

$$Wh = 371 \text{ mm and } Wc = 207.85 \text{ mm.}$$

The additional area is too large to be provided by any combination of the largest angle sizes which are commonly available to us. The alternative therefore, is to use thickened roof and shell plates within the compression zone.

Following the same method used in the design example in Section 3.7.10.1, the following result is obtained.

$$\text{Corroded area required} = 20,897.2 \text{ mm}^2$$

$$\text{Try } t_r = 18 \text{ mm}$$

$$\text{and } t = 16 \text{ mm}$$

$$\text{c.a.} = 0 \text{ mm}$$

$$\text{Roof slope} = 1 \text{ in } 5$$

$$\text{Tank diameter} = 30 \text{ m}$$

$$\text{Roof radius} = 76.486 \text{ m}$$

$$(t_r - \text{c.a.}) = 18 \text{ mm}$$

$$(- \text{c.a.}) = 16 \text{ mm}$$

Recalculate:

$$Wh = 0.6 \sqrt{1000 \times 76.485 \times 18} = 704 \text{ mm}$$

and

$$Wc = 0.6 \sqrt{1000 \times 15 \times 16} = 294 \text{ mm}$$

Also the Code allows the participating roof plate to overhang the shell by $16t$ which in this case is $16 \times 16 = 256 \text{ mm}$.

By trial and error it is found that the roof plate dimensions of $Wh = 700 \text{ mm}$ plus a shell overhang of 210 mm , give a roof plate area of $(700 + 210) \times 18 = 16380 \text{ mm}^2$.

Using the allowable shell length of $294 \text{ mm} \times 16 \text{ mm}$, then the area for the shell section is 4704 mm^2 .

The total area is therefore $16,380 + 4,704 = 21,084 \text{ mm}^2$ and this meets the requirement.

Also it can be found that the centroid of the two plate sections lies 7.64 mm above the corner formed by the two participating plates.

From Section 3.7.11 the maximum distance for the position of the centroid of area, either above or below the corner is:

$$1.5 (t_r + t) / 2 = 25.5 \text{ mm}$$

The chosen arrangement satisfies this requirement.

The weight of this composite section is $15,594 \text{ kg}$.

The compression zone will be constructed as shown in Figure 3.73.

3.9.4.7 Roof plating

The roof plate thickness was selected as 5 mm , which is the minimum to the Code, and, as is normal for this type of roof, the lapped joints between the plates are welded on the top side only.

The suitability of this thickness and joint type has to be proved in accordance with equation 5.3 in Chapter 5. The reason for this is that, as the roof plating is only attached to the tank at its periphery then, under pressure it can lift off its support structure and act as a membrane and so its suitability in this condition has to be verified.

From equation 5.3 in Chapter 5, the thickness of the roof plate to resist pressure:

$$t_r = \frac{p \cdot R_1}{10 \cdot S \cdot \mu}$$

where:

$$p = 56 \text{ mbar}$$

$$R_1 = 76.485$$

$$S = 18.33 \text{ N/mm}^2$$

$$\mu = 0.35 \text{ for a single side-welded lap joint}$$

Then

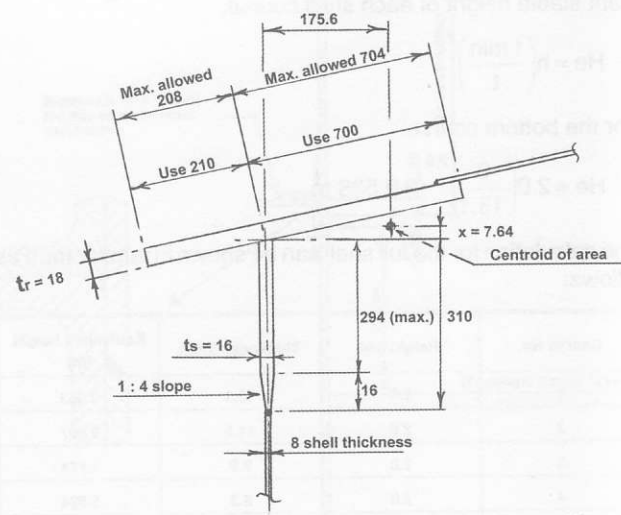


Figure 3.73 Compression zone construction

$$t_r = \frac{56 \times 76.485}{10 \times 183.33 \times 0.35} = 6.68 \text{ mm}$$

The roof plating is not acceptable at 5 mm thick, with single lap-welded joints. Three solutions to this situation are possible:

- 1) Lose 7 mm roof plating (which is a non-standard thickness, therefore 8 mm would probably be selected.)
- 2) Weld the underside as well as the top side of the lap welds. This would increase the joint efficiency factor p to 0.5. The required design thickness would then reduce to 4.67 mm and then 5 mm plate would be acceptable.

However, welding the underside laps on such a large area of roof would be an expensive and labourious task.

- 3) Re-design the roof as an umbrella roof when the roof radius can be selected to accept 5 mm plate.

In practice solution 3) would be the most favourable option, but for the purposes of this exercise we will continue with the cone roof and select to use 8 mm plate. A further effect of this decision is to increase the weight on the roof structure by about 17 tonnes (24 kg/m²) and hence the design of the structure will have to cater for this additional load.

The weight of the 8 mm roof plating is found to be 45,270 kg.

3.9.4.8 Roof structure

The various types of roof structures are dealt with in Chapter 5, where it will be seen that they are designed to structural engineering standards, which are not exhaustively dealt with in the tank standards. For the tank in question, the structure will be of the internal truss type and from previous experience it is found that the net weight of such a structure is in the region of 31,000 kg, after allowing for the thicker than usual roof plating at 8 mm.

3.9.4.9 Anchorage calculation

Enough information is now available to calculate the effective weight G_a of the tank for the anchorage calculation and this is summarised as follows:

	kg
Shell	110,681
Wind girder	1,150
Shell-to-roof compression zone	15,594
Roof structure	31,000
Roof plating	45,270

$$G_a = 203,695 = 1997.6 \text{ kN}$$

Note: The floor weight is excluded from the effective tank weight.

The forces acting on the tank which can cause anchorage to be required will now be considered.

3.9.4.10 Overturning moment due to wind action only

Referring to Section 3.3.3.4 for the theory used in CP3 : Chapter V : Part 2, the following is found:

Using equation 3.15, the design wind speed has been established as 44.16 m/sec.

From equation 3.17 the dynamic pressure:

$$q = 0.613 \cdot V_s^2$$

$$q = 0.613 \times 44.16^2$$

$$q = 1195.4 \text{ N/m}^2$$

The tank height to diameter ratio = $16/30 = 0.533$; and from Figure 3.12 the coefficient $C_f = 0.7$.

The wind force normal to the shell from equation 3.20:

$$F_s = C_f \cdot q \cdot D \cdot H$$

$$F_s = 0.7 \times 1195.4 \times 30 \times 16$$

$$F_s = 401,654 \text{ N}$$

The wind force normal to the roof from equation 3.21:

$$F_r = C_f \cdot q \cdot \frac{1}{2} D \cdot h$$

$$F_r = 0.7 \times 1195.4 \times (30/2) \times 3$$

$$F_r = 37,655.1 \text{ N}$$

The resulting wind moment on the tank is found from equation 3.19:

$$M_w = [F_s \cdot H/2] + [F_r(H + h/3)]$$

$$M_w = [401,654.4 \times 16/2] + [37,655.1 (16 + 3/3)]$$

$$M_w = 3,853,371.9 \text{ N or } 3,853.37 \text{ kNm}$$

Whilst it is not specifically mentioned in BS 2654, it is advisable to apply a factor of safety to the tank overturning moment. Guidance on this is given in BS 449: Part 2 "The use of structural steel in building". Clause 10b of this Code states "When considering wind loads, the restoring moment shall not be less than 1.4 times the overturning moment due to dead loads and wind loads, nor less than 1.2 times the overturning moment due to the combined effects of dead, imposed and wind loads". Therefore a factor of 1.4 will be used.

The value for M_w used in the anchorage calculation then increases to $3,853.37 \times 1.4 = 5,394.72 \text{ kNm}$.

The counteracting righting moment on the tank is given by multiplying the effective weight of the tank W less the uplift on the roof due to the wind passing over it, which is usually taken as $0.6 \times q \times \text{area}$, by the moment arm measured between the polar axis of the tank and the tank shell.

$$M_{r1} = [W - (0.6 \times q \times \text{area}) \times D/2]$$

The uplift in this case is $0.6 \times 1195.4 \times \frac{\pi}{4} \times 30^2 = 507 \text{ kN}$

$$M_{r1} = (1997.6 - 507) \times 30/2$$

$$M_{r1} = 22.359 \text{ kNm}$$

As $M_{r1} > M_w$, anchorage is not required.

3.9.4.11 Overturning moment due to wind action while in service

The Code requires the tank stability to be checked when it is empty, but subjected to its internal design pressure together with the external wind load and this is performed as follows:

The upthrust on the roof due to the internal pressure is:

$$U_p = \pi/4 \cdot D^2 \cdot p$$

$$U_p = \pi/4 \times 30^2 \times 56 \times 0.01$$

$$U_p = 3958.42 \text{ kN}$$

The resultant downward load is:

$$G_a - U_p = 1,997.6 - 3,958.42 = -1,960.82$$

Then the righting moment for this case:

$$M_{r2} = (G_a - U_p)D/2$$

$$M_{r2} = -1,960.82 \times 30/2$$

$$M_{r2} = -29,412.3 \text{ kNm}$$

The wind moment M_w is the same as before for this condition and hence for this case $M_{r2} < M_w$ and therefore anchorage is required.

Note: There is provision in the Code for the tank user to stipulate that there will always be a certain amount of product in the tank at all times whilst the tank is in service. For such cases the applicable weight of this product can be added to the weight of the tank to counteract the upthrust due to the internal pressure. This, in some cases, can negate the requirement for anchorage to be provided.

3.9.4.12 Design of the anchorage

To determine the load induced in the anchorage by the overturning moment, consider the following approach.

From the fundamental theory of bending it is known that:

$$\frac{M}{I} = \frac{f}{y}$$

where:

- M = in this case the wind overturning moment
- I = moment of inertia of the cross-section of the tank
- f = stress in cross-section
- y = maximum distance from the axis of the section to the outer fibres, in this case the radius r of the tank shell

It is also known that:

$$\frac{I}{y} = Z \text{ the modulus of the cross-section.}$$

and therefore:

$$f = \frac{M}{Z} \quad \text{equ 3.81}$$

Also:

$$\text{Stress} = \frac{\text{load}}{\text{area}} \quad \text{or} \quad f = \frac{L}{A}$$

The cross-sectional area of a thin cylinder is given as:

$$A = \pi \cdot D \cdot t$$

where:

- D = diameter of the tank
- t = shell thickness

Then:

$$f = \frac{L}{\pi \cdot D \cdot t} \quad \text{equ 3.82}$$

Equations 3.81 and 3.82 can now be equated:

$$\frac{M}{Z} = \frac{L}{\pi \cdot D \cdot t}$$

By definition, Z , the section modulus for a thin walled cylinder is given by:

$$\pi \cdot r^2 \cdot t = \frac{\pi}{4} \cdot D^2 \cdot t$$

Then:

$$\frac{M}{\pi/4 \cdot D^2 \cdot t} = \frac{L}{\pi \cdot D \cdot t} \quad \text{or} \quad \frac{4 \cdot M}{\pi \cdot D^2 \cdot t} = \frac{L}{\pi \cdot D \cdot t}$$

Hence:

$$L = \frac{4 \cdot M}{D}$$

L is the total load in all the anchors, so if the number of anchors is N , then the load in each anchor is:

$$\text{Load/anchor} = \frac{4 \cdot M}{D \cdot N}$$

The force W , resisting this load is that due to the shell, shell stiffening and that part of the roof structure and plating which is supported by the shell, (all after the deduction of any corrosion allowance), minus p , the simultaneous uplift from operating conditions such as the internal pressure on the roof.

This uplift may in certain cases be more than the weight of the tank and in such cases the load is added to the load due to the overturning moment.

Then:

$$W = (w - p) \text{ and the load per bolt} = \frac{W}{N}$$

The load in each anchor is therefore is

$$t_b = \frac{4 \cdot M}{D \cdot N} - \frac{W}{N} \quad \text{equ 3.83}$$

This is the expression, which appears in API 650, clause 3.111.3 for anchorage, except that D is shown as the anchor circle diameter.

Adopting the nomenclature used in Section 3.9.4.10 and 11, then equation 3.83 can be written:

$$t_b = \frac{4 \cdot Mw}{D \cdot N} - \frac{(Ga - Up)}{N} \quad \text{equ 3.84}$$

The BS Code does not give a method for calculating the anchorage loading but leaves this to the individual designer to formulate.

The BS Code does stipulate that the spacing of anchors shall be between 1 and 3 metres (see Section 3.8.10.3) and also that the minimum cross-sectional area of an anchor shall not be less than 500 mm², excluding any corrosion allowance.

It is often convenient to arrange the anchors such that there are an equal number on each shell plate, in this way clashes between anchor positions and vertical course welds can be avoided.

For the 30 m diameter tank in question, the maximum number of anchors is 94 and the minimum 32. As there are 12 plates per course, then 36 anchors will be selected, giving 3 per shell plate.

Assume the use of anchor bolts as shown in Figure 3.69 (a) and a pitch circle diameter of 30.32 m.

From equation 3.80 the load per bolt

$$t_b = \frac{4 \times 5,394.72}{30.32 \times 36} - \frac{(-1,960.82)}{36} = 74.2 \text{ kN/bolt}$$

Selected from the worked example in Section 3.8.10.3 is an anchor bolt material having a minimum tensile strength of 430 N/mm² and a minimum yield strength of 255 N/mm² and hence an allowable tensile stress of 127.5 N/mm² based on 50% of the yield strength.

A bolt diameter of 36 mm will be selected, having a core stress area of 817 mm² and this excludes a corrosion allowance.

The tensile stress in the anchor bolts will be:

$$\frac{74.24 \times 1000}{817} = 90.87 \text{ N/mm}^2$$

This actual stress is less than 127.5 N/mm² and is therefore acceptable.

3.9.4.13 Check for frangibility

If the tank were required to have a frangible roof joint, then the calculation given in Section 3.8.10.3 would be based on the anticipated roof failure pressure and performed as follows:

From Section 3.9.4.6 the total area of the compression zone is 21,084 mm².

From Section 3.9.4.7 the roof plating is 8 mm thick

From Section 3.9.4.9 the effective weight of the tank (excluding the roof plates) is 158,425 kg = 1553.64 kN.

From equation 3.79

$$p = \frac{A \cdot \sigma_c \cdot \tan \theta}{50 \cdot R^2} + 0.77 \cdot tr$$

$$= \frac{21,084 \times 220 \times 0.2}{50 \times 15^2} + 0.77 \times 8$$

$$= 88.62 \text{ millibar or } 8.862 \text{ kN/m}^2$$

The upthrust on the roof:

$$U_p = \pi \times 15^2 \times 8.862$$

$$= 6,264.18 \text{ kN}$$

The net uplift on the roof is:

$$6,264.18 - 1,553.64 = 4,710.54 \text{ kN.}$$

The tank is to have 36 off 36 mm diameter bolts, each having a core cross-sectional diameter of 817 mm².

The load in each bolt:

$$= \frac{4,710.54}{36} = 130.85 \text{ kN}$$

The stress in each bolt:

$$= \frac{130.85 \times 1000}{817} = 160.16 \text{ N/mm}^2$$

This is greater than the allowable stress of 127.5 N/mm² and is therefore unacceptable.

Try using 42 mm diameter bolts with a core cross-sectional area of 1112 mm².

The stress in each bolt:

$$= \frac{130.85 \times 1000}{1112} = 117.67 \text{ N/mm}^2 \text{ and this is acceptable}$$

It can be seen then, that whilst the tank anchorage of 36 off 36 mm diameter bolts was acceptable for wind and service loading, for the frangible roof condition the bolt diameter had to be increased to 42 mm.

Alternatively, the number of bolts could have been increased if there was a desire to maintain a bolt diameter of 36 mm.

As mentioned in Section 3.8.10.4 the stress in the shell plating must be checked at the roof failure pressure.

3.9.4.14 Wind loading to API 650

The American Code uses a different method to establish the wind loading on a tank.

In clause 3.11.1 of the Code, specific wind pressures are published, based on a wind speed of 100 mph, (160 km/h) and these are:

1.4 kPa (30 lbf/ft²) on vertical plane surfaces.

0.86 kPa (18 lbf/ft²) on projected areas of cylindrical surfaces.

0.72 kPa (15 lbf/ft²) on projected areas of conical and double curved surfaces.

These pressures can be adjusted for other wind velocities by multiplying them by (V/160)² for SI units, or (V/100)² for Imperial units, where V is the wind speed in km/h or mph respectively.

The value Mw, the overturning wind moment, is then calculated using the above figures.

The American Code chooses a safety factor of 1.5 (it was 1.4 for the British Code) and therefore for an unanchored tank:

1.5 Mw must be less than or just equal to the effective weight of the tank W x D/2.

This is actually shown in the Code as:

$$M_w \leq \frac{2}{3} \left(\frac{W \cdot D}{2} \right)$$

The load in each anchor tb is found from equation 3.79 except that it is presented in the Code as:

$$t_b = \frac{4 \cdot M}{d \cdot N} - \frac{W}{N}$$

where:

d = diameter of the anchor circle (m)

3.10 Tanks produced in stainless steel materials

The BS and API Codes are written around the use of carbon steel materials. However for many years the petrochemical industry has required tanks made in stainless steel materials. Accordingly designers have used the existing Codes and adapted them for stainless steel materials.

Stainless steel does not strain under load in the same way that carbon steel does, as it does not have a distinct yield point. The alternative is to use the value of the "proof stress" as the yield stress and usually the value for the 1 % proof stress is used.

In 1998 API 650 introduced Appendix S into the Code and this gives recommendations for designing tanks in austenitic stainless steel grades 304, 304L, 316, 316L, 317 and 317L.

The Appendix gives many recommendations, the important ones being in the following areas:

- Lists of acceptable materials to be used for plates and structural sections, piping, forgings and bolting materials.
- Design information — This is very similar to that given in the main body of the Code but for the shell design it includes the use of a joint efficiency, the value of which is dependant upon the level of radiographic inspection of the shell welds.
- Tables for the allowable stresses and "yield stresses" for tank shells at various design temperatures for the range of steel grades covered by the Code.
- A table giving values for the modulus of elasticity of stainless steel over a range of temperatures.
- A list of other Appendices which require modification when used for austenitic stainless steels.

The BS Code does not yet give advice on the use of stainless steels for tank construction.

The European code prEN 14015 -1 does include references to the use of stainless steel and these can be briefly summarised as follows:

- A list of acceptable austenitic and austenitic-ferritic steels to EN 10088 -1 is given
- The allowable stress levels have to be determined by the designer from EN 10088 -1
- Minimum floor plate thicknesses are given as:

Lap-welded floors 5 mm
(compared to 6 mm for carbon steel)

Butt-welded floors 3 mm
(compared to 5 mm for carbon steel))

- The minimum allowable nominal shell thicknesses are given as:

Tank diameter 'D'	Stainless steel (mm)	Carbon steel (mm) for comparison
D < 6	2	5
6 to < 10	3	5
10 to < 15	4	5
15 to < 30	5	6
30 to < 45	6	8
≥ 45	By agreement between the purchaser and the contractor	Up to 12 depending on diameter

- Minimum roof plate thickness 3 mm
(compared to 5 mm for carbon steel)
- Minimum thickness of structural roof members 3 mm
(compared to 5 mm for carbon steel)
- Shell nozzle barrel thicknesses:

O.D. of Nozzle (mm)	Stainless steel (mm)	Carbon steel (mm) for comparison
≤ 50	3.5	5
> 50 ≤ 75	5	5.5
> 75 ≤ 100	6	7.5
> 100 ≤ 150	7	8.5
> 150 ≤ 200	8	10.5
> 200	9	12.5

- Roof nozzle barrel thicknesses:

n.b. of nozzle (mm)	Stainless steel (mm)	Carbon steel (mm) for comparison
25	2.7	3.4
50	2.7	3.9
80	3	5.5
100	3	6
150	3.4	7.1
200	3.7	8.2
250	4	9.3
300	4.5	9.5

3.11 Semi-buried tanks for the storage of aviation fuel

An interesting design of storage tank has become the Standard for the storage of aviation fuel at most military air bases and some commercial airports.

These are vertical cylindrical tanks which are cased in reinforced concrete and either fully or semi-buried. In the case of military establishments, the reason is based on security from aerial or ground attack. A series of these tanks under construction is shown in Figure 3.74

The tanks are supported on a cone down to the centre reinforced concrete foundation with a slope of 1:25 and a central bottom liquid outlet. The bottom is usually butt-welded and around 12 mm thick. It is important to ensure that there are no voids between the floor plating and the foundation in order to give a firm seating for the support columns. Also, the floor must

freely drain to the centre sump. It is therefore important to ensure that the plates do not distort during welding and the use of strongbacks is essential as shown in Figure 3.75.



Figure 3.74 Semi-buried tanks under construction
Courtesy of McTay

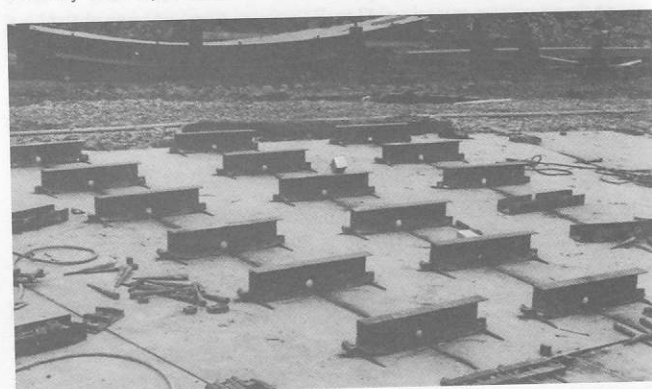


Figure 3.75 The use of strongbacks during welding to stop plate distortion
Courtesy of McTay

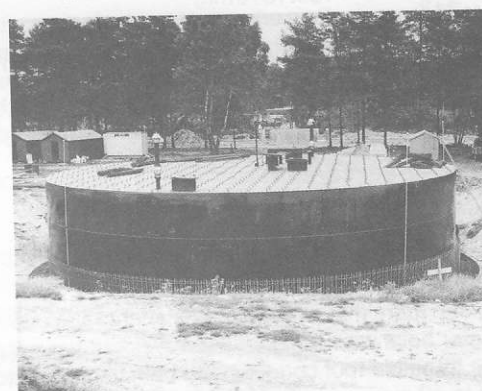


Figure 3.76 The tank shell is coated with bitumen-based paint system
Courtesy of Whessoe

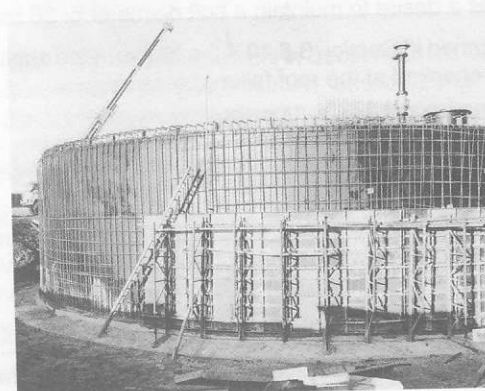


Figure 3.77 The tank is clad in reinforced concrete
Courtesy of Whessoe

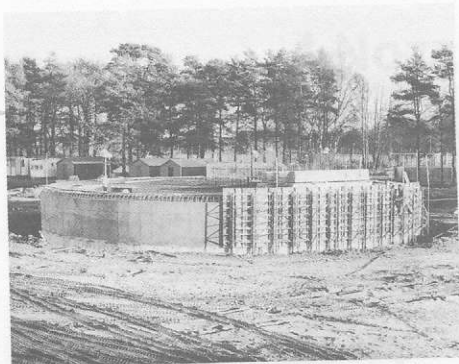


Figure 3.78 The tank is clad in reinforced concrete
Courtesy of Whessoe



Figure 3.79 The tank roof is clad in reinforced concrete
Courtesy of Whessoe



Figure 3.80 The whole structure is partially or completely buried
Courtesy of Whessoe

The tanks' bottoms were originally designed to resist an external pressure arising from the ground water of around 1 m head (hence the 12 mm thickness) but for later tanks this requirement was removed (although the 12 mm thickness was maintained).

The tank shells are butt-welded and the tank roof is flat supported by internal columns. Following construction of the metallic parts, the tank shell is coated with a bitumen-based paint system, see Figure 3.76, and is clad in reinforced concrete (Figures 3.77 and 3.78) as is the tank roof (Figure 3.79). In some cases a concrete combined pump house and control room is

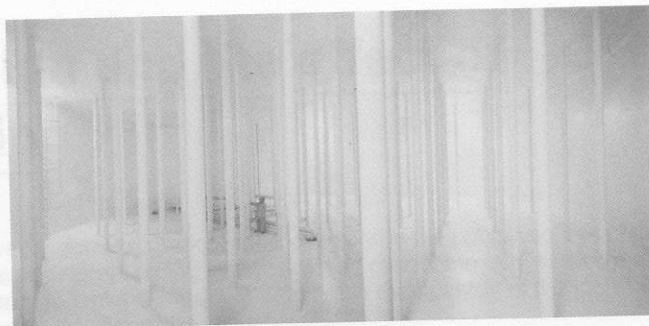


Figure 3.81 Lining of internal surfaces with an appropriate epoxy-based paint system

Courtesy of Mc Tay

constructed on top of the tank. The whole structure is then partially, or completely buried and grassed over to make its presence less obvious. This is shown in Figure 3.80.

These tanks are made of carbon steel, up to 33 m in diameter. The internal surfaces are lined with an appropriate epoxy based paint system for reasons of product cleanliness as shown in Figure 3.81.

3.12 References

- 3.1 *A Review of the Development of Fracture Safe Designs and Codes for Oil and LPG Storage Tanks*, H. C. Cotton, Consultant and J. B. Denham, BP International Ltd.
- 3.2 *Formulas for Stress and Strain*, by R. J. Roark and W. C. Young, published by McGraw Hill.
- 3.3 *Transcript in the American Society of Engineers, Volume 53 (15)107, 1931*, H. E. Saunders and A. F. Windenberg.
- 3.4 *Shell-to-Base Joint Design, Inspection & Repair*, H. Kroon, Paper presented at the Storage Tank Design and Inspection Seminar, University College Stockton, UK, 1999.
- 3.5 *Beams on Elastic Foundations*, M. , The University of Michigan Press and Oxford University Press, 1946. (This reference is contained within the H. Kroon paper.)
- 3.6 *Stability of API Standard 650 Tank Shells*, R. V. McGrath, Proceedings of the American Petroleum Institute, Section III - Refining.
- 3.7 *Frangibilite, etude sur la rupture eventuelle dun reservoir cylindrique vertical a toit conique soumis a une surpression provoquee par une deflagration accidentelle*, R. Perono, SNCT Publications, (circa 1980).
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4 Nozzle design and the effect of applied loading

The majority of piping systems connect into a tank at a low elevation in the bottom course of the shell plating. The design of these external piping systems which connect to thin-walled, large diameter, cylindrical vertical storage tanks can pose a problem in the analysis of the interface between the piping system and the shell nozzles.

The designer must consider the stiffness of the tank shell and the radial deflection and meridional rotation of the shell nozzle resulting from the product head, pressure and uniform or differential temperature between the shell and the bottom. The work of the piping designer and the tank designer must be coordinated to ensure that the piping loads imposed on the shell nozzles by the piping are within safe limits.

This Chapter elaborates on the method of analysis given in Appendix P of API 650.

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4.1 Nozzle design

4.1.1 The scope of the nozzles analysed

4.1.1.1 The loading on the nozzle

4.1.1.2 Definition of stiffness coefficients

4.1.1.3 Shell deflection and rotation

4.1.1.4 Determination of loads on the nozzle

4.1.2 The assessment of nozzle loadings

4.1.2.1 Determination of allowable loads according to the API 650 approach

4.1.2.2 Construction of the nomograms

4.1.2.3 Determination of allowable loads

4.1.3 Concluding comments

4.1.4 Method of analysis example

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4.1.4.2 The solution

4.1.4.3 The stiffness coefficients for the nozzle-tank connection

4.1.4.4 Unrestrained shell deflection and rotation at the nozzle centreline

4.1.5 Assessment of the nozzle loading example

4.1.5.1 Determination of the non-dimensional quantities

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4.2 References

4.1 Nozzle design

Grateful acknowledgment is given to the late Professor A. S. Tooth, Professor of Mechanical Engineering at University of Strathclyde, Glasgow, for the following elaboration of the application of the theory.

Many large diameter cylindrical tanks are constructed with low entry nozzles in the shell close to the base plate - illustrated in Figure 4.1. The location of these enables bulk liquid storage systems to make use of gravity feed for discharge. In view of this, and of their small diameter compared to the tank diameter and the fact that the tank radius/wall thickness (R/t) ratio is large, it is not possible to make use of the charts provided in BS 5500 and WRC Bulletin 107 (or WRC 297) to determine the stiffness coefficients for the nozzles when subjected to local loading. The above references are primarily designed for the analysis of pressure vessels, rather than storage vessels, and are limited to vessel geometries within the range appropriate for high pressure service.

To cope with this, a simplification is often made when carrying out an overall pipework analysis, in which the tank is assumed to be a rigid anchor. However, ignoring the local flexibility of the nozzle-shell connection in the piping flexibility analysis can result in a significant overestimation of the rigidity of the piping system and of the "end reactions" at the pipe-to-nozzle junction. This can often lead to unnecessary redesign of the piping system and the nozzle-shell attachment to handle the higher loads, which are predicted by the analysis.

The API 650 Code Appendix P addresses this problem, whereby the local stiffness coefficients can be obtained. These are given for a range of R/t values, nozzle radius/shell radius ratio values (a/R), and ratios of distance from the base/nozzle diameter ($L/2a$), which are appropriate for these large storage vessels. The nozzle restraints can thus be more accurately modelled and included in any conventional piping analysis program, to determine the actual loads on the nozzle and from these the resulting stresses in the vessel. The method is how-

ever only to be applied to tanks whose diameter is larger than 36 m.

The approach, by Billimoris and Hagstrom, (*Reference 4.1*) was incorporated into API 650 Appendix P in November 1988. The purpose of the method is to provide local stiffness coefficients for the nozzle-shell connection that can be used in the design of the piping system. The restraint of the nozzle connection can be simulated by including these coefficients in any conventional piping flexibility analysis program. Then from a compatibility analysis of the piping system, the value of the loads on the nozzle can be determined and, thereafter evaluated to see if they can be safely carried by the tank.

4.1.1 The scope of the nozzles analysed

Two types of reinforced nozzle connections are considered in API 650. These are:

- **Reinforcing in the nozzle only** by an increase in the nozzle wall thickness, in which case the tank is not reinforced by a pad plate or insert.
- **Reinforcing of the shell** by means of a pad plate or an insert plate. The width of the reinforcing zone on each side of the nozzle centre-line is prescribed as $2a$ and the thickness of the reinforcing plate is assumed equal to the tank thickness.

For both types of nozzle connections, the distance from the tank bottom L , (see Figure 4.52), is described in terms of $L/2a$. Two cases are examined, viz., $L/2a = 1.0$ and $1.5L$.

Curves for determining the stiffness coefficients are given for R/t ratios from 300 to 3000 and a/R ratios from 0.005 to 0.04. For intermediate values of R/t and a/R , the stress values can be found by interpolation from the curves. It is considered that the ranges of the ratios R/t and a/R given in the Code should adequately encompass the majority of low-type fittings. Other values of $L/2a$ can be approximated.

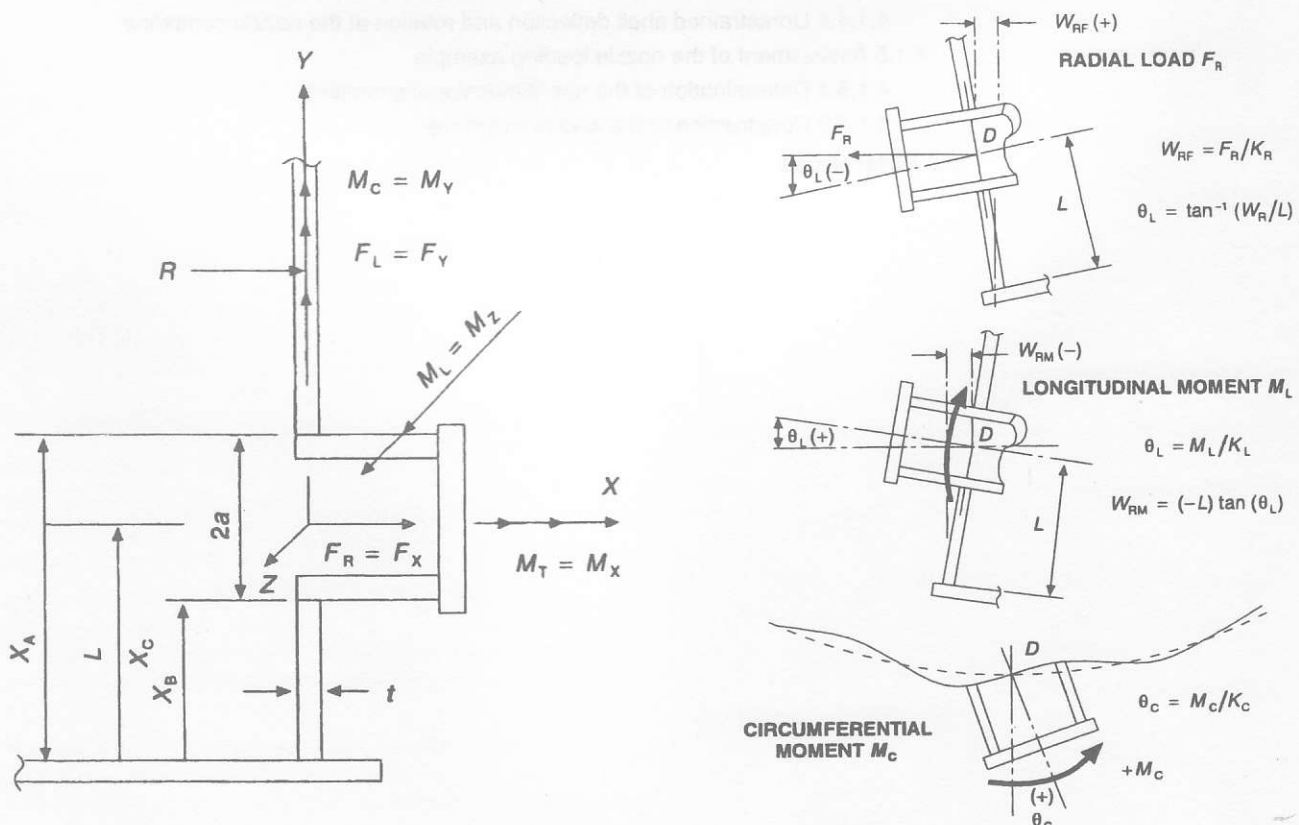


Figure 4.1 API 650 nomenclature for piping loads and deformation on nozzle together with three types of loading

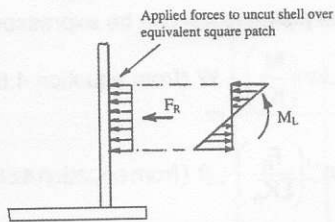


Figure 4.2 Diagrammatic presentation of pressure load distributions

4.1.1.1 The loading on the nozzle

Under the most general movement of the piping system, the nozzle will be subject to three forces and three moments acting in and about the orthogonal axes. However, only one force and two moments are considered significant in causing shell deformations.

These three types of loading are shown in Figure 4.1; they are: the radial thrust F_R , longitudinal moment M_L , applied in a vertical plane through the centre of the nozzle, and circumferential moment M_C , applied in a horizontal plane through the centre of the nozzle.

The above nozzle loadings were modelled assuming the nozzle radial load was uniformly distributed over an equivalent square patch of the uncut shell. That is the hole, the nozzle penetration and the nozzle geometry are ignored. The moment loadings were assumed to apply a triangular interface pressure load to the square patch of the uncut shell. These distributions are shown diagrammatically on Figure 4.2.

NOTE: This simplified approach, by which the nozzle local loading is transferred to the uncut vessel, is that used in WRC Bulletin 107 and BS 5500. However, in the WRC Bulletin 297 a more rigorous approach is adopted whereby the actual nozzle and shell are analysed, that is to say the shell is penetrated.

In addition to the deformations due to piping loads, there will be free-body deflections and rotations of the tank shell.

4.1.1.2 Definition of stiffness coefficients

The relationship between the elastic deformation of the tank shell nozzle connection and the external loads are expressed on Figure 4.1 in the following linear form:

$$F_R = K_R \times W_{RF} \quad \text{equ 4.1}$$

$$M_L = K_L \times \theta_L \quad \text{equ 4.2}$$

$$M_C = K_C \times \theta_C \quad \text{equ 4.3}$$

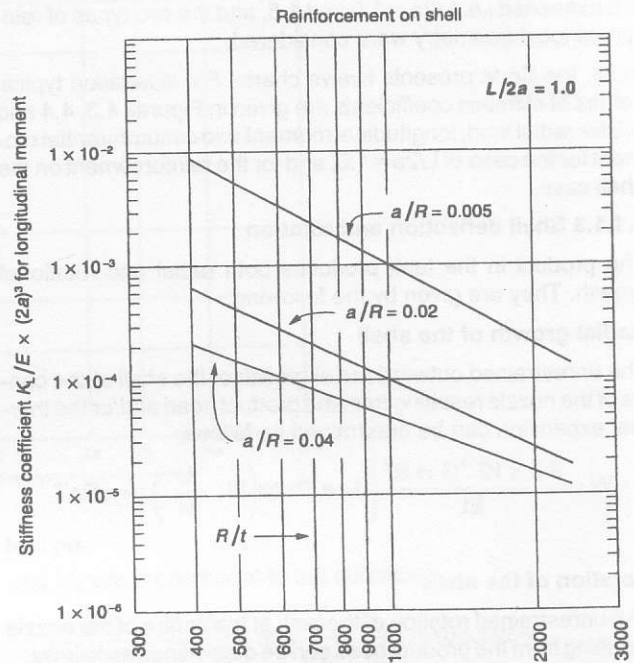
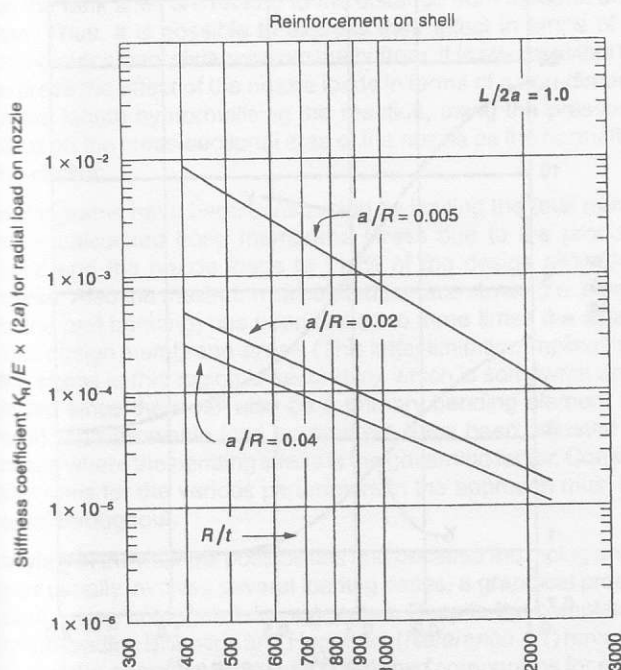
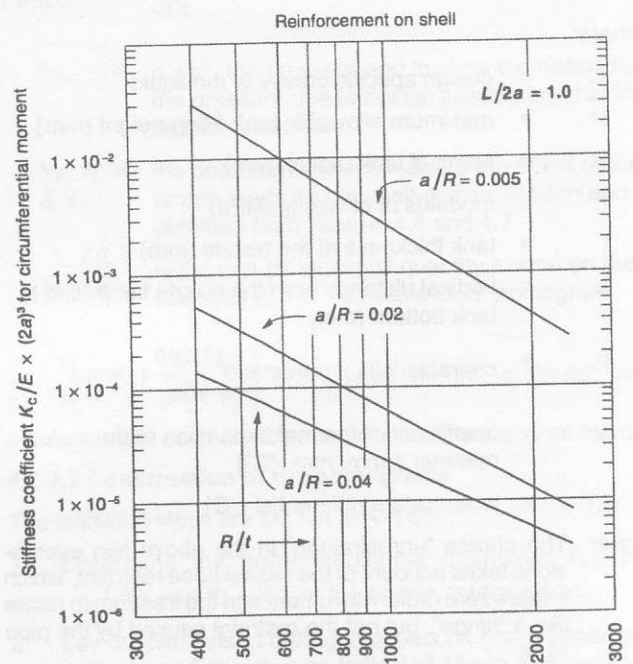
where:

K_R, K_L = stiffness coefficients
& K_C

W_{RF} = radial deflection of the tank at the nozzle connection

θ_L = rotation of the tank meridian in a vertical plane at the nozzle connection

θ_C = rotation in the horizontal plane at the nozzle connection due to a circumferential moment

Figure 4.4 Stiffness coefficient for longitudinal moment: Reinforcement in nozzle neck only ($L/2a = 1.0$)Figure 4.3 Stiffness coefficient for radial load: Reinforcement on shell ($L/2a = 1.0$)Figure 4.5 Stiffness coefficient for circumferential moment: Reinforcement on shell ($L/2a = 1.0$)

- F_R = radial thrust
 M_L = longitudinal moment
 M_C = circumferential moment

In relation to the equations 4.1 to 4.3, it should be noted that radial deflections and meridian rotations arise from both the radial thrust F_R and the longitudinal moment M_L . The resultant compatibility equations are given in Section 4.1.1.4, equations 4.6 to 4.8, which make this point clear. In API 650 there is no distinction between the displacements caused by the individual nozzle forces and the resultant displacements caused by all effects, the same symbol is used for both.

A computer program based upon the work of Kalnins was used to derive these stiffness coefficients, which are given in the Code in non-dimensional form. As indicated in Section 4.1.1, two values of the ratio, distance from the base/nozzle diameter, are examined i.e. $L/2a = 1.0$ and 1.5 , and the two types of reinforced local geometry were considered.

In all, the Code presents twelve charts. For illustration typical values of stiffness coefficients are given in Figures 4.3, 4.4 and 4.5 for radial load, longitudinal moment and circumferential moment for the case of $L/2a = 1.0$, and for the reinforcement on the shell case.

4.1.1.3 Shell deflection and rotation

The product in the tank produces both radial and rotational growth. They are given by the following:

Radial growth of the shell

The unrestrained outward radial growth of the shell at the centre of the nozzle resulting from the product head and/or the thermal expansion can be determined as follows:

$$W = \frac{9.8 \times 10^{-6} G H R^2}{Et} \left[1 - e^{-\beta L} \cos\left(\beta L - \frac{L}{H}\right) \right] + \alpha R \Delta T \quad \text{equ 4.4}$$

Rotation of the shell

The unrestrained rotation of the tank at the centre of the nozzle resulting from the product head can be determined as follows:

$$\theta = \frac{9.8 \times 10^{-6} G \cdot H \cdot R^2}{E \cdot t} \left(\frac{1}{H} - \beta e^{-\beta L} (\cos(\beta \cdot L) + \sin(\beta \cdot L)) \right) \quad \text{equ 4.5}$$

where:

- G = design specific gravity of the liquid
 H = maximum allowable tank filling height (mm)
 R = nominal tank radius (mm)
 E = modulus of elasticity (MPa)
 t = tank thickness at the nozzle (mm)
 L = vertical distance from the nozzle centreline to tank bottom (mm)
 β = characteristic parameter = $\frac{1.285}{\sqrt{Rt}}$ (1/mm)
 α = coefficient of thermal expansion of the tank material, [(mm/mm °C)]
 ΔT = temperature differential (°C)

Note: The phrase "unrestrained" in the above two expressions takes account of the vessel base restraint, which implies zero radial movement and the freedom to rotate like a "hinge", but not the restraint caused by the pipe work.

4.1.1.4 Determination of loads on the nozzle

The relationship between the elastic deformation of the nozzle

and the external piping loads can be expressed as follows:

$$W_R = \frac{F_R}{K_R} - L \tan\left(\frac{M_L}{K_L}\right) + W \quad \text{(from equation 4.67)} \quad \text{equ 4.6}$$

$$\theta_L = \frac{M_L}{K_L} - \tan^{-1}\left(\frac{F_R}{LK_R}\right) + \theta \quad \text{(from equation 4.68)} \quad \text{equ 4.7}$$

$$\theta_C = \frac{M_C}{K_C} \quad \text{equ 4.8}$$

W_R , θ_L and θ_C are the resultant radial deflection (in mm) and rotation (in radians) of the tank at the nozzle opening resulting from the piping loads F_R , M_L and M_C and the product head, pressure and uniform or differential temperature between the tank shell and the tank bottom.

In the above equations the deflections W and θ can be obtained from equations 4.4 and 4.5. The resultant deflection and rotations on the left-hand side of equations 4.6 to 4.8 must be equal to those from the connecting piping system, which can be obtained from a pipe work analysis. The problem, therefore, comes down to the solution of three simultaneous equations, where the unknowns are the three piping loads, F_R , M_L and M_C . The problem, therefore, is solved.

The assessment of these loads as given in API 650 are outlined in Section 4.1.2 and in Section 4.1.4 the details of the approach in Appendix P is shown by means of an example.

4.1.2 The assessment of nozzle loadings

4.1.2.1 Determination of allowable loads according to the API 650 approach

API 650 Appendix P provides a linear interaction diagram to establish an allowable load criterion for any "low type" nozzle configuration when several loads act together. The hoop stress due to the product head is taken into consideration in formulating the criteria. When the nozzle loads are acting to produce ten-

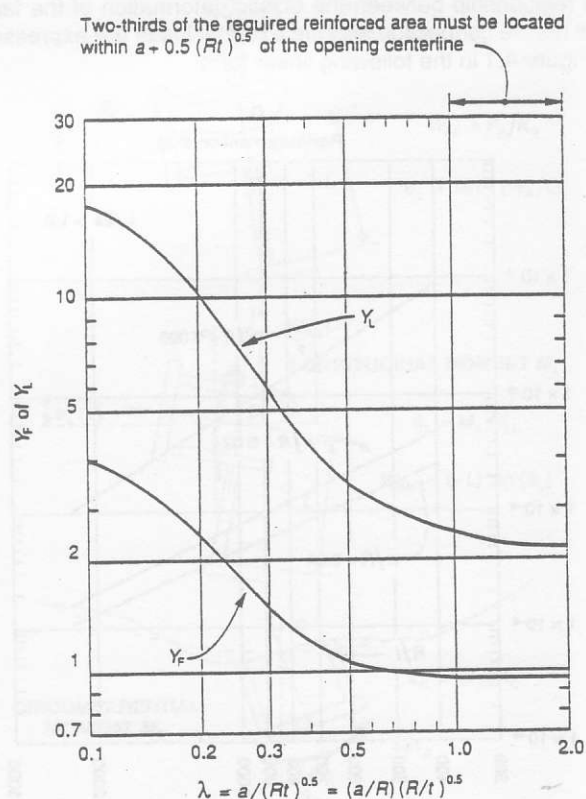
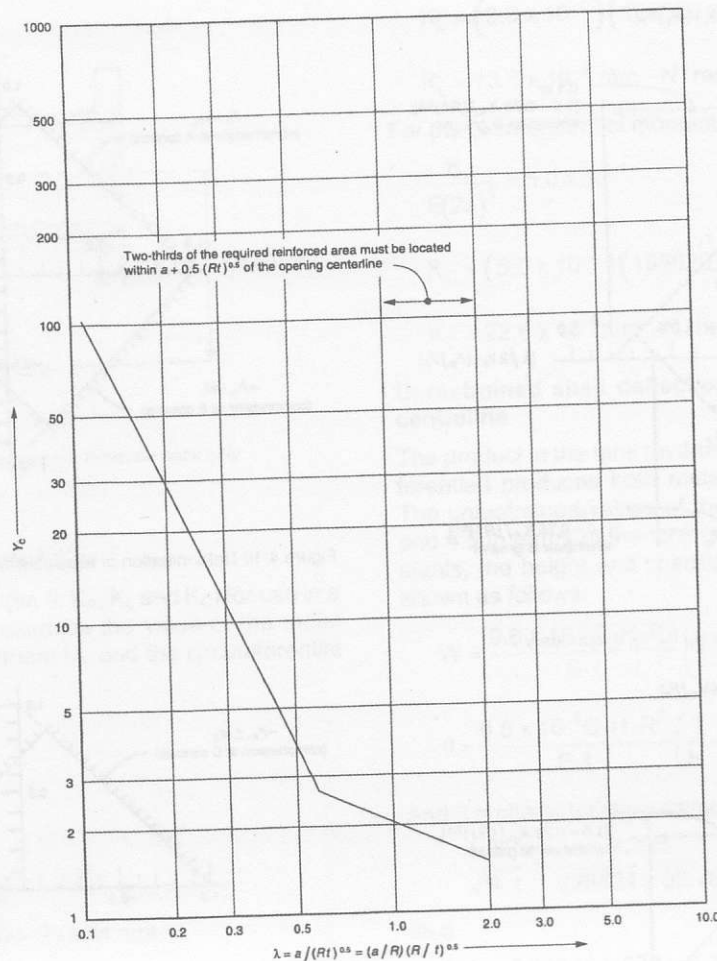


Figure 4.6 The coefficients Y_F and Y_L

Figure 4.7 Obtaining coefficient Y_c

sion in the areas of the tank shell which experience hoop tensile stresses due to product head, the criteria for allowable nozzle loads are more restrictive than when these nozzle reactions act in the opposite direction and their effect is mitigated by the product head.

The stresses due to the product head at a particular elevation on the tank shell are related to the distance from the tank bottom. Thus, it is possible to express their effect in terms of a non-dimensional distance from the bottom. It is also possible to express the effect of the nozzle loads in terms of a non-dimensional length by normalising the reaction, using the pressure force on the cross-sectional area of the nozzle as the normalising divisor.

Homograms have been constructed by limiting the total maximum calculated hoop membrane stress due to the product head and the nozzle loads to 110% of the design allowable stress. Also the maximum calculated surface stress (i.e. membrane and bending) has been limited to three times the allowable design membrane stress. (This latter limitation implies that the stress in this region is secondary, which is somewhat optimistic since there will also be a primary bending element as well). The allowable load parameters have been adjusted in cases where the bending stress is the governing factor. Consistent units for the various parameters in the approach must be used throughout.

In view of the multiple possibilities and because the piping analysis usually involves several loading cases, a graphical procedure, using nomograms is suggested. Despite the complexity of the loading Billimoris and Hagstrom (*Reference 4.1*) have reduced the approach to the use of only two nomograms for each nozzle configuration.

The non-dimensional stresses due to the piping loads F_R , M_L

and M_C are proportional to the quantities:

$$\frac{\lambda}{2Y_F} \left(\frac{F_R}{F_P} \right), \frac{\lambda}{aY_L} \left(\frac{M_L}{F_P} \right) \text{ and } \frac{\lambda}{aY_C} \left(\frac{M_C}{F_P} \right) \text{ respectively,}$$

where:

$$\lambda = \frac{a}{\sqrt{Rt}}$$

$$F_P = p \pi a^2, \text{ the pressure end load on the nozzle for the pressure due to design product head at the nozzle centreline}$$

$$Y_F, Y_L \text{ and } Y_C = \text{the coefficients which indicate the effect of the nozzle loads on the shell-nozzle junction and obtained from Figures 4.6 and 4.7}$$

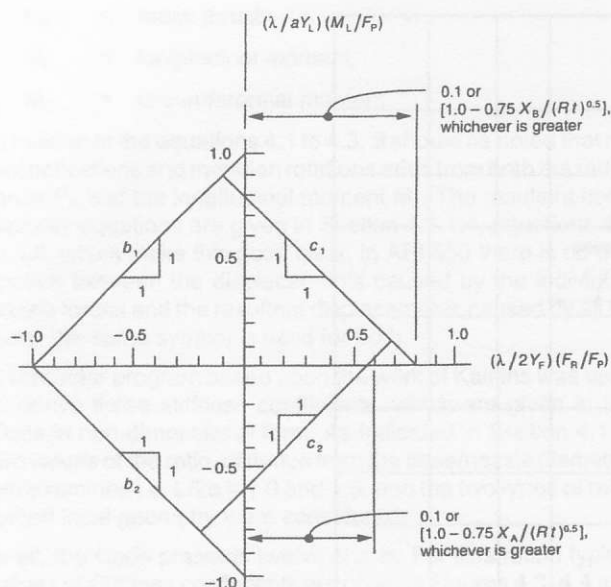
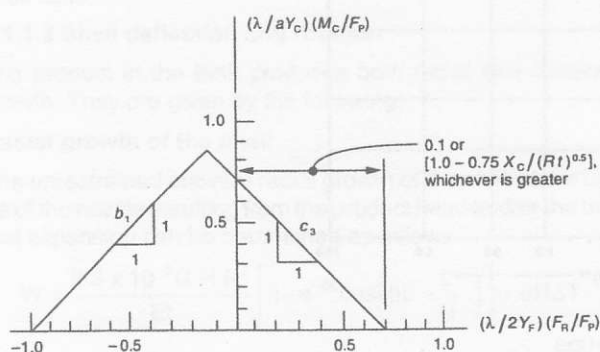
$$\frac{\lambda}{2Y_F} \left(\frac{F_R}{F_P} \right) = \text{the non-dimensional quantity plotted on the abscissa of the "allowable load" nomogram}$$

$\frac{\lambda}{aY_L} \left(\frac{M_L}{F_P} \right)$ and $\frac{\lambda}{aY_C} \left(\frac{M_C}{F_P} \right)$ are plotted on the ordinate; one nomogram for each combination of radial load and moment.

4.1.2.2 Construction of the nomograms

The following steps are set out in API 650:

1. Determine the non-dimensional quantities $X_A/\sqrt{Rt}$, $X_B/\sqrt{Rt}$ and $X_C/\sqrt{Rt}$ for the nozzle configuration.
2. Lay out two sets of orthogonal axes on graph paper with ordinate and abscissa as indicated above and shown in Figures 4.8 and 4.9.
3. Construct four boundaries for Figure 4.8 and two bound-

Figure 4.8 Construction of nomogram for b_1 , b_2 , c_1 , c_2 boundaryFigure 4.9 Construction of nomogram for b_1 , c_3 boundary

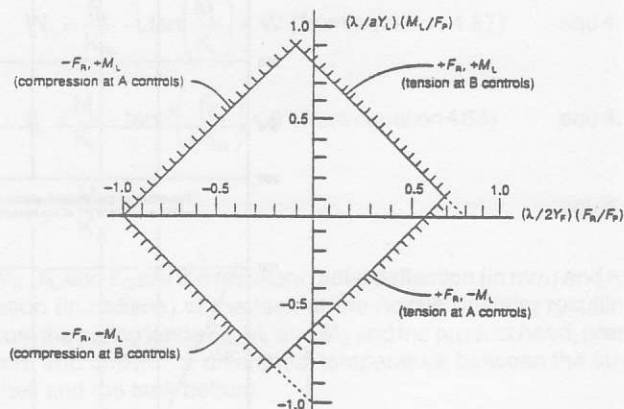
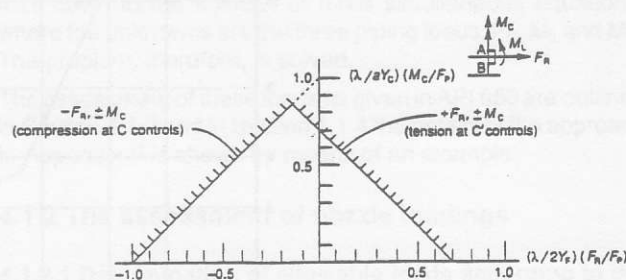
aries for Figure 4.9. Boundaries b_1 and b_2 are constructed as lines at 45° angles between the abscissa and the ordinate. Boundaries c_1 , c_2 , and c_3 are constructed as lines at 45° angles passing through the calculated values indicated on Figures 4.8 and 4.9. The shift in the 45° lines reflects the points made earlier concerning the necessity of restricting the tensile stresses when they are additive.

4.1.2.3 Determination of allowable loads

1. From the values of the local nozzle loading F_R , M_L and M_C , and the other parameters, the following quantities can be obtained:

$$\frac{\lambda}{2Y_F} \left(\frac{F_R}{F_P} \right), \frac{\lambda}{aY_L} \left(\frac{M_L}{F_P} \right) \text{ and } \frac{\lambda}{aY_C} \left(\frac{M_C}{F_P} \right).$$

2. Plot the point corresponding to $\frac{\lambda}{2Y_F} \left(\frac{F_R}{F_P} \right)$, $\frac{\lambda}{aY_L} \left(\frac{M_L}{F_P} \right)$ on the nomogram constructed as shown in Figure 4.8, redrawn in Figure 4.10.
3. Plot the point corresponding to $\frac{\lambda}{2Y_F} \left(\frac{F_R}{F_P} \right)$, $\frac{\lambda}{aY_C} \left(\frac{M_C}{F_P} \right)$ on the nomogram constructed as shown in Figure 4.9, redrawn in Figure 4.11.
4. For the piping loads to be acceptable both points must lie within the boundaries of the nomograms shown in Figures 4.10 and 4.11.

Figure 4.10 Determination of allowable loads from nomogram: F_R and M_L Figure 4.11 Determination of the allowable loads from nomogram: F_R and M_C

4.1.3 Concluding comments

The method set out in API 650 provides a method for determining the stiffness characteristics of the tank shell-to-nozzle junction, which can be used in a thorough piping analysis to determine the piping loads.

Having determined the piping loads, their magnitudes can be assessed by means of an interaction diagram set out in API 650. The ordinates of two nomograms are normalised with respect to the end pressure on the nozzle. Design limitations consistent with the various piping loads are built into these diagrams to provide the required design safety.

Note: Such an analysis is not provided in BS 2654. It could well be that this reflects a degree of uncertainty as to the validity, or value, of the newer methods of analysis. Perhaps further assessment of these methods is required.

4.1.4 Method of analysis example

The example given in API 650 Appendix P is used to illustrate the method of analysis to determine the forces which arise on a 610 mm (24") nozzle located near the bottom of the tank when connected to a simple pipework layout.

4.1.4.1 The problem

This is presented as follows:

A tank is 79.24 m (260 ft) in diameter and 19.506 m (64 ft) high, and its bottom course is 33.78 mm (1.33 in) thick. The tank has a low type nozzle with an outside diameter of 610 mm in accordance with API 650, and the nozzle centreline is 630 mm (24.75 in) up from the bottom plate, with reinforcement on the

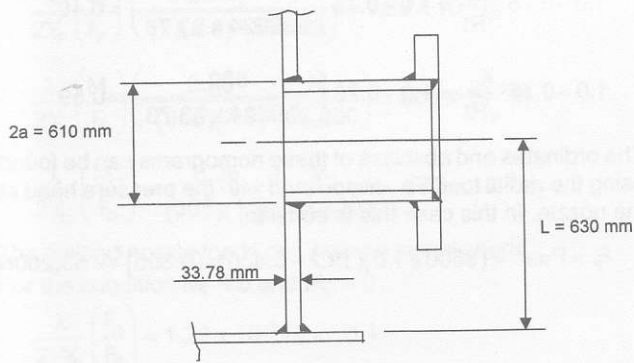


Figure 4.12 Low type nozzle with reinforcement in nozzle neck only

From API 650, Appendix P, figure P-6

opening (neck) only (see Figure 4.12).

Determine the end conditions (W , θ , K_R , K_L and K_C) for use in a piping analysis and hence determine the value of the radial thrust F_R , the longitudinal moment M_L and the circumferential moment M_C .

where:

$$\begin{aligned} a &= 305 \text{ mm} \\ L &= 630 \text{ mm} \\ H &= 19,506 \text{ mm} \\ \Delta T &= 93 - 21 = 72^\circ\text{C} \\ R &= 79.24/2 \times 1000 = 39,624 \text{ mm} \\ t &= 33.78 \text{ mm} \\ E &= 198,620 \text{ N/mm}^2 \\ G &= 1.0 \\ \alpha &= 0.0000012 \text{ mm/}^\circ\text{C} \end{aligned}$$

4.1.4.2 The solution

In the first instance API 650 Appendix P is used to determine the stiffness coefficients and the unrestrained shell deflection and rotation at the nozzle resulting from the hydrostatic head and the temperature differential. Thereafter these values are used in a pipework analysis to determine the thrust and moments at the nozzle. An assessment is made, using the method given in Appendix P, to determine the acceptability of the design.

The stiffness coefficients:

For the nozzle-tank connection

$$R/t = 39624 / 33.78 = 1173$$

$$a/R = 305 / 39624 = 0.008$$

$$L/2a = 630 / 610 \approx 1.0$$

For the radial load from Figure 4.13

$$\frac{K_R}{E(2a)} = 3.1 \times 10^{-4}$$

$$K_R = (3.1 \times 10^{-4})(198620 \times 610)$$

$$K_R = 37559 \text{ N/mm}$$

For the longitudinal moment from Figure 4.14

$$\frac{K_L}{E(2a)^3} = 3.0 \times 10^{-4}$$

$$K_L = (3.0 \times 10^{-4})(198620 \times 610^3)$$

$$K_L = 13.6 \times 10^{-9} \text{ mm-N/radian}$$

For the circumferential moment from Figure 4.15

$$\frac{K_C}{E(2a)^3} = 5.0 \times 10^{-4}$$

$$K_C = (5.0 \times 10^{-4})(198620 \times 610^3)$$

$$K_C = 22.6 \times 10^{-9} \text{ mm-N/radian}$$

Unrestrained shell deflection and rotation at the nozzle centreline

The product in the tank (hydrostatic head and temperature differential) produces both radial and rotational displacement. The unrestrained values of these are given by equations 4.4 and 4.5 in terms of the tank geometry, the tank material constants, the height and specific gravity of the liquid contents, shown as follows:

$$W = \frac{9.8 \times 10^{-6} G \cdot H \cdot R^2}{E \cdot t} \left(1 - e^{-\beta L} \cos \left(\beta \cdot L - \frac{L}{H} \right) \right) + \alpha \cdot R \cdot \Delta T$$

$$\theta = \frac{9.8 \times 10^{-6} G \cdot H \cdot R^2}{E \cdot t} \left(\frac{1}{H} - \beta e^{-\beta L} (\cos(\beta \cdot L) + \sin(\beta \cdot L)) \right)$$

And β = characteristic parameter

$$= \frac{1.285}{\sqrt{R \cdot t}} = \frac{1.285}{\sqrt{39624 \times 33.78}} = 0.00111$$

thus

$$\beta \cdot L = 0.00111 \times 630 = 0.7 \text{ rad}$$

Substituting into equation 4.4:

$$W = \frac{9.8 \times 10^{-6} \times 1.0 \times 19506 \times 39624^2}{198620 \times 33.78}$$

$$\left(\left(1 - e^{-\beta L} \cos 0.7 \right) - \frac{630}{19506} \right)$$

$$+ (12 \times 10 \times 39624 \times 72)$$

$$W = 44.73 \times (1 - (0.4966 \times 0.7648) - 0.0323) + 34.23$$

$$W = 60.53 \text{ mm (API gives 59.7 mm, using rounded values)}$$

Substituting into equation 4.5:

$$\theta = \frac{9.8 \times 10^{-6} G \cdot H \cdot R^2}{E \cdot t}$$

$$\left(\frac{1}{H} - \beta \cdot e^{-\beta L} (\cos(\beta \cdot L) + \sin(\beta \cdot L)) \right)$$

$$\theta = \frac{9.8 \times 10^{-6} \times 1.0 \times 19506 \times 39624^2}{198620 \times 33.78}$$

$$\left(\frac{1}{19506} - 0.00111 \times 0.4966 (\cos 0.7 + \sin 0.7) \right)$$

$$\theta = 44.73 \times$$

$$\{ 0.00005127 - (0.00111 \times 0.4966 \times (0.7648 + 0.6442)) \}$$

$$\theta = 44.73 \times -0.0007254$$

$$\theta = -0.032 \text{ radians (as given in API) } = 1.833 \text{ degrees}$$

4.1.5 Assessment of the nozzle loading example

As indicated in Section 4.1.2, using the approach in API 650

Appendix P provides an interaction diagram to establish the allowable loads. The background to the criteria and details of the method of construction of the nomograms has already been given. The example given here uses the method and plots the four cases on the resulting nomograms.

4.1.5.1 Determination of the non-dimensional quantities

From the nozzle illustrated in Figure 4.12 the following values can be found:

$$X_A = 935 \text{ mm at the top of the nozzle}$$

$$X_B = 325 \text{ mm at the bottom of the nozzle}$$

$$X_C = 630 \text{ mm at the centre line of the nozzle}$$

Using these the following non-dimensional quantities are as follows:

$$\frac{X_A}{\sqrt{Rt}} = \frac{935}{\sqrt{39624 \times 33.78}} = 0.81$$

$$\frac{X_B}{\sqrt{Rt}} = \frac{325}{\sqrt{39624 \times 33.78}} = 0.28$$

$$\frac{X_C}{\sqrt{Rt}} = \frac{630}{\sqrt{39624 \times 33.78}} = 0.54$$

$$\lambda = \frac{a}{\sqrt{Rt}} = \frac{305}{\sqrt{39624 \times 33.78}} = 0.26$$

From Figures 4.6 and 4.7, the values of Y_L , Y_F and Y_C can be found.

$$Y_L = 7.8$$

$$Y_F = 1.9$$

$$Y_C = 15.0$$

4.1.5.2 Construction of the load nomograms

From these values a nomogram can be constructed.

$$1.0 - 0.75 \frac{X_B}{\sqrt{Rt}} = 1.0 - 0.75 \left(\frac{325}{\sqrt{39624 \times 33.78}} \right) = 0.79$$

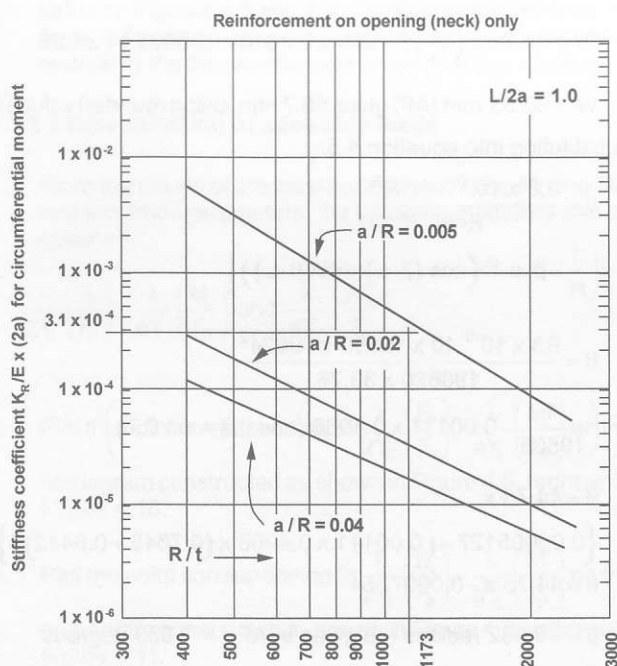


Figure 4.13 Stiffness coefficient for radial load: Reinforcement in nozzle neck only ($L/2a = 1.0$)

From API 650, figure P-2G

$$1.0 - 0.75 \frac{X_A}{\sqrt{Rt}} = 1.0 - 0.75 \left(\frac{935}{\sqrt{39624 \times 33.78}} \right) = 0.40$$

$$1.0 - 0.75 \frac{X_C}{\sqrt{Rt}} = 1.0 - 0.75 \left(\frac{630}{\sqrt{39624 \times 33.78}} \right) = 0.59$$

The ordinates and abscissa of these nomograms can be found using the radial load F_p , associated with the pressure head at the nozzle. In this case this is equal to:

$$F_p = P\pi a^2 = (9800)(1.0)(19.2 - 0.630)\pi(0.305)^2 = 53,200\text{N}$$

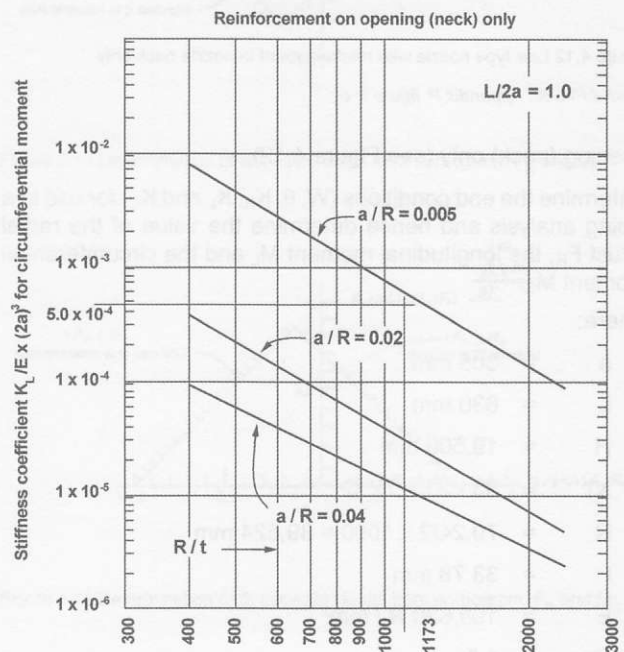


Figure 4.14 Stiffness coefficient for longitudinal moment: Reinforcement in nozzle neck only ($L/2a = 1.0$)

From API 650, figure P-2H

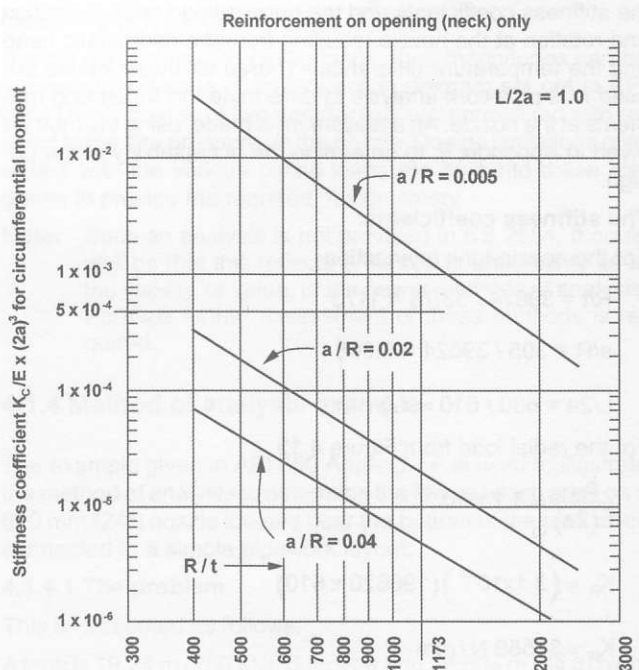


Figure 4.15 Stiffness coefficient for circumferential moment: Reinforcement in nozzle neck only ($L/2a = 1.0$)

From API 650, figure P-2I

$$\frac{\lambda}{2Y_F} \left(\frac{F_R}{F_P} \right) = \frac{0.26}{(2)(2.0)} \left(\frac{F_R}{53,200} \right) = 1.22 \times 10^{-6} F_R$$

$$\frac{\lambda}{2Y_L} \left(\frac{M_L}{F_P} \right) = \frac{0.26}{(305)(7.8)} \left(\frac{M_L}{53,200} \right) = 2.05 \times 10^{-9} M_L$$

$$\frac{\lambda}{2Y_C} \left(\frac{M_C}{F_P} \right) = \frac{0.26}{(305)(15)} \left(\frac{M_C}{53,200} \right) = 1.07 \times 10^{-9} M_C$$

The limiting nozzle loads can now be established.

For the condition $M_L = 0$ and $M_C = 0$

$$\frac{\lambda}{2 \cdot Y_F} \left(\frac{F_R}{F_P} \right) = 1.22 \times 10^{-6} F_R \leq 0.4$$

and hence

$$F_{R_{\max}} = \frac{0.4}{1.22 \times 10^{-6}} = 328,000 \text{ N (tension at 'A' controls)}$$

For the condition $M_L = 0$ and $F_R = 0$

$$\frac{\lambda}{a \cdot Y_C} \left(\frac{M_C}{F_P} \right) = 1.07 \times 10^{-9} M_C \leq 0.59$$

and hence

$$M_{C_{\max}} = \frac{0.59}{1.07 \times 10^{-9}} = 550 \times 10^6 \text{ N.mm}$$

(tension at 'C' controls)

For the condition $M_C = 0$ and $F_R = 0$

$$\frac{\lambda}{a \cdot Y_L} \left(\frac{M_L}{F_P} \right) = 2.05 \times 10^{-9} M_L \leq 0.4$$

and hence

$$M_{L_{\max}} = \frac{0.4}{2.05 \times 10^{-9}} = 195 \times 10^6 \text{ N.mm}$$

(tension at 'A' controls)

A summary of the limiting nozzle loadings are:

$$F_{R_{\max}} = 328,000 \text{ N (tension at 'A' controls)}$$

$$M_{C_{\max}} = 550 \times 10^6 \text{ N.mm (tension at 'C' controls)}$$

$$M_{L_{\max}} = 195 \times 10^6 \text{ N.mm (tension at 'A' controls)}$$

4.2 References

- 4.1 *Stiffness Coefficients and Allowable Loads for Nozzles in Flat Bottom Storage Tanks*, H. D. Billimoris and J. Hagstrom, ASME Jn Pres Ves Techn 100 (4), 1978 p. 389.

5 The design of tank roofs - fixed

The design of fixed roofs for atmospheric storage tanks has not undergone any radical change for a considerable period of time. Designs are based almost entirely on the practices and experiences of tank users in the petrochemical industry over many years and the design rules which are laid down in the various Codes.

The most influential and widely used tank Code is American API 650. This Code was first published as API 12C in 1936 and since the early 60s the design rules for tank roofs have not changed significantly.

The British Standard for atmospheric storage tanks BS 2654 has taken a different approach to the American Code in many areas of tank design, but in terms of tank roof design, it has followed the API rules almost exactly. The design of floating roofs is discussed in Chapter 6.

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5.1 The design of tank roofs

This is an area of design which has been effectively fossilised for some 40 years. This is perhaps largely because the existing designs work well giving little incentive for innovation and that the savings to be made are modest in comparison with the perceived risks of new and untried designs being used.

Tank roofs perform the basic function of keeping the elements – and possibly the occasional bird – out of the stored product, and, with varying degrees of success, keeping product vapours out of the atmosphere. The various types of roofs are outlined below.

5.1.1 Basic types

There are two main types of tank roof and these are illustrated in Figure 5.1.

- The first type is the fixed roof
- The second type is the floating roof

Both fixed and floating roofs are available in a number of different forms. Fixed roofs are discussed in this Chapter and floating roofs are discussed in Chapter 6.

5.1.2 Differences between fixed and floating roofs

One of the disadvantages of the fixed roof tank, especially with the more volatile products, is the loss of product vapour which occurs for two reasons.

Firstly, the diurnal changes in atmospheric temperature cause "breathing losses".

Secondly, the import and export of product to and from the tank causes "filling" losses.

The emission of large volumes of product vapour into the atmosphere is both costly and environmentally undesirable. This problem is largely solved by the floating roof tank where the roof sits on the surface of the product and moves up and down as product is imported and exported and thus the majority of the vapours are contained under the roof.

There is also a hybrid of these two main types of roof and that is where an internal floating cover, which is of a much lighter construction than the normal floating roof, is fitted within a fixed roof tank. This internal cover may be fitted to the tank when it is first built, or it may be retro-fitted at a later date since the components for these types of cover are designed to fit through a standard 24" (610 mm) shell manhole.

These internal covers are used for the following reasons:

- Where a tank service is changed to the storage of a more volatile product.
- Where changes to either environmental or safety considerations require the reduction of vapour emissions.
- Where the vapours of a highly volatile product have to be contained and also there is a need to ensure that the product is kept dry and not contaminated with rainwater.

5.2 Fixed roofs

5.2.1 Design basis

The basic design parameters are laid down in the most widely used Codes BS 2654, API 650 and the proposed European Code prEN 14015. There are other national and company specific Standards, which may partially supersede or augment parts of these tank Codes, but they are too numerous to be in-

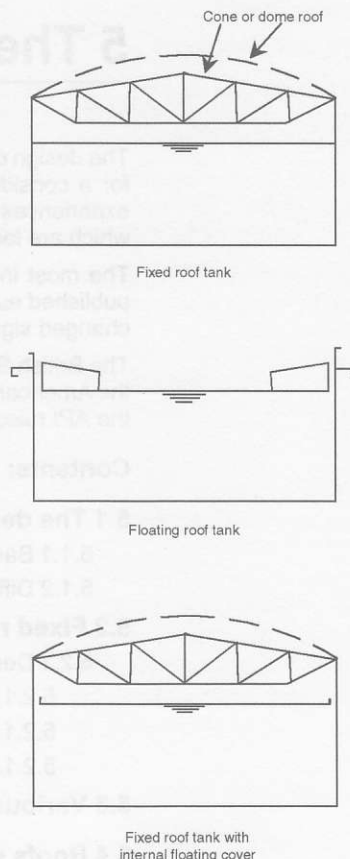


Figure 5.1 Types of tank roof

cluded here and would be applied by the designer as directed by the tank purchaser, on a job-by-job basis.

5.2.1.1 Design loadings

- An external superimposed load of a minimum of 1.2 kN/m² (25 lb/in²).

In the case of the American Code, this load is deemed to include dead load plus a uniform live load.

For the British Code, this load is the sum of either internal vacuum and snow load, or, internal vacuum and live load.

This loading generally dictates the thickness of the roof sheeting for roofs without supporting structures, and dictates the nature of the supporting structure for roofs which have such structures.

- Internal pressure. The British Code states that this can be between 7.5 and 56 mbar.

It is usual to specify a modest design pressure, but in special circumstances, higher pressures can be used (see Chapter 4, Section 4.3.2.8). As the pressure increases, so does its influence not only on the thickness of the shell and roof plating, but also on the size of the compression area at the roof-to-shell junction (see Chapter 4, Section 3.7) and on the requirements for anchorage to prevent tank uplift (see Chapter 3, Section 3.8.10 and 3.9).

The American Code is based on the tank operating at atmospheric pressure, or that internal pressure which equates to the weight of the $\frac{3}{16}$ " (4.76 mm) roof plates i.e. 4 mbar. The exception to this is covered by Appendix F of the Code which gives the requirements for tanks operating at up to 2½ lbs/in².g (172 mbar).

- Exceptional loadings. These may include the possibility of an internal explosion or sudden overpressure due to abnormal causes. For such cases it is usual to specify a frangible shell-to-roof joint which fails preferentially to relieve the high internal pressure, whilst continuing to contain the stored product. (see Chapter 3, Section 3.8).

5.2.1.2 Design methods

a) Roof plating

Apart from exceptional circumstances, the minimum roof sheet thickness allowable is specified in the Codes.

The British Code requires a minimum thickness of 5 mm, whilst the American Code calls for $\frac{3}{16}$ " (4.76 mm).

Apparently these minimum thicknesses are based on N.E.P.A. 78 Lightning Protection Code which states "— steel sheet less than $\frac{3}{16}$ " (4.76mm) in thickness may be punctured by severe strikes and shall not be relied upon as protection for direct lightning strikes".

b) Roof framing

The British Code refers to the Structural Steel design Code BS 449.

The American Code contains its own rules taken from various publications (*References 5.3, 5.4 and 5.5*).

5.2.1.3 Code requirements

The rules for designing and detailing tank roofs are covered fully in both the British and American Codes and these should be followed carefully during the design process. Some of the major requirements are given here as follows:

From the British Code

- The spacing of roof plate supporting members for cone roof tanks shall be such that the span between them does not exceed 2 metres where one edge of the panel is supported by the top curb angle. Where this support is not present, the span shall not exceed 1.7 metres.
- For dome roofs this spacing may be increased as agreed between the tank purchaser and the manufacturer.
- The roof plating shall be continuously welded to the shell curb angle.
- For tanks exceeding 12.5 metres diameter, roof plates shall not be attached to the roof supporting structure.
- The roof plates are normally lapped by a minimum of 25mm and fillet-welded on the top side only. The laps should be arranged such that the lower edge of the uppermost plate is beneath the upper edge of the lower plate (the opposite way to that of tiles on the roof of a building) in order to minimise the possibility of moisture due to condensation on the underside of the plates entering the internal lap joint.

Note: The American Code shows the laps the opposite way to this, presumably to allow the roof to shed rain water. Depending upon the stored product, it may be sometimes necessary for the lap joint to be welded on both sides or made as a butt joint.

- The slope of cone roofs is generally 1:5 or for column-supported roofs 1:16. The radii of domed roofs is generally between 0.8D and 1.5D, where D is the tank diameter.
- The minimum thickness for structural sections shall be 5mm (excluding any corrosion allowance) but this does not apply to the webs of rolled steel joists, channels or packings, or to structures where special provisions against corrosion have been made.
- Roof plate joints are considered to have the following joint efficiencies:
 - 1.0 for butt-welded joints.
 - 0.35 for lapped joints with fillet welds on one side.
 - 0.5 for lapped joints with fillet welds on both sides.
- The allowable stress shall be taken as $\frac{2}{3}$ of the minimum specified yield strength of the roof plate material. In special circumstances, increases in joint efficiency may be permitted,

by agreement between the tank purchaser and the manufacturer, provided that this can be justified by special procedure tests simulating the actual configuration to be used on site.

- Cross bracing shall be provided in the plane of the roof, in at least in two bays, i.e. between two pairs of adjacent rafters, on all roofs more than 15 metres in diameter. Sets of bracing shall be equi-spaced around the tank circumference.
- Vertical bracing on trussed roof structures only shall be provided in an approximate vertical plane between trusses as follows:

For roofs more than 15 metres diameter — 1 ring.

For roofs more than 25 metres diameter — 2 rings.

These ring(s) shall be at the end of the trusses which are near to the tank shell.

From the American Code

- Roof plates shall be attached to the top angle of the tank by a continuous fillet weld on the top side only. Figure 3-3A in the Code shows the roof plates laps to be the same configuration as tiles on the roof of a building. i.e. opposite to the British Code.
- All internal and external structural members shall have a minimum nominal thickness of 4.3 mm (0.17") in any component. The method of providing a corrosion allowance, if any, for the structural members shall be a matter of agreement between the purchaser and the manufacturer.
- The minimum thickness of any structural member, including any corrosion allowance on the exposed side or sides, shall not be less than 6mm (0.25"), for columns, knee braces and beams or stiffeners which by design normally resist axial compressive forces, or 4.3 mm (0.17") for any other structural member.
- Roof plates of supported cone roofs shall not be attached to the supporting members.
- For all types of roofs, the plates may be stiffened by sections welded to the plates but may not be stiffened by sections welded to the supporting rafters or girders
- When the purchaser specifies lateral loads that will be imposed on the roof supporting columns (when used), the columns must be proportioned to meet the requirements for combined axial compression and bending as specified in the Code.
- The slope of supported cone roofs shall be 19 mm in 300 mm ($\frac{3}{4}$ " in 12") or greater if specified by the purchaser.

Note: This slope of 1 in 16 is fairly flat and is usually used for column-supported roofs. Roofs which are supported by radial rafters or trusses and without internal columns, normally have a slope of 1 in 6 (the maximum allowable to this Code for a frangible roof). This is because the steeper slope favours the production of a more economical rafter or truss design.

- Main roof supporting members of column-supported roofs, which are in contact with the roof plates, (excluding radial rafters carrying dead loads only) shall be considered as receiving no lateral support from the roof plates and shall be laterally braced, if necessary, by other acceptable methods. Radial rafters carrying dead loads plus live loads, which are in contact with the roof plates applying the live loading to the rafters, may be considered as receiving adequate lateral support from the friction between the roof plates and the compression flanges of the rafters, with the following exceptions:

- a) Trusses and open web joints used as rafters.

- b) Rafters with a nominal depth greater than 375mm.
- c) Rafters with a slope greater than 1 in 6.
- Rafters for supported cone roofs shall be spaced so that in the outer ring, their centres are not more than 0.6π metres = 1.885 metres (2π ft = 6.283ft) apart, measured along the circumference of the tank. Spacing on inner rings shall not be greater than 1.7 metres ($5\frac{1}{2}$ ft). When specified by the purchaser, for tanks located in areas subject to earthquakes, 19mm ($\frac{3}{4}$ ") diameter tie rods (or their equivalent) shall be placed between the rafters in the outer rings. These tie rods may be omitted if I-sections or H-sections are used as rafters.
- Self-supporting cone roofs shall have a minimum thickness of 5 mm ($\frac{3}{16}$ ") and a maximum of 12.5 mm ($\frac{1}{2}$ ") excluding any corrosion allowance.
- The slope of self-supporting roofs shall be within the range of 9.5° to 37° which is (1: 6 to 1: 1.333). The method of calculating the required thickness for a self-supporting cone roof is described later in Section 5.4.1.4.
- The requirements for roofs in the draft form of European Code for prEN 14015 - 1, are basically the same as that given in BS 2654.

5.3 Various forms of fixed roofs

Figure 5.2 summarises the various types of fixed roofs in common use.

5.4 Roofs with no supporting structure

5.4.1 Cone roofs

The British Code states that the slope of the roof shall comply with the requirements specified by the purchaser or shall be 1 in 5.

1 Roofs with no supporting structures

a) Cone roofs

- i) Self supporting cone
- ii) Folded plate petal type

b) Dome roofs

- i) Simple dome
- ii) Umbrella type

2 Roofs with supporting structures

a) Cone roofs

- i) Radial rafter type
- ii) Trussed frame type
- iii) Externally framed type

b) Dome roofs

- i) Radial rafter type
- ii) Externally framed type
- iii) Other types

3 Column-supported roofs

The American Code is more specific and says that the slope shall be within the range of 9.5° to 37° which is (1 in 6, to 1 in 1.333).

5.4.1.2 Thickness of roof plating

The British Code states that the minimum thickness of roof plating shall be 5 mm, excluding any corrosion allowance.

The American Code states that self-supporting cone roofs shall have a minimum thickness of 5 mm ($\frac{3}{16}$ ") and a maximum of 12.5 mm ($\frac{1}{2}$ ") excluding any corrosion allowance.

5.4.1.3 Self-supporting cone (or membrane roof)

The design loadings for self-supporting cone roofs are sustained entirely by the roof sheeting itself, without any supporting structure. Generally this type of roof is confined to smaller tanks, up to say 8 metres diameter.

The lack of an internal structure makes the roof ideal for:

- Tanks which require the application of an internal lining, where a internal structure would hamper the lining process.
- Tanks where a high internal corrosion allowance is specified, thus avoiding the requirement for a support structure in very thick steel sections.
- Tanks where stainless steel roof materials are required. There is a limited range of stainless steel sections which are available and therefore a membrane roof obviates the need for any support structure.

5.4.1.4 British Code — Design requirements

Equations 5.1 and 5.4 for the thickness of a self-supporting cone roof, are based on work done by the late Professor A.S. Tooth, see *Reference 5.1* and are derived as follows:

The membrane stress for a conical roof under internal pressure occurs in the circumferential direction at the roof-to-shell junction and is given by:

$$f = \frac{p \cdot r_s}{t_{rc} \cdot \sin \theta}$$

and therefore:

$$t_{rc} = \frac{p \cdot r_s}{f \cdot \eta \cdot \sin \theta} \quad \text{equ 5.1}$$

where:

- f = membrane stress (N/mm²)
- p = internal pressure (mbar)
- r_s = radius of tank shell (m)
- t_{rc} = thickness of cone roof plating (m)
- θ = the slope of the roof measured from the horizontal (degrees)
- η = joint efficiency. For self-supporting roofs the BS Code only allows butt-welded roof joints where $\eta = 1.0$, or double lap-welded joints where $\eta = 0.5$.

To express equation 5.1 in terms of the radius of the cone roof ' r_c ' at the point where it meets the shell, instead of the shell radius ' r_s ', it can be seen from Figure 5.3 that:

$$\sin \theta = \frac{r_s}{r_c}$$

Substituting for ' $\sin \theta$ ' in equation 5.1 then:

$$t_{rc} = \frac{p \cdot r_c}{f \cdot \eta} \quad \text{equ 5.2}$$

This equation has to be adjusted to accept the varying units as follows:

Figure 5.2 Various types of fixed roofs

$$t_{rc} = \frac{p \cdot r_c \cdot 10^3}{10^4 \cdot f \cdot \eta} = \frac{p \cdot r_c}{10 \cdot f \cdot \eta} \quad \text{equ 5.3}$$

This is the equation which is given in the British Code for the thickness of unsupported cone roofs.

The roof must also be checked to withstand the external pressure due to the roof loading and vacuum. This is achieved by referring to the classical theory for buckling pressure for a perfect sphere and adapting this for the cone roof.

The buckling pressure for a perfect sphere is

$$q' = \frac{2 \cdot E \cdot t_{rd}^2}{r_d^2 \cdot \sqrt{3(1-\nu^2)}} \quad \text{equ 5.4}$$

where:

- q' = the buckling pressure (mbar)
- r_d = the radius of the sphere (m)
- E = Young's Modulus (N/mm²)
- t_{rd} = the thickness of the roof plate (m)
- ν = Poisson's ratio

Using a value of 0.3 for Poisson's ratio the equation becomes:

$$q' = \frac{1.21 \cdot E \cdot t_{rd}^2}{r_d^2} \quad \text{equ 5.5}$$

This expression only applies to a perfect sphere and does not allow for imperfections in fabrication or for a factor of safety. The British Code applied a factor of approximately 20 to equation 5.5.

This then gives an equation for the safe allowable external pressure 'Pe':

$$Pe = \frac{0.0625 \cdot E \cdot t_{rd}^2}{r_d^2} \quad \text{equ 5.6}$$

Rearranging this equation for t_{rd} we obtain:

$$t_{rd} = \sqrt{\frac{Pe \cdot r_d^2}{0.0625 \cdot E}} = 4 \cdot r_d \sqrt{\frac{Pe}{E}}$$

where::

- Pe = allowable safe external pressure (kN/m²)
- r_d = spherical radius of the dome (m)
- E = Young's Modulus (N/mm²)

Writing the equation for these units gives:

$$t_{rd} = 4 \cdot 1000 \cdot r_d \sqrt{\frac{10 \cdot Pe}{10,000 \cdot E}} \quad \text{equ 5.7}$$

$$t_{rd} = 40 \cdot r_d \sqrt{\frac{10 \cdot Pe}{E}}$$

For a cone roof tank 'rd' is the radius at the point where the roof joins the shell and is given the notation 'rc' and from Figure 5.3:

$$r_c = \frac{r_s}{\sin \theta}$$

Substituting for 'rd' in equation 5.7 gives 't_{rc}' for cone roof tanks as:

$$t_{rc} = \frac{40 \cdot r_s}{\sin \theta} \sqrt{\frac{10 \cdot Pe}{E}} \quad \text{equ 5.8}$$

t_{rc} shall not be less than 5 mm, excluding corrosion allowance.

The form of this equation given in the British Code is that of equation 5.7, as in this form it can be used for both cone and

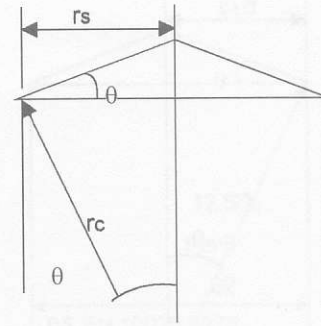


Figure 5.3 Equation 5.1 derivation

dome roofs simply by inserting the relevant value for the roof radius.

5.4.1.5 American Code — Design requirements

Self-supporting cone roofs shall have a minimum thickness of 5 mm ($\frac{3}{16}$ ") and a maximum of 12.5 mm ($\frac{1}{2}$ ") excluding any corrosion allowance.

The slope of self-supporting roofs shall be within the range of 9.5° to 37° which is (1 : 6 to 1 : 1.333).

The API 650 Code is based on tanks working at atmospheric pressure and the section which deals with self-supporting cone roofs (Section 3.10.5, in the Code) therefore, only deals with the calculation for external pressure considerations. For cases where tanks have to be designed for internal pressures, the designer is required to refer to Appendix F, Clause F.7.3 of the Code, which in turn refers to API 620 for such designs.

For external pressures the theory for buckling given above in equation 5.7 applies, except that in the American Code the following values are assumed:

- The value of Young's Modulus $E = 29 \times 10^6$ lb/in² (200,000 N/mm²)
- The external roof loading is taken as, a live load of 25 lb/ft² (1.2 kN/m²) plus a dead load of 20 lb/ft² (approximately 1.0 kN/m²), which is the self-weight of $\frac{1}{2}$ " (12.5 mm) roof plating – the maximum thickness allowed.

Also the American Code uses the tank diameter rather than the roof radius in its equation.

From equation 5.8

$$t_{rc} = \frac{40 \cdot r_s}{\sin \theta} \sqrt{\frac{10 \cdot Pe}{E}}$$

as:

$$r_s = D/2$$

and:

$$Pe = 2.2 \text{ kN/m}^2 \text{ and } E = 200,000 \text{ kN/m}^2$$

Then equation 5.8 becomes:

$$t_{rc} = \frac{40 \cdot D}{2 \cdot \sin \theta} \sqrt{\frac{10 \cdot 2.2}{200,000}}$$

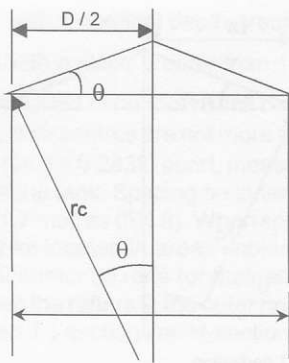
$$t_{rc} = \frac{0.20976 \cdot D}{\sin \theta}$$

$$t_{rc} = \frac{D}{4.8 \sin \theta} \quad \text{equ 5.9}$$

where:

D is in metres

t_{rc} can only be a minimum of 5mm, and a maximum of 12.5 mm, excluding corrosion allowance.



Equation 5.9 is given in the American Code.

Note: When the sum of the live and dead loads exceeds 2.2 kN/m^2 , the minimum thickness shall be increased by the following ratio:

$$\sqrt{\frac{\text{live load} + \text{dead load}}{2.2 \text{ kN/m}^2}}$$

The American Code also states that: — "Self-supporting roofs, to the American Code, whose roof plates are stiffened by sections welded to the plates need not conform to the minimum

thickness requirements, but the thickness of the roof plates shall not be less than $5 \text{ mm} (\frac{3}{16} \text{ inch})$ when so designed by the manufacturer, subject to the approval of the purchaser".

This means that a membrane roof whose thickness calculates to be more than the maximum allowable of $12.5 \text{ mm} (\frac{1}{2} \text{ inch})$ can be re-designed by other means to allow for the inclusion of stiffeners which are welded to the roof plates. Because storage tanks are generally designed for small internal pressures, the thickness of the unsupported roof is usually determined by the external, rather than the internal pressure to which the tank is subjected.

5.4.1.6 Folded plate type cone roof

This type of construction was originally devised by the Shell International Petroleum Company and is included in its *Tank Design and Engineering Practice Manual*.

For this type of roof, illustrated in Figure 5.4, one edge of each of the radial roof plate panels is flanged into the form of a channel section to form an integral supporting structure. This type of roof construction is limited by the British Code to tanks up to 12.5 metres diameter.

Normally the plate folds are internal, but for specific cases where a smooth interior surface is required for the application of an internal lining, the petals can be external to the tank.

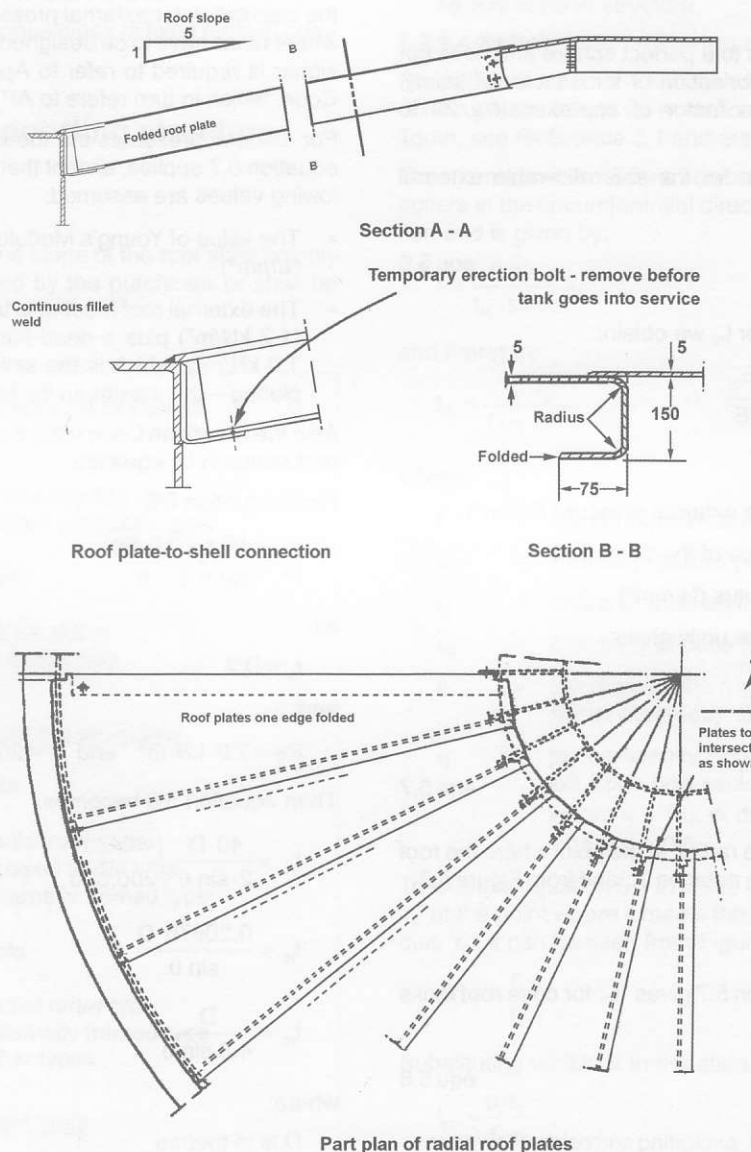


Figure 5.4 Folded plate type cone roof design

DESIGN FOR A FOLDED PLATE PETAL CONE ROOF.
DESIGNED TO BSS 449 & "FORMULAS FOR STRESS
& STRAIN" 5th EDITION BY ROARK & YOUNG.

TANK DIA.	12.5 m
ROOF SLOPE	1 in ?
No. OF PETALS	32
MAT'L. TYPE	BS EN 10025 S275
YIELD or 1% PROOF STRESS	275 N/mm ²
PLATE THKS.	5 mm
CORR. ALLOWANCE	0 mm
DESIGN PLATE THKS. (SEE FOLDED SECTION BELOW)	5 mm
O.D. OF ROOF PLATING = 12500 + (2 x 25mm) LAPS OVER SHELL	12550 mm
SLOPE LENGTH OF CONE ROOF (Incl. 25mm lap over Shell)	6399 mm
LENGTH OF FOLDED SECTION	5701 mm
FLANGE WIDTH OF FOLDED PLATE	75 mm
WEB DEPTH OF FOLDED PLATE	150 mm
OVERALL O.D. OF CENTRE CROWN PLATE	1344 mm
O.D. OF CROWN RING WEB PLATE	850 mm
GAP BETWEEN LOWER RAFTER FLANGES	59.5 mm O.K.
HEIGHT OF CROWN RING WEB PLATE	125 mm
THKS. OF CROWN RING PLATING	10 mm
DESIGN THKS. OF CROWN PLATING	10 mm
SUPERIMPOSED LOAD	1.2 kN/m ²
INSULATION (IF ANY) (0.25 kN/m ²)	0 kN/m ²
 TOTAL LOAD = ROOF PLATING	48.51 kN
+ ROOF FOLDS	15.80 kN (Corroded)
+ CROWN RING	2.33 kN
+ INSULATION	0 kN
+ S'IMPOSED LOAD	147.26 kN
TOTAL LOAD	213.91 kN
 LOAD PER PETAL "Q" 213.91kN / 32 PETALS	6.68 kN
REACTION AT CROWN "Rb" = 1/3x"Q"	2.23 kN
COMP.N. IN FOLD "P"=Rb / sin theta	11.37 kN

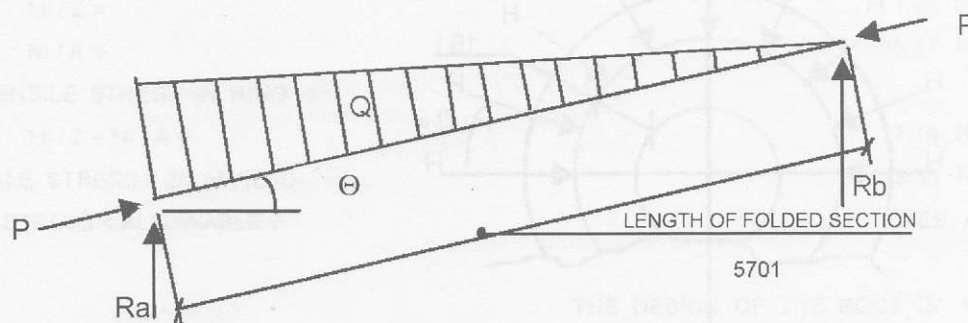
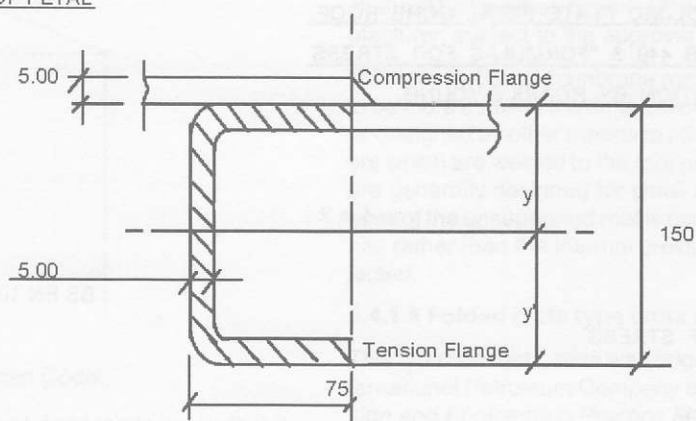


Figure 5.5 Design example for folded plate petal cone roof - page 1

SECTION OF FOLDED ROOF PETAL



EFFECTIVE FLANGE WIDTHS

C.S.A. OF FOLD	"A"	1450 mm ²
SECOND M.O.A. OF FOLD	"I"	5087083.3 mm ⁴
SECTION MODULUS	"Z" = I / y	67827.8 mm ³
RAD of GYRATION	"R _{xx} "	59.2 mm
SLENDERNESS RATIO	"L/R _{xx} " = 6399 / 59.2 =	108.0
RATIO	"D/T"	30
MAX BENDING MNT. = 0.128*Q*L		5.5 kNm
MAX. BENDING STRESS $F_{bc} = B.M. / Z$		80.7 N/mm ²
ALL'BL BEND'G STRESS "P _{bc} " N/mm ²		117 BS 449 Table 3a
MAX. COMPRESSIVE STRESS $F_c = P/A$		7.8 N/mm ²
ALL'BL COMPR. STRESS "P _c " N/mm ²		73 BS 449 Table 17a
$F_{bc}/P_{bc} + F_c/P_c \leq 1.0$		0.8 ACCEPT
DEFLECTION = $(0.01304 * Q * L^3) / E * I$		21.7 ACCEPT
ALLOWABLE DEFLN. From Table 5 BS 5950: Part 1 = L / 200		32.0 mm

CROWN RING DESIGN FROM:

ROARK 5th EDITION TABLE 17-7

$2 \times \alpha$ = ANGLE B'TWEEN RAFTERS	11.25 DEGREES
α = 1/2 ANGLE B'TWEEN RAFTERS	5.625 DEGREES
$1/\alpha = 360/2\pi \times \alpha$	10.186 RADIANS
$1/\sin \alpha$	10.202
$1/\tan \alpha$	10.153
HORIZ. LOAD ON RING "H" = $P \cos \theta$	11.148 kN

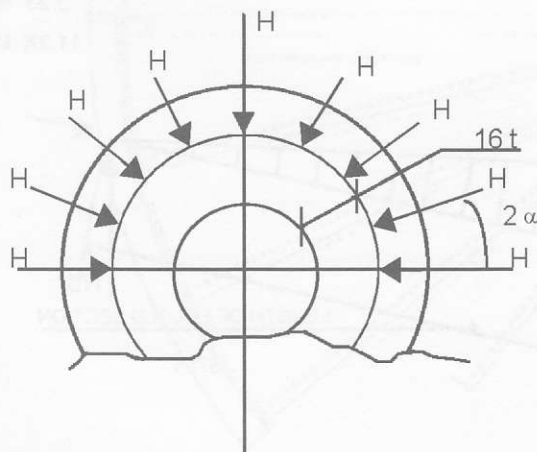
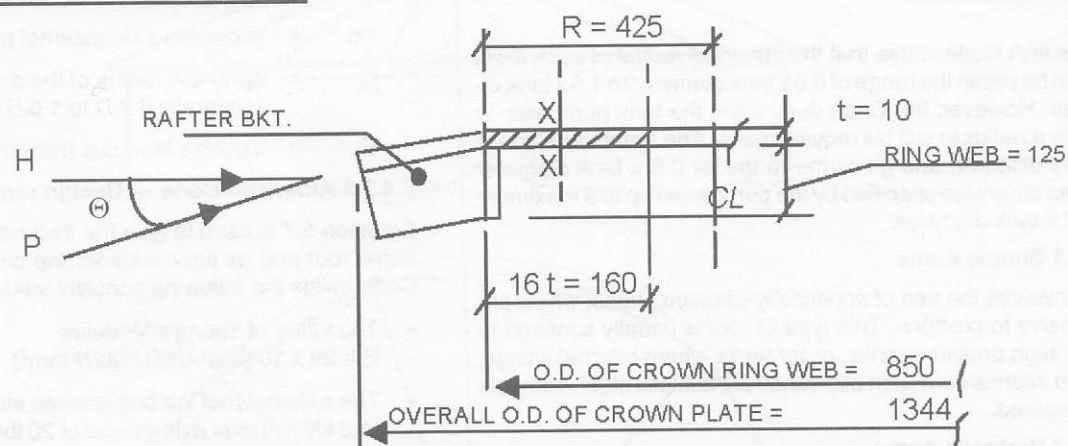


Figure 5.5 Design example for folded plate petal cone roof - page 2

TYPICAL DETAIL OF CROWN RING.



PROPERTIES OF RING :

RADIUS OF RING "R" 425 mm

WIDTH OF RING = 16* THICKNESS 160 mm

C.S.A. OF THIS ANNULAR RING "A" 1600 mm²

SECOND M.O.A. ON AXIS 'XX' PERP. TO 'H' 3413333.33 mm⁴

SECTION MODULUS $Z = I / y$ 42666.67 mm³

MOMENT BETWEEN FORCES "H" is :

$M_o = H \cdot R / 2 (1 / \sin \alpha - 1 / \alpha)$ 54782.23 Nmm

COMPRESSION IN RING is :

$N_o = H / 2 (1 / \sin \alpha)$ 56866.15 N

$M_o / Z =$ 1.28 N/mm²

$N_o / A =$ 35.54 N/mm²

TOTAL COMP. STRESS IN RING is :

$M_o / Z + N_o / A =$ 36.83 N/mm²

ALLOWABLE STRESS = 2/3 of YIELD =

183.33 N/mm²

COMP. STRESS < ALLOWABLE ?

YES ACCEPT

MOMENT AT FORCES "H" is :

$M_i = H \cdot R / 2 (1 / \alpha - 1 / \tan \alpha)$ 77572.00 Nmm

TENSION IN RING is :

$N_i = H / 2 (1 / \tan \alpha)$ 56592.33 N

$M_i / Z =$ 1.82 N/mm²

$N_i / A =$ 35.37 N/mm²

TOTAL TENSILE STRESS IN RING is :

$M_i / Z + N_i / A =$ 37.19 N/mm²

ALLOWABLE STRESS = 2/3 of YIELD

183.33 N/mm²

TENSILE STRESS < ALLOWABLE ?

YES ACCEPT

THE DESIGN OF THE ROOF IS ACCEPTED

Figure 5.5 Design example for folded plate petal cone roof - page 3

A design example for this type of roof is given in Figure 5.5.

5.4.2 Dome roofs

The British Code states that the spherical radius of such roofs should be within the range of 0.8 x tank diameter to 1.5 x tank diameter. However, the Code does allow the tank purchaser to specify a radius to suit his requirements. The American Code is slightly different, and gives the range as 0.8 x tank diameter (unless otherwise specified by the purchaser) up to a maximum of 1.2 x tank diameter.

5.4.2.1 Simple dome

This involves the use of spherically-pressed plates, which are expensive to produce. This type of roof is usually confined to small, high pressure tanks, or for tanks where internal linings, and an internal corrosion allowance or stainless steel materials are required.

5.4.2.2 Umbrella dome

This is a cheaper version of the simple dome and again is generally used only on small diameter tanks. The roof petal plates in this case are rolled in the radial direction only and when they are assembled the appearance of the roof is like that of an umbrella – hence the name. (See Figure 5.6.)

5.4.2.3 British Code — Design requirements

The membrane stress in a spherical shell is given by the standard expression:

$$f = \frac{p \cdot r_d}{2 \cdot t_{rd}} \quad \text{equ 5.10}$$

where:

- P = internal pressure (mbar)
- r_d = spherical radius (m)
- t_{rd} = thickness of the domed roof plating (mm)

Rearranging for t_{rd} then:

$$t_{rd} = \frac{p \cdot r_d}{2 \cdot f} \quad \text{equ 5.11}$$

As was the case for the self supported cone roof, the Code uses the same joint efficiencies η as follows:

- η = 1.0 for butt-welded joints
- = 0.35 for lapped joints with fillet welds on one side.
- = 0.5 for lapped joints with fillet welds on both sides

Rationalising the units, the equation becomes:

$$t_{rd} = \frac{p \times r_d \times 10^3}{10^4 \times 2 \times f \times \eta}$$

$$t_{rd} = \frac{p \cdot r_d}{20 \cdot f \cdot \eta} \quad \text{equ 5.12}$$

This is the form of equation which is found in the British Code for the thickness of a spherical roof under pressure.

The roof must also be checked to withstand the external pressure due to the roof loading and vacuum and by reference to the previous equations 5.4, 5.5, 5.6 and 5.7, which are all based on the theory for a domed roof, it can be seen from equation 5.7 that:

$$t_{rd} = 40 \cdot r_d \sqrt{\frac{10 \cdot P_e}{E}}$$

where:

- t_{rd} = thickness of the domed roof plating (mm)
(not less than 5mm excluding corrosion allowance)
- P_e = allowable safe external pressure (kN/m²)
- r_d = spherical radius of the dome (m)
(generally 0.8.D to 1.5.D)
- E = Young's Modulus (N/mm²)

5.4.2.4 American Code — Design requirements

Equation 5.7 is used to give the thickness for an unsupported dome roof and as previously for the cone roof, the American Code builds the following constant values into the equation:

- The value of Young's Modulus
 $E = 29 \times 10 \text{ lb/in}^2$ (200,000 N/mm²)
- The external roof loading is taken as, a live load of 25 lb/ft² (1.2 kN/m²) plus a dead load of 20 lb/ft² (approximately 1.0 kN/m²), which is the self-weight of 1/2" (12.5 mm) roof plating – the maximum thickness allowed.

Equation 5.7 then becomes:

$$t_{rd} = 40 \cdot r_d \sqrt{\frac{10 \times 2.2}{200,000}}$$

$$t_{rd} = \frac{r_d}{2.4}$$

This equation is given in the American Code.

As for the unsupported cone roof, the following applies to unsupported dome roofs:

When the sum of the live and dead loads exceeds 2.2 kN/m², the minimum thickness shall be increased by the following ratio:

$$\sqrt{\frac{\text{live load} + \text{dead load}}{2.2 \text{ kN/m}^2}}$$

The American Code also states that:

"Self-supporting roofs, whose roof plates are stiffened by sections welded to the plates, need not conform to the minimum thickness requirements, but the thickness of the roof plates shall not be less than 5 mm (3/16") when so designed by the manufacturer, subject to the approval of the purchaser."

Observations on the unsupported cone and dome roof thickness equations

- 1) By comparing equation 5.3 for the cone roof

$$t_{rc} = \frac{p \cdot r_c}{10 \cdot f \cdot \eta}$$

with equation 5.12 for the domed roof

$$t_{rd} = \frac{p \cdot r_d}{20 \cdot f \cdot \eta}$$

it can be seen that for a given roof construction, roof radius and internal pressure then the thickness of a cone roof is twice that for a dome roof.

- 2) By comparing the expression for the stress in a cylinder from equation 4.6

$$f = \frac{p \times D}{2 \times t}$$

with the expression for the stress in a spherical roof from equation 5.10

$$f = \frac{p \cdot r_d}{2 \cdot t_{rd}}$$

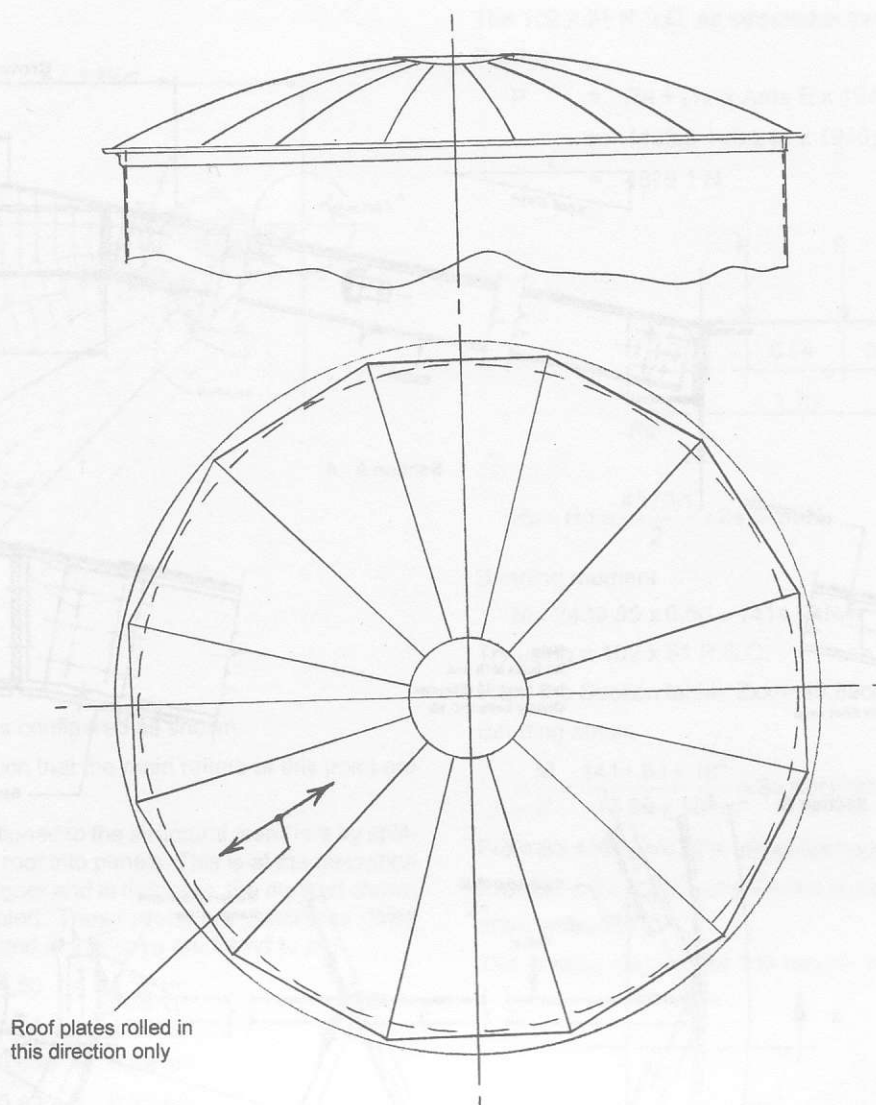


Figure 5.6 Umbrella type dome roof

then it can be seen, that for a constant thickness shell and spherical roof, and hence equating 't' and 't_d'

$$\frac{p \times D}{2 \times f} = \frac{p \times r_d}{2 \times f}$$

and for this condition then, $D = r_d$

Then for a dome roof thickness to be the same as that of the top course of shell plating, the radius of the dome is equal to the diameter of the tank.

The American Code adopted this approach for setting the limits for the maximum and minimum radii for domed roofs but allows a $\pm 20\%$ variation thus giving the range for roof radii to be:

$$r_d = 0.8.D \text{ to } 1.2.D$$

which has been given earlier.

5.5 Roofs with supporting structures, supported from the tank shell

5.5.1 Cone roofs

The usual slope for this type of roof is 1 in 5 for the British Code and 1 in 6 for the American Code. Unless the internal pressure dictates otherwise it is usual for the roof plating to be 5mm ($\frac{3}{16}$ ") thick and is single lap-welded on the top side. The Codes do not permit the roof plating to be attached to the supporting framework.

5.5.1.1 Radial rafter type

This type of roof is supported by a radial rafter framework composed of structural sections. It is illustrated in plan form in Figure 5.7.

These structures are usually confined to tanks with diameters less than 15 metres.

5.5.1.2 Design example

One method of designing such a structure using the British Codes is as follows:

Assume a tank diameter of 12.5 m

No. of main rafters $R_1 = 8$

No. of secondary rafters $R_2 = 16$

Superimposed load = 1200 N/m²

Dead load (structure and roof plating)
(Derived from experience) = 740 N/m²

Total loading = 1940 N/m²

Roof slope is 1 in 5.

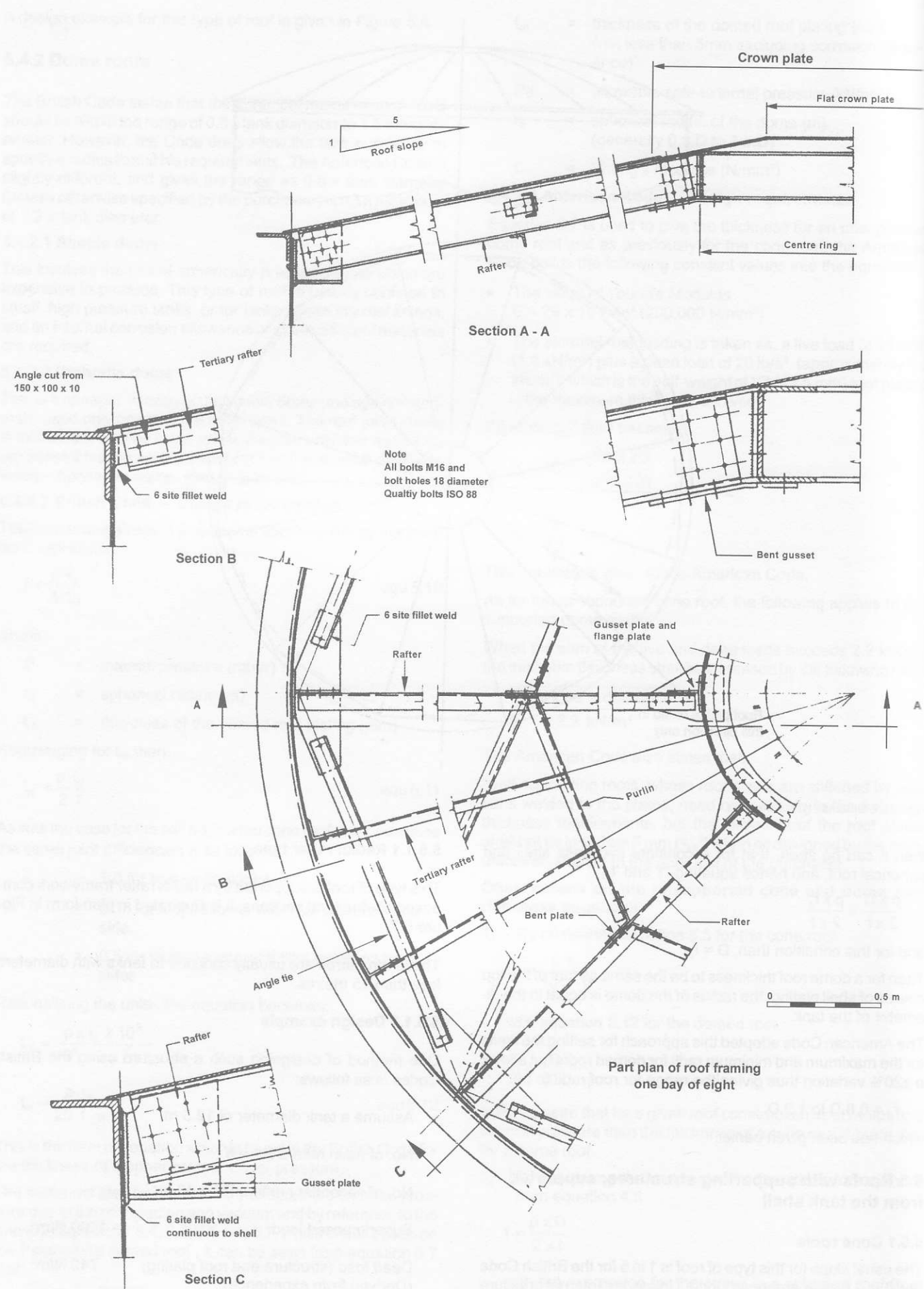
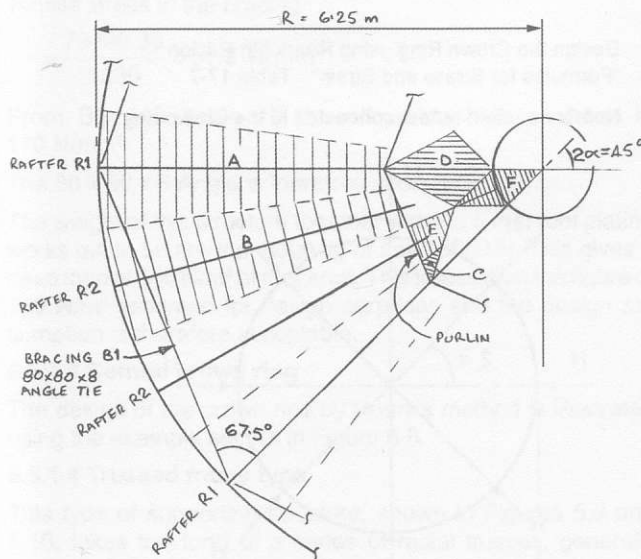


Figure 5.7 Plan arrangement of radial rafter type cone roof structure



The loading diagram is configured as shown.

The Purlin length is such that the main rafters at this point are 1.7m apart.

The roof load is apportioned to the structural members by splitting the surface of the roof into panels. This is at the discretion of each individual designer and in this case, the method shown above has been adopted. These areas are calculated using geometrical methods and in this case are found to be:

Area	A	1	x	4.50	=	4.50 m ²
	B	2	x	4.54	=	9.08 m ²
	C	2	x	0.045	=	0.09 m ²
	D	1	x	0.82	=	0.82 m ²
	E	2	x	0.245	=	0.49 m ²
	F	1	x	0.36	=	0.36 m ²
						15.34 m ²

Check the sector area = $\frac{1}{8} \times \pi/4 \times 12.5^2 = 15.34 \text{ m}^2$ O.K.

Secondary rafter R2:

Plan length of rafter is found to be 4.18 m

Slope length of rafter is

$$\sqrt{26/5} \times 4.18 = 4.263 \text{ m}$$

Load on rafter

$$= (\frac{1}{2} \times \text{Area B} \times 1940) = 4.54 \times 1940 = 8807.6 \text{ N}$$

Reactions at ends of rafter Ra and Rb

$$= 8807.6/2 = 4403.8 \text{ N}$$

Bending moment in rafter

$$M = \frac{W \cdot L}{8} = \frac{8807.6 \times 4.263}{8} = 4693.35 \text{ Nm}$$

Try using a 102 x 51 R.S.C.

From the Section tables $Z_{xx} = 40.89 \text{ cm}^3$

Bending stress

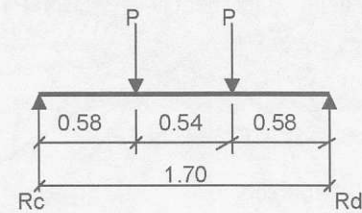
$$= \frac{M}{Z} = \frac{4693.35 \times 10^3}{40.89 \times 10^3} = 114.78 \text{ N/mm}^2$$

From BS 449 Table 2 the allowable bending stress is 180 N/mm²

The 102 x 51 R.S.C. as selected is therefore acceptable.

Purlin:

$$\begin{aligned} P &= R_a + (\frac{1}{2} \times \text{Area E} \times 1940) \\ &= 4403.8 + (0.245 \times 1940) \\ &= 4879.1 \text{ N} \end{aligned}$$



$$R_c = R_d = \frac{4879.1}{2} = 2439.55 \text{ N}$$

Bending moment

$$M = 2439.55 \times 0.58 = 1414.94 \text{ Nm}$$

Try using a 102 x 51 R.S.C.

From the Section tables $Z_{xx} = 40.89 \text{ cm}^3$

Bending stress

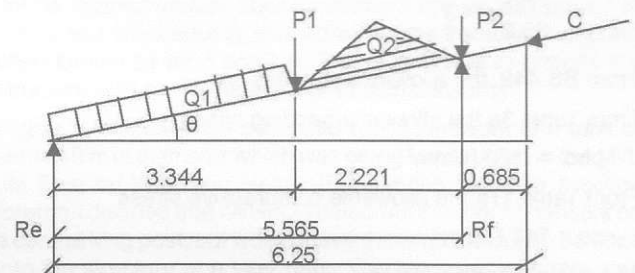
$$= \frac{M}{Z} = \frac{1414.94 \times 10^3}{40.89 \times 10^3} = 34.60 \text{ N/mm}^2$$

From BS 449 Table 2 the allowable bending stress is 180 N/mm

The 102 x 51 R.S.C. as selected is therefore acceptable.

Main rafter R1:

The loading diagram for this rafter is as follows:



$$\tan \theta = 1/5 = 0.2 = 11.31^\circ \text{ and } \sin \theta = 0.1961$$

$$\begin{aligned} P1 &= (2 \times P) + (\text{Area C} \times 1940) \\ &= (2 \times 4879.1) + (0.09 \times 1940) = 9932.80 \text{ N} \\ P2 &= \text{Area F} \times 1940 = 0.36 \times 1940 = 698.40 \text{ N} \\ Q1 &= \text{Area A} \times 1940 = 4.50 \times 1940 = 8730.00 \text{ N} \\ Q2 &= \text{Area D} \times 1940 = 0.82 \times 1940 = 1590.80 \text{ N} \end{aligned}$$

Taking moments about Re

$$\begin{aligned} (Q1 \times 1.672) + (P1 \times 3.344) + (Q2 \times 4.4545) + (P2 \times 5.565) &= R_f \times 5.565 \\ (8730 \times 1.672) + (9932.8 \times 3.344) + (1590.8 \times 4.4545) + (698.4 \times 5.565) &= R_f \times 5.565 \\ 1459.66 + 33215.28 + 7086.22 + 3886.60 &= R_f \times 5.565 \\ 58784.66 &= R_f \times 5.565 \end{aligned}$$

$$R_f = \frac{58784.66}{5.565} = 10563.3 \text{ N}$$

$$R_e = 20952.0 - 10563.3 = 10388.7 \text{ N}$$

Note: The compressive stress transmitted to the shell by this load shall be minimised by mounting the rafter fixing bracket on to a doubler plate welded to the shell.

The maximum bending moment is at position P1.

Taking moments about P1

$$(R_e \times 3.344) - (Q_1 \times 1.672)$$

$$(10388.7 \times 3.344) - (8730 \times 1.672)$$

$$= 34739.81 - 14596.56$$

$$= 20143.25 \text{ Nm}$$

The compressive force C in the rafter is found as follows:

$$\begin{aligned} C &= R_f / \sin \theta \\ &= 10563.3 / 0.1961 \\ &= 53862.47 \text{ N} \end{aligned}$$

Try a 203 x 76 R.S.C.

From the Section tables:

$$\text{C.S.A.} = 3034 \text{ mm}^2$$

$$Z = 192 \times 10^3 \text{ mm}^3$$

$$r_{xx} = 80.2 \text{ mm}$$

$$D/T = 18.2$$

Maximum slope length of rafter between fixing points

$$L = 3.344 \text{ m} \times \sqrt{26/5} = 3.41 \text{ m}$$

$$\frac{L}{r_{xx}} = \frac{3410}{80.2} = 43$$

From BS 449, the allowable stresses are:

From Table 3a the allowable bending stress

$$p_{bc} = 180 \text{ N/mm}^2$$

From Table 17a the allowable compressive stress

$$p_c = 148 \text{ N/mm}^2$$

The actual bending stress

$$f_{bc} = \frac{M}{Z} = \frac{20143.25 \times 10^3}{192 \times 10^3} = 104.91 \text{ N/mm}^2$$

The actual compressive stress

$$f_c = \frac{C}{\text{c.s.a.}} = \frac{53862.47}{3034} = 17.75 \text{ N/mm}^2$$

BS 449 states that $\frac{f_{bc}}{p_{bc}} + \frac{f_c}{p_c}$ must not be more than 1.0

$$\frac{104.91}{180} + \frac{17.75}{148} = 0.58 + 0.12 = 0.7 < 1.0 \text{ O.K.}$$

The 203 x 76 R.S.C. as selected is therefore acceptable.

Bracing B1:

The load B1 in the bracing is found using Lami's theorem:

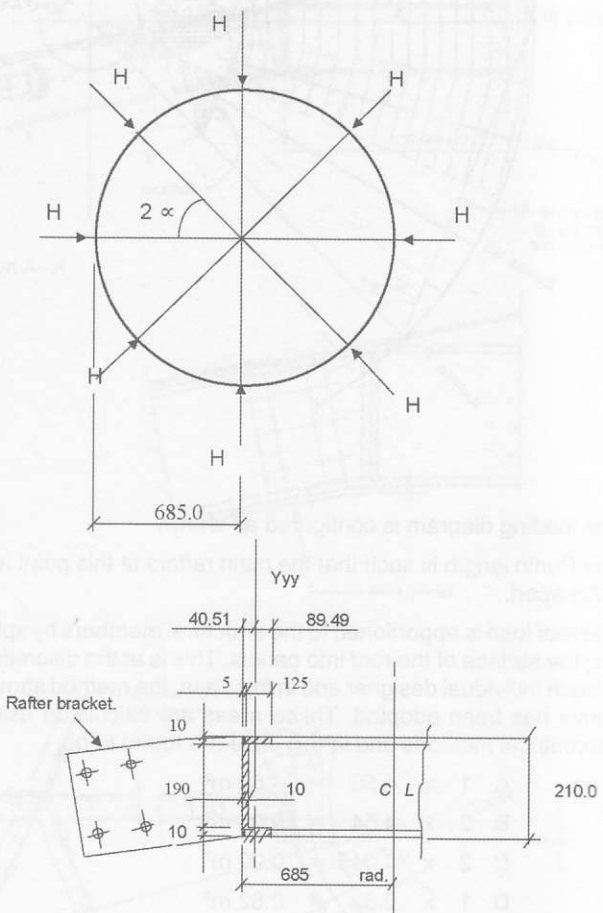
$$B1 = C \times \frac{\sin 67.5^\circ}{\sin 45^\circ}$$

$$B1 = 53862.47 \times \frac{0.9238}{0.7071} = 70369.33 \text{ N}$$

Try using a 80 x 80 x 8 Angle.

Design the Crown Ring using Roark 5th Edition
"Formulas for Stress and Strain" - Table 17-7

Number of Main rafters connected to the Crown ring = 8



$2 \times \alpha$ = Angle Between Rafters	45.00 degrees
α = 1/2 Angle Between Rafters	22.50 degrees
$1/\alpha$ = $360 / (2\pi \times \alpha)$	2.55 radians
$1/\sin \alpha$	2.61
$1/\tan \alpha$	2.41
Horiz. Load on Ring "H" = $C \times \cos \theta$	52.82 kN

C.S.A. of corroded Crown Ring	A = 4400.00 mm ²
Position of Yyy axis from outer face of ring	= 40.51 mm
Moment of Inertia on Axis thro' centroid	I _{yy} = 7488016.10 mm ⁴
Max. 'Y' value	= 89.49 mm
Section Modulus	Z _{yy} = 83675.61 mm ³
Radius of Gyration	r _{yy} = 41.25 mm
Radius of corroded Crown Ring	R = 685.00 mm

Moment between Forces "H" is "Mo"

$M_o = (H \times R / 2) \times (1/\sin \alpha - 1/\alpha)$	1205697.99 N
Compression in Ring is "No" = $H/2 (1/\sin \alpha)$	69012.66 N
M_o / Z	14.41 N/mm ²
No / A	15.68 N/mm ²
Total Comp. Stress in Ring = $M_o/Z + No/A$	30.09 N/mm ²
Allowable Design Stress =	183.330 N/mm ²
Is Total Comp. Stress < Allowable Design Stress?	Yes accept

Moment at Forces "H" is "Mi"

$M_i = (H \times R / 2) \times (1/\alpha - 1/\tan \alpha)$	2392795.81 N
Tension in Ring is "Ni" = $H/2 (1/\tan \alpha)$	63759.38 N
M_i / Z	28.60 N/mm ²
Ni / A	14.49 N/mm ²
Total Tension in Ring = $M_i/Z + Ni/A$	43.09 N/mm ²
Allowable Design Stress =	183.33 N/mm ²
Is Total Tensile Stress < Allowable Design Stress?	Yes accept.

The design of the Crown ring is accepted

This completes the design of the roof structure

Figure 5.8 Crown ring design example using Roark's method

$$\text{C.S.A.} = 1230 \text{ mm}^2$$

Tensile stress in the bracing

$$= \frac{70369.33}{1230} = 57.2 \text{ N/mm}^2$$

From BS 449 Table 19 the allowable tensile stress is 170 N/mm²

The 80 x 80 x 8 Angle is therefore acceptable.

The weight of this structure together with the 5 mm roof plating works out to be around 8000 kg or say 78500 N. This gives a dead load of 640 N/m² of roof area. This is less than the figure of 740 N/m² assumed for design purposes and the design assumption is therefore acceptable.

5.5.1.3 Central crown ring

The design of the crown ring by Roark's method is illustrated using the example set out in Figure 5.8.

5.5.1.4 Trussed frame type

This type of supporting structure, shown in Figures 5.9 and 5.10, takes the form of a series of radial trusses, generally made up from steel angle sections. Between these trusses are circumferentially arranged members providing stability and support for the roof plating.

The British Code requires that vertical ring bracing shall be provided under the outer circumferential purlins. This shall be, one

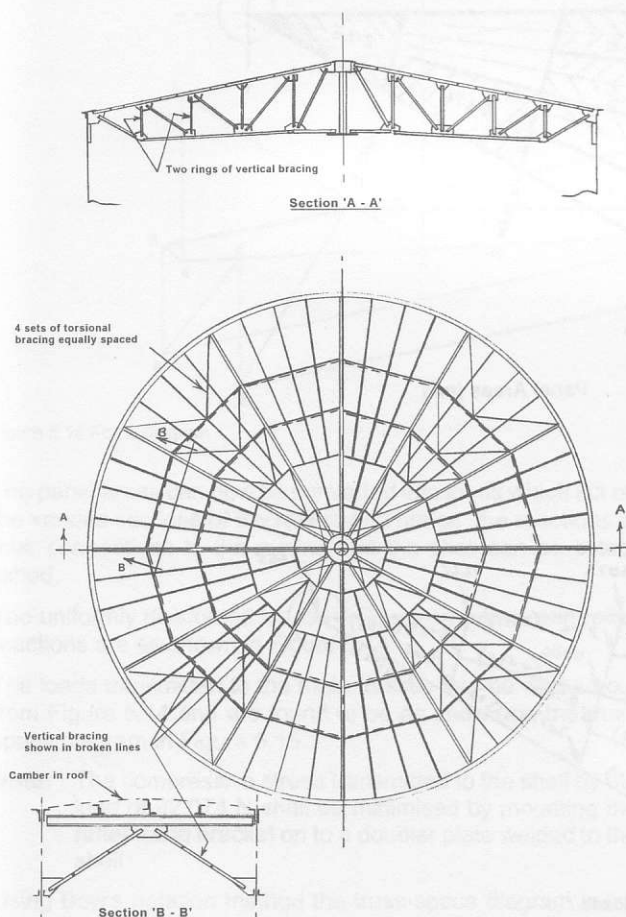


Figure 5.9 Trussed frame type roof

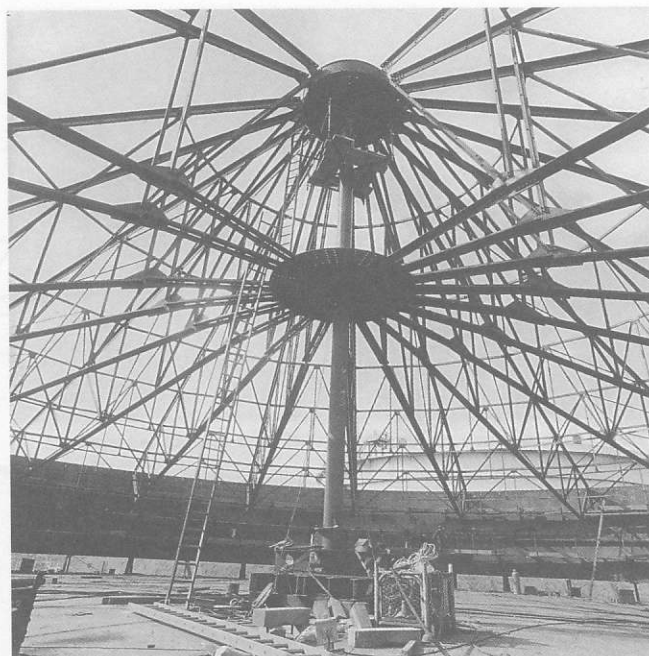


Figure 5.10 A 39 metre trussed frame type structure under construction
Courtesy of McTay

ring for roofs over 15 m and up to 25 m diameter and two rings for roofs over 25 m diameter. Also it requires cross-bracing to be provided in the plane of the roof surface, in at least two bays, between two pairs of adjacent trusses for roofs over 15 m diameter. These sets of bracings have to be evenly spaced around the tank circumference and are to give torsional stability to the structure.

The importance of the diagonal bracing members which occur in most types of roof supporting structures where the framework is within the tank and not attached to the roof plating cannot be overestimated. These members are usually placed in two or four bays equally spaced within the framework and are often known as wind bracing. Their function is to provide the structure with some measure of torsional stability.

Figure 5.11 shows the collapsed roof framework of a tank of some 40 m in diameter which was being constructed in the Middle East by Whessoe Heavy Engineering Ltd. The erection foreman decided that he would construct the roof framework on a central king post, but would leave the wind bracing to be fitted into the structure at a later date. The king post was removed and the roof collapsed. The spiral nature of the failure is clear to see.

The roof did not fail immediately, which was fortunate as this would have resulted in serious injury to the operatives within the tank at the time that the central support was removed, but was

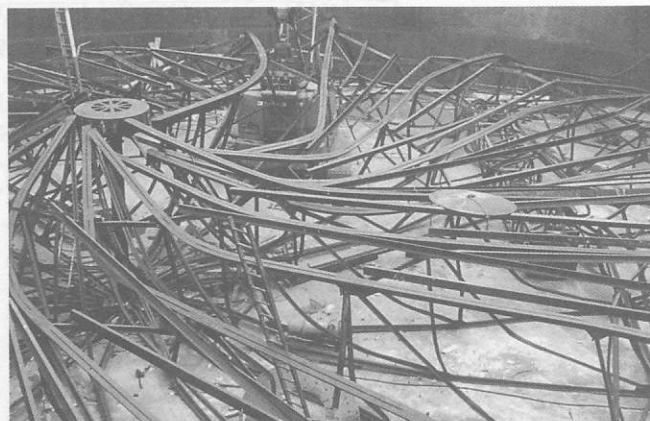


Figure 5.11 The collapsed roof framework
Courtesy of Whessoe



Figure 5.12 A view from outside the tank shell when the roof had failed
Courtesy of Whessoe

kind enough to wait until they had gone for lunch. Figure 5.12 shows a view from outside the tank shell which was forced into a curious, but quite regular shape by the action of the main trusses pulling inwards as the roof failed.

The American Code does not specifically mention these bracing requirements, but nevertheless, it is generally thought to be good practice to include them in roofs of this type.

This type of roof is commonly used within the range of 15 m to 60 m diameter.

The lower part of the trusses generally protrude down below the level of the top of the tank shell and hence can become submerged in the stored product. In certain circumstances, or for some corrosive stored products, this may be an undesirable feature.

5.5.1.5 Design example

These days there are computer-aided design packages available for structural designers to use, but for this example, the tried and tested "hand-cranked" method is demonstrated.

The exercise will demonstrate how the sizes of the members of a 30 m diameter roof structure are calculated.

The arrangement of rafters and purlins in one of 12 bays of the structure is shown in Figure 5.13. The three intermediate rafters per bay are supported at their outer end by the shell and by three purlins in the plane of the roof. The rafters lie on top of the purlins which in turn transmit the rafter loads to the main trusses. The load on the sections of rafters is determined by dividing the roof sector into panels as shown in Figure 5.13, the size of these panels is calculated using simple geometric methods.

The numbers in Figures 5.13, represent plan areas in m^2
As before:

Superimposed load	= 1200 N/m ²
Dead load (structure and roof plating) (Derived from experience)	= 740 N/m ²
Total loading	= 1940 N/m ²

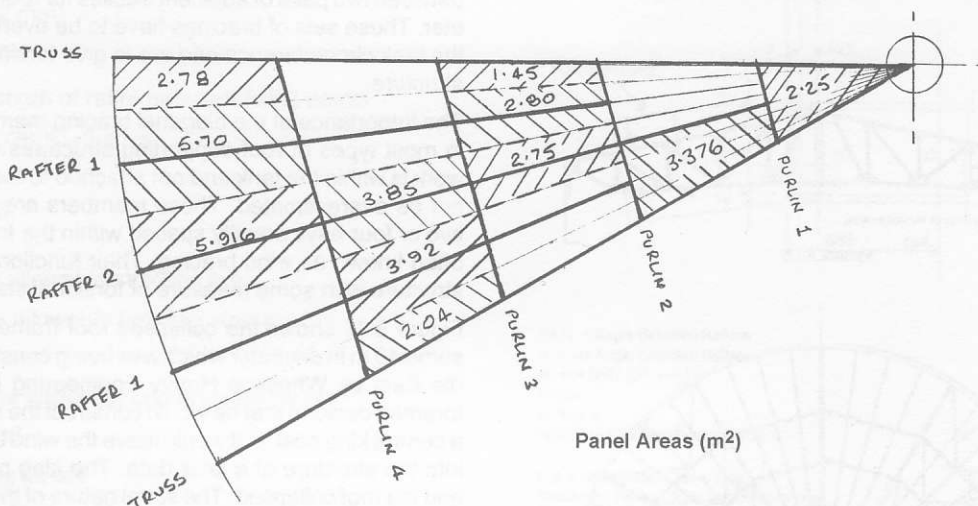


Figure 5.13 Arrangements of rafters and purlins

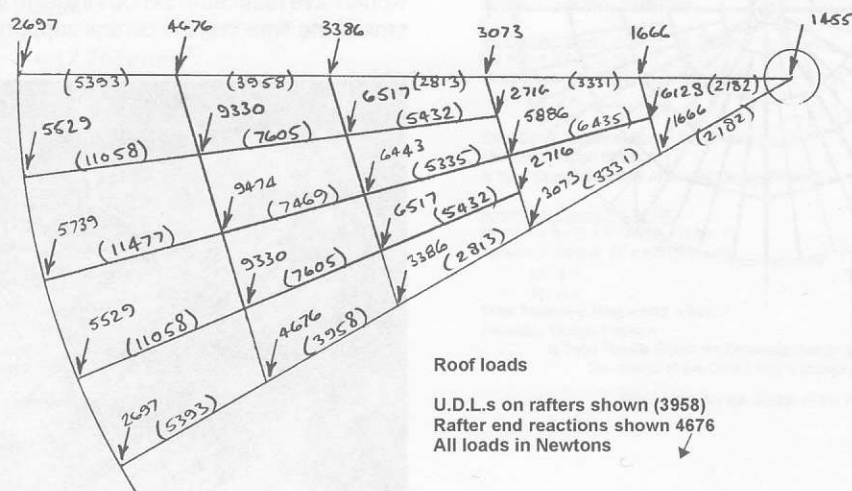


Figure 5.14 Uniformly distributed rafter loads and rafter reactions

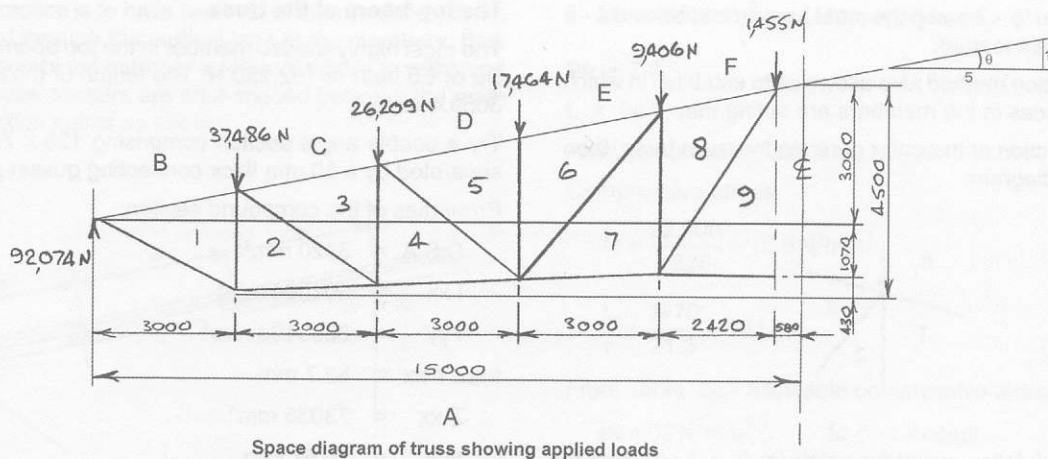


Figure 5.15 Space diagram of truss showing applied loads

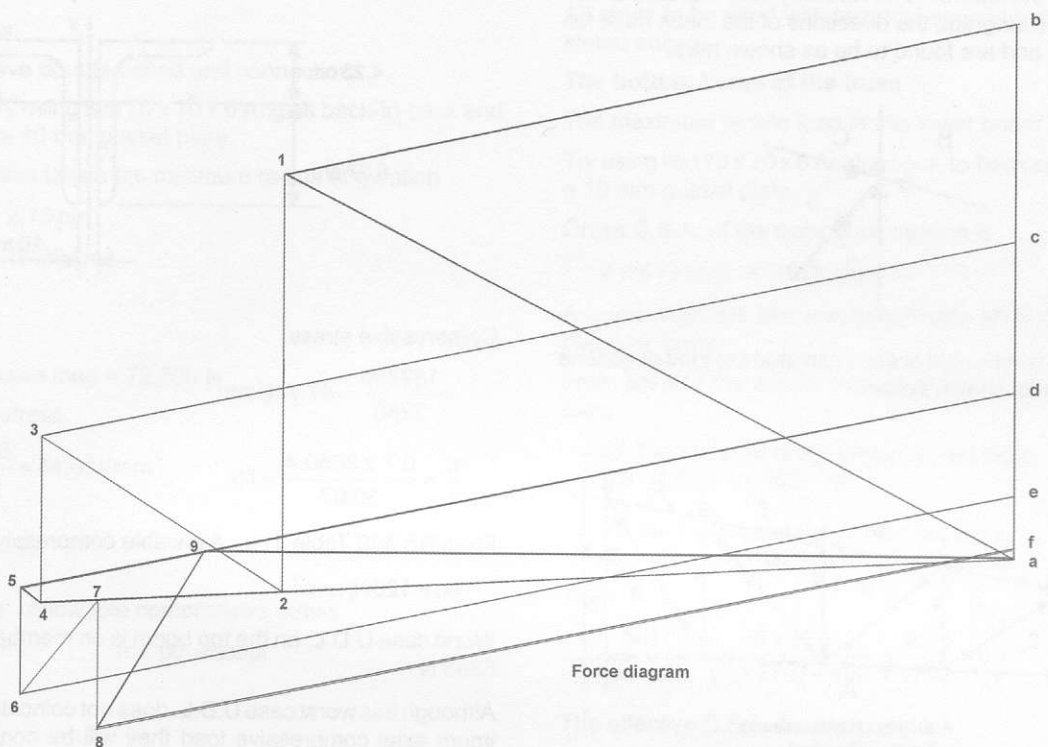


Figure 5.16 Force diagram

The panel areas can now be converted into loads which act on the various sections of the rafters and hence the reactions at their connections to the purlins and the shell can be established.

The uniformly distributed loads (U.D.L.s) on rafters and rafter reactions are as shown in Figure 5.14.

The loads transmitted to the main trusses can be worked out from Figure 5.14 and are found to be as shown on the truss space diagram in Figure 5.15.

Note: The compressive stress transmitted to the shell by the load of 92,074 N shall be minimised by mounting the rafter fixing bracket on to a doubler plate welded to the shell.

Using Bow's notation method the truss space diagram is lettered A to F and numbered 1 to 9 and a force diagram is produced to a suitable scale.

The force diagram in Figure 5.16 is produced as follows:

The loads 'b' to 'c', 'c' to 'd', 'd' to 'e', 'e' to 'f', and 'f' to 'a' are drawn to scale down the right-hand side of the diagram.

Draw a line parallel to the slope of the main truss through 'b' representing member 'b' - 1.

Through point a draw a line parallel with the lower outer member 'a' - 1. Where these two lines meet is point 1 and the scale length of these lines represents the axial load carried by members 'b' - 1 and 'a' - 1.

Through point 1 draw a vertical line representing member 1 - 2 and through point a draw a line parallel to member 'a' - 2. Where these two lines meet gives us point 2 and hence the axial loads in members 1 - 2 and 'a' - 2.

This procedure is continued until the diagram is completed as shown.

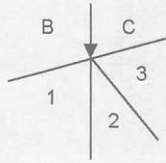
By scaling off the diagram the axial loads in all the members can be found.

The same results could be found mathematically using geometrical methods but the force diagram gives a good pictorial appreciation of the magnitude of the loadings on the various truss members.

- i.e. 'd' - 5 and 'd' - 6 being the most heavily loaded and 4 - 5 being the least loaded.

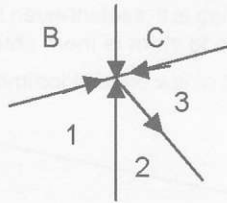
The Bow's notation method also allows us to establish in which direction the forces in the members are acting.

Take the connection of the outer purlin to the main truss, then from the force diagram.

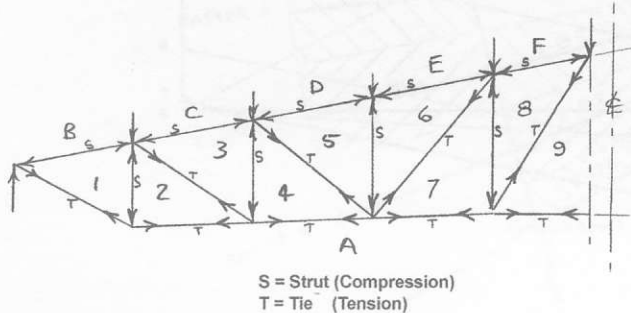


- Starting at 'b', follow round the points 'c', 3, 2, 1 and back to 'b'.

The direction of the load 'b' - 'c' is vertically downwards, then following round the diagram, the directions of the loads must follow this pattern and are found to be as shown here.



This procedure is repeated at each joint and the load directions are established as shown below.



The axial load in each member is given in Figure 5.17, showing also if the member is a strut or a tie.

Having found all the loadings, then suitable section sizes for the members can be found using the requirements of BS 449.

For expediency, the numbers and sizes of bolts required for the many and various connections in the truss will not be calculated here because, although this is a fairly simple task it is quite labourious. All connections will assume M20 bolts in 22 mm diameter holes.

B1	135,500 N	Strut	A1	148,250 N	Tie	1-2	72,500 N	Strut
C3	178,750 N	Strut	A2	133,500 N	Tie	2-3	50,000 N	Tie
D5	182,250 N	Strut	A4	175,500 N	Tie	3-4	29,000 N	Strut
E6	182,250 N	Strut	A7	165,500 N	Tie	4-5	4,250 N	Tie
F8	168,500 N	Strut	A9	147,000 N	Tie	5-6	18,250 N	Strut
						6-7	21,000 N	Tie
						7-8	22,000 N	Strut
						8-9	36,250 N	Tie

Figure 5.17 The axial load in each member

The top boom of the truss

The most highly-loaded member in the top boom of the truss is D5 or E5 both at 182,250 N. The length of these members is 3059.4 mm

Try a double angle section comprising 125 x 75 x 10 angles separated by a 10 mm thick connecting gusset plate.

Properties of the compound section:

$$\text{C.S.A.} = 3820 \text{ mm}^2$$

$$I_{xx} = 604000 \text{ mm}^4$$

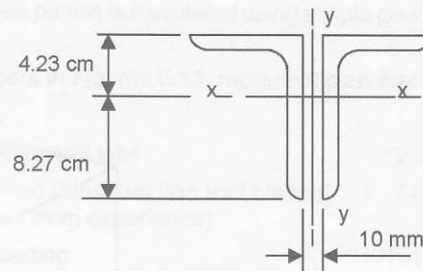
$$I_{yy} = 3593103 \text{ mm}^4$$

$$\text{Max } Y_{xx} = 82.7 \text{ mm}$$

$$Z_{xx} = 73035 \text{ mm}^3$$

$$r_{yy} = 30.67 \text{ mm}$$

$$D/t = 12.5$$



Compressive stress

$$= \frac{182250}{3280} = 47.7 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{0.7 \times 3059.4}{30.67} = 70$$

From BS 449 Table 17a - Allowable compressive stress

$$p_c = 123 \text{ N/mm}^2$$

Worst case U.D.L. on the top boom is on member B1 and is 2 x 5393 N

Although this worst case U.D.L. does not coincide with the maximum axial compressive load they will be combined here to prove that the chosen section for the top boom is adequate.

Bending moment

$$M = \frac{W \cdot L}{8} = \frac{2 \times 5393 \times 3059.4}{8} = 4,124,836 \text{ Nmm}$$

Bending stress

$$f_{bc} = \frac{M}{Z_{xx}} = \frac{4,124,836}{73035} = 56.5 \text{ N/mm}^2$$

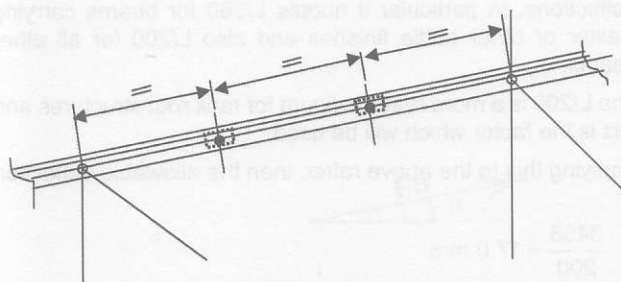
From BS 449 Table 3a - Allowable bending stress = 172 N/mm²

$\frac{f_c}{p_c} + \frac{f_{bc}}{p_{bc}}$ must be less than 1.0 for the selected member section to be acceptable.

$$\frac{47.7}{123} + \frac{56.5}{172} = 0.39 + 0.33 = 0.72 < 1.0 \text{ Accept}$$

If by combining the two worst case loads acting on the top boom member, as shown above, the member was proved to be inadequate, then each of the members making up the top boom would have to be separately analysed using their own individual, axial and U.D.L.s. This can result in the selected section for the top boom being found to be adequate.

The normal practice is to have two sets of intermediate 10 mm packers bolted through the vertical legs of the members, thus affording the combined member additional rigidity to withstand axial load. These packers are equi-spaced between the main bolted connection points as shown:



Vertical struts

All struts to have double-bolted end connections.

For all struts try using two 70 x 70 x 6 Angles back-to-back and separated by a 10 mm gusset plate.

From the Section tables the minimum radius of gyration

$$r = 2.13 \text{ cm}$$

$$\text{C.S.A.} = 16.26 \text{ cm}^2$$

Strut 1-2

$$L = 1200 \text{ mm}$$

$$\text{Axial compressive load} = 72,500 \text{ N}$$

Compressive stress

$$f_c = \frac{72,500}{1626} = 44.6 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{2100}{21.3} = 99$$

From Table 3a - Allowable compressive stress

$$p_c = 84 \text{ N/mm}^2 \quad f_c < p_c \text{ Accept}$$

Strut 3-4

$$L = 2557 \text{ mm}$$

$$\text{Axial compressive load} = 29,000 \text{ N}$$

Compressive stress

$$f_c = \frac{29,000}{1626} = 18 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{2577}{21.3} = 120$$

From Table 3a — Allowable compressive stress

$$p_c = 62 \text{ N/mm}^2 \quad f_c < p_c \text{ Accept}$$

Strut 5-6

$$L = 3013 \text{ mm}$$

$$\text{Axial compressive load} = 18,250 \text{ N}$$

Compressive stress

$$f_c = \frac{18,250}{1626} = 11.2 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{3013}{21.3} = 141$$

From Table 3a - Allowable compressive stress

$$p_c = 46 \text{ N/mm}^2 \quad f_c < p_c \text{ Accept}$$

Strut 7-8

$$L = 3470 \text{ mm}$$

$$\text{Axial compressive load} = 22,000 \text{ N}$$

Compressive stress

$$f_c = \frac{22,000}{1626} = 13.5 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{3470}{21.3} = 163$$

From Table 3a - Allowable compressive stress

$$p_c = 35 \text{ N/mm}^2 \quad f_c < p_c \text{ Accept}$$

All the above struts are acceptable using two 70 x 70 x 6 Angles back-to-back and separated by a 10 mm gusset plate.

All struts to be fitted with two equi-spaced bolted packers (as stated above).

The bottom boom of the truss

The maximum tensile load in this lower boom is 175,500 N

Try using two 70 x 70 x 6 Angles back-to-back and separated by a 10 mm gusset plate.

Gross C.S.A. of the compound section is

$$2 \times 813 \text{ mm}^2 = 1626 \text{ mm}^2$$

Assume that the ties are bolted with M20 bolts in 22 mm diameter holes.

From BS 449 the effective areas of the angle legs are as follows:

$$a_2 \text{ the net area of the unconnected leg is } (70 - 6/2) \times 6 = 402 \text{ mm}^2$$

$$a_1 \text{ the net area of the connected leg is } 402 - (22 \times 6) = 270 \text{ mm}^2$$

Then

$$\frac{5a_1}{5a_1 + a_2} = \frac{5 \times 270}{(5 \times 270) + 402} = \frac{1350}{1752} = 0.77$$

The effective C.S.A. for each angle is

$$270 + (0.77 \times 402) = 579.5 \text{ mm}^2$$

and for the compound section is therefore 1159 mm²

The maximum tensile stress in the tie

$$\frac{175,500}{1159} = 151.4 \text{ N/mm}^2$$

From Table 19 - the allowable stress is 170 N/mm²

The compound section is therefore acceptable.

Diagonal ties

The most highly-loaded tie is 2-3 at 50,000 N

Try using two 50 x 50 x 6 Angles back-to-back and separated by a 10 mm gusset plate.

Gross C.S.A. of the compound section is $2 \times 569 \text{ mm}^2 = 1138 \text{ mm}^2$

Assume that the ties are bolted with M20 bolts in 22 mm diameter holes.

From BS 449 the effective areas of the angle legs are as follows:

$$a_2 \text{ the net area of the unconnected leg is } (50 - 6/2) \times 6 = 282 \text{ mm}^2$$

a1 the net area of the connected leg is
 $282 - (22 \times 6) = 150 \text{ mm}^2$

Then

$$\frac{5a_1}{5a_1 + a_2} = \frac{5 \times 150}{(5 \times 150) + 282} = \frac{750}{1032} = 0.73$$

The effective C.S.A. for each angle is

$$150 + (0.73 \times 282) = 355 \text{ mm}^2$$

and for the compound section is therefore

$$2 \times 355 = 710 \text{ mm}^2$$

The maximum tensile stress in the tie

$$= \frac{50,000}{710} = 70 \text{ N/mm}^2$$

From Table 19 the allowable the allowable stress is 170 N/mm^2

The compound section is therefore acceptable.

Crown ring

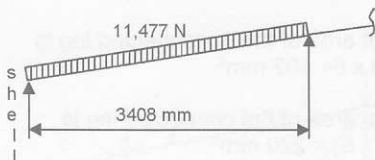
The central crown ring is designed as for the previous example using Roark's method. See Figure 5.18.

Intermediate rafters

The longest intermediate rafter at 3408 mm, is the one at the centre of the bay, running between the shell and Purlin No.4. This rafter is also the most heavily-loaded, carrying a total U.D.L of 11,477 N. The design for all the intermediate rafters will be based on this worst case.

Loading diagram

The maximum bending moment is given by



$$M = \frac{W \cdot L}{8} = \frac{11477 \times 3408}{8}$$

$$= 4.9 \times 10^6 \text{ Nmm}$$

Try using a 127 x 64 R.S.C.

$$Z_{xx} = 75.99 \text{ cm}^3$$

$$I_{xx} = 482.5 \text{ cm}^4$$

$$r_{yy} = 1.88 \text{ cm}$$

$$\frac{D}{t} = 13.8$$

Bending stress

$$f_{bc} = \frac{M}{Z_{xx}} = \frac{4.9 \times 10^6}{75.99 \times 10^3} = 64.48 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{3408}{13.8} = 247$$

From Table 3a the allowable bending stress p_{bc} is 89 N/mm^2

The stress in the beam is acceptable.

Check for deflection.

Deflection is given by

$$\frac{5 \cdot W \cdot L^3}{384 \cdot E \cdot I} = \frac{5 \times 11477 \times 3408^3}{384 \times 207,000 \times 482.5 \times 10^4} = 5.9 \text{ mm}$$

The allowable deflection given in BS 449 is

$$\frac{L}{360}$$

This factor is to ensure, among other reasons, that there will be no damage to building finishes, which is not a concern when designing tank roof structures.

BS 5950: Part 1, Table 5 gives several alternatives for allowable deflections. In particular it quotes $L/360$ for beams carrying plaster or other brittle finishes and also $L/200$ for all other beams.

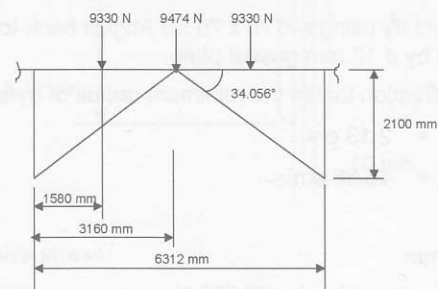
The $L/200$ is a more realistic figure for tank roof structures and this is the factor which will be used.

Applying this to the above rafter, then the allowable deflection is:

$$\frac{3408}{200} = 17.0 \text{ mm}$$

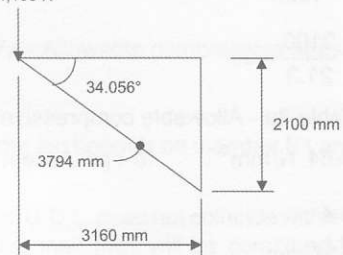
Hence the chosen beam size is acceptable for the stress level and deflection.

Purlin No. 4



Design of diagonal bracing

$$9474 + 9330 / 2 = 14,139 \text{ N}$$



Load in diagonal bracing

$$= 14,139 \div \sin 34.056 = 25,248 \text{ N}$$

Try using two 80 x 80 x 6 Angles back-to-back and separated by a 10 mm gusset plate.

$$\text{C.S.A.} = 1870 \text{ mm}^2$$

$$\text{Min. } r = 24.5 \text{ mm}$$

Compressive stress

$$f_c = \frac{25,248}{1870} = 13.5 \text{ N/mm}^2$$

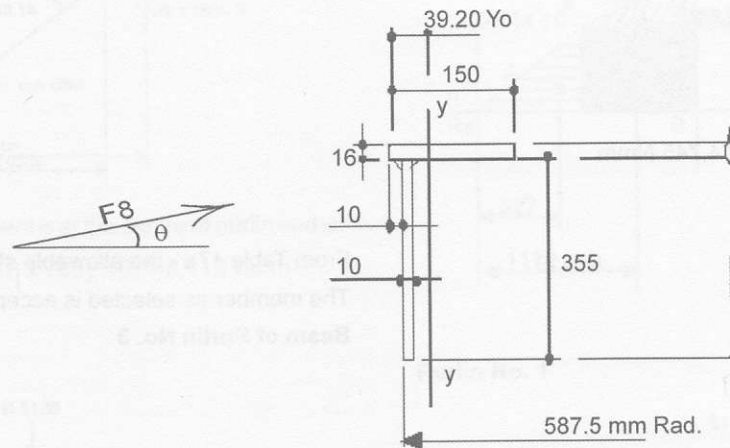
$$\frac{L}{r} = \frac{3794}{24.5} = 153$$

From Table 17a - the allowable stress $p_c = 40 \text{ N/mm}^2$

The member as selected is acceptable.

Crown ring

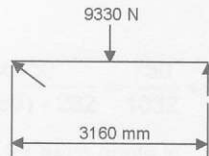
Central crown ring design using Roark's method



Tank dia.	30.00	m
Number of rafters	12.00	
Crown ring dia	1175.00	mm
Roof slope 1 in ?	5.00	
Compressive load in rafter	168.50	kN
Design stress	183.33	N / mm ²
2 x α = Angle Between Rafters	30.00	degrees
α = 1/2 Angle Between Rafters	15.00	degrees
$1/\alpha = 360 / (2\pi \times \alpha)$	3.820	radians
$1 / \sin \alpha$	3.864	
$1 / \tan \alpha$	3.732	
Horiz. Load on Ring "H" = $F8 \times \cos \theta$	165.228	kN
Properties of Ring		
C.S.A. of corroded Crown Ring	A =	5950.00 mm ²
Moment of Inertia on Axis thro' centroid	I yy =	9684541.32 mm ⁴
Section Modulus	Z yy =	87406.93 mm ³
Radius of Gyration	R yy =	40.34 mm
Radius of Crown Ring	R =	587.50 mm
Moment between Forces "H" is "Mo"		
$Mo = (H \times R / 2) \times (1/\sin \alpha - 1/\alpha)$	2134825.88	N
Compression in Ring is "No" = $H/2(1/\sin \alpha)$	319195.68	N
	Mo / Z =	24.42 N/mm ²
	No / A =	53.65 N/mm ²
Total Comp. Stress in Ring = $Mo/Z + No/A$	78.07	N/mm ²
Allowable Design Stress =	183.333	N/mm ²
Is Total Comp. Stress < Allowable Design Stress?	Yes	accept
Moment at Forces "H" is "Mi"		
$Mi = (H \times R / 2) \times (1/\alpha - 1/\tan \alpha)$	4255017.50	N
Tension in Ring is "Ni" = $H/2(1/\tan \alpha)$	308319.35	N
	Mi / Z =	48.68 N/mm ²
	Ni / A =	51.82 N/mm ²
Total Tension in Ring = $Mi/Z + Ni/A$	100.50	N/mm ²
Allowable Design Stress =	183.33	N/mm ²
Is Total Tensile Stress < Allowable Design Stress	Yes	accept
The complete Roof Design is --	accepted	

Figure 5.18 Central crown ring design calculation using Roark's method

Beam of Purlin No. 4



Bending moment

$$\frac{W \cdot L}{4} = \frac{9.330 \times 3.106}{4} = 7,244,745 \text{ Nmm}$$

Try using a 127 x 64 R.S.C.

$$Z_{xx} = 75.99 \text{ cm}^3$$

$$I_{xx} = 482.5 \text{ cm}^4$$

$$r_{yy} = 1.88 \text{ cm}$$

$$\frac{D}{t} = 13.8$$

Bending stress

$$f_{bc} = \frac{M}{Z_{xx}} = \frac{7,244,745}{75,990} = 95.3 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{1580}{18.8} = 84$$

From Table 3a - the allowable bending stress p_{bc} is 148 N/mm²

The stress in the beam is acceptable.

Check for deflection

Deflection is given by

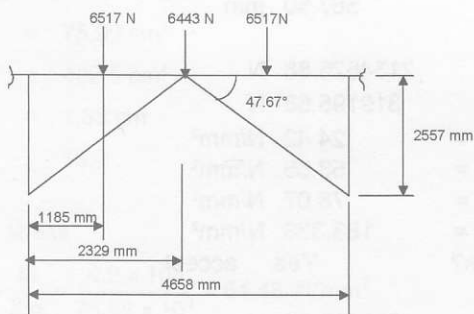
$$\frac{W \cdot L^3}{48 \cdot E \cdot I} = \frac{9.330 \times 3.106^3}{48 \times 207,000 \times 4825 \times 10^4} = 5.83 \text{ mm}$$

The allowable deflection is:

$$\frac{3106}{200} = 15.5 \text{ mm}$$

Hence the chosen beam size is acceptable for the stress level and deflection.

Purlin No. 3



Design of diagonal bracing.

Load in diagonal bracing = $9702 \div \sin 47.67 = 13,124 \text{ N}$

Try using two 80 x 80 x 6 Angles back-to-back and separated by a 10 mm gusset plate.

$$C.S.A. = 1870 \text{ mm}^2$$

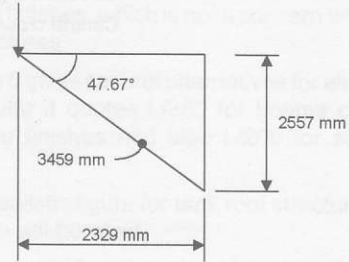
$$\text{Minimum } r' = 24.5 \text{ mm}$$

Compressive stress

$$f_c = \frac{13,123}{1870} = 7.0 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{3459}{24.5} = 141$$

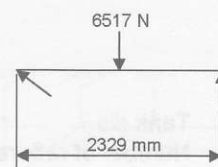
$$6443 + 6517 / 2 = 9702 \text{ N}$$



From Table 17a - the allowable stress $p_c = 46 \text{ N/mm}^2$

The member as selected is acceptable.

Beam of Purlin No. 3



Bending moment

$$M = \frac{W \cdot L}{4} = \frac{6517 \times 2329}{4} = 3,794,523 \text{ Nmm}$$

Try using a 127 x 64 R.S.C.

$$Z_{xx} = 75.99 \text{ cm}^3$$

$$I_{xx} = 482.5 \text{ cm}^4$$

$$r_{yy} = 1.88 \text{ cm}$$

$$\frac{D}{t} = 13.8$$

Bending stress

$$f_{bc} = \frac{M}{Z_{xx}} = \frac{3,794,523}{75,990} = 50.0 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{1185}{18.8} = 63$$

From Table 3a the allowable bending stress p_{bc} is 175 N/mm²

The stress in the beam is acceptable.

Check for deflection.

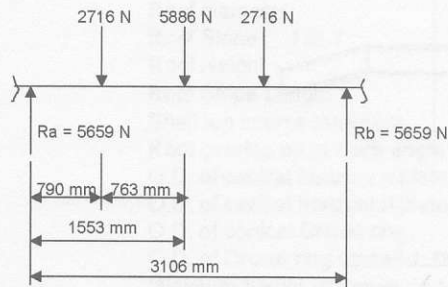
Deflection is given by:

$$\frac{W \cdot L^3}{48 \cdot E \cdot I} = \frac{6517 \times 2329^3}{48 \times 207,000 \times 4825 \times 10^4} = 1.7 \text{ mm}$$

The allowable deflection is

$$\frac{2329}{200} = 11.6 \text{ mm}$$

Hence the chosen beam size is acceptable for the stress level and deflection.

Purlin No. 2

The maximum bending moment is at the centre of purlin and is:

$$M = (5659 \times 1553) - (2716 \times 763) = 6,716,119 \text{ Nmm}$$

Try using a 127 x 64 R.S.C.

$$Z_{xx} = 75.99 \text{ cm}^3$$

$$I_{xx} = 482.5 \text{ cm}^4$$

$$r_{yy} = 1.88 \text{ cm}$$

$$\frac{D}{t} = 13.8$$

Bending stress

$$f_{bc} = \frac{M}{Z_{xx}} = \frac{6,716,119}{75,990} = 88.4 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{790}{18.8} = 42$$

From Table 3a - the allowable bending stress p_{bc} is 180 N/mm²

The stress in the beam is acceptable.

Check for deflection.

As the beam is loaded symmetrically, Mohr's area method will be used to determine the maximum deflection in the beam.

The deflection measured at R_a , from a tangent at the centre of the deflected beam is equal to:

The first moment of area of the bending moment diagram between R_a and the centre of the beam, divided by the modulus of elasticity and the second moment of area of the beam section.

$$\text{or Deflection} = \frac{1^{\text{st}} \text{ m.o.a. of B.M. diag. (Ra to centre)}}{E \times I}$$

1st m.o.a.:

$$A = \frac{790}{2} \times 4.471 \times 10^6 \times 527 = 93.1 \times 10^{10}$$

$$B = 763 \times 4.471 \times 10^6 \times 1172 = 399.8 \times 10^{10}$$

$$C = \frac{763}{2} \times 2.245 \times 10^6 \times 1299 = 111.3 \times 10^{10}$$

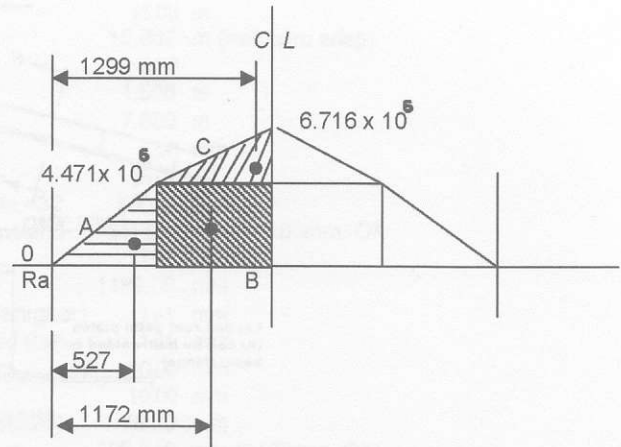
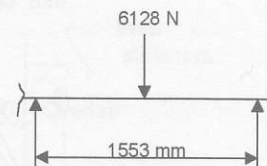
Total 1st m.o.a. of B.M. diag. between R_a & C.L. = $604.2 \times 10^{10} \text{ N/mm}^2$

$$\text{Deflection} = \frac{604.2 \times 10^{10}}{207,000 \times 482.5 \times 10^4} = 6.1 \text{ mm}$$

Allowable deflection is

$$\frac{3106}{200} = 15.53 \text{ mm}$$

Hence the chosen beam size is acceptable for the stress level and deflection.

**Purlin No. 1**

Maximum bending moment

$$M = \frac{W \cdot L}{4} = \frac{6128 \times 1553}{4} = 2.38 \times 10^6 \text{ N/mm}$$

Try using a 102 x 51 R.S.C.

$$Z_{xx} = 40.89 \text{ cm}^3$$

$$I_{xx} = 207.7 \text{ cm}^4$$

$$r_{yy} = 1.48 \text{ cm}$$

$$\frac{D}{t} = 13.3$$

Bending stress

$$f_{bc} = \frac{M}{Z_{xx}} = \frac{2.38 \times 10^6}{40890} = 58.2 \text{ N/mm}^2$$

$$\frac{L}{r} = \frac{776.5}{18.3} = 53$$

From Table 3a - the allowable bending stress p_{bc} is 180 N/mm²

The stress in the beam is acceptable.

The deflection in the beam

$$= \frac{W \cdot L^3}{48 \cdot E \cdot I} = \frac{6128 \times 1553^3}{48 \times 207,000 \times 207.7 \times 10^4} = 1.1 \text{ mm}$$

Allowable deflection is

$$\frac{1553}{200} = 7.8 \text{ mm}$$

Hence the chosen beam size is acceptable for the stress level and deflection.

Cross bracings

As mentioned earlier, the British Code requires that cross bracing shall be provided in the plane of this size of roof, to give the structure torsional stability. This bracing shall be in at least two bays of the roof, between two pairs of adjacent rafters.

In practice, it has been found that designers have often provided four sets of bracing in 30 metre diameter structures, as

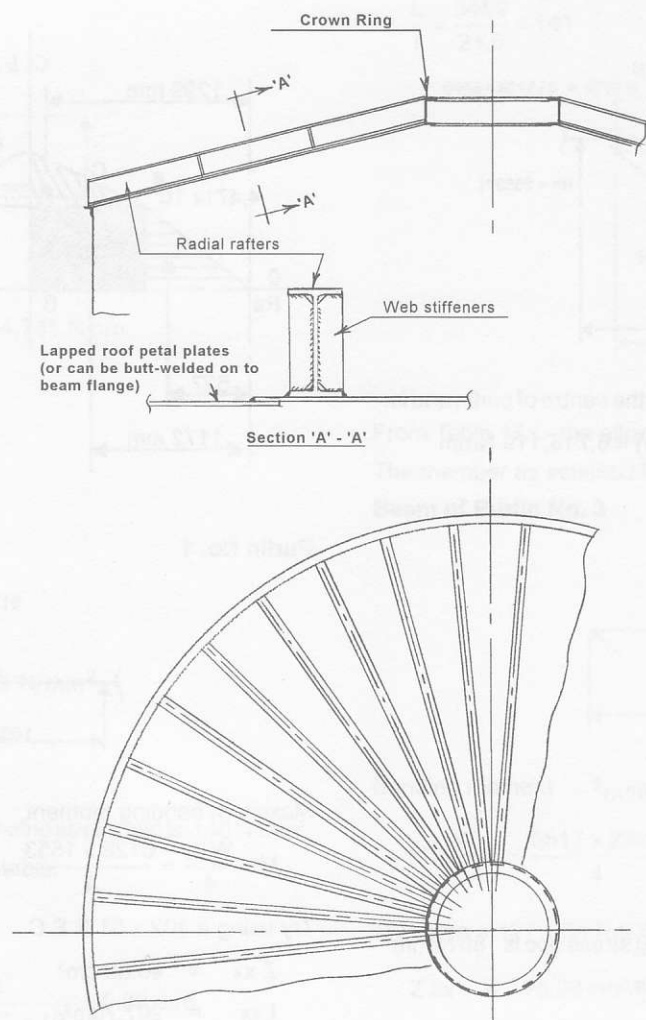


Figure 5.19 Externally-framed cone roof type arrangement

this has the advantage of giving added rigidity to the structure during the construction of the roof.

The selection of the section size for these bracings usually relies on the experience of the individual designer because there are no specific loads to work with. Hence the length of the bracing is considered with regard to the sag which is likely to occur due to self-weight, and a suitable angle section is normally chosen against this criteria.

For the structure designed above a bracing angle section of 70 x 70 x 6 has been chosen.

The weight of the finished structure can be calculated and in this case it is found to be 24,300 kg. Adding the weight of the roof plating, 29,000 kg, to this gives a total of 53,300 kg or 522713 N which gives a overall dead load of 739.5 N/mm² which equates favourably to the figure of 740 N/mm² used for design purposes.

This concludes the design for the trussed frame type structure.

5.5.1.6 Externally-framed roofs

This type of supporting structure consists of a series of radial steel sections. The roof petal plate sections are welded to the underside of the lower flange of each beam. The arrangement is shown in Figure 5.19.

The design calculation for this type of structure based on a 15 metre diameter tank is given in Figure 5.20.

5.5.2 Dome roofs

5.5.2.1 Radial rafter type

This structure consists of a series of curved radial steel beam sections connected to the shell at their outer end and to a centre crown ring at the centre of the tank. A series of circumferential rings provide lateral support for the beams and cross bracing in the plane of the roof is provided in some bays to give the structure torsional stability. This type of roof can be used in all sizes of tank and has an advantage over the truss type of structure when dealing with tanks over say 50 metres in diameter where the truss type structure becomes quite massive.

There is a further advantage because, unlike the truss type structure, the domed structure is completely clear of the stored product. Also, if an internal floating cover is to be installed in the tank, there is no loss of tank capacity.

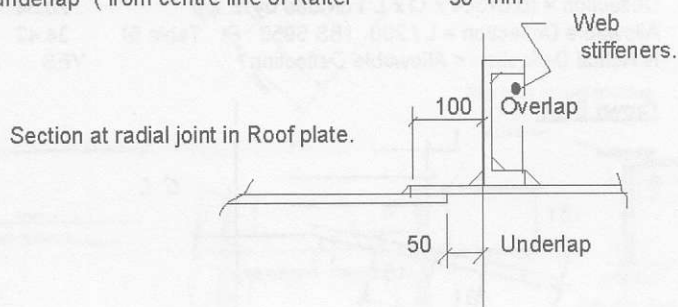
One disadvantage is that this type of roof is not frangible and therefore if frangibility is a desirable feature then it can not be used.

Details of this type of structure and an illustration showing a roof under construction are given in Figures 5.21 and 5.22 respectively.

Figure 5.23 (8 pages, at the end of this Chapter, pages 144 - 151), provides a typical design calculation for this type of structure, using a 39 metre diameter tank as the basis.

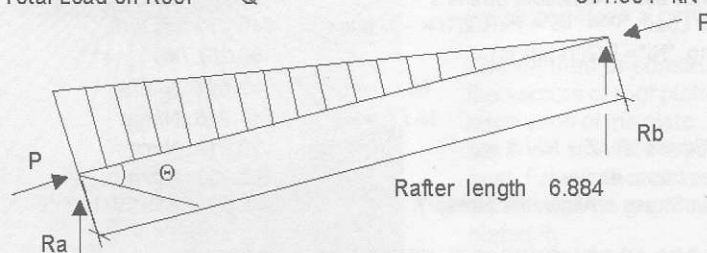
There are also software packages available such as STAAD or ANSYS which enable the complete roof structure to be modelled.

Tank diameter	15.00 m
Roof diameter	15.062 m (incl. curb o/lap)
Roof Slope 1 in ?	5.00
Roof Height	1.506 m
Roof Slope Length	7.680 m
Shell top course thickness	6.00 mm
Roof overlap on to Curb angle	25.00
O.D. of central horizontal plate of Crown ring. (min.=32	500.00 mm
O.D. of central horizontal plate to i.d. of Crown upstand	341.000 mm =>100mm, OK
O.D. of conical Crown ring.	1757 mm
O.D. of Crown ring upstand.	1189.00 mm
Minimum height of Crown ring upstand - (can be higher)	191 mm
Max. depth of Rafter fixing bracket to suit selected Raft	161 mm
Thickness of Rafter fixing bracket	10.00 mm
Thickness of Crown plating	10.00 mm
Flange width of Rafter (see below)	76.20 mm
Space between toes of adjacent Rafters at Crown	195.719 mm (>100mm, OK)
Rafter overlap on to Crown Ring (usually =>100 mm)	100 mm (>100mm, OK)
Gap between Rafter end & Crown upstand (say 190 mm)	190 mm
Petal plate edge "overlap" (from centre line of Rafter)	100 mm
Petal plate edge "underlap" (from centre line of Rafter	50 mm



Tan of Roof Angle	0.2000
Sin of Roof Angle	0.1961
Cos of Roof Angle	0.9806
Roof Angle (Theta)	11.310 degrees
Roof plate steel Type CS or SS ?	CS
Roof plate Yield or 1% Proof Stress	275.00 N/mm ²
Roof plate design Stress = 2/3 x Yield or 1% Proof Stre	183.33
Roof Plate Thks.	5.00 mm
Corrosion Allowance on Roof plating.	0.00 mm
Roof Plate Design Thks.	5.00 mm
Weight of Roof Plating	69.290 kN uncorroded
Weight of insulation	0.00 kN/m ²
Weight due to Insulation	0.000 kN

No. of Beams	16.00
Corrosion allowance off each face of Rafter	0.000 mm
Total corrosion allowance is therefore	0.000 mm
Unit weight of Beams	23.82 kg/m uncorroded
Weight of Structure detailed above	25.738 kN
Weight of Crown Ring	3.062 kN uncorroded
Superimposed Load (normally 1.2kN/m ²)	1.20 kN/m ²
Superimposed Load	213.814 kN
Total Load on Roof 'Q'	311.904 kN



Load per Rafter "Q" = Total Load/No. of Beams	19.494 kN
Vertical Load @ Roof Centre = 1/3 x "Q" =	6.498 kN
Load down axis of Rafter = "P" = "Rb"/sin Theta	33.133 kN

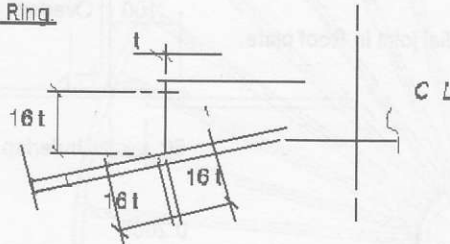
Figure 5.20 Design calculation for externally-framed cone roof type - page 1

Try using a Rafter Section: 203 x 76 x 23.82 kg/m R.S.C.

The relevant properties of the uncorroded Rafters are as follows : --

Depth of Section		203.20 mm
Flange width		76.20 mm
Flange thickness		11.200 mm
Weight of Rafter		23.820 kg / m
Cross sectional Area "A"	Uncorroded properties	30.34 cm ²
Moment of Inertia	I xx	1950.00 cm ⁴
Elastic Modulus	Z xx	192.00 cm ³
Radius of Gyration	R xx	8.02 cm
Ratio D/T		18.20
Length of Rafter	L	6.884 m
Slenderness Ratio L/R xx (Beam restrained by roof plate)		85.8
Modulus of Elasticity "E"		207.000 kN/mm ²
Max Bending Mnt. = BM = 0.128 x Q x L		17.177 kN.m
Max Bending Stress "fbc" = BM / Z		89.464 N/mm ²
Max Compressive Stress "fc" = P / A		10.921 N/mm ²
Allowable Bending Stress "pbc" (BS 449 Tables 2 & 3a)		150.0 N/mm ²
Allowable Comp. Stress "pc" (BS 449 Table 17a)		101.0 N/mm ²
fbc/pbc + fc/pc must be < 1.0 Actual value is :-		0.705 ACCEPTABLE
Deflection = (0.01304 x Q x L ³) divided by E x I		20.54 mm
Allowable Deflection = L / 200 (BS 5950 : Pt. Table 5)		34.42
Is Actual Deflection < Allowable Deflection?		YES ACCEPTABLE

Crown Ring



Effective regions of Ring = $16 \times 't'$ or the actual available dimension whichever is the smaller.

Upstand =	160 mm
Inner conical section =	160 mm
Outer conical section =	160 mm
Load on Crown Ring "P" =	33.133 kN
Section Modulus of Ring "Z" =	174.811 cm ³
C.S.A. of Ring "A" =	4837.858 mm ²
Radius of Crown Ring "R" =	594.500 mm
From "Roark 5th Edition Table 17-7"	
Angle between Rafters = $2 \times \alpha$	22.500 degrees
$1/2 \alpha = \alpha$	11.250 degrees
$1/\theta = (360 / 2 \times \pi \times \alpha)$	5.093 radians
$1/\sin \alpha =$	5.126
$1/\tan \alpha =$	5.027
Moment between Loads "P" = "Mo" = $P \times R / 2(1/\sin \alpha - 1/\alpha)$	323.761 kN.mm
Compression in Ring "No" = $P / 2(1/\sin \alpha)$	84.918 kN
Mo / Z =	1.852 N/mm ²
No / A =	17.553 N/mm ²
Total Compressive Stress Mo/Z + No/A =	19.405 N/mm ²
Allowable Stress from earlier is	183.333 N/mm ²
Is Total Comp. Stress < Allowable Stress ?	YES ACCEPTABLE
Moment under Load "P" = "Mi" = $P \times R / 2(1/\alpha - 1/\tan \alpha)$	646.273 kN.mm
Tension in Ring "Ni" = $P / 2(1/\tan \alpha)$	83.287 kN
Mi / Z =	3.697 N/mm ²
Ni / A =	17.216 N/mm ²
Total Tensile Stress Mi / Z + Ni / A =	20.913 N/mm ²
Allowable Stress from earlier is	183.333 N/mm ²
Is Total Tensile Stress < Allowable Stress ?	YES ACCEPTABLE

THE ROOF AS DESIGNED IS THEREFORE ACCEPTED

Number of plates required to cut Petal plates from is : - 8 OFF

8478 X 2937 X 5

Figure 5.20 Design calculation for externally-framed cone roof type - page 2

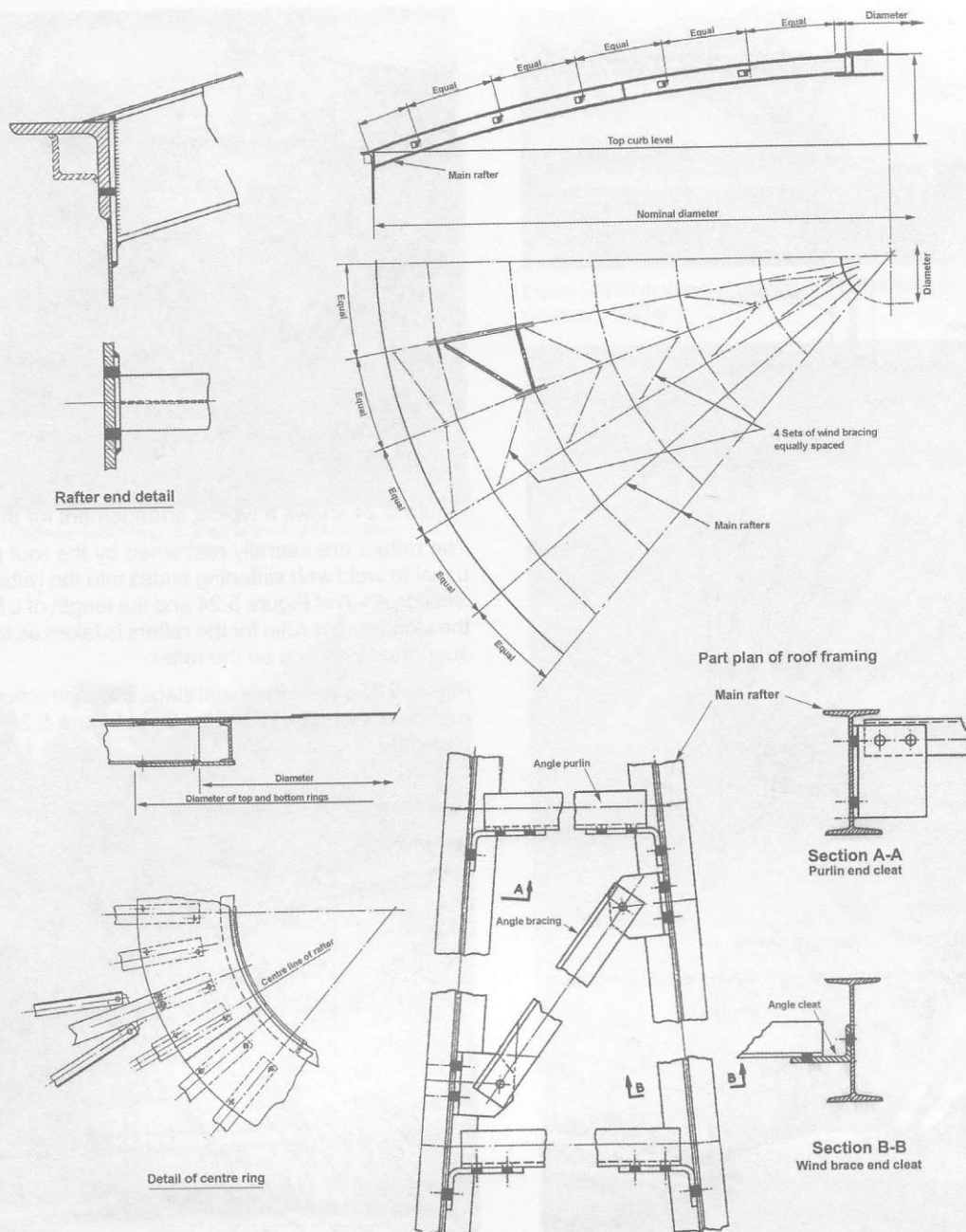
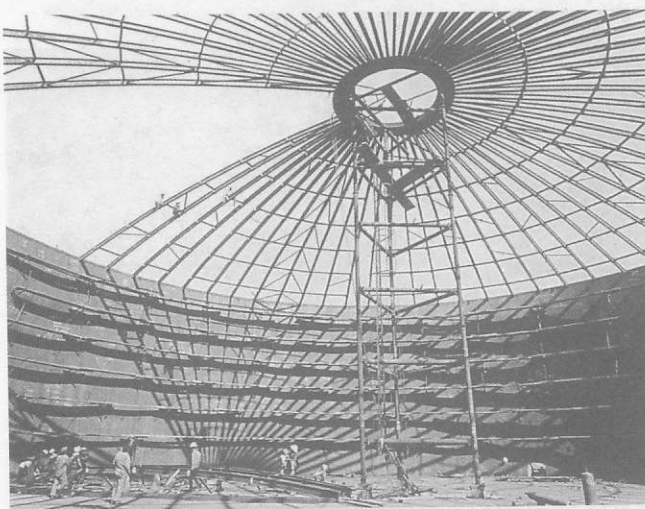


Figure 5.21 Details of rafter type dome roof

Figure 5.22 Radial rafter dome roof under construction
Courtesy of Whessoe

5.5.2.2 Externally-framed type

This again consists of a supporting structure composed of a series of curved radial rafters. In this case the roof sheeting is attached to the underside of the supporting rafters. This type of arrangement is ideal for internally-lined or stainless steel tanks, which can have a carbon steel external structure.

The method of construction used here was to shop-fabricate the sectors of roof plating with a radial beam already welded to each edge of the plate. The photograph shows the first four petals in place and supported at the centre by a temporary king post. Every other petal plate sector was then lifted into position and finally the gaps between the pre fabricated sectors were plated in.

The design of this type of structure is similar to that of the internally domed structure but as the roof plates are welded to the lower flange of the radial rafters, the rafters are "tied" together and hence there is no horizontal load transmitted to the shell from the rafters and hence the reinforced curb angle arrangement is not required.

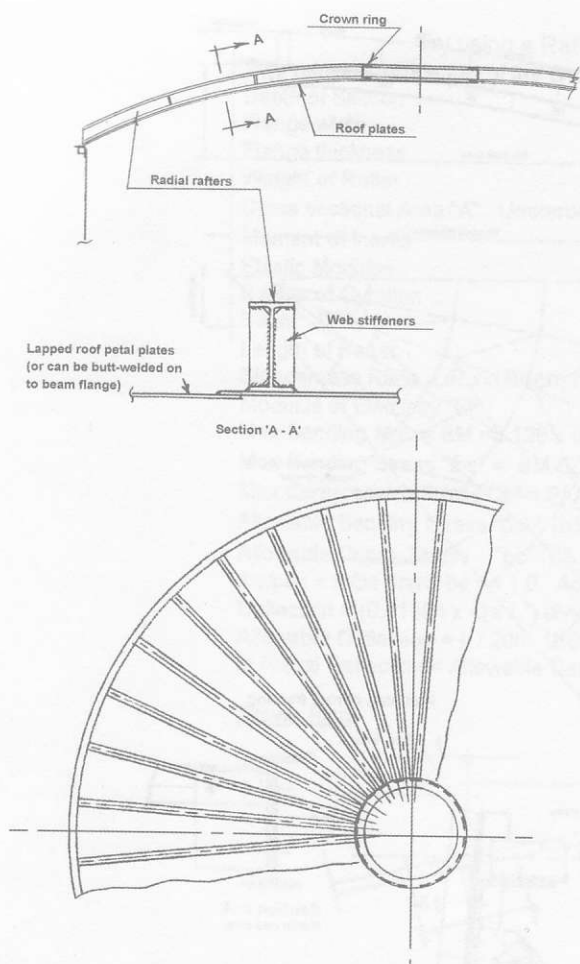


Figure 5.24 Externally-framed dome roof type arrangement

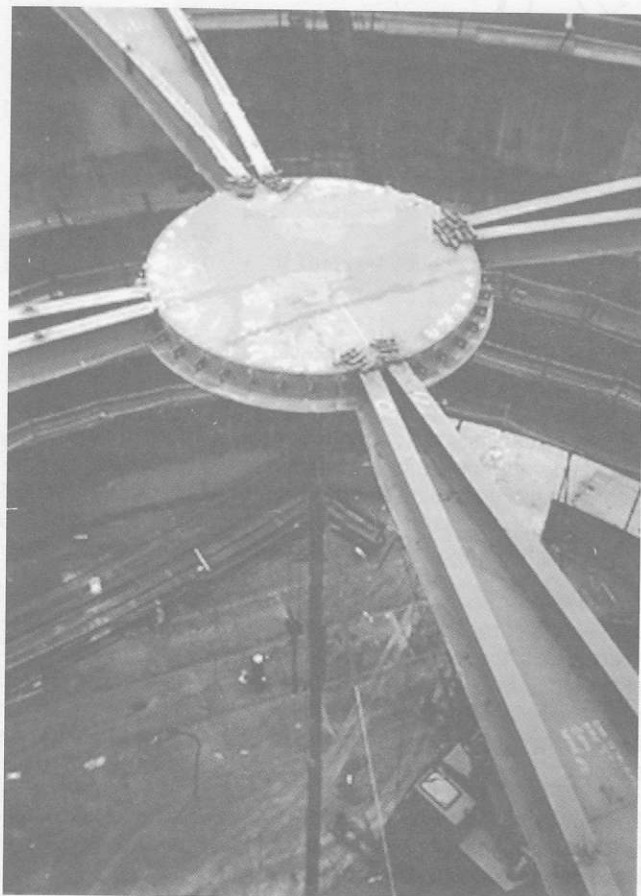


Figure 5.25 Initial stage of construction of externally-framed dome roof of a 44 m diameter tank
Courtesy of McTay

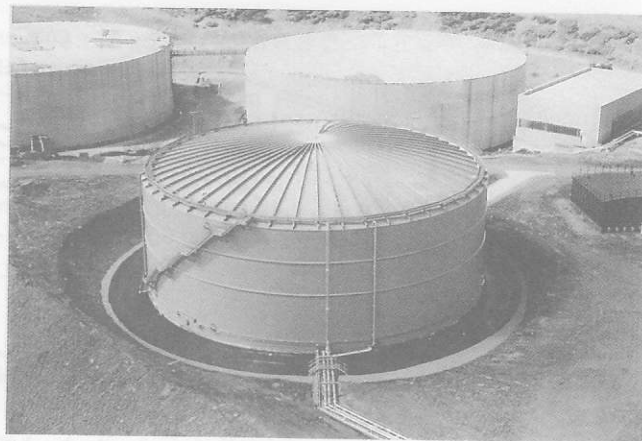


Figure 5.26 Completed externally-framed dome roof tank
Courtesy of Whessoe

Figure 5.24 shows a typical arrangement for this type of roof. The rafters are laterally restrained by the roof plating but it is usual to weld web stiffening plates into the rafters as shown in Section A – A of Figure 5.24 and the length of L for determining the slenderness ratio for the rafters is taken as the greatest unsupported distance on the rafter.

Figure 5.25 shows the initial stage of construction of this type of roof on a 44 metre diameter tank. Figure 5.26 shows a completed 90 m diameter tank roof.

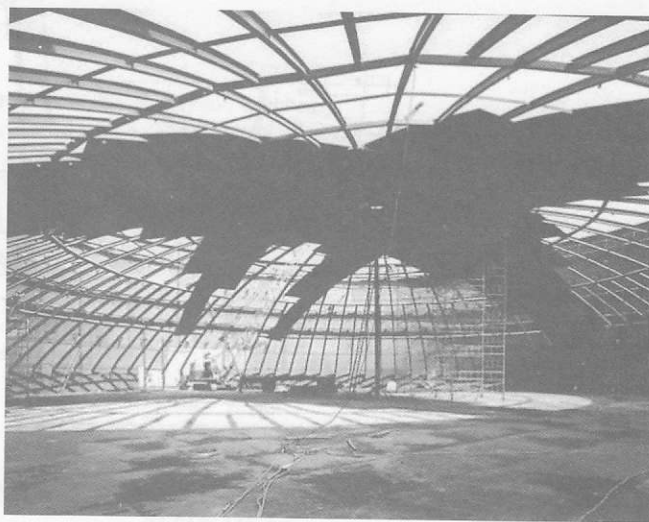


Figure 5.27 90 m diameter internally-framed dome roof under construction



Figure 5.28 90 m diameter internally-framed dome roof completed and ready to be air-lifted (note the stabilisation cables attached to the centre of the floor)



Figure 5.29 A 90 m diameter roof being air-lifted to the top of the tank



Figure 5.30 A 90 m diameter roof being secured into place

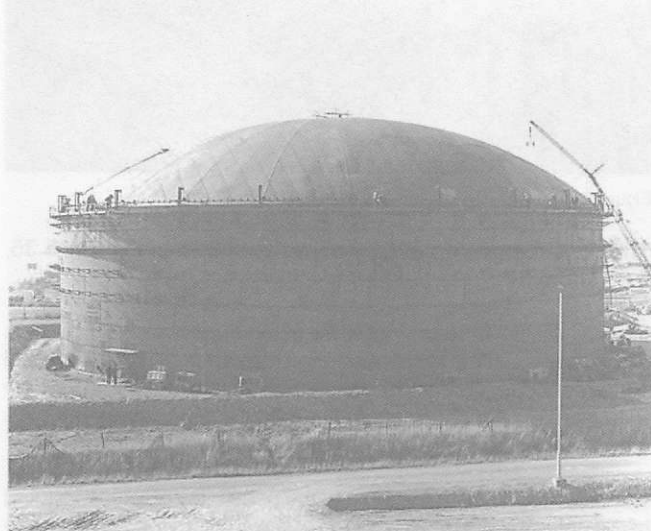


Figure 5.31 A 90 m diameter roof in its final position and ready for welding to the shell compression plate

5.5.3 Other types

There are a number of methods available for designing domed roofs and in some instances the circumferential rings are deemed to take tensile loads, thus decreasing the load in the



Figure 5.32 33 m diameter geodesic dome roof being built alongside a tank
Courtesy of McTay



Figure 5.33 A 33 m diameter aluminium geodesic dome roof being lifted into position on a tank
Courtesy of McTay

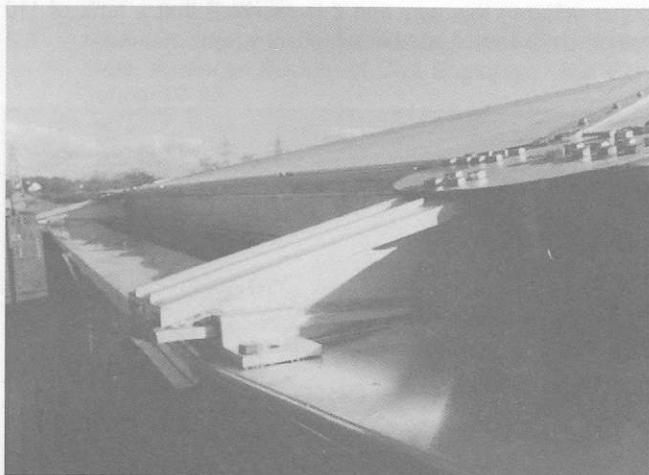


Figure 5.34 A 33 m aluminium geodesic dome roof in position on the tank ready for the final peripheral flashings to be put into place
Courtesy of McTay

main rafters. In particular for very large diameters say above 80 metres, *Reference 5.2* should be consulted.

For ease of construction, these very large diameter roofs are often constructed inside the shell on the floor of the tank, see Fig-

ures 5.26, and then lifted to the top of the tank under air pressure. The small gap between the rim of the completed roof and the shell is sealed with a temporary flexible membrane which is secured to the roof rim. The pressure under the roof which is required to lift it is surprisingly small.

Take a 90 m diameter roof having an all-up weight of 620 tonnes. The pressure equalling this weight over the area of the tank is equivalent to 9.6 mbar and this pressure can be delivered by large volume fans attached to the shell manholes. The roof is stabilised during its ascent by cables attached to the floor which pass through the crown of the roof and across the outer surface to sheaves at the rim, finally these cables are anchored at points above the rim of the shell. Figures 5.27, 5.29, 5.30 and 5.31 show a 90 m diameter roof constructed and lifted in this way.

5.5.3.1 Geodesic dome roofs

This type of roof is a fully triangulated, spherical, space frame structure, generally designed to be self-supporting from its periphery with an integral peripheral tension ring to take the horizontal forces. They are usually constructed in reinforced plastic or aluminium, Figure 5.32 to 5.34 show a 33 m diameter roof of this type under construction and being lifted into position.

They are particularly suited to water and wastewater applications where their corrosion resistant properties are a distinct advantage, also these relatively lightweight structures lend themselves to being retrofitted to existing tanks for the containment of vapour, gasses and odours, as they can be erected alongside a tank and lifted into position in one piece.

They are also used in the petrochemical industry, again for the containment of vapours or as weatherproof covers for floating roof tanks containing moisture sensitive products.

5.6 Column-supported roofs

As an alternative to providing a structure which is supported only by the tank shell, the column-supported roof introduces a series of vertical supporting columns. These are arranged in a series of circumferential rings around a single centre column. The rings of columns are circumferentially linked by girders which in turn support radial rafters on which the roof plating is laid. It is usual to adopt a shallow conical shape (1 in 16) and in theory there is no limit to the size of the tank roof which can be constructed in this way and it is reported that a tank of 110 metres in diameter has been built.



Figure 5.35 Column-supported roof tanks under construction

Courtesy of MB Engineering Services Ltd



Figure 5.36 Column-supported cone roof tanks under construction
Courtesy of Whessoe

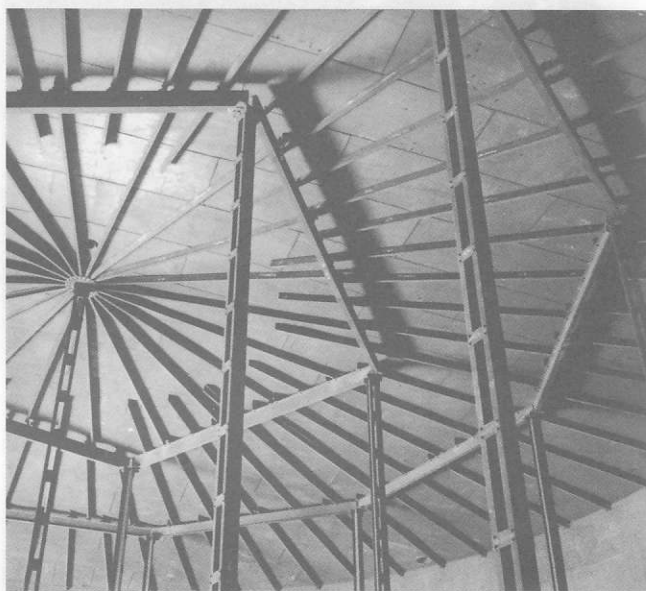


Figure 5.37 Completed column-supported roof structure

The construction of this type of roof is shown in Figures 5.35, 5.36 and 5.37.

Clearly, careful thought has to be given in cases where there is a possibility that the tank foundation may be prone to differential settlement due to poor soil conditions, which can result in differential settlement of the columns, thus causing undesirable increase stresses in the roof members and their connections.

Consideration has to be given to the possibility of lateral loading of the columns due to the motion of the stored product when designing for a seismic condition. The column bases should, under all conditions, be restrained in position on the tank floor. The bases should not be attached to the floor but shall be prevented from moving by welding angle cleats to the floor at the edges of the column bases.

Figure 5.37 shows the rafters projecting beyond the support beams, this is done to ensure that the maximum allowed spacing of 1.7m (5.5 ft) between the rafters is maintained.

To provide torsional stability in the plane of the roof it is necessary to provide cross bracing in at least two bays of the structure for roofs exceeding 15m in diameter. These sets of bracing should be spaced evenly around the tank circumference. The bracings are normally thin flat tie bars welded to the top flanges of the rafters or may be tie rods connected between the webs of the rafters.

The shallow roof slope makes this type of roof unsuitable for internal pressures much in excess of the self-weight of the roof plating itself (usually 4 mbar).

For column-supported roof structures which are designed to the British Code then the recommendations of the Structural Steel Code BS 449 shall apply.

For tanks designed to the American Code then the applicable Structural Steel Codes which apply to the country in which the tank is being built shall apply.

For tanks which are built in America the AISC Code, (see *Reference 5.3*), shall be used together with the overriding requirements of API 650 given in the Code, clause 3.10.3.3. for slenderness ratios and clause 3.10.3.4, for the allowable compression in columns.

The design of column-supported roofs is fairly straightforward and may be approached as follows:

- Split up the area of the roof and apportion the resulting loads to the individual radial rafters. These rafters are treated as simply supported beams with a U.D.L.
- The girders connecting the tops of the columns together take the point loads from the radial rafters, remembering that the girders support half the load from an inner ring of rafters, plus half the load from an outer ring of rafters. Again the girders are considered as simply supported beams with multi-point loads.
- Half the load from each of the two adjacent girders in a circumferential ring is carried by the connected column and the design of the columns is subject to the applicable Structural Steel design Code.

5.6.1 Column selection

The selection of the type of column section to be used excites the imagination inasmuch as the columns are usually quite tall and hence the minimum radius of gyration through any axis of the column must be as large as possible in order to arrive at the greatest value obtainable for L/r . The obvious answer is to use a tubular section for the columns, which of course has only one

value for its radius of gyration but there is often reluctance to using tubes because of the possibility of internal corrosion damage which cannot be detected, also tubes are often more expensive than other sections or combination of sections.

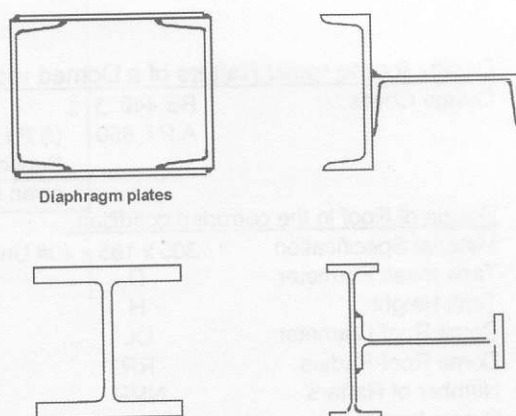


Figure 5.38 Examples of other sections used for columns in column-supported roofs

Other sections which have been used are shown in Figure 5.38.

5.7 References

- 5.1 *Structural stability of the tank-code requirements*, Professor A.S. Tooth, Department of Mechanical Engineering, University of Strathclyde.
- 5.2 *A design philosophy for large storage tank braced dome roofs*, *The Structural Engineer*, G. Thompson, G. K. Schleyer and Prof. A. S. Tooth, 1987.
- 5.3 *Specification for Structural Steel Buildings, Manual of Steel Construction, Allowable Stress Design*, The American Institute of Steel Construction (AISC), (Note that Chapter 'N' on the use of plastic design in Part 5 Allowable Stress Design of this latter Specification is specifically not allowed.)
- 5.3 *Steel Plate Engineering Data Series, Useful Information - Design of Plate Structures, Volume II.*, American Iron & Steel Institute (AISI).
- 5.5 *Minimum design loads for Buildings and other Structures*, American Society of Civil Engineers (ASCE) - Standard 7-93.

Design for the radial Rafters of a Domed roof.

Design Codes :-

BS 449

A.P.I. 650

(A.P.I. 650 does not give all of the specific requirements for Supported Dome or Umbrella Roofs therefore the guidance given in Clause 3.10.2.7. applies to this design.)

Design of Roof in the corroded condition.

Material Specification	305 x 165 x 40# Universal Beam to B.S.4. in BS En 10025 S275 Material		
Tank mean Diameter	D	39 m	
Tank Height	H	22 m	
Dome Roof Diameter	DL	39	
Dome Roof Radius	RR	58.5 m. RR / DL = 1.50	OK
Number of Rafters	NMR	44	
Super. load		1.2 kN/m ²	
Rafter weight		31 kg/m	
Roof plate thickness		5 mm	
Roof plate corrosion allowance		0 mm	
Other uniform roof load	Purlins	0.031 kN/m ²	
Crown ring	4.65 kN	0.004 kN/m ²	
Design load for roof	TL	1.86 kN/m ²	
Radius to inner end of Rafter	RU	1250 mm	
Dia. to inner end of Rafter	RD	2500 mm	
For lateral restraint the Rafter is split into	5	sections by fitting Purlins. (Actually there are 6 sections, but the outer one is not at the same spacing as the others therefore is ignored here.)	

Dome Roof Design

1. Determine load applied by the structure Crown Ring.

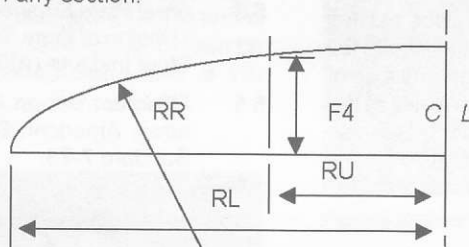
$$PCL = \frac{RD^2 \cdot \pi \cdot TL}{4 \cdot NMR}$$

Where : RD = Diameter of Crown Ring. (2 x RU)

TL = Roof loading.

NMR = Number of main Rafters.

2. Determine geometry of any section.



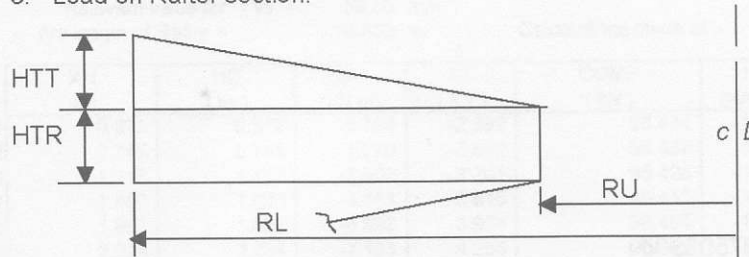
RL = Rad. at outer end of Rafter

RU = " " inner " " "

RR = Rad. of dome.

F1 = Angle subtended by RU $\text{Arcsine} (RU / RR)$ F2 = Angle subtended by RL $\text{Arcsine} (RL / RR)$ F3 = Angle subtended by section $F2 - F1$ F4 = Rise in height of section $\{(1 - \cos F2) - (1 - \cos F1)\} \cdot RR$ Arc = Arc length RL to F4 $F3 \cdot RR$

3. Load on Rafter section.



Where :

$$HTR = \frac{RU \cdot 2 \cdot \pi \cdot TL}{NMR}$$

$$HTT = \frac{(RL - RU) \cdot 2 \cdot \pi \cdot TL}{NMR}$$

4. Reactions at lower end of Rafter section.

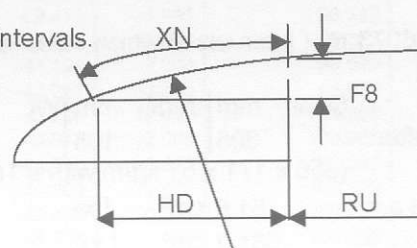
Horizontal Reaction.

$$HTH = \{ (HTR \cdot (RL - RU)^2 / 2 + (HTT \cdot (RL - RU)^2 / 6 + PCL \cdot (RL - RU)) / F4$$

Vertical Reaction.

$$VTH = HTR \cdot (RL - RU) + (HTT \cdot (RL - RU)) / 2 + PCL$$

5. Calculations at 50 No. intervals.



XN = Present arc distance from upper end of section.

$$F7 = XN / RR$$

$$HD = \text{Horizontal distance. } \{ \sin (F1 + F7) \cdot RR \} - RU$$

$$F8 = \text{Vertical distance at point } \{ (1 - \cos (F1 + F7)) - (1 - \cos F1) \} \cdot RR$$

6. Bending moment at the above intervals.

$$BM1 = -HTH \cdot F8 + \frac{HTT \cdot HD^3}{(RL - RU) \cdot 6} + \frac{HTR \cdot HD^2}{2} + PCL \cdot HD$$

7. Shear force at above intervals.

F10 = Vertical load at any point considered.

$$= \frac{HTT \cdot HD^2}{(RL - RU) \cdot 2} + HTR \cdot HD + PCL$$

SF = Shear load at point considered.

$$= F10 \cdot \cos (F1 + F7) + (-HTH \cdot \sin (F1 + F7))$$

8. Compression at above intervals.

$$COM = F10 \cdot \sin (F1 + F7) + HTH \cdot \cos (F1 + F7)$$

9. Stress at above intervals.

$$f_c = -COM / \text{Area of Rafter}$$

$$f_b = BM / Z \text{ of Rafter}$$

$$\text{Stress in top flange} = f_c + f_b$$

$$\text{Stress in bottom flange} = f_c - f_b$$

Figure 5.23 Design calculation for radial rafter dome roof type - page 2

Design calculations.

1. PCL = 0.2075058 kN
2. F1 = 0.0213691 Rads.
F2 = 0.3398369 Rads.
F3 = 0.3184678 Rads.
F4 = 3.3323148 m
ARC = 18.630364 m
3. HTR = 0.3320092 kN/m
HTT = 4.8473347 kN/m
4. HTH = 98.476427 kN (Horiz. load at shell)
VTH = 50.498603 kN (Vert. load at shell)
5. XN = 0.3726073 m (intervals at which calcs. are made along the Rafter.)

Beam section to be used for the Rafter :-

Depth mm	Width mm	Wt. kg/m	Sect. type
305	165	40	U.B.

(356 x 171 x 51 kg/m with a 1mm c.a. off each face.)

Properties of Rafter :-

c.s.a.=	51.5 cm ²	I xx	8523 cm ⁴
Z xx =	561.2 cm ³	I yy :	763 cm ⁴
D/T=	29.9	r yy	3.85 cm

Is the Rafter welded to the Roof plating ? NO (i.e. Internal or external structure ?)

Purlin Section size is :- 90 x 90 x 10 R.S.A.

Thickness of Roof plating 5 mm

Roof plating corr. allowance 0 mm

Roof plating design thicknes 5 mm

Properties of Rafter incl plate

c.s.a.=	67.75 cm ²	I xx	11491.00 cm ⁴
(For external structures only)	Z xx =	605.82 cm ³	D/T=
			29.9
		r yy	5.69 cm

For this case :-

Use bare Rafter properties only

c.s.a.=	51.5 cm ²	I xx	8523.00 cm ⁴
Z xx =	561.20 cm ³	D/T=	29.9
		r yy	3.85 cm

The value of 'r' to be used is r yy = 3.85 cm (for lateral restraint for the Beam)

Figure 5.23 Design calculation for radial rafter dome roof type- page 3

Cross sectional area = 5150 mm² Zxx = 561200 mm³ D/T = 29.9
 Relevant value for 'r yy' = 38.50 mm
 Arc length of Rafter = 18.630 m Calculations made at :- 50 intervals along Rafter

	XN arc	HD (m)	BM (kN.m)	SF (kN)	COMP (kN)	f c (N/mm ²)	f b (N/mm ²)	Top (N/mm ²)	Btm (N/mm ²)
1	0.373	0.372	-0.798	-2.382	98.448	-19.116	-1.422	-20.539	-17.694
2	0.745	0.745	-1.770	-2.830	98.437	-19.114	-3.155	-22.269	-15.959
3	1.118	1.117	-2.903	-3.241	98.426	-19.112	-5.172	-24.284	-13.940
4	1.490	1.490	-4.181	-3.616	98.415	-19.110	-7.451	-26.560	-11.659
5	1.863	1.862	-5.593	-3.954	98.405	-19.108	-9.966	-29.074	-9.142
6	2.236	2.234	-7.123	-4.255	98.398	-19.106	-12.693	-31.799	-6.413
7	2.608	2.606	-8.759	-4.520	98.392	-19.105	-15.608	-34.714	-3.497
8	2.981	2.977	-10.487	-4.749	98.390	-19.105	-18.688	-37.793	-0.417
9	3.353	3.349	-12.294	-4.941	98.392	-19.105	-21.907	-41.012	2.801
10	3.726	3.720	-14.165	-5.098	98.399	-19.107	-25.241	-44.348	6.135
11	4.099	4.091	-16.088	-5.218	98.411	-19.109	-28.668	-47.777	9.559
12	4.471	4.462	-18.050	-5.303	98.429	-19.112	-32.163	-51.275	13.050
13	4.844	4.833	-20.036	-5.352	98.454	-19.117	-35.702	-54.819	16.585
14	5.217	5.203	-22.034	-5.366	98.486	-19.123	-39.262	-58.386	20.139
15	5.589	5.574	-24.031	-5.345	98.525	-19.131	-42.820	-61.951	23.689
16	5.962	5.944	-26.013	-5.289	98.574	-19.141	-46.353	-65.493	27.212
17	6.334	6.313	-27.968	-5.199	98.632	-19.152	-49.836	-68.988	30.684
18	6.707	6.683	-29.883	-5.074	98.700	-19.165	-53.248	-72.413	34.083
19	7.080	7.052	-31.745	-4.915	98.779	-19.180	-56.566	-75.746	37.386
20	7.452	7.420	-33.541	-4.722	98.868	-19.198	-59.767	-78.965	40.570
21	7.825	7.788	-35.260	-4.496	98.970	-19.218	-62.829	-82.047	43.612
22	8.197	8.156	-36.888	-4.237	99.084	-19.240	-65.730	-84.970	46.490
23	8.570	8.524	-38.413	-3.945	99.212	-19.264	-68.448	-87.712	49.183
24	8.943	8.891	-39.823	-3.620	99.353	-19.292	-70.961	-90.253	51.669
25	9.315	9.258	-41.107	-3.263	99.509	-19.322	-73.248	-92.570	53.926
26	9.688	9.624	-42.251	-2.875	99.679	-19.355	-75.287	-94.643	55.932
27	10.060	9.990	-43.245	-2.455	99.865	-19.391	-77.059	-96.450	57.667
28	10.433	10.356	-44.077	-2.005	100.068	-19.431	-78.541	-97.971	59.110
29	10.806	10.721	-44.735	-1.523	100.287	-19.473	-79.714	-99.187	60.241
30	11.178	11.085	-45.209	-1.012	100.523	-19.519	-80.557	-100.076	61.038
31	11.551	11.449	-45.486	-0.471	100.777	-19.568	-81.051	-100.619	61.483
32	11.923	11.812	-45.556	0.099	101.050	-19.621	-81.176	-100.797	61.555
33	12.296	12.175	-45.408	0.698	101.341	-19.678	-80.913	-100.591	61.235
34	12.669	12.538	-45.032	1.325	101.653	-19.738	-80.243	-99.981	60.505
35	13.041	12.900	-44.417	1.981	101.984	-19.803	-79.147	-98.950	59.344
36	13.414	13.261	-43.553	2.663	102.335	-19.871	-77.607	-97.478	57.736
37	13.786	13.622	-42.429	3.372	102.708	-19.943	-75.605	-95.548	55.662
38	14.159	13.982	-41.037	4.108	103.102	-20.020	-73.123	-93.143	53.103
39	14.532	14.341	-39.365	4.869	103.518	-20.101	-70.145	-90.245	50.044
40	14.904	14.700	-37.405	5.656	103.957	-20.186	-66.652	-86.838	46.466
41	15.277	15.058	-35.147	6.467	104.419	-20.275	-62.629	-82.904	42.353
42	15.650	15.416	-32.583	7.302	104.904	-20.370	-58.059	-78.429	37.689
43	16.022	15.772	-29.703	8.161	105.413	-20.468	-52.927	-73.395	32.458
44	16.395	16.128	-26.498	9.043	105.946	-20.572	-47.216	-67.788	26.644
45	16.767	16.484	-22.960	9.948	106.504	-20.680	-40.913	-61.593	20.233
46	17.140	16.839	-19.082	10.874	107.087	-20.794	-34.002	-54.795	13.208
47	17.513	17.193	-14.854	11.822	107.695	-20.912	-26.469	-47.380	5.557
48	17.885	17.546	-10.270	12.790	108.329	-21.035	-18.300	-39.334	-2.735
49	18.258	17.898	-5.321	13.778	108.990	-21.163	-9.481	-30.644	-11.682
50	18.630	18.250	0.000	14.785	109.677	-21.297	0.000	-21.297	-21.297

Maximum values are : - 109.677 21.297 81.176
 Comp kN f c N/mm² f b N/mm²

Compare max. bending stresses against allowable to BS 449.

The Rafter is not welded to the roof plating, therefore the relevant value of 'r yy' is to be used based upon the effective length between purlins.

Length of Rafter Lr = 18.360 m.

The Beam is split into:- 5 sections by web stiffeners or purlins

Lr/5 = L

L = 3.726 m

L/r = 97

D/T = 29.9

Figure 5.23 Design calculation for radial rafter dome roof type - page 4

Table 17a of BS 449	$p_c =$	86 N/mm ²
Table 3a of BS 449	$p_{bc} =$	127 N/mm ²
Actual comp'v'e. stress	$f_c =$	21.297 N/mm ²
Actual bend'g stress	$f_{bc} =$	81.176 N/mm ²
	$f_c / p_c + f_{bc} / p_{bc} =$	0.89 < 1, OK

10. Crown Ring design

From Roark 5th edition Table 17 Ref. No. 7

Angle between Rafters	$= 2\alpha =$	0.143 rads.
1/2 angle between Rafters	$= \alpha =$	0.071 rads.
	$1/\alpha =$	14.006
	$1/\sin \alpha =$	14.018
	$1/\tan \alpha =$	13.982

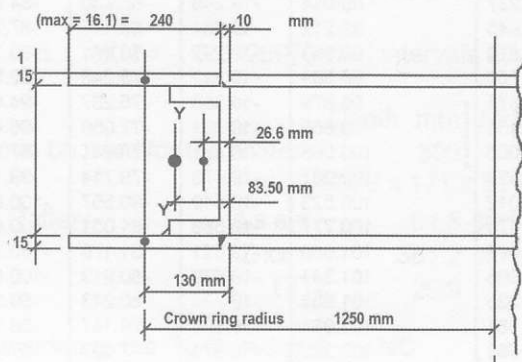
Selection of Crown Ring properties

Enter requirements Y or N

From Sheet 'B' of this Prog.

From another source (give details):

Y/N ?	Details as applicable	Sect Mod'ls Z (cm ³)	Area A (cm ²)
N	See below	319.5	137.52
		445.48	130.83
		(cm ³)	(cm ²)



Properties of Channel:

Size: 305 x 102 x	46.18 kg/m
c.s.a. =	58.83 cm ³
I =	499.50 cm ⁴
cy =	2.66 cm

Areas:

Channel	58.83
Plate rings	72.00
Total	130.83 cm ²

1st m.o.a. from back of Channel:

Channel	156.49
Plate rings	936.00
Total	1092.49 cm ³

Weight of Crown ring =	332.69 kg Channel +	141.3 kg Top & Btm plates
=	473.99 kg which is	4.65 kN

Position of centroid of section =	$\frac{1092.49}{130.83} =$	8.35 cm
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Figure 5.23 Design calculation for radial rafter dome roof type - page 5

2nd m.o.a. about centroid of section:

Channel	2404.479 cm ⁴	
1 gg for plate	$\frac{B \cdot D^3}{12}$	1728 cm ⁴
1 yy for plate	2506.263 x 2 =	5012.527 cm ⁴
Total 2nd m.o.a. =	2404.48 + 5012.53 =	7417.01 cm ⁴
y max. =	16.65 cm	
y min. =	8.35 cm	
Z yy =	$\frac{7417.01}{16.65}$ =	445.48 cm ³
Cross sectional area A =	130.830 cm ²	
Section modulus Z =	445.480 cm ³	
Total weight W =	474.0 kg	or = 4.65 kN
Horizontal load = HTH = H =	98.476 kN	
BM between loads on Ring =	Mo = H x R/2 (1/sin ∞ - 1/(1/∞)) =	732.853 kN.mm
Compression in Ring is =	No = H/2 (1/sin ∞) =	690.199 kN
	Mo/Z =	2.086 5.602 N/mm ²
	No/A =	15.493 N/mm ²
Total comp. stress in Ring =	Mo/Z + No/A =	21.095 N/mm ²
Allowable stress to BS 449 =		180.000 N/mm ²
	Is the actual stress in the Ring acceptable?	YES
BM at loads on Ring =	Mi = H x R/2(1/∞ - 1/tan ∞) =	1465.332 kN.m
Tension in ring is =	Ni = H/2 (1/tan ∞) =	688.440 kN
	Mi/Z =	11.200 N/mm ²
	Ni/A =	15.454 N/mm ²
Total tensile stress in Ring =	Mi/Z + Ni/A =	26.654 N/mm ²
Allowable stress to BS 449 =		180.000 N/mm ²
	Is the actual stress in the Ring acceptable?	YES

Deflections in the Ring due to load from Rafters

Radial displacement at each load point = $\frac{H \times R^3}{2 \times E \times I} [1/\sin^2 \infty (1/2 \infty + 1/2 \sin \infty \cos \infty) - 1/\infty]$

1/sin ² ∞ =	196.491
1/2 ∞ =	0.0357
1/2 sin ∞ cos ∞ =	0.036
1/∞ =	14.006
E =	207000 N/mm ²
I =	6104.4 cm ⁴

Figure 5.23 Design calculation for radial rafter dome roof type- page 6

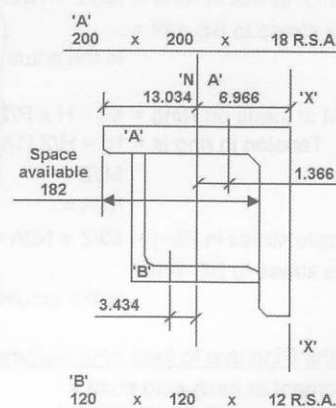
Radial displacement at each load point =	0.000062 mm (inwards)
Acceptable displacement = Length between loads/200	0.892 mm
Is displacement acceptable?	YES
Radial displacement between each load point =	$\frac{H \times R^3}{4 \times E \times I} [2/\alpha - 1/\sin\alpha - [\alpha \times (\cos\alpha/\sin\alpha^2)]]$
$2/\alpha =$	28.011
$\cos\alpha =$	0.997
$\sin\alpha^2 =$	0.005
$\alpha \times (\cos\alpha/\sin\alpha^2) =$	13.994
Radial displacement between each load point =	0.000054 mm (outwards)
Is displacement acceptable?	YES

The design of the roof is accepted

In the above design method, the main rafters are deemed to carry all the loadings and the circumferential rings are there to give lateral support to the rafters but they do not take any appreciable load. This means that the rafters exert an appreciable horizontal load at their attachment point to the shell and the top of the shell must be reinforced to take this load. From the above calculation this load is seen to be HTH at 98.47 kN and the necessary reinforcement in this case is provided by a double angle arrangement which is designed as follows:

Design of a Ring, (Curb Angle) having multiple equispace
Design based on Roark theory Table 17 Case 7

Try two angles forming a box section
 200 x 200 x 24 R.S.A. and a 120 x 120 x 12 R.S.A.



Number of equispaced loads acting on the Ring.	44
Horizontal Load on Crown Ring HTH = "H" =	98.476 kN (from Sht. 'A')
Radius of Ring "R" =	19500 mm
C.S.A of Ring "A" =	9660 mm ²
Moment of Inertia of Ring "I" =	3421.227 cm ⁴
Section of Modulus of Ring "Z" =	262.494 cm ³

Figure 5.23 Design calculation for radial rafter dome roof type - page 7

From "Roark 5th Edition Table 17-7

Angle between Rafters = $2 \times \infty$ =	8.181818 degrees
1/2 Angle between Rafters = ∞ =	4.090909 degrees
1/Theta = $(360/2 \times \pi \times \infty)$ =	14.00563 radians
1/sin ∞ =	14.01754
1/tan ∞ =	13.98183
Moment between Loads "H" = "Mo" = $H \times R/2(1/\sin \infty - 1/\tan \infty)$ =	11432.5 kN.mm
Tension in Ring "No" = $H/2(1/\sin \infty)$ =	690.1987 kN/mm
Mo/Z =	43.55336 N/mm ²
No/A =	71.44914 N/mm ²
Total Tension Stress Mo/Z + No/A =	115.0025 N/mm ²
Allowable Stress from BS 449 is:	180 N/mm ²
Is Total Tensile Stress < Allowable Stress? YES	ACCEPTABLE
Moment under Load "H" = "Mi" = $H \times R/2(1/\infty - 1/\tan \infty)$ =	22859.17 kN.mm
Compression in Ring "Ni" = $H/2(1/\tan \infty)$ =	688.4402 kN
Mi/Z =	87.08452 N/mm ²
Ni/A =	71.2671 N/mm ²
Total Comprehensive Stress Mi/Z + Ni/A =	158.3516 N/mm ²
Allowable Stress from BS 449 is:	180 N/mm ²
Is Total Comprehensive Stress < Allowable Stress? YES	ACCEPTABLE

Deflections in the Ring due to load from Rafters

Radial displacement at each load point =

$$\frac{H \times R^3}{2 \times E \times I} [1/\sin^2 \infty (1/2 \infty + 1/2 \cdot \sin \infty \cdot \cos \infty) - 1/\infty]$$

1/sin ² ∞ =	196.4915
1/2 ∞ =	0.0357
1/2 · sin ∞ cos ∞ =	0.035579
1/ ∞ =	14.00563
E =	207000 N/mm ²
I =	3421.227 cm ⁴

Radial displacement at each load point =	0.417 mm (outwards)
Acceptable displacement = Length between Loads/200 =	13.923 mm
Is displacement acceptable? YES	

Radial displacement between each load point =

$$\frac{H \times R^3}{4 \times E \times I} [2/\infty - 1/\sin \infty - [\infty \times (\cos \infty / \sin \infty^2)]]$$

2/ ∞ =	28.01127
1/sin ∞ =	14.01754
cos ∞ =	0.997452
sin ² ∞ =	0.005089
$\infty \times (\cos \infty / \sin \infty^2)$ =	13.99371

Radial displacement between each load point =	0.365265 mm (inwards)
Is displacement acceptable? YES	

The design of the Ring is accepted

Figure 5.23 Design calculation for radial rafter dome roof type - page 8

6 The design of tank roofs - floating

A floating roof greatly reduces vapour losses due to changes in climatic conditions and during tank filling operations. These losses are particularly significant where volatile organic compounds are stored in tanks which are subject to high filling and emptying cycles. The two types of floating roofs are discussed: the external floating roof and the internal floating roof and variations on these. A review of floating roof accessories or equipment is made and examples of many appurtenances given.

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6.2 The principal of the floating roof

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6.5.15 Sample/dip hatch

6.5.16 Foam dam

6.5.17 Electrical continuity

6.1 Introduction

The realisation that a great deal of product was being lost by evaporation from fixed roof petroleum tanks lead research into developing a roof which floated directly on the surface of the product thus reducing these evaporation losses.

The development of this technology began shortly after the first World War by Chicago Bridge & Iron Company (CB & I), which undertook full scale floating roof fire tests in the presence of prominent leaders in the petroleum and insurance industries to convince them that storing volatile products in floating roof tanks was a viable proposition.

A series of tests were carried out in 1923, see Figure 6.1, whereby gasoline was poured on to a floating roof and its seals and fittings, and was then ignited. The fire was readily extinguished without damage to tank or its contents of gasoline, see Figure 6.2. The original CB & I floating roof designs, and some variant of them, have been in regular use ever since.



Figure 6.1 CB & I Floating Roof fire test in 1923
Courtesy of



Figure 6.2 CB & I Floating Roof fire test for invited audience of petroleum industry leaders — hats compulsory!
Courtesy of

The use of a floating roof also greatly reduces vapour losses due to changes in climatic conditions and during tank filling operations. These losses are particularly significant where volatile organic compounds are stored in tanks which are subject to high filling and emptying cycles.

Figure 6.3 illustrates very simply the loss mechanisms experienced in fixed roof tanks.

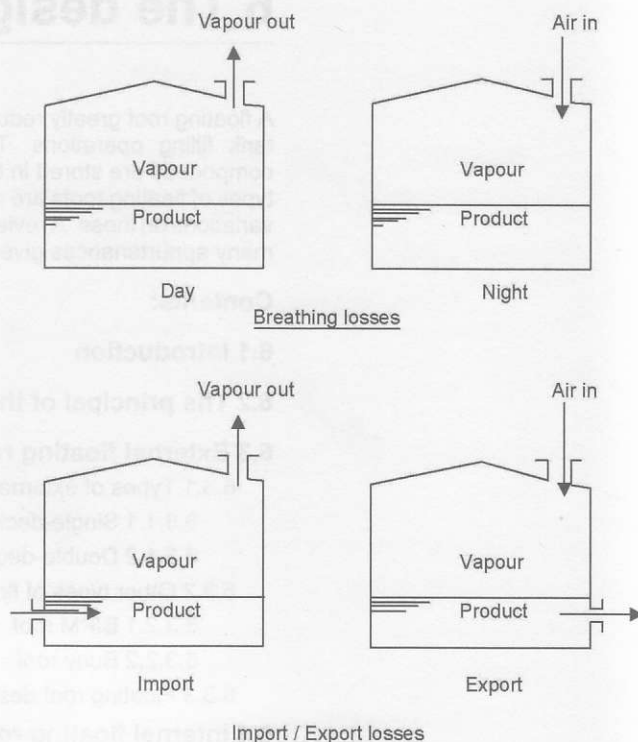


Figure 6.3 The loss mechanisms experienced in fixed roof tanks

6.2 The principal of the floating roof

The floating roof is a circular steel structure which is provided with built-in buoyancy allowing it to float on top of the stored product in a closed or open top tank. Due to the limits of accuracy in constructing large circular structures, the overall diameter of the floating roof is generally about 400 mm smaller than the inside tank diameter thus allowing it to rise and fall on the product without binding on the tank shell, rather like a piston in a cylinder. The gap between the outer rim of the roof and the inside of the tank shell is closed by means of a flexible sealing system, of which there are many types available and these are discussed later in Section 6.5. The seal also serves to centralise the position of the roof in the tank.

There are two types of floating roof:

- The external floating roof**, where the roof sits on the product in an open top tank and the roof is open to the elements.
- The internal floating roof** where the roof floats on the product in a fixed roof tank. The roof and product in this arrangement are protected from the ingress of rain and snow and also from the effect of wind. This type of roof, being protected from the elements, is usually of much lighter construction.

6.3 External floating roofs

The single-deck pontoon type and the double-deck type of roof are the most commonly used type of designs, although there are other variants available.

The design rules laid down in API 650, BS 2654 and the proposed European Code prEN 14015-1 are essentially the same and these are:

- The roof design shall be such that the roof will remain afloat on a product of specific gravity of 0.7 with two adja-

cent pontoon compartments punctured (additionally for the single-deck pontoon type roof only, that the centre deck is also punctured).

- b) The roof design shall be such that the roof will remain afloat on a product of specific gravity of 0.7 carrying a load of 250 mm of rainfall over the entire roof area with the primary roof drain considered inoperative.

6.3.1 Types of external floating roof

6.3.1.1 Single-deck pontoon type

This type of roof, illustrated in Figure 6.4, derives its principal buoyancy from an outer annular pontoon which is divided radially into liquid tight compartments. The centre deck is formed by a membrane of steel plates lap welded together (usually on the top side only) and connected to the inner rim of the pontoons. This centre deck is normally 5 mm or $\frac{3}{16}$ " thick.

This type of roof is used in tanks up to about 65 metres in diameter. Roofs that are larger than this have been known to suffer from wind-excited fatigue which can cause cracking in the welded joints of the centre deck. (Attempts to prevent this by introducing stiffening on the underside of the deck has not always been entirely successful.) Also, because of the flexibility of a large centre deck, the natural rise in the deck when floating can make drainage of rainwater from the deck a problem. Vapour can also become trapped in the space thus formed under the deck, which can promote corrosion in this area.

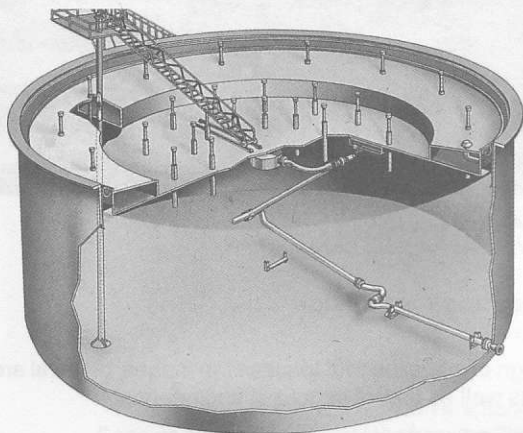


Figure 6.4 Single-deck pontoon type roof
Courtesy of Whessoe

6.3.1.2 Double-deck type

This type of roof, shown in Figure 6.5, consists of an upper and lower steel membrane (usually in 5mm plate) separated by a series of circumferential bulkheads which are subdivided by radial bulkheads. The outer ring of the compartments so formed are the main liquid tight buoyancy for the roof. This type of roof

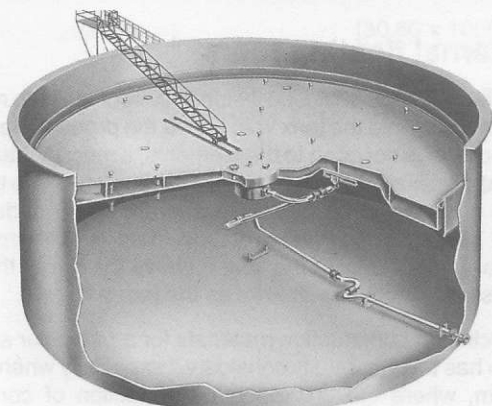


Figure 6.5 Double-deck type roof
Courtesy of Whessoe

is of much heavier construction (and hence more expensive) but this more rigid design allows better drainage from the top of the roof, which usually has a minimum slope of 1:64 and the lower membrane is more likely to stay in contact with the stored product and hence there is less likelihood of static vapour pockets forming under the roof. Also, the air gap between the upper and lower plates has an insulating effect against solar heat reaching the stored product which can be advantageous when storing volatile products in hot climates.

The rigidity of this type of roof mainly (although not completely) overcomes wind-excited cracking problems.

This type of roof is favoured for small tanks under, say 10 metres in diameter, where if the single-deck pontoon type were used, would only leave a very small centre deck area. It is also used for tanks above, say 65 metres in diameter, where the more rigid construction mainly eliminates the drainage, under-deck corrosion and deck cracking problems. The double-deck roof has more buoyancy available compared with the single-deck type which is advantageous in satisfying the design requirement in a) above, especially for large diameter roofs.

Figure 6.6 shows a double-deck floating roof under construction. The bottom deck has been laid, the circumferential and radial bulkheads fitted and the top deck stiffeners are in place ready to receive the top deck plating.



Figure 6.6 A double-deck floating roof under construction
Courtesy of McTay

6.3.2 Other types of floating roof

6.3.2.1 BIPM roof

The BIPM type of roof designed by Shell, the Netherlands, consists of both annular pontoons and radial box girders which offer additional buoyancy for the punctured condition. These box girders also stiffen the centre deck membrane. The design is illustrated in Figure 6.7.

This design was an attempt to produce a floating roof which was stiffer than the single-deck pontoon type without incurring the cost and weight penalties associated with the double-deck roof. The reason for this initiative was in the main associated with the need to produce an economic roof with good resistance to wind induced fatigue problems. In this respect the design was successful. However, other problems bedevilled this design as the radial ribs were prone to buckling in service, which was thought to be related to:

- The initial periphery to centre construction preset.
- Foundation settlement giving uneven support to the roof in the landed condition.
- Changes in the stored product specific gravity.

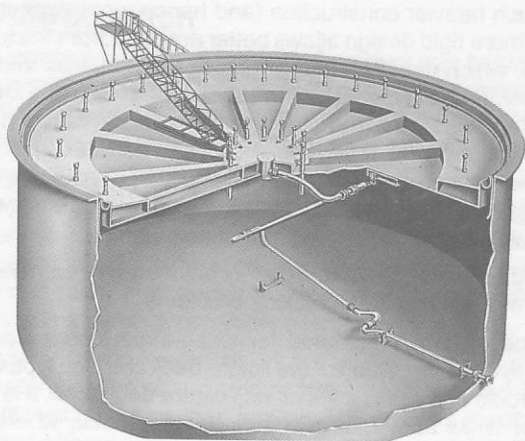


Figure 6.7 A type of roof consisting of both annular pontoons and radial box girders

Courtesy of Whessoe

The resulting buckling of the ribs led to numerous failures in service and the use of this design was discontinued and it is not known if any roofs of this type are still in service.

6.3.2.2 Buoy roof

Of the two mandatory Code design conditions a) and b) given in earlier, it has been found through experience that for the single-deck pontoon roof, the most onerous design condition is when the two adjacent pontoon compartments and the deck are punctured. In this condition the flooded deck plating exerts radial loads on to the pontoons which cause compressive stresses in the pontoon structure. Also, as the tank diameter increases, the weight of the centre deck to be supported increases, and the buoyancy required from the peripheral pontoons increases.

The obvious answer may be to increase the width of the pontoon ring which will increase buoyancy and reduce the size of the centre deck. However it has been established that the relatively thin upper and lower pontoon plates offer little resistance to the induced compressive stresses and they can buckle at relatively low stress levels. The area of the pontoons which offer most resistance is found to be the inner and outer rim plates and a short section of the upper and lower pontoon plating immediately adjacent to the rim plates. The remainder of the upper and lower plates therefore require stiffening by using structural sections, thus increasing the weight and cost of the roof.

The principal problem with the single-deck pontoon roof is the lack of buoyancy in the centre deck and in the early 1970s an American tank constructor produced a roof design which overcame this problem. It was called it the "Buoy roof". This design incorporates a series of liquid-tight buoyancy units arranged in a grid pattern on the top of the centre deck. These units give buoyancy to the centre deck when in the punctured condition. They can be circular, square, rectangular, or of any shape to suit the width of the plates used to form the centre deck. Generally the deck support legs (described later) are housed through the centre of the units, which has the advantage of offering stiffening to the units concerned and vertical stiffness to the legs themselves.

A further advantage of the buoy roof is that the cross-section of the peripheral pontoons is dramatically reduced as it only has to provide enough buoyancy for itself and a short section of the centre deck plating immediately adjacent to it. The overall advantage of this type of roof design is for tanks having diameters larger than, say, 65 metres.

This roof design appeared in the UK at a time when site construction was beset by problems of labour militancy, high costs

and poor quality. The buoy roof allowed an increased level of shop fabrication which was helpful in controlling quality, time and cost. It was usual to arrange for shop-fabricated units consisting of the buoy, the supporting leg and the single-deck immediately surrounding the buoy to be supplied to site where only the closing seams were required to be completed.

This design suffered from problems with wind-excited fatigue cracking, particularly around the buoy units where the stiffness of the buoy and the deck were very different. Also problematic was the draining of rainwater because the majority of the centre deck floated flat and consequently there was no natural slope to the drainage sumps. Rain would accumulate on the roof away from the drains, this then caused low points attracting more rain which formed non-draining ponds on the roof. In some cases drainage channels were fabricated into the roof in an attempt to alleviate the problem but this added more weight to the roof which was undesirable.

A typical buoy roof is shown in Figure 6.8. It is a 96 m diameter roof at the Phillips Seal Sands Facility for crude oil storage.



Figure 6.8 A typical buoy roof

Courtesy of Phillips Petroleum Company

6.3.3 Floating roof design

The design of a floating roof touches the fringes of naval architecture as well as that of structural engineering.

Where the Codes give guidance on designing say, secondary wind girders or shell-to-roof connections, we are left to our own devices with regard to the detail design of floating roofs. Hence, each tank designer has developed his own approach in order to satisfy the requirements of the Code.

One such approach is given for the design of a single-deck roof, and is shown in Figure 6.9, "Design of a single-deck Floating Roof for a Storage Tank designed to API 650 Appendix C and/or BS 2654".

6.4 Internal floating roofs

Internal floating roofs are used inside fixed roof tanks to reduce vapour emission into the tank void above the product. Because this type of roof is not open to the elements, a much lighter form of construction in aluminium or plastic can be used. Also the rim seals do not have to be as robust and are often made from moulded flexible closed cell urethane foam in the form of a wiper seal where the tip of the seal is above the rim as the roof descends and flips below the rim as the roof ascends.

The selection of construction materials for a particular service condition has to be carefully considered especially when using aluminium, where the unexpected introduction of corrosive traces in the product can cause serious damage to the roof components.

Design of a Single deck Floating Roof for a Storage Tank.
 Designed to A.P.I. 650 10th edition, Nov 1998 Appendix 'C' and / or
 B.S. 2654 : 1989 + amd 1997 Clause 9

Tank size : 35.00 m i / dia. x 15.00 m high

Specific gravity of Product = 0.70

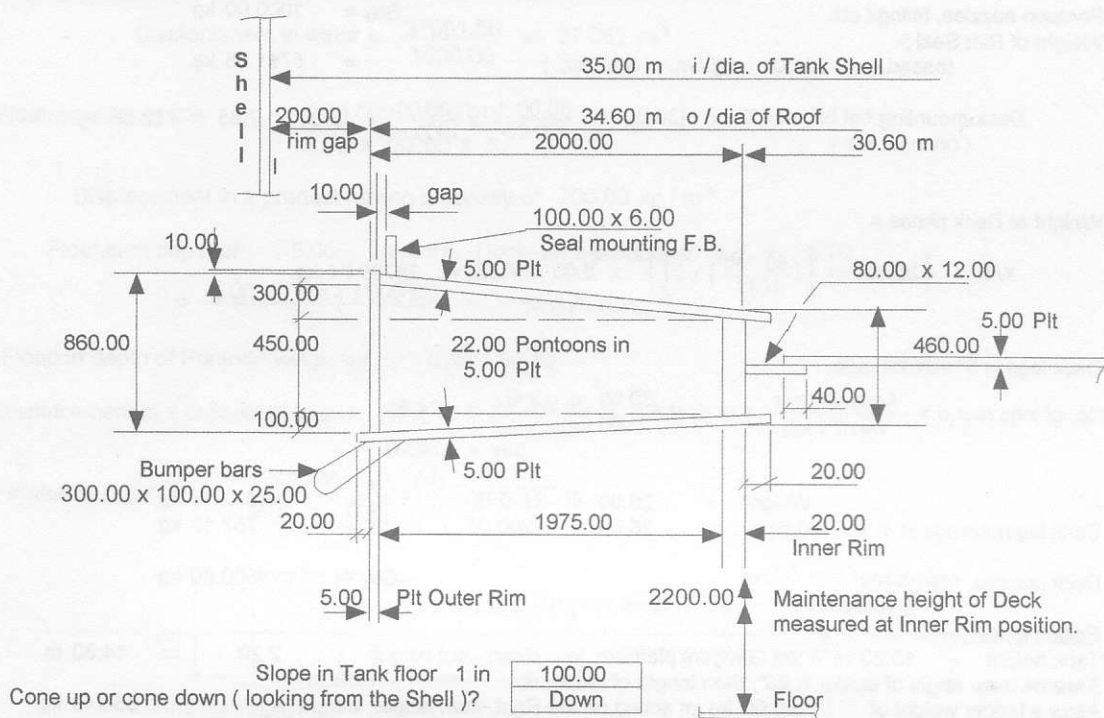
The Code requires the Roof to be designed for a specific gravity of :- 0.70

However, this complete calculation may be repeated if necessary using the actual product s.g. in order to determine actual floatation levels.

Yield stress of steel being used = 275.00 N / mm²

Modulus of Elasticity of the steel = 209000.00 N / mm²

Pontoon geometry. (All dimensions in 'mm' unless otherwise stated.)



Weight of Floating Roof.

$$22.00 \text{ Compartment plates} = \frac{0.85 + 0.45}{2.00} \times 1.98 \times 5.00 \times 7.85 = 1108.52 \text{ kg.}$$

$$\text{Top pontoon plate} = \pi \times 17.50^2 \times 15.46^2 \times 5.00 \times 7.85 = 8282.43 \text{ kg.}$$

$$\text{Btm pontoon plate} = \pi \times 17.34^2 \times 15.30^2 \times 5.00 \times 7.85 = 8221.08 \text{ kg.}$$

$$\text{Inner rim} = \pi \times \frac{[30.60 \times 1000.00] + 20.00}{1000.00} \times 0.45 \times 20.00 \times 7.85 = 6796.22 \text{ kg.}$$

$$\text{Outer rim} = \pi \times \frac{[34600.00 - 5.00]}{1000.00} \times 0.87 \times 5.00 \times 7.85 = 3689.94 \text{ kg.}$$

$$\text{Seal mounting F.B.} = \pi \times 34.56 \times 0.10 \times 6.00 \times 7.85 = 511.44 \text{ kg.}$$

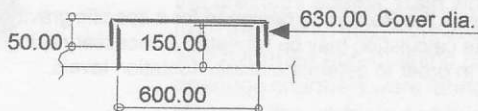
Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 1

$$\text{Bumper bars} = 22.00 \times 0.30 \times 0.10 \times 25.00 \times 7.85 = 129.53 \text{ kg.}$$

$$\text{Pontoon legs in 3" sch. 80 Pipe} = \frac{22.00}{2} \times 3070.00 \times 15.40^{\#} = 520.06 \text{ kg.}$$

$$\text{Pontoon leg housings in 4" sch. 80 pipe} = \frac{22.00}{2} \times 1310.00 \times 22.40^{\#} = 322.78 \text{ kg.}$$

600.00 dia. Pontoon Hatches, in
6.00 mm Plt.



$$\text{Weight} = \text{Neck} = 13.32 + \text{Cover} = 19.34$$

$$\text{Total} = 32.66 \text{ kg.} \times 22.00 = 718.53 \text{ kg.}$$

Pontoon nozzles, fittings etc.

$$\text{Say} = 1000.00 \text{ kg}$$

Weight of Rim Seal :-

$$\text{(based on 53.00 kg./m. of Rim circ.)} = 5761.05 \text{ kg.}$$

$$\text{Deck mounting flat bar} = \pi \times \left(30.60 - \frac{80.00}{1000.00} \right) \times \frac{80.00}{1000.00} \times 12.00 \times 7.85 = 722.56 \text{ kg.}$$

(on inner rim)

Weight of Deck plates =

$$\pi/4 \times \left\{ 30.60 - \left[\left(\frac{80 - 40}{1000} \right) \times 2 \right] \right\}^2 \times 5.00 \times 7.85 = 28789.64 \text{ kg.}$$

Deck legs in 3" sch. 80 pipe.

$$\text{No. of legs reqd.} = \frac{\text{Area of deck}}{30 \text{ m}^2 / \text{leg}} = \frac{30.60^2 \times \pi / 4}{3.00 \text{ } 0.00} = 24.51$$

Say = 26.00

$$\text{Weight} = 26.00 \times 3327.00 \times 15.40^{\#} = 1332.13 \text{ kg.}$$

$$\text{Deck leg housings in 4" sch. 80 pipe} = 26.00 \times 1300.00 \times 22.40^{\#} = 757.12 \text{ kg.}$$

Deck nozzles, fittings etc.

$$\text{Say} = 1500.00 \text{ kg}$$

Rolling ladder.

$$\text{Tank height} = 15.00 \text{ m} + 2 \text{m Gaugers platform, less clean-out height } 2.20 = 14.80 \text{ m}$$

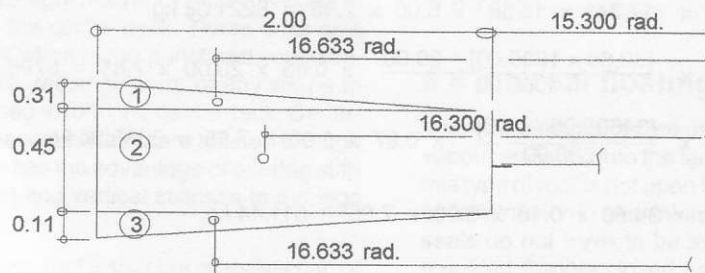
Assume max. angle of ladder is 60°, then length of ladder is :- 17.09 m

$$\text{Allow a ladder weight of } 50.00 \text{ kg /m acting on the Roof then ladder weight is :- } 854.48 \text{ kg.}$$

The worst case eccentricity for the ladder is at 8.76 m. from the Tank centre line.
(to be used for a later calculation.)

Summary of weights :- Pontoon components :- 37061.58
Deck components :- 33955.93
Total weight of Floating Roof 'W' = 71017.51 kg.

Volume of Pontoons.



$$\begin{aligned}
 \text{Volume ①} &= 0.31 \times \frac{2.00}{2.00} \times 33.27 \times \pi = 31.876 \text{ m}^3 \\
 \text{Volume ②} &= 0.45 \times 2.00 \times 32.60 \times \pi = 92.174 \text{ m}^3 \\
 \text{Volume ③} &= 0.11 \times \frac{2.00}{2.00} \times 33.27 \times \pi = 10.974 \text{ m}^3 \\
 &135.024 \text{ m}^3
 \end{aligned}$$

Operational floatation levels.

Floatation depth of 5.00 mm thk. Centre Deck, on water.

$$= \frac{5.00 \times 7.85}{1000.00} \times 1000.00 = 39.25 \text{ mm}$$

Floatation depth of Pontoon weighing 37061.58 kg

$$\text{Displacement in water} = \frac{37061.58}{1000.00} = 37.062 \text{ m}^3$$

$$\text{Floatation depth 'd'} = \frac{(37.062 - 10.974)}{2.00 \times 32.60 \times \pi} = 0.127 \text{ m}$$

Displacement in a product having a density of 700.00 kg / m³

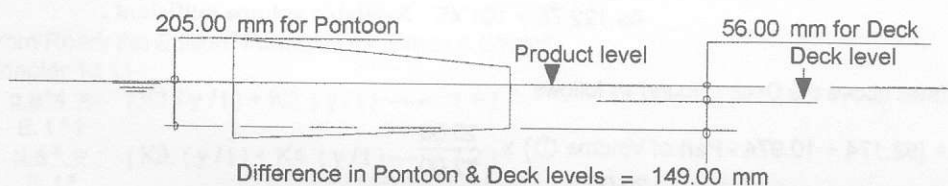
Floatation depth of 5.00 mm thk. Deck, on a product of s.g. = 0.70

$$= \frac{5.00 \times 7.85 \times 1000.00}{700.00} = 56.071 \text{ mm}$$

Floatation depth of Pontoon weighing 37061.58 kg

$$\text{Displacement in a product of s.g.} = 0.70 = \frac{37061.58}{700.00} = 52.945 \text{ m}^3$$

$$\text{Floatation depth 'd'} = \frac{(52.945 - 10.974)}{2.00 \times 32.60 \times \pi} = 0.205 \text{ m}$$



Set Deck at 149.00 mm up from inner corner of Pontoon and the underside of the Deck will still be 'wetted'.

$$\text{Freeboard available above Deck level and the top outer corner of the Pontoon} = 450.00 - 149.00 + 305.00 = 606.00 \text{ mm}$$

The normal operational level for the Roof is :-

Weight of Roof 71017.513 kg

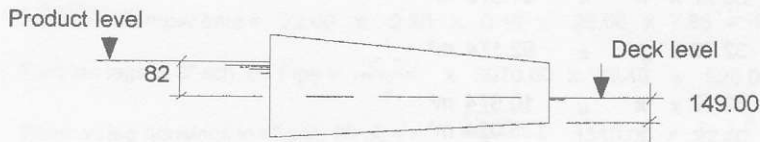
$$\text{This equates to a volume of product of : } - \frac{71017.513}{700.00} = 101.454 \text{ m}^3$$

Then the depth of floatation above the Deck is resolved as follows :-

$$101.454 = 10.974 + 92.174 \times \frac{149.00}{450.00} + (\pi/4 \times 30.60^2 \times \text{depth})$$

$$\text{depth} = \frac{101.454 - 10.974 - 30.520}{735.415} \times 1000 = 81.532 \text{ mm}$$

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 3



Deck to support 250mm (10") of rainwater.

Volume of rainwater collected over the area of the Tank =

$$\pi / 4 \times 35.00^2 \times 0.25 = 244.377 \text{ m}^3$$

Volume to be displaced on a product density of 700.00 kg / m³

$$\frac{71017.51}{700.00} + \frac{244.38}{0.70} = 450.563 \text{ m}^3$$

Assuming the Deck stays level.

With the Deck set at 149.00 mm from lower inner corner of the Pontoon,

then the max. volume available is :- $10.974 + (92.174 \times \frac{0.15}{0.45}) + (0.79 \times 34.60^2 \times 0.61) = 610.995 \text{ m}^3$

As the volume available > than volume required, the calculation is accepted

The Roof must still float with the Centre Deck & two Pontoon compartments punctured.

Volume available with two out of 22.00 compartments punctured = $135.024 \times \frac{20.00}{22.00} = 122.75 \text{ m}^3$

Minimum volume required to meet design requirements = $\frac{71017.51}{700.00} = 101.454 \text{ m}^3$

As $122.75 > 101.45$ Available volume sufficient .

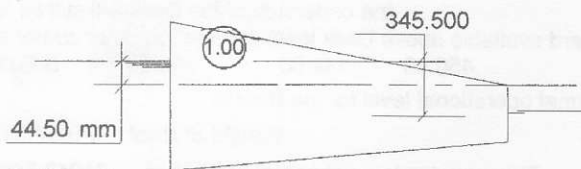
Product liquid level above the Deck is found as follows :-

$$101.454 = (92.174 + 10.974 - \text{Part of Volume } \textcircled{1}) \times \frac{20.00}{22.00}$$

$$101.45 - 93.77 = \text{Pt. vol. } \textcircled{1} \times \frac{20.00}{22.00}$$

$$\text{Pt. vol. } \textcircled{1} = 8.451 \text{ m}^3$$

Product level above base of Section $\textcircled{1}$
is found by iteration using method given
overleaf Enter a value here →



This gives a Part volume for $\textcircled{1}$ of
This is close enough to

8.487 m³
8.451 m³ to be acceptable.

'Freeboard' of Pontoon above the product

level for the punctured condition is $305.00 - 44.50 = 260.50 \text{ mm}$ This is acceptable

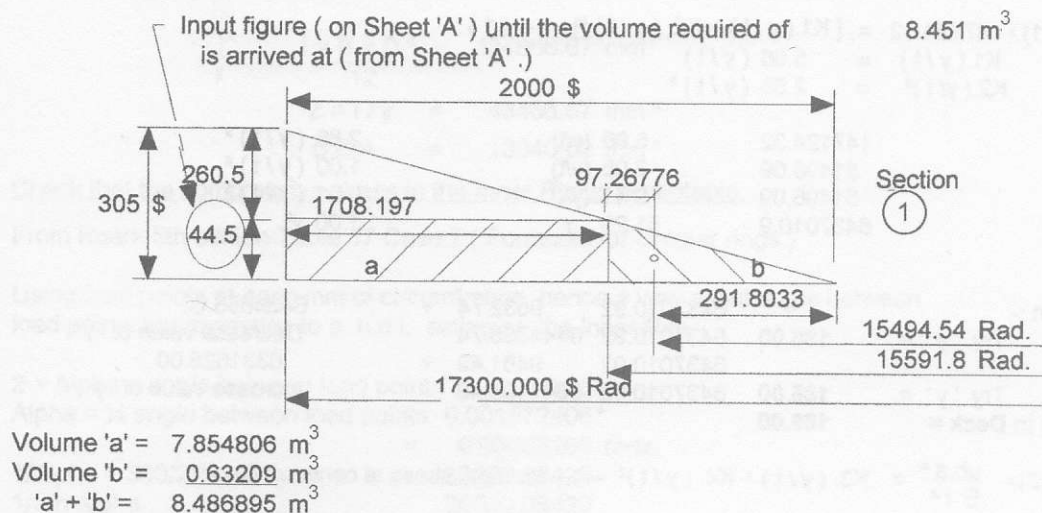
Level of product above the Deck = 345.50 mm

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 4

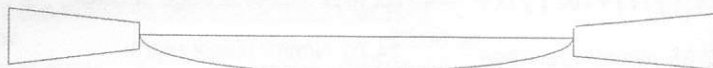
Method to find the level by which a Single deck Floating Roof sinks due to two compartments being punctured.

The loss of buoyancy will cause the product to rise in top section (1) of the Pontoon cross - section and this iteration method determines that level.

'\$' denotes dimensions automatically inputted from the design sheet.



Check the stresses and deflection in the Centre Deck and the adequacy of the Inner Rim with a punctured Centre Deck.



From Roark 5th Edition "Formulas for Stress & Strain"
Chapter 10.11

$$\frac{q \cdot a^4}{E \cdot t^4} = [K1 \cdot (y/t) + K2 \cdot (y/t)] \text{ --- (1)}$$

$$\frac{\mu \cdot a^2}{E \cdot t^2} = [K3 \cdot (y/t) + K4 \cdot (y/t)] \text{ --- (2)}$$

Where q = unit load of Deck. (N/mm²)

$$5.00 \cdot (7.85 - 0.70) \times 9.81 \times 10^6 = 0.000351$$

where:- t = Deck plate thks. (mm)

Thks. of Inner Rim plate (mm)

Width of Deck mounting flat bar (mm)

T = Thks. of Deck mounting flat bar (mm)

a = radius of Tank (mm)

ν = poisson's ratio (0.3)

E = youngs modulus (N/mm²)

plate yield stress (N/mm²)

allowable stress = $2/3 \times$ Yield (N/mm²)

μ_b = bending stress (N/mm²)

μ_d = diaphragm stress (N/mm²)

μ = total stress $\mu_b + \mu_d$

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 5

Condition :-

Fixed & Held. $K1 = \frac{5.33}{(1 - v^2)} = 5.86$ $K2 = \frac{2.6}{(1 - v^2)} = 2.86$

At the centre $K3 = \frac{2}{(1 - v)} = 2.86$ $K4 = 0.98$

At the edge $K3 = \frac{4}{(1 - v^2)} = 4.40$ $K4 = 0.48$

Equation (1) : $147124.32 = [K1.(y/t) + K2.(y/t)^3] \text{ ---- (1)}$

$K1.(y/t) = 5.86 (y/t)$

$K2.(y/t)^3 = 2.86 (y/t)^3$

147124.32	5.86 (y/t)	2.86 (y/t) ³
51496.09	2.05 (y/t)	1.00 (y/t) ³
51496.09	0.41 (y/t)	0.01 (y/t) ³
6437010.9	51.25 y	1.00 y ³

By iteration :-

Try 'y' =	186.00	6437010.92	9532.74 +	6434856.00
		6437010.92	6444388.74	Decrease value of 'y'
		6437010.92	9481.49 +	6331625.00
Try 'y' =	185.00	6437010.92	6341106.49	Increase value of 'y'
Sag in Deck =	186.00			

Equation (2) = $\frac{\mu_b a^2}{E. t^2} = K3. (y/t) + K4. (y/t)^2 \text{ ---- for max. stress at centre of Deck.}$

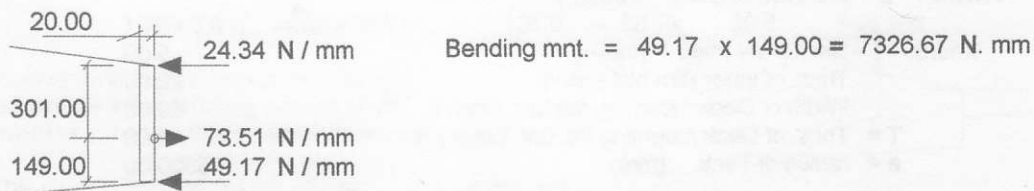
μ_b at centre = 2.37 N/mm^2 (bn'dg.) 30.15 N/mm^2 (Diaphragm)
 = 32.52 N/mm^2 (total stress)
 Acceptable

Equation (2) = $\frac{\mu_b a^2}{E. t^2} = K3. (y/t) + K4. (y/t)^2 \text{ ---- for max. stress at edge of Deck.}$

μ_b at edge = 3.65 N/mm^2 (bending) 14.70 N/mm^2 (Diaphragm)
 = 18.35 N/mm^2 (total stress)

It is the diaphragm stress at the edge which causes the tension at the outer edge of the Deck and hence the stress in the Inner Rim.

Then radial force on Inner Rim = $14.70 \times 5.00 = 73.51 \text{ N / mm. circ.}$



Section modulus = $\frac{B \times D^2}{6} = \frac{1 \times 20.00^2}{6.00} = 66.67 \text{ mm}^3$

Then bending stress in Rim plate = $\frac{7326.67}{66.67} = 109.90 \text{ N / mm}^2$
 Acceptable

Find Section Modulus of the Inner Rim using an area of 16 x thks. as the Section boundaries.

$$I = \frac{B \times D^3}{12} = 434666.67 \text{ mm}^4$$

$$Z = I / y = 43466.67 \text{ mm}^3$$

$$\text{C.S.A.} = 13040.00 \text{ mm}^2$$

Check that the compressive stress in the Inner Rim is acceptable.

From Roark 5th edition Table 17 Case 7 (Formulas for circular rings)

Using load points at each mm of circumference, hence a very small angle between load points approximates to a u.d.l. acting on the Inner Rim.

$$2 \times \text{Alpha} = \text{angle between load point} = 0.00^\circ$$

$$\text{Alpha} = \frac{1}{2} \text{ angle between load points} = 0.001872406^\circ$$

$$= 0.00003268 \text{ rads.}$$

$$1/\text{Alpha} = 360/2\pi \times \text{Alpha} = 30600.08429$$

$$1/\sin \text{Alpha} = 30600.08429$$

$$1/\tan \text{Alpha} = 30600.08428$$

$$\text{Load / mm of Rim circumference} = 73.51 \text{ N/mm}$$

$$\text{No. of load points on the circum'ce} = 96133.00 \text{ (one / mm of circ.)}$$

$$\text{Horiz. load on Inner Rim 'H'} = 0.074 \text{ kN / Load Point}$$

Properties of the effective section of the Inner Rim

$$\text{Rim diameter} = 30.60 \text{ m}$$

$$\text{Radius of Inner Rim 'R'} = 15300.00 \text{ mm}$$

$$\text{C.S.A. of the effective section 'A'} = 13040.00 \text{ mm}^2$$

$$\text{Section modulus } Z = I / y \text{ (in plane of load)}$$

$$= 434666.667 \text{ mm}^3$$

Moment between loads 'H' is :-

$$M_o = H \times R / 2 (1/\sin \text{Alpha} - 1/\text{Alpha}) = 3.063 \text{ Nmm}$$

Compression in Inner Rim is :-

$$N_o = H / 2 (1/\sin \text{Alpha}) = 1124757.498 \text{ N}$$

$$M_o / Z = 0.00000705 \text{ N/mm}^2$$

$$N_o / A = 86.254 \text{ N/mm}^2$$

Total compressive stress in Inner Rim is :-

$$M_o / Z + N_o / A = 86.254 \text{ N/mm}^2$$

$$\text{Allowable stress} = 183.333 \text{ N/mm}^2$$

Is comp. stress < Allowable stress ? Yes accept

Moment at loads 'H' is :-

$$M_i = H \times R / 2 (1/\text{Alpha} - 1/\tan \text{Alpha}) = 6.126 \text{ Nmm}$$

Tension in Inner Rim is :-

$$N_i = H / 2 (1/\tan \text{Alpha}) = 1124757.498 \text{ N}$$

$$M_i / Z = 0.00001409 \text{ N/mm}^2$$

$$N_i / A = 86.254 \text{ N/mm}^2$$

Total tension in Inner Rim is :-

$$M_i / Z + N_i / A = 86.254 \text{ N/mm}^2$$

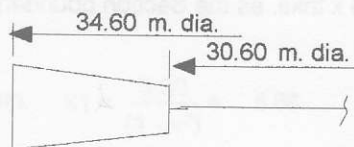
$$\text{Allowable stress} = 183.333 \text{ N/mm}^2$$

Is tensile stress < Allowable stress ? Yes accept

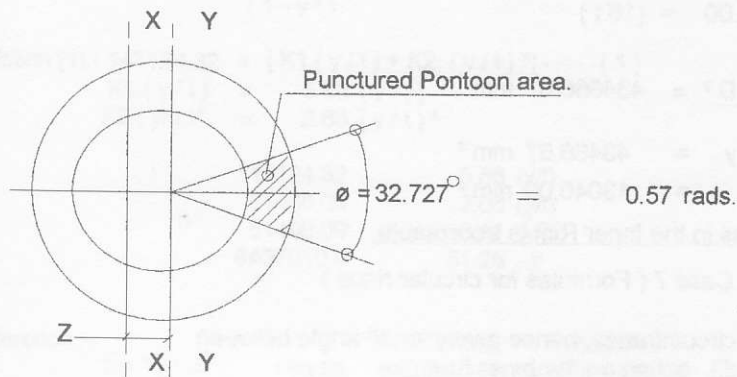
The stresses are accepted

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 7

Consider the effect of two punctured Pontoons and Centre Deck on the stability of the Floating Roof.



$$\text{Area of Pontoon} = \pi / 4 \times (34.60^2 - 30.60^2) = 204.832 \text{ m}^2$$



$$\text{Remaining Pontoon area} = 204.832 \times \frac{360 - 32.727}{360.00} = 186.211 \text{ m}^2$$

$$Z = \frac{2 \sin \theta / 2 (R^3 - r^3)}{3 \times A \text{ rem.}} = \frac{2 \sin 32.727/2 (17.300^3 - 15.300^3)}{3 \times 186.211} = 1.610 \text{ m}$$

Moment of Inertia of remaining Pontoon area :-

$$I_{yy} = \left(\frac{R^4 - r^4}{8.00} \right) [2 \pi - (\theta/180 \times \pi) - \sin \theta]$$

$$= (17.300^4 - 15.300^4) [2 \pi - (32.727/180 \times \pi) - \sin 32.727] = (4347.05) \times (6.283 - 0.571 - 0.541) \\ = 22480.08 \text{ m}^4$$

$$I_{xx} = I_{yy} + (A \text{ rem.} \times Z^2)$$

$$= 22480.08 + (186.211 \times 1.610^2) = 22962.73 \text{ m}^4$$

$$\begin{aligned} \text{Listing moment} &= \text{Weight of Roof 'W'} \times Z \\ &= 71.018 \times 1.610 \\ &= 114.335 \text{ Tonnes} \cdot \text{m} \end{aligned}$$

Compare to actual eccentric loads :-

$$\begin{aligned} \text{Deck} &= 33.101 \times \frac{2.00}{22.00} \times 15.300 = 46.041 \text{ tonnes} \cdot \text{m} \\ \text{Pontoons} &= 37.062 \times \frac{2.00}{22.00} \times 16.300 = 54.919 \text{ tonnes} \cdot \text{m} \\ \text{Ladder} &= 0.854 \times \quad \quad \times 8.755 = 7.481 \text{ tonnes} \cdot \text{m} \\ \text{Total} &= 108.441 \text{ tonnes} \cdot \text{m} \end{aligned}$$

As 108.44 is less than 114.33 The Roof is O.K.

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 8

Additional submersion on punctured side :-

$$d' = \frac{ML (R + Z)}{I_{xx} \times \text{s.g.}} = \frac{114.335 \times (17.300 + 1.610)}{22962.729 \times 0.700} = 0.135 \text{ m}$$

Reduced depth on opposite side :-

$$d'' = \frac{ML (R - Z)}{I_{xx} \times \text{s.g.}} = \frac{114.335 \times (17.300 - 1.610)}{22962.729 \times 0.700} = 0.112 \text{ m}$$

Nominal floatation depth is 345.50 above Deck (from earlier calculation)

$$\text{Max. submersion} = 0.346 + 0.135 = 0.480 \text{ m}$$

As this is < 0.606 (ie. there is "freeboard" of 0.126 **the Roof still floats.**

$$\text{Min. submersion} = 0.346 - 0.112 = 0.234 \text{ m}$$

$$\text{Angle of Roof} = \tan^{-1} \frac{0.135}{17.300} = 0.445^\circ$$

Consider the influence of 10" (254mm) of rainwater on the Deck.

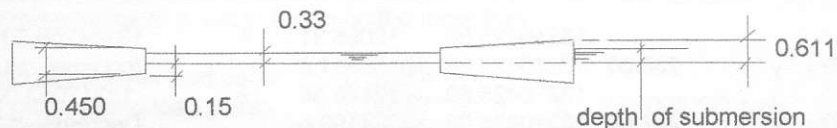
Volume of rainfall (from previous calculation) :- 244.377

Volume of displacement (from previous calc.) :- 450.563

$$\text{Area of total Roof} = \pi/4 \times 34.60^2 = 940.247 \text{ m}^2$$

$$\text{Area of Deck only} = \pi/4 \times 30.600^2 = 735.415 \text{ m}^2$$

$$h' = \text{height of rainwater above deck} \quad 244.38 / 735.42 = 0.33 \text{ m}$$



$$\begin{aligned} \text{Depth of submersion} &= \frac{450.563 - 10.974 - [92.174 \times (0.149/0.450)]}{940.25} \\ &= \frac{450.563 - 10.974 - 30.520}{940.247} = 0.435 \text{ m} \end{aligned}$$

$$\text{Load due to steel Deck \& rainwater} = 33955.932 + 244376.639 = 278332.571$$

$$\text{Upward force of product on u/s Deck} \quad 735.415 \times 0.435 \times 700.00 = 223967.93$$

$$\begin{aligned} \text{Nett downward force} &= 278332.57 - 223967.934 = \frac{54364.64 \text{ kg}}{735.42 \text{ m}^2} = 73.92 \text{ kg/m}^2 \\ &= 725.19 \text{ N/m}^2 \end{aligned}$$

The Centre Deck deflects downwards due to the additional weight of water on the Deck.
This deflection is found from Roark 5th Edition "Formulas for Stress & Strain" Chapter 10.11 (page 406)

$$\frac{q \cdot a^4}{E \cdot t^4} = [K1 \cdot (y/t) + K2 \cdot (y/t)] \text{ --- (1)}$$

$$\frac{\mu \cdot a^2}{E \cdot t^2} = [K3 \cdot (y/t) + K4 \cdot (y/t)] \text{ --- (2)}$$

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 9

μ_b = bending stress (N/mm²)
 μ_d = diaphragm stress (N/mm²)
 μ = total stress $\mu_b + \mu_d$

304223.09	5.86 (y/t)	2.86 (y/t) ³
106483.41	2.05 (y/t)	1.00 (y/t) ³
106483.41	0.41 (y/t)	0.01 (y/t) ³
13310426	51.25 y	1.00 y ³

μ_b at centre = 3.02 N/mm² (b'ndg.) 48.95 N/mm² (Diaphragm)
 = 51.97 N/mm² (total stress)

$$\text{Bending mnt.} = 49.17 \times 149.00 = 7326.67 \text{ N. mm}$$

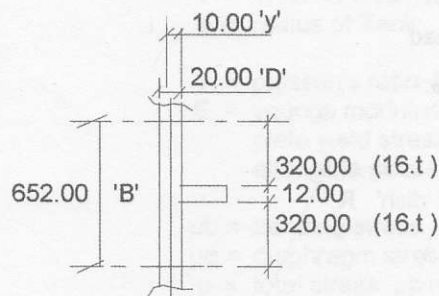
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$$\text{Section modulus} = \frac{B \times D^2}{6} = \frac{1 \times 20.00^2}{6.00} = 66.67 \text{ mm}^3$$

$$\text{Then bending stress in Rim plate} = \frac{7326.67}{66.67} = 109.90 \text{ N/mm}^2$$

Acceptable

Find Section Modulus of the Inner Rim using an area of 16 x thks. as the Section boundaries.



$$I = \frac{B \times D^3}{12} = 434666.67 \text{ mm}^4$$

$$Z = I / y = 434666.67 \text{ mm}^3$$

$$\text{C.S.A.} = 13040.00 \text{ mm}^2$$

Check that the compressive stress in the Inner Rim is acceptable.

From Roark 5th edition Table 17 Case 7 (Formulas for circular rings)

Using load points at each mm of circumference, hence a very small angle between load points approximates to a u.d.l. acting on the Inner Rim.

2 × Alpha = angle between load point:	0.00374°	
Alpha = ½ angle between load points	0.00187°	= 0.00003268 rads.
1/Alpha = 360/2Pi × Alpha	30600.08429	
1/Sin Alpha	30600.08429	
1/Tan Alpha	30600.08428	
Load / mm of Rim circumference	119.35 N/mm	
No. of load points on the circumference	96133.00 (one / mm of circ.)	
Horiz. load on Inner Rim 'H'	0.119 kN / Load Point	

Properties of the effective section of the Inner Rim

Rim diameter	30.60 m
Radius of Inner Rim 'R'	15300.00 mm
C.S.A. of the effective section 'A'	13040.00 mm ²
Section modulus Z = I / y (in plane	434666.667 mm ³

Moment between loads 'H' is :-

$$M_o = H \times R / 2 (1/\sin \text{Alpha} - 1/\text{Alpha}) = 4.973 \text{ Nmm}$$

Compression in Inner Rim is :-

$$\text{No} = H / 2 (1/\sin \text{Alpha}) = 1826121.630 \text{ N}$$

$$M_o / Z = 0.00001144 \text{ N/mm}^2$$

$$\text{No} / A = 140.040 \text{ N/mm}^2$$

Total compressive stress in Inner Rim is :-

$$M_o / Z + \text{No} / A = 140.040 \text{ N/mm}^2$$

$$\text{Allowable stress} = 183.333 \text{ N/mm}^2$$

$$\text{Is comp. stress} < \text{Allowable stress?} \quad \text{Yes accept}$$

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 11

Moment at loads 'H' is :-

$$M_i = H \times R / 2 (1/\text{Alpha} - 1/\tan \text{Alpha}) \quad 9.946 \text{ Nmm}$$

Tension in Inner Rim is :-

$$N_i = H / 2 (1/\tan \text{Alpha}) \quad 1826121.629 \text{ N}$$

$$M_i / Z = \quad 0.00002288 \text{ N/mm}^2$$

$$N_i / A = \quad 140.040 \text{ N/mm}^2$$

Total tension in Inner Rim is :-

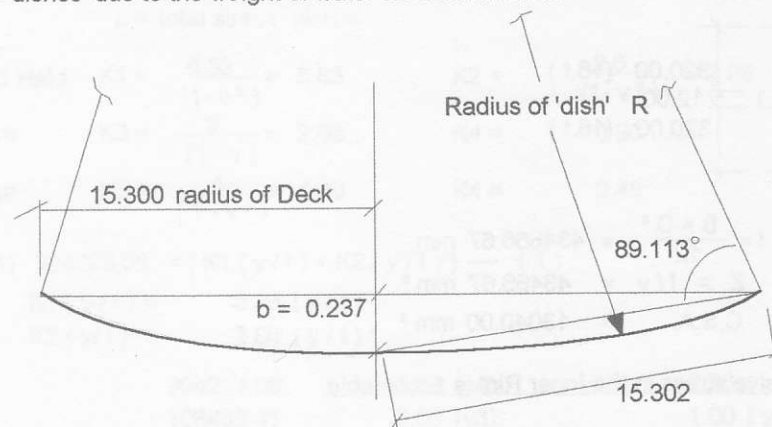
$$M_i / Z + N_i / A = \quad 140.040 \text{ N/mm}^2$$

$$\text{Allowable stress} = \quad 183.333 \text{ N/mm}^2$$

Is tensile stress < Allowable stress ? Yes accept

The stresses are accepted

The Deck 'dishes' due to the weight of water as shown below:-



Solving the above geometry the radius of the 'dished' Deck is 493.979 m

$$\text{Vol. of dished Deck} = \pi / 3 \times b^2 (3R - b) = 87.15 \text{ m}^3$$

$$\text{Depth 'h'} = \frac{244.377 - 87.154}{735.415} = 0.214 \text{ m}$$

$$\text{To find revised submersion depth 'd'} = \frac{450.56 - 10.97 - 30.520 - 87.15}{940.247} = 0.342 \text{ m}$$

Find nett load acting on the Deck.

$$\text{Weight of steel Deck} = 33955.93 \text{ kg}$$

$$\text{Weight of rain water} = \frac{244376.64}{278332.57} \text{ kg}$$

$$\text{Total upward force on Deck.} = [87.154 + (735.415 \times 0.342)] \times 700.00 = 237258.37 \text{ kg}$$

$$\text{Nett downward force} = 278332.57 - 237258.371 = 41074.200 \text{ kg}$$

$$\text{This represents a pressure of} \quad \frac{41074.200 \times 9.810}{735.415} = 547.91 \text{ N/m}^2$$

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 12

Check again to ensure that the stresses in the Inner Rim are acceptable in this revised condition.

$$\frac{q \cdot a^4}{E \cdot t^4} = [K1 \cdot (y/t) + K2 \cdot (y/t)^3] \text{--- (1)}$$

$$\frac{\mu \cdot a^2}{E \cdot t^2} = [K3 \cdot (y/t) + K4 \cdot (y/t)^2] \text{--- (2)}$$

Where q = unit load of Deck. (N/mm²)

$$547.905 \times 10^{-6} =$$

0.00

where:- t = Deck plate thks. (mm)

5.00

Thks. of Inner Rim plate (mm)

20.00

Width of Deck mounting flat bar (mm)

80.00

T = Thks. of Deck mounting flat bar (mm)

12.00

a = radius of Tank (mm)

15300.00

ν = poisson's ratio (0.3)

0.30

E = youngs modulus (N/mm²)

209000.00

plate yield stress (N/mm²)

275.00

allowable stress = $2/3 \times \text{Yield}$ (N/mm²)

183.333

μ_b = bending stress (N/mm²)

μ_d = diaphragm stress (N/mm²)

μ = total stress $\mu_b + \mu_d$

Condition :-

Fixed & Held. $K1 = \frac{5.33}{(1-\nu^2)} = 5.86$ $K2 = \frac{2.6}{(1-\nu^2)} = 2.86$

At the centre $K3 = \frac{2}{(1-\nu)} = 2.86$ $K4 = 0.98$

At the edge $K3 = \frac{4}{(1-\nu^2)} = 4.40$ $K4 = 0.48$

$$\text{Equation (1)} = 229850.16 = [K1 \cdot (y/t) + K2 \cdot (y/t)^3] \text{--- (1)}$$

$$K1 \cdot (y/t) = 5.86 (y/t)$$

$$K2 \cdot (y/t)^3 = 2.86 (y/t)^3$$

229850.16	5.86 (y/t)	2.86 (y/t) ³
80451.58	2.05 (y/t)	1.00 (y/t) ³
80451.58	0.41 (y/t)	0.01 (y/t) ³
10056447	51.25 y	1.00 y ³

By iteration :-

	10056447.11	11019.03	+	9938375.00
Try 'y' =	215.00	10056447.11	9949394.03	Increase value of 'y'
	10056447.11	11070.28	+	10077696.00
Try 'y' =	216.00	10056447.11	10088766.3	Decrease value of 'y'

Sag in Deck = 216.00

$$\text{Equation (2)} = \frac{\mu_b \cdot a^2}{E \cdot t^2} = K3 \cdot (y/t) + K4 \cdot (y/t)^2 \text{--- for max. stress at edge of Deck.}$$

$$\begin{aligned} \mu_b \text{ at edge} &= 4.24 \text{ N/mm}^2 \text{ (bendin)} & 19.83 \text{ N/mm}^2 \text{ (Diaphragm)} \\ &= 24.07 \text{ N/mm}^2 \text{ (total stress)} \end{aligned}$$

$$\text{Equation (2)} = \frac{\mu_b \cdot a^2}{E \cdot t^2} = K3 \cdot (y/t) + K4 \cdot (y/t)^2 \text{--- for max. stress at centre of Deck.}$$

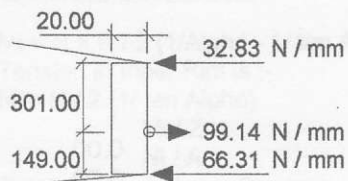
$$\begin{aligned} \mu_b \text{ at centre} &= 2.75 \text{ N/mm}^2 \text{ (bn'dg.)} & 40.66 \text{ N/mm}^2 \text{ (Diaphragm)} \\ &= 43.41 \text{ N/mm}^2 \text{ (total stress)} \end{aligned}$$

Acceptable

It is the diaphragm stress at the edge which causes the tension at the outer edge of the Deck and hence the stress in the Inner Rim.

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 13

Then radial force on Inner Rim = $19.83 \times 5.00 = 99.14 \text{ N/mm. circ.}$



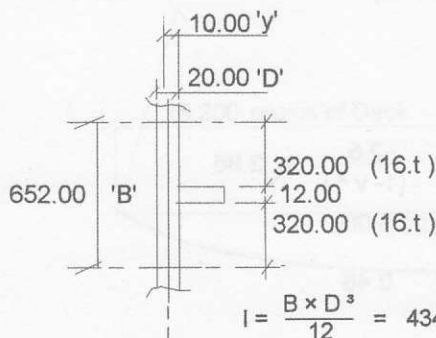
Bending mnt. = $49.17 \times 149.00 = 7326.67 \text{ N. mm}$

$$\text{Section modulus} = \frac{B \times D^2}{6} = \frac{1 \times 20.00^2}{6.00} = 66.67 \text{ mm}^3$$

Then bending stress in Rim plate = $\frac{7326.67}{66.67} = 109.90 \text{ N/mm}^2$

Acceptable

Find Section Modulus of the Inner Rim using an area of 16 x thks. as the Section boundaries.



$$I = \frac{B \times D^3}{12} = 434666.67 \text{ mm}^4$$

$$Z = I / y = 43466.67 \text{ mm}^3$$

$$\text{C.S.A.} = 13040.00 \text{ mm}^2$$

Check that the compressive stress in the Inner Rim is acceptable.

From Roark 5th edition Table 17 Case 7 (Formulas for circular rings)

Using load points at each mm of circumference, hence a very small angle between load points approximates to a u.d.l. acting on the Inner Rim.

$2 \times \text{Alpha} = \text{angle between load point: } 0.00374481^\circ$
 $\text{Alpha} = \frac{1}{2} \text{ angle between load points } 0.001872406^\circ = 0.00003268 \text{ rads.}$

$1/\text{Alpha} = 360/2\pi \times \text{Alpha} \quad 30600.08429$

$1/\sin \text{Alpha} \quad 30600.08429$

$1/\tan \text{Alpha} \quad 30600.08428$

Load / mm of Rim circumference 99.14 N/mm

No. of load points on the circumference $96133.00 \text{ (one / mm of circ.)}$

Horiz. load on Inner Rim 'H' $0.099 \text{ kN / Load Point}$

Properties of the effective section of the Inner Rim

Rim diameter 30.60 m

Radius of Inner Rim 'R' 15300.00 mm

C.S.A. of the effective section 'A' 13040.00 mm^2

Section modulus $Z = I / y \text{ (in plane)} \quad 434666.667 \text{ mm}^3$

Moment between loads 'H' is :-

$M_o = H \times R / 2 (1/\sin \text{Alpha} - 1/\text{Alpha}) \quad 4.131 \text{ Nmm}$

Compression in Inner Rim is :-

$N_o = H / 2 (1/\sin \text{Alpha}) \quad 1516842.578 \text{ N}$

$M_o / Z = \quad 0.00000950 \text{ N/mm}^2$

$N_o / A = \quad 116.322 \text{ N/mm}^2$

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 14

Total compressive stress in Inner Rim is :-

$$\begin{aligned} M_o / Z + N_o / A &= 116.322 \text{ N/mm}^2 \\ \text{Allowable stress} &= 183.333 \text{ N/mm}^2 \\ \text{Is comp. stress} < \text{Allowable stress?} &\quad \text{Yes accept} \end{aligned}$$

Moment at loads 'H' is :-

$$M_i = H \times R / 2 (1/\alpha - 1/\tan \alpha) = 8.262 \text{ Nmm}$$

Tension in Inner Rim is :-

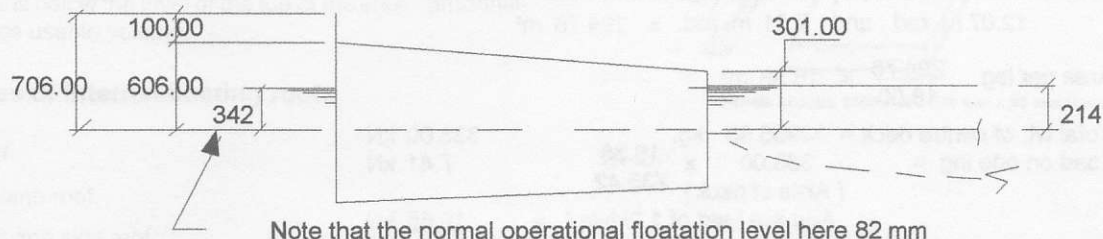
$$\begin{aligned} N_i &= H / 2 (1/\tan \alpha) = 1516842.578 \text{ N} \\ M_i / Z &= 0.00001901 \text{ N/mm}^2 \\ N_i / A &= 116.322 \text{ N/mm}^2 \end{aligned}$$

Total tension in Inner Rim is :-

$$\begin{aligned} M_i / Z + N_i / A &= 116.322 \text{ N/mm}^2 \\ \text{Allowable stress} &= 183.333 \text{ N/mm}^2 \\ \text{Is tensile stress} < \text{Allowable stress?} &\quad \text{Yes accept} \end{aligned}$$

The stresses are accepted

Resulting state of floatation.



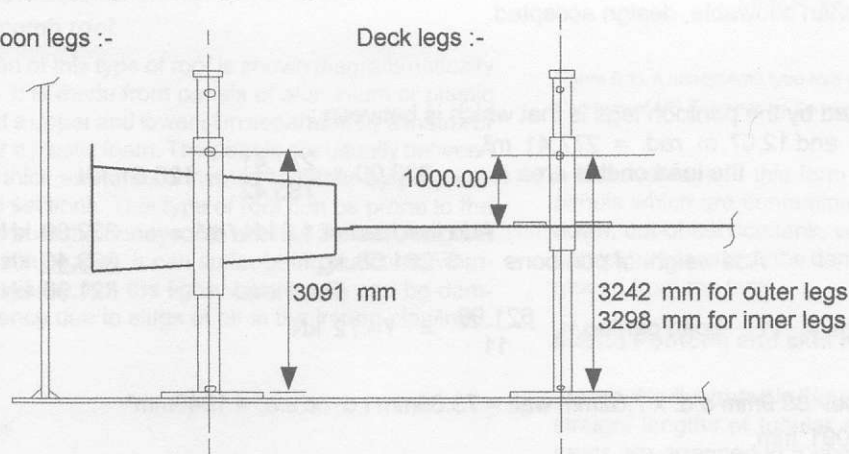
Design of the supporting legs.

Note that the legs are to be designed to carry only the weight of the roof and not the weight of any accumulated rain water on the deck. To this end it is important to ensure that when the tank is out of service, the drain bungs must be removed from the deck to allow any rain water to drain to the tank floor.

There are two types of support legs.

Pontoon legs :-

Deck legs :-



8 Inner deck legs are on a
18 Outer deck legs are on a
11 Pontoon legs are on a

4.42 m. radius.
10.00 m. radius.
16.45 m. radius.

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 15

Inner deck legs.

Area of deck supported by the inner legs is 7.21 m. rad. = 163.25 m²

$$\text{Area per leg} = \frac{163.25}{8.00} = 20.41 \text{ m}^2$$

$$\text{Total wt. of centre deck} = 33955.93 \text{ kg.} \quad 333.00 \text{ kN}$$

$$\text{Load on one leg} = \frac{333.00}{735.42} \times 20.41 = 9.24 \text{ kN}$$

$$\text{Add live load of } 1.2 \text{ kN/m}^2 = 24.49 \text{ kN}$$

$$\text{Load on one leg} = 33.73 \text{ kN}$$

Use 3" nb. sch 80 pipe. 88.9mm o.d. x 7.62mm wall = 73.66mm i.d. cc.s.a. = 1948mm²

Length of leg 3298 mm

$$f_c = L / A = \frac{33727.38}{1948} = 17.31 \text{ N / mm}^2$$

$$L / r = \frac{3298}{28.96} = 113.88 \text{ From BS 449 Table 17a Allowable stress} = 66.00 \text{ N / mm}^2$$

Actual stress is less than allowable, design accepted.

Outer deck legs.

Area of deck supported by the outer legs is that which is between :-

$$12.07 \text{ m. rad. and } 7.21 \text{ m. rad.} = 294.76 \text{ m}^2$$

$$\text{Area per leg} = \frac{294.76}{18.00} = 16.38 \text{ m}^2$$

$$\text{Total wt. of centre deck} = 33955.93 \text{ kg.} \quad 333.00 \text{ kN}$$

$$\text{Load on one leg} = \frac{333.00}{735.42} \times 16.38 = 7.41 \text{ kN}$$

$$\text{Add live load of } 1.2 \text{ kN/m}^2 = 19.65 \text{ kN}$$

$$\text{Load on one leg} = 27.07 \text{ kN}$$

Use 3" nb. sch 80 pipe. 88.9mm o.d. x 7.62mm wall = 73.66mm i.d. cc.s.a. = 1948mm²

Length of leg 3242 mm

$$f_c = L / A = \frac{27065.37}{1948} = 13.89 \text{ N / mm}^2$$

$$L / r = \frac{3242}{28.96} = 111.95 \text{ From BS 449 Table 17a Allowable stress} = 66.00 \text{ N / mm}^2$$

Actual stress is less than allowable, design accepted.

Pontoon legs.

Area of deck supported by the pontoon legs is that which is between :-

$$15.30 \text{ m. rad. and } 12.07 \text{ m. rad.} = 277.41 \text{ m}^2,$$

$$\text{the load on this area is } 333.00 \times \frac{277.41}{735.42} = 125.61 \text{ kN}$$

$$\text{Add live load of } 1.2 \text{ kN / m}^2 = 332.89 \text{ kN}$$

$$\text{Add weight of pontoons } 37061.58 \text{ kg} \quad 363.46 \text{ kN}$$

$$\text{Total load} = 821.96 \text{ kN}$$

$$\text{No. of pontoon legs} = 11 \quad \text{Load per leg} = \frac{821.96}{11} = 74.72 \text{ kN}$$

Use 3" nb. sch 80 pipe. 88.9mm o.d. x 7.62mm wall = 73.66mm i.d. cc.s.a. = 1948mm²

Length of leg 3091 mm

$$f_c = L / A = \frac{74723.43}{1948} = 38.36 \text{ N / mm}^2$$

$$L / r = \frac{3091}{28.96} = 106.73 \text{ From BS 449 Table 17a Allowable stress} = 72.00 \text{ N / mm}^2$$

Actual stress is less than allowable, design accepted.

Figure 6.9 Design of a single-deck floating roof for a storage tank designed to API 650 Appendix C and/or BS 2654 - page 16

Internal roofs either float directly on the product, and therefore there is no vapour space, or, the sealing membrane is carried above the product on pontoons and so there is a confined vapour space. The likelihood of an explosion or fire in this space is improbable as the saturated vapour will be too rich to support combustion.

An important issue, which is relevant to the use of internal floating roofs, is that the free space above the roof must be adequately vented to prevent an accumulation of a potentially explosive weak vapour and air mixture, and this is usually achieved by fitting large purpose made vent cowls around the periphery of the tank roof, together with a vent at the crown of the roof. These vents encourage the scouring of this space by wind action.

The usage of capacity of the tank is governed by the limit of travel of the roof within the tank. The lowest level is determined by the roof not fouling any floor piping or shell fittings which protrude into the tank. Also for maintenance purposes, personnel will require access to the underside of the roof via the shell manhole.

The upper limit is governed by the type of roof structure and/or the depth of the shell brackets supporting the roof structure. Large diameter tanks which have a truss type roof structure which extends below the level of the top of the shell can significantly reduce usable volume.

6.4.1 Types of internal floating roofs

- Pan roof
- Honeycomb roof
- Pontoon and skin roof

6.4.1.1 Pan roof

The pan roof, shown diagrammatically in Figure 6.10, consists of a circular membrane with a vertical outer rim plate on to which the rim gap seal is mounted. This type of roof is prone to sinking because it does not have any closed buoyancy compartments. Leakage on to the roof can cause it to capsize and sink. Hence, whilst cheap to construct, the operational disadvantage of this type of roof means that it is rarely, if ever used.

6.4.1.2 Honeycomb roof

The construction of this type of roof is shown diagrammatically in Figure 6.11. It is made from panels of aluminium or plastic which consist of an upper and lower skin separated by a matrix of internal cells, or a plastic foam. The panels are usually between 25 and 80mm thick and are connected together by purpose-made extruded sections. This type of roof can be prone to the skin separating from the honeycomb but has the advantage of natural inherent buoyancy. It can suffer being punctured without losing buoyancy, but the light construction can be damaged by turbulence due to slugs of air in the import pipeline.

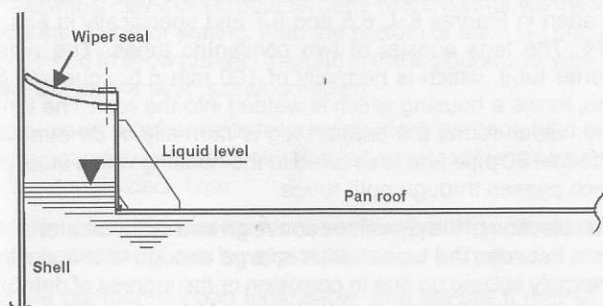
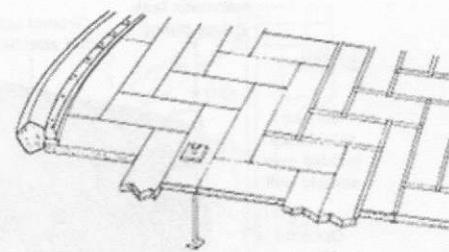
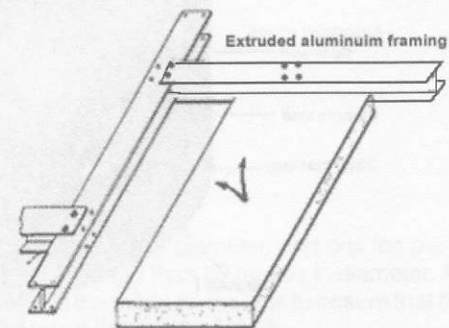


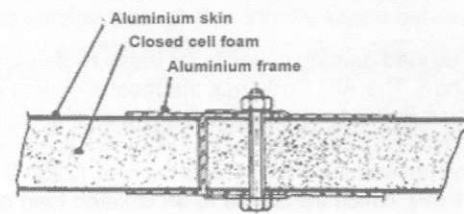
Figure 6.10 A pan roof shown diagrammatically



Panel arrangement



Panels approx. 2500 mm x 500 mm x 50 mm thick



Cross section of panel and framing

Figure 6.11 A honeycomb type roof construction
Courtesy of MB Engineering Services Ltd

A disadvantage in this form of construction is that punctured panels which are contaminated with product make a drained down, out-of-service tank, very difficult to gas free for maintenance purposes until the damaged panels are identified and removed from the tank.

6.4.1.3 Pontoon and skin roof

This roof is illustrated in Figure 6.12 and consists of a number of straight lengths of tubular aluminium pontoons. These pontoons are arranged in a ring around the periphery of the roof with parallel rows of pontoons connecting from one side of the ring to the other. The rows of pontoons are connected together by purpose-made aluminium extruded sections set at right angles to the lines of pontoons the ends being joined to the outer pontoon ring.

Attached to the matrix formed by these sections is a thin aluminium skin which forms the vapour barrier. The skin sits above the product by about 150 to 200 mm and the gap is sealed at the periphery of the roof by a vertical rim plate, the lower end of which is immersed in the product. The peripheral rim gap is sealed with a preformed flexible wiper seal.

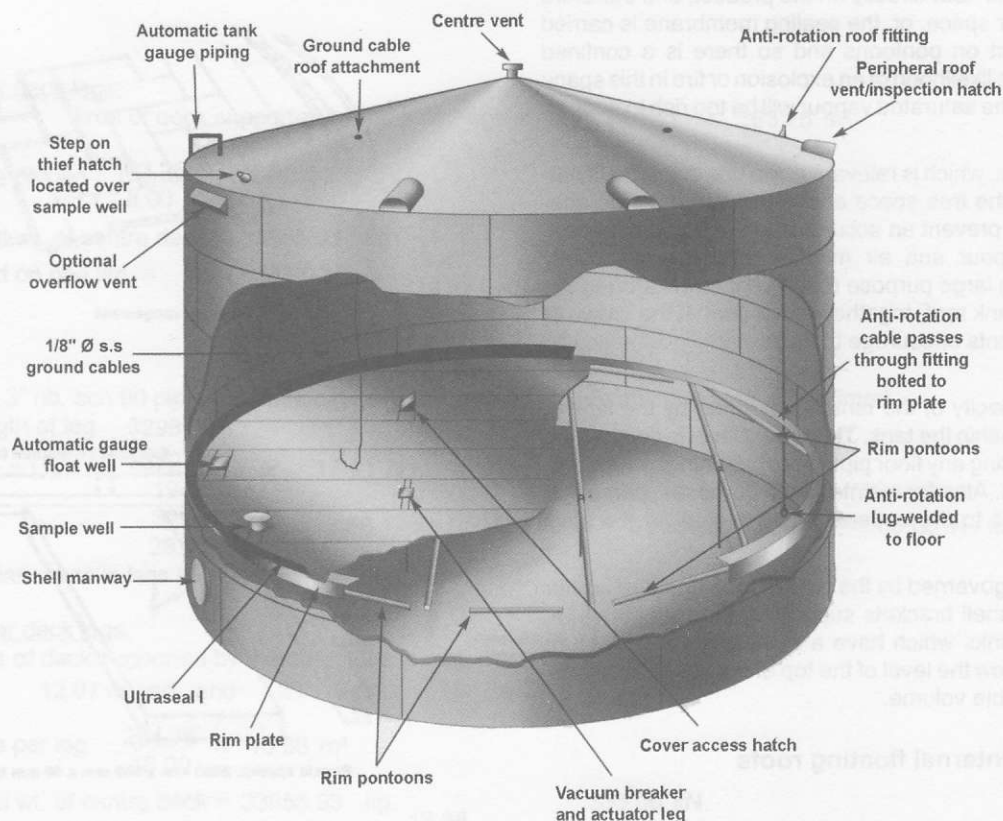


Figure 6.12 The pontoon and skin roof - showing the normal appurtenances for an internal floating roof
Courtesy of Ultraflo Corporation

The required load bearing capacity for these roofs varies from Code to Code. The API Code has the most stringent requirement, which requires the roof when floating or when supported on its legs, to be able to safely carry a load which is equivalent to at least two men walking anywhere on the roof, (2200 N over an area of 0.1 m^2), which translates to an isolated load of 22 kN over 1 m^2 .

Similarly the BS and European Codes require that at least three men should be supported over an area of 3 m^2 which is an equivalent isolated load of only 1 kN over 1 m^2 .

The appurtenances provided on these type of roofs are also shown in Figure 6.12.

Ensuring electrical continuity between the deck and the tank is very important in order to allow any charges of static electricity which are transmitted to the deck from the product to be released safely. All conductive surfaces of the roof must be electrically connected and bonded to the shell either by electrical shunts in the seal (a minimum of four to API) or in the case of the BS or European Codes by multi-stranded flexible cables attached to the top surface of the deck and the tank roof or shell. Two cables are required on tanks up to 20m diameter, and four for larger sizes. The European Code recommends that the minimum cross sectional area of each stranded cable should be 80 mm^2 . Care must be taken to ensure that the cables do not snag on any of the roof fittings during the operation of the roof and it may be that spring loaded cable reels can be used to keep the cables tensioned at all times.

The full Code design requirements can be found in the following publications:

BS 2654 Appendix E

API 650 Appendix H

prEN 14015 -1 2000 Annex C

These internal decks are usually proprietary designs and so all design work for them is completed by the specific manufacturer. They are usually designed so that all the component parts can be passed through a 24" (610mm) diameter manhole. This allows them to be retro fitted to existing tanks.

6.5 External floating roof appurtenances

The diagram shown in Figure 6.13 shows the principle appurtenances which are required for the operation of a external floating roof. The diagram depicts a single-deck roof but the principles are basically the same for all roofs.

6.5.1 Roof support legs

When the tank is empty, the floating roof needs to be supported at some distance above the tank floor. This is necessary so that the roof does not foul any heating coils, drain lines, shell-mounted propeller mixers etc. Also access will be required via the shell manholes for the maintenance personnel.

The roof is therefore provided with support legs and these can be seen in Figures 6.4, 6.5 and 6.7 and specifically in Figure 6.14. The legs consist of two concentric tubes. The outer, shorter tube, which is normally of 100 mm n.b. schedule 80 pipe, forms a housing which is welded into the roof. The inner tube, which forms the support leg is normally of 80 mm n.b. schedule 80 pipe and is secured to the housing with a steel pin which passes through both tubes.

The selection of the pipe sizes above gives a radial clearance of 4 mm between the tubes which is large enough to prevent the assembly seizing up due to corrosion or the ingress of detritus. The legs normally have two pin location holes, one giving a leg length for operational conditions and the other allowing a longer leg which is used when the tank is coming out of service. This additional length increases headroom under the roof for

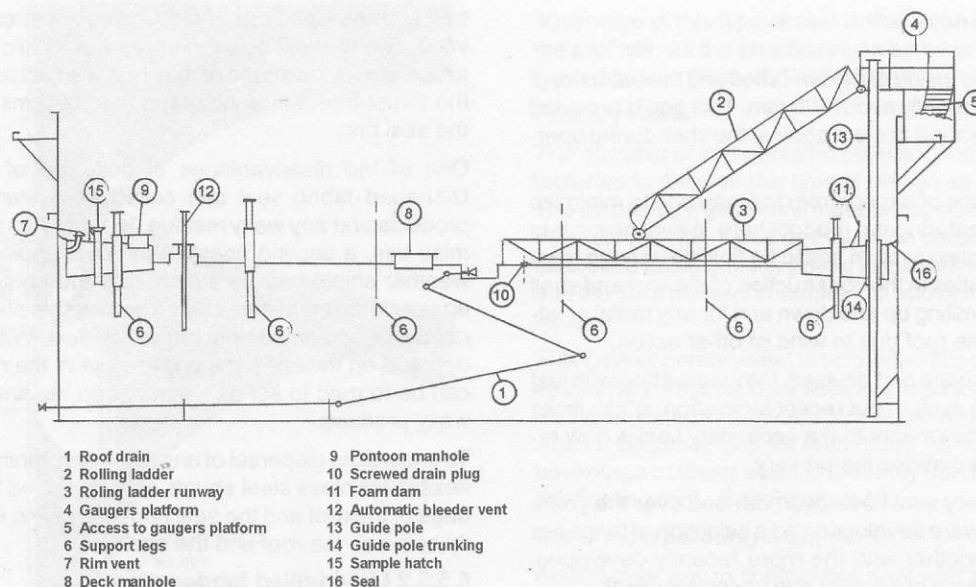


Figure 6.13 Principal floating roof appurtenances

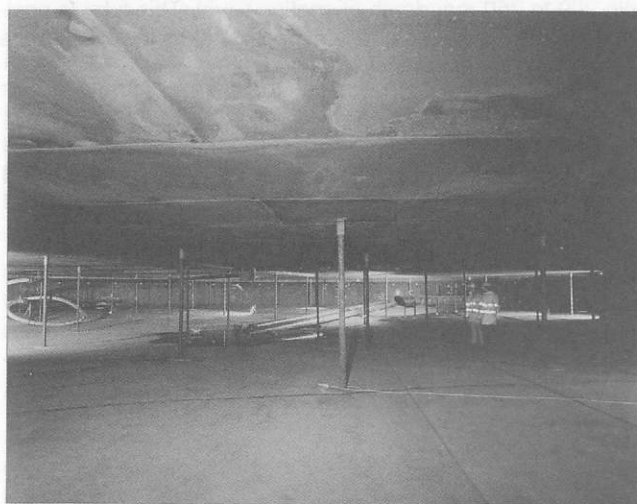


Figure 6.14 The underside of a floating roof showing the support legs and internal pipework
 Courtesy of McTay

maintenance personnel. The adjustment of the leg pin position is made manually, while the roof is floating, and hence it is recommended that the leg size is limited to 80 mm n.b. as a larger size would be too heavy to handle.

Where the leg housings are welded into single-decks which are lap-welded on the top side only, it is recommended to stitch-weld the underside laps to give added strength in the area of the housing connection.

The area of the floor on which the legs land is normally reinforced with a fully welded doubler plate which distributes the leg loads into the floor plating. Also the bottom of each leg should be notched to allow product trapped in the leg during service, to drain out as the tank is drained down.

The support requirements for a single-deck pontoon type roof require careful consideration, as this type of roof is not as rigid as the double-deck type.

An initial calculation for the number of support legs required for a single-deck roof can be approximated as follows:

For the pontoon support legs, allow one leg per 6 metres of tank circumference.

The number of centre deck legs can be roughly calculated by allowing one leg per 34 square metres of centre deck area for

tanks up to 60 metres in diameter, and one leg per 26 square metres for tanks larger than 60 metres in diameter. A structural design check is made on the legs to ensure that they are capable of carrying the required loads.

The centre deck legs are located as near as possible on equi-spaced radii between the tank centre and the inner rim of the pontoon.

The concentric tube construction of the legs allows product vapour to escape through the annular space between the leg and its housing and also through the leg location pin holes. This can be prevented by covering each leg with a non-permeable fabric tube, closed off at the top and tightly clamped around the leg housing at the bottom. They are known in the tank industry as "leg socks".

6.5.2 Guide pole

A vertical guide pole is situated about one metre inside the tank shell and its purpose is to prevent the floating roof from rotating in the tank. The pole is usually made from 300 to 450 mm n.b. pipe. The lower end is connected to the tank floor (or lower shell) and at the top to the gaugers platform, which is an extension to the tank top access stair. Only one of the connections can be rigid and it is normal for this to be the lower one, the top of the pole passing through a large diameter ring at platform level which has three adjusting screws for plumbing the pole.

The pole passes through a trunking in the roof pontoons, the top cover of which is fitted with rollers to prevent lateral movement of the roof in the trunking. Radial movement of the roof is not restrained here as this is provided by the roof seal system which tends to centralise the roof in the tank. Excessive escape of vapour from the radial elongated slot in the cover of the trunking is limited by the use of a brass plate, which is a snug fit on the pole but is allowed to slide radially across the cover of the trunking, thus sealing the slot in the cover.

The guide pole is very often used to house level-indicating equipment. To ensure that the product level in the pole is the same as the level in the tank, slots are cut in the pole to allow the liquid levels to equalise. This has the disadvantage in that the slots allow the escape of vapour into the atmosphere, although this may be minimised by the use of a tubular fabric concertina type sealing system on the outside of the pole.

6.5.3 Roof seals

The gap between the inside of the tank shell and the outer rim of the floating roof is normally about 200 mm. This gap is provided to ensure that the roof will not jam against the shell during operation.

To prevent the escape of vapour from this gap and to minimise the amount of rain entering the product here, a sealing system is required. This sealing system has to be flexible enough to allow for any irregularities in the construction of the roof and shell when the roof is travelling up and down and for any radial or lateral movement of the roof due to wind or other action.

When floating roofs were first devised, they were fitted with just one primary sealing system but recent legislation, which limits vapour emissions, has meant that a secondary seal is now required to be mounted above the primary.

Many types of primary seal have been devised over the years since floating roofs were developed and a selection of these are discussed below together with the more recently developed compression plate type of primary and secondary seal.

6.5.3.1 Mechanical seals

This type of seal has been in use for many years and its robust construction gives years of maintenance free service, Figure 6.15 illustrates such a seal.

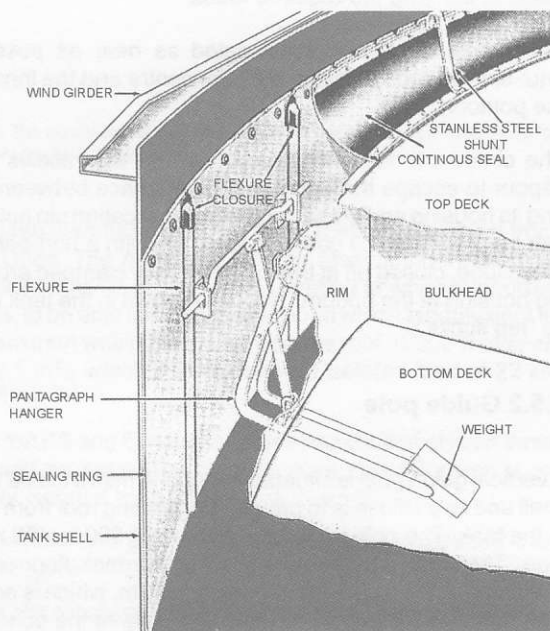


Figure 6.15 Mechanical seal
Courtesy of Chicago Bridge & Iron Company (CB & I)

The seal consists of a ring of thin galvanised or stainless steel plates, each about 4 metres long and 1.2 metres deep, bolted together with sealing strips and countersunk bolts.

This ring of sealing plates is kept in close contact with the shell by a series of weighted or spring-loaded pantograph mechanisms mounted on the outer rim of the pontoons. The lower edge of the plates is immersed in the product and the upper edge is roughly level with the top rim of the pontoons. The gap between the plates and the pontoons is sealed by a flexible U-shaped fabric which is connected to the top of the ring of plates and to the pontoon rim by clamp bars and bolts.

Vapour can escape however where irregularities in the shape of the shell allow gaps between the plates and the shell. To alleviate this problem the seal ring can be made to accommodate such changes in shape by the introduction of flexure points in the seal plates. These flexure points are formed by vertical shallow V-shaped creases in the plates at about 560 mm cen-

tres and the open top of these creases is capped to prevent vapour emission. The creases, as well as allowing the seal ring to conform to the shape of the shell, also act as stiffeners where the thrust from the pantograph mechanisms is transmitted to the seal ring.

One of the disadvantages of this type of seal is that the U-shaped fabric seal can collect rainwater, shell corrosion products and any waxy residue deposited on the shell. To minimise this, a second ring of short overlapping plates called a weather shield can be attached to the pontoon rim and rest against the shell at about 60°. This weather shield helps to shed rainwater and any detritus from the seal. With regard to waxy deposits on the shell, the upper edge of the ring of seal plates can be formed to act as a scraper on the shell to remove any waxy products.

To ensure the dispersal of any static or lightning, a series of thin flexible stainless steel shunts are connected between the bolt rings of the roof and the seal ring thus giving electrical continuity between the roof and the shell.

6.5.3.2 Liquid-filled fabric seal

The liquid-filled fabric seal, see Figure 6.16, consists of a petroleum and abrasion resistant synthetic rubber type tube filled with 200 to 250 mm depth of sealing liquid. This tube is positioned in the rim space and is supported at its lower end by a bottom ring on a hanger system.

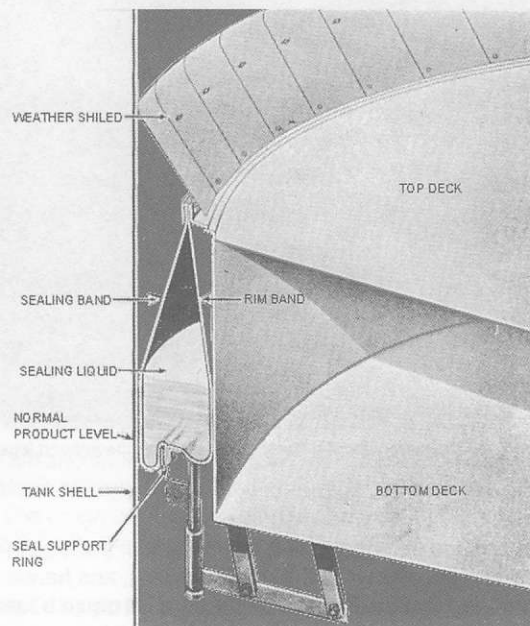


Figure 6.16 Liquid-filled fabric seal
Courtesy of Chicago Bridge & Iron Company (CB & I)

The sealing liquid ensures close contact of the tube on the tank shell and the outer rim of the floating roof. The liquid may be fuel oil or the same liquid as that stored in the tank. In non-freezing climates water may be used as the sealing liquid. The sealing liquid makes the tube take up whatever rim space is available around the circumference and automatically compensates for discontinuities in the shell or roof rim profile. The fixed diameter flexible bottom ring is supported by a hanger system which incorporates bumper bars to limit the minimum rim gap and prevents pinching of the tube material. This flexible ring has a fixed circumference and therefore automatically aligns to any discontinuities in the major or minor axes of the tank and roof. The usual rim space range is plus or minus 100 mm on a nominal rim gap of 200 mm.

6.5.3.3 Resilient foam-filled seal

This type of seal, shown in Figure 6.17, is similar to the liquid-filled seal except that the tube is filled with pre-formed blocks of resilient urethane foam, rather than a liquid and there-

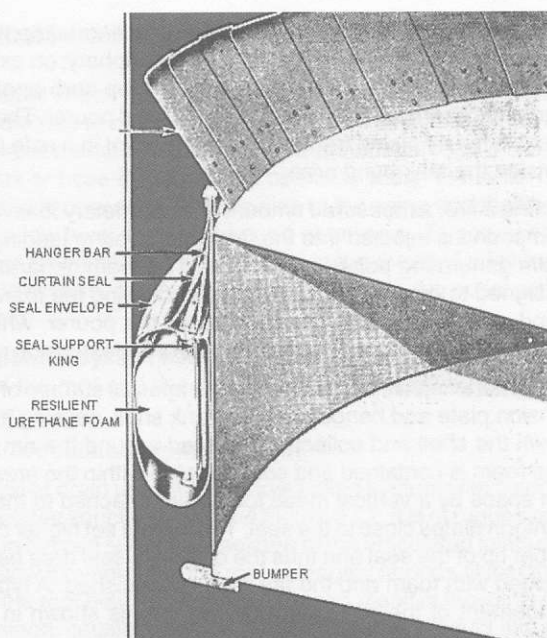


Figure 6.17 Resilient foam-filled seal
Courtesy of Chicago Bridge & Iron Company (CB & I)

fore does not require a bottom hanger support system. The resilient foam blocks ensure a good contact of the tube on the shell and roof outer rim gap of 200 mm. The seal allows variations of ± 100 mm in the rim space and excessive pinching of the seal tube is prevented by limiting bumper bars mounted on the lower edge of the outer rim of the roof.

Advantages of this type of seal are that when it is mounted just above the liquid level in the rim gap, any small tears or abrasions in the tube will not cause a serious collapse of the seal. Also, when replacement is finally necessary, this may be done entirely from above the roof.

6.5.3.4 Compression plate type seals

In terms of the timescale of the evolution of floating roofs, the compression plate type of seal is a more recent innovation and these are described as follows.

Secondary seals

Demanding environmental requirements required seal manufacturers to develop seals which would significantly reduce even further the vapour or odour losses from floating roof tanks. It was found that even properly maintained primary seals, operating in geometrically accurate tank shells, permitted vapour losses from the rim gap due to the swirling, scouring action of the wind within the tank. To counter this, independently mounted spring action compression plate secondary seals, formed from thin galvanised steel or stainless steel sheet, were mounted above the primary seal thus excluding the wind from the rim gap.

The number and size of the plates are custom-made to suit the profile of the shell, roof and the rim gap and the bolting pitch is made to suit the existing vertical or horizontal seal mounting ring on the outer rim of the roof. The spring action, due to the induced compression in the plates ensures a close seal between the abrasion resistant polymer seal tip and the shell. The tip is bolted to the edge of the plate and the joints between adjacent lengths of tip are overlapped with a scarfed joint and bonded with an adhesive compound.

The joints between adjacent compression plates are bolted and sealed with a soft gasket and allow relative movement between the plates whilst preserving an impervious seal. In some cases the plates are not bolted and sealed, but instead a continuous flexible vapour barrier fabric is fitted behind the plates attached to the seal tip and the seal mounting ring on the roof. A further

advantage of this type of seal is that it can be fitted from above the roof without the tank having to be taken out of service. This type of seal is illustrated in Figure 6.18.

Primary seals

The success of compression plate secondary seals led manufacturers to develop this type of design as a primary seal also. The technology, geometry, materials of construction and the fixing method is the same as that of the secondary seal, the main difference being that the primary seal deflects downwards such that the tip of the seal is usually just above the level of the stored liquid.

This type of primary seal is very often fitted in conjunction with its counterpart secondary seal. It is used for new tanks and also as the replacement system for the older type of existing seals when it becomes due for retirement. As mentioned earlier, an advantage of these seals is that they can be fitted from above the floating roof. See Figure 6.19.

Seals incorporating foam dams

An effective way to contain and deal with a potential fire in the rim space of a floating roof tank is to provide a foam dam at the outer rim of the roof. This short vertical steel wall ensures that

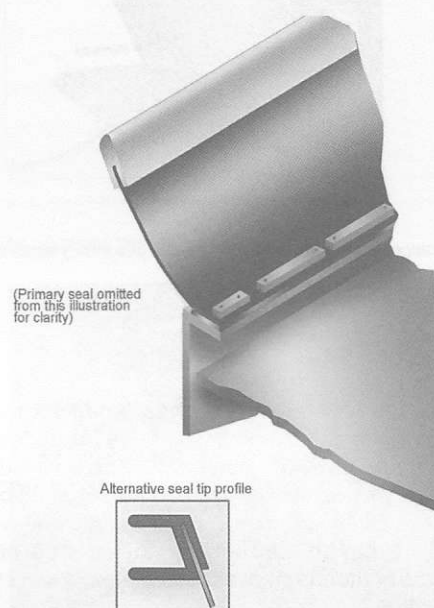


Figure 6.18 Compression plate type secondary seal
Courtesy of McTay

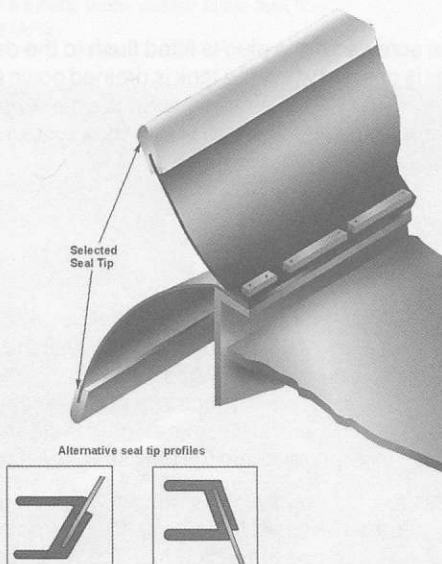


Figure 6.19 Compression plate type primary and secondary seals
Courtesy of McTay

as the top-injected fire fighting foam spills down the inside face of the shell, the foam dam contains and concentrates the foam within the rim space and does not allow it spill out over the surface of the roof. Some of the older floating roof tanks were not provided with foam dams and a further refinement, which can be included when fitting the compression plate type of seals, is the inclusion of a purpose-made foam dam. The design is such that no hot work is required to fit it as it bolts on to the seal fixing ring. Again, the tank does not have to be taken out of service to have this refinement fitted. See Figure 6.20.

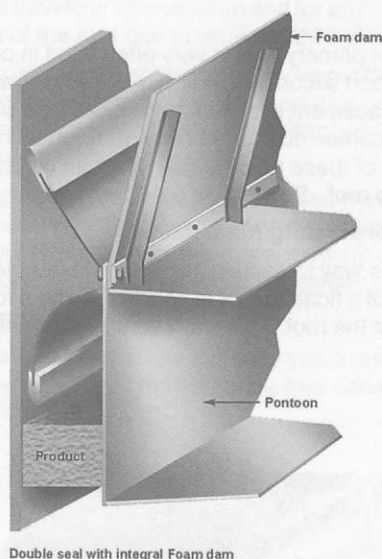


Figure 6.20 Compression plate type primary and secondary seals with a foam dam
Courtesy of McTay

6.5.4 Rim vents

Depending upon how a tank receives product, there are instances where entrained vapour may be released into the tank from the filling pipeline. This surge of vapour would seek release from the tank via the rim gap and the resulting build-up of pressure could cause damage to the sealing fabric. To prevent this, a vent tube may be fitted between the outer rim and the upper deck of the pontoon where either a pressure relief valve or a free vent is fitted.

6.5.5 Drain plugs

At least one screwed drain plug is fitted flush to the deck of the roof and this is opened when the tank is drained down and out of service. The open drain allows rainwater to drain from the surface of the roof on to the tank floor and thus relieves the roof support legs of any additional load.

6.5.6 Fire fighting

Fires in floating roof tanks are usually limited to the area between the shell and the rim of the floating roof i.e. the rim space. However, fires in this area are fairly rare, because the available sources of ignition are generally limited to that of a lightning strike, or a discharge of static electricity between the roof and the shell. The latter is virtually eliminated by the earthing systems which are incorporated into the tank structure and seals.

Nevertheless fire fighting systems are provided on tanks and one such system is designed to deliver a flame smothering expanded foam mixture into the tank rim space which quickly extinguishes the fire.

Such a system may be set up in the following way:

Several sets of foam generating and injection equipment are provided, equi-spaced around the tank periphery, on extension plates set above and bolted to the shell top curb angle. This equipment consists of a foam generator and pourer. The equipment is fed by piping from a fire fighting point in a safe position outside the tank bund area.

During a fire, a measured amount of a proprietary foam making compound is injected into the fire water system leading to the foam generating points on the tank. The foam generators are designed to draw air into the mixture, causing the foam to expand as it is injected into the tank via the pourer, which is a downward facing cowling on the inside of the extension plate.

This pourer injects the foam on to the internal surface of the extension plate and hence on to the tank shell, causing it to flow down the shell and collect and spread around the rim space. The foam is contained and concentrated within the area of the rim space by a vertical metal foam dam attached to the upper pontoon plates close to the seal. This dam is set higher than the upper tip of the seal and thus the complete seal area becomes flooded with foam and the fire thus extinguished. A typical arrangement of the equipment on the tank is shown in Figure 6.21.

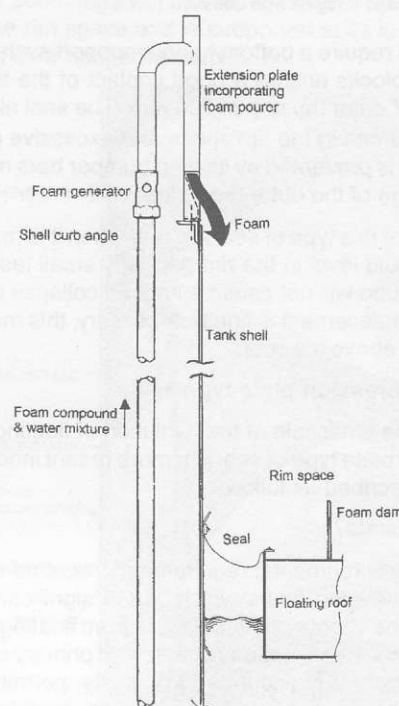


Figure 6.21 Foam fire fighting system
Courtesy of Angus Fire

6.5.6.1 Rim fire detection

The fire fighting equipment can be triggered to operate by a detection system which is in the rim space. This can take the form of a small bore plastic tube which runs around the whole circumference of the rim area.

This tube is connected into a more substantial piping system in both flexible and hard piping, which is connected into a fire fighting alarm or initiation control unit on the gaugers platform. The rim tubing is subjected to an internal pressure and in the event of a fire, the tubing melts releasing the pressure thus triggering an alarm and/or actuating the fire fighting system.

Another method is to have a series of tensioned wires with fusible links arranged around the rim space. Again, in the event of a fire a fused link would cause the alarm to be raised.

6.5.7 Roof drains

The rainfall which accumulates on the surface of the floating roof is drained to one or more sumps set into the low points of the top roof membrane. The sump is drained through a closed pipework or hose system which operates within the tank. The upper end is connected into the side of the sump and the lower end to a low level shell nozzle and gate valve. To prevent the roof from being flooded with product in the event of a failure in the drain system, a non-return valve is fitted to the outlet within the sump.

The pipework system has to be flexible to allow for the movement of the roof and this can be accommodated by using the following:

6.5.7.1 Articulated piping system

This type of drain uses a solid steel piping system with a series of articulated knuckle joints, see Figure 6.22. It is of rugged construction but can suffer from seizure of the articulated joints due to the slow movement of the roof or lengthy periods of inactivity due to the roof being stationary. This can result in the joints being strained causing them to fail and allowing product into the drain system.

However, a variation of this type of joint has been devised whereby a two-piece steel bracket, pivoting in one plane and housing a short length of armoured flexible hose connected to the face of each bracket, is used as the flexible joint.

6.5.7.2 Armoured flexible hose

This type of system eliminates the need for articulated joints, but it has been known for the hose to snag on internal tank fittings or for it to be trapped under a roof support leg as the roof grounds on the tank floor.

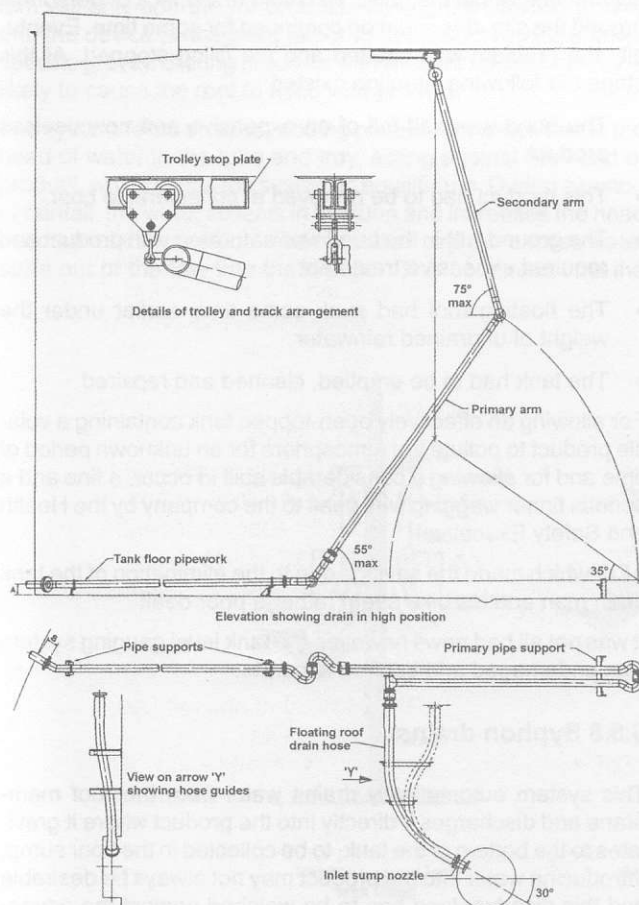


Figure 6.22 Articulated pipe drainage system for floating roof tanks

The hose system is outlined in Figure 6.23, and Figure 6.24 shows a tubular frame welded to the tank floor which is designed to guide the hose away from the leg landing area.

6.5.7.3 Helical flexible hose

The helical hose (see Figure 6.25), is a refinement of the straight hose as it is designed to take up the form of a helical spring, the idea being that it maintains a constant repeatable lay-down pattern on the tank floor, expanding and contracting with the rise and fall of the roof.

Hoses can of course sustain damage due to malfunctions in service and if punctured allow the stored product into the drain system.

The gate valve on the drain nozzle at the shell of the tank is always kept closed except when draining water from the roof and it is important to regularly monitor the roof for the accumulation of water, which must be drained off leaving the system dry, es-

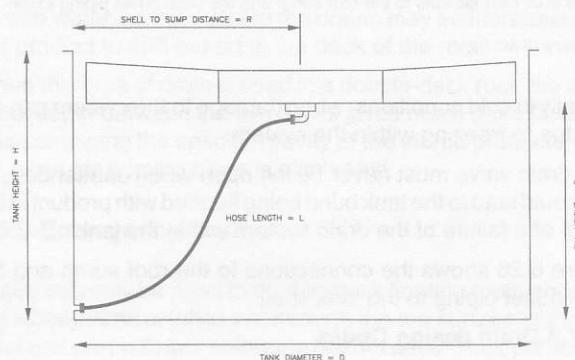


Figure 6.23 An armoured flexible hose



Figure 6.24 A tubular frame welded to the tank floor
Courtesy of McTay

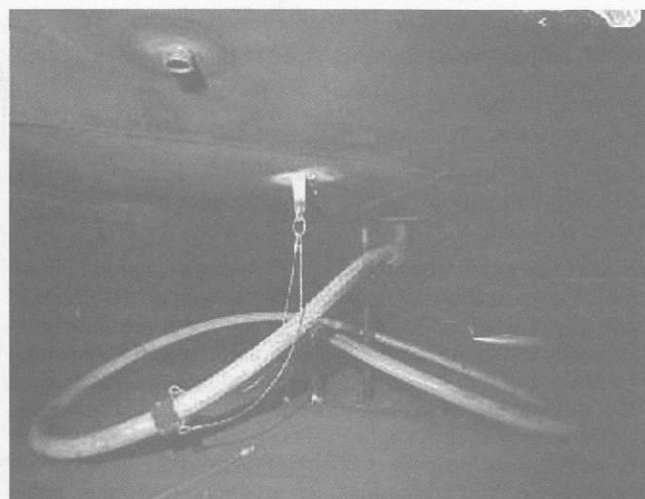


Figure 6.25 Helical flexible hose
Courtesy of McTay

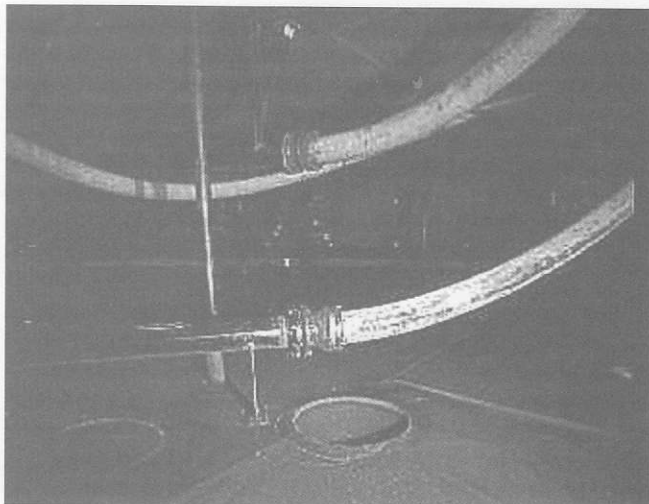


Figure 6.26 Connections to the roof sump and the steel outlet piping to the tank shell

Courtesy of McTay

pecially in cold conditions, when damage to the system can occur due to freezing within the system.

The drain valve must never be left open when unattended, as this could lead to the tank bund being flooded with product in the event of a failure of the drain system within the tank.

Figure 6.26 shows the connections to the roof sump and the steel outlet piping to the tank shell.

6.5.7.4 Drain design Codes

The design Codes require that at least one roof drain shall be provided as follows:

API Code

The drain diameter should be:

- at least 3" (80 mm) diameter, for tanks ≤ 36 m diameter.
- at least 4" (100 mm) diameter for tanks > 36 m diameter.

BS Code

The drain diameter should be:

- 75 mm diameter, for tanks ≤ 30 m diameter.
- 100 mm diameter, for tanks > 30 m diameter.
- 150 mm diameter, for tanks > 60 m diameter.

European Code

The drain diameter should be:

- 75 mm diameter, for tanks < 30 m diameter.
- 100 mm diameter, for tanks 30 to 60 m diameter.
- 150 mm diameter, for tanks > 60 m diameter.

6.5.7.5 "The man who drained the floating roofs"

— A cautionary tale:

A large refinery located in the UK, which shall remain nameless, had a large number of floating roof tanks storing crude oil and refined products.

It is necessary to remove the accumulated rainwater from floating roof tanks as they are only designed to support 10 inches of water whilst floating. To achieve this the roofs are fitted with drains which take the rainwater from a sump or series of sumps on the floating roof down through the product to a lower shell outlet connection which is fitted with an external drain valve. This valve was always kept closed because of concern at that time, about the possibility of failure of the roof drain, within the product liquid. In this circumstance an open drain valve would mean that the tank would dump most of its contents into the bund.

At this particular refinery the roof drainage was achieved by an employee who, armed with a bicycle, would cycle from tank to tank. He would climb the radial or circumferential tank stairway and look down at the floating roof. If accumulated rainwater was present, he would descend and drain the water into the site drains using the external valve. During his visit to the tank he would check to see that no oil was present in the drained water, indicating the beginnings of an internal drain problem. He would also look to see if the roof drain sump outlet was clear and not blocked by sundry debris or seagulls' nests and that the tank bund was not being undermined by the local rabbit population. In addition to performing a useful purpose and having a pleasant outdoor life, the combination of cycling several miles each day and climbing several hundred feet up tank stairways kept our friend as fit as a butcher's dog.

Sadly this idyllic state of affairs was not to be allowed to continue. New management, equipped with the cost cutting gene were installed. The tank drain man and his bicycle were seen as being rather old-fashioned and were removed from the payroll. Half-hearted attempts to use clever drainage valves which could discriminate between rainwater and oil, and consequently allow the tank drain valves to remain constantly open, were made but this is an expensive and problematic area and was consequently soon forgotten.

Some little time later, one of the tanks came to the attention of the facility management. It was exhibiting contradictory symptoms. The rolling ladder was inclined at an angle which indicated that the tank was empty whereas the level indication system indicated that the tank was full. It was decided that the rolling ladder could not lie whilst the level indication could, as in the past it had occasionally failed to register the correct situation.

Without examining the tank further, filling was commenced. Product soon poured over the top of the tank shell and began to accumulate within the bund. Because of the lack of personnel around the site, this situation continued for some time. Eventually the problem was spotted and the filling stopped. At this stage the following situation existed:

- The bund was half full of an expensive and now useless product
- This product had to be removed at considerable cost
- The ground within the bund was saturated with product and required expensive treatment
- The floating roof had sunk some time earlier under the weight of undrained rainwater
- The tank had to be emptied, cleaned and repaired

For allowing an effectively open-topped tank containing a volatile product to pollute the atmosphere for an unknown period of time and for allowing a considerable spill to occur, a fine and a serious finger wagging was dealt to the company by the Health and Safety Executive!

All of which made the savings due to the elimination of the tank drain man and his bike seem rather a poor deal!

It was not all bad news however, the tank level gauging system was undamaged and spot-on accurate.

6.5.8 Syphon drains

This system automatically drains water from the roof membrane and discharges it directly into the product where it gravitates to the bottom of the tank, to be collected in the floor sump. Introducing water into the product may not always be desirable and this disadvantage has to be weighed against the advantage of rainwater being automatically removed from the roof without the need for any manual operations.

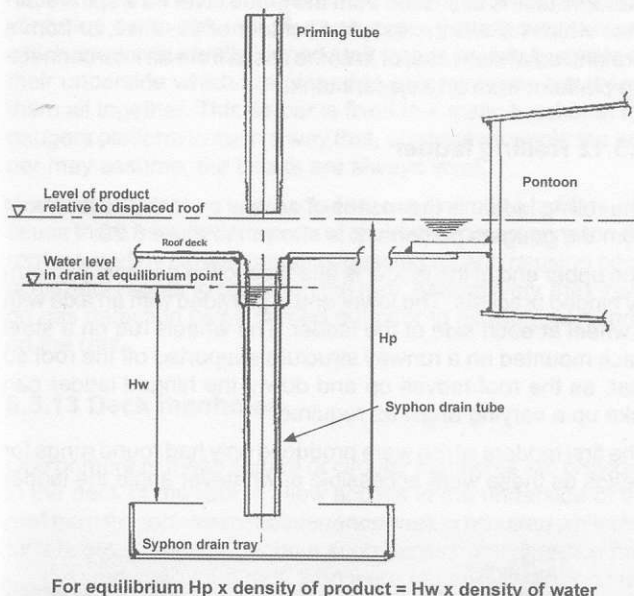


Figure 6.27 A syphon drain fitted to a single deck floating roof

A diagrammatic representation of a syphon drain fitted to a single-deck floating roof is as shown in Figure 6.27.

The device which is built into the construction of the floating roof, consists of a length of tube (usually 50 or 80 mm bore) set flush with the top surface of the roof membrane and extending vertically into the product below the roof level. The lower end of the tube sits in a open top tray which is supported off the tube. The length of the tube and the position of the tray is critical and is calculated to suit the specific gravity of the stored product and the displacement of the roof within the stored product. The system will only operate for products having the specific gravity that the device is designed for, or for a product having a lower specific gravity. Storing products with a higher specific gravity is likely to cause the roof to flood with product.

The system relies on always being primed with water, as it is the head of water in the tube and tray, acting against the head of product, which keeps the system in equilibrium. During a period of rainfall, the water collects in the tube and increases the head over that of the constant head of product and the excess water spills out of the tray into the product. This continues until the

rain stops, when the excess head of water decreases and the system returns to equilibrium.

As mentioned earlier the syphon drains must always be primed with water. This means that when a tank with a single-deck roof is filled from being empty, the drains have to be fitted with an extension tube to prevent product escaping on to the deck through the drain points, due to the natural displacement of the roof. This is achieved by temporarily screwing a priming pipe into the top of each drain tube and when the roof is floating, water is poured into the priming pipe until the level of water in the syphon tube is below deck level. The priming pipes are then removed.

Also when a single-deck roof tank is on hydrotest the priming pipes must be fitted to prevent the roof being flooded with water. In this case the pipes remain in position throughout the test and are only removed after the priming operation mentioned above.

During periods of hot dry weather the drains should be topped up with water, as the water in the drains may evaporate and allow product to spill out on to the deck of the roof.

When this type of drain is used in a double-deck roof, the additional depth between the two decks gives much more flexibility when changing the specific gravity of the stored products, also the need for priming pipes is eliminated.

6.5.9 Emergency drains

These can only be fitted to double-deck floating roofs and they are simply vertical tubes set through the top surface of the top deck and protrude just below the bottom deck. Their purpose is to allow natural drainage of rainwater in the event of malfunction of the primary drains. The top of these drains are normally provided with a mesh screen to prevent them being blocked by detritus from the deck. The use of this type of drain has waned because the open drain allows vapour to escape from the tank, which is unacceptable nowadays.

6.5.10 Bleeder vents

This vent only comes into operation either when the floating roof is being landed, and the tank is drained down, or when an empty tank is being filled. Its purpose is to vent the area below the landed roof in its stationary position, allowing air to enter the space under the roof as the product is evacuated from the tank, thus avoiding a vacuum in the space and then to allow the air

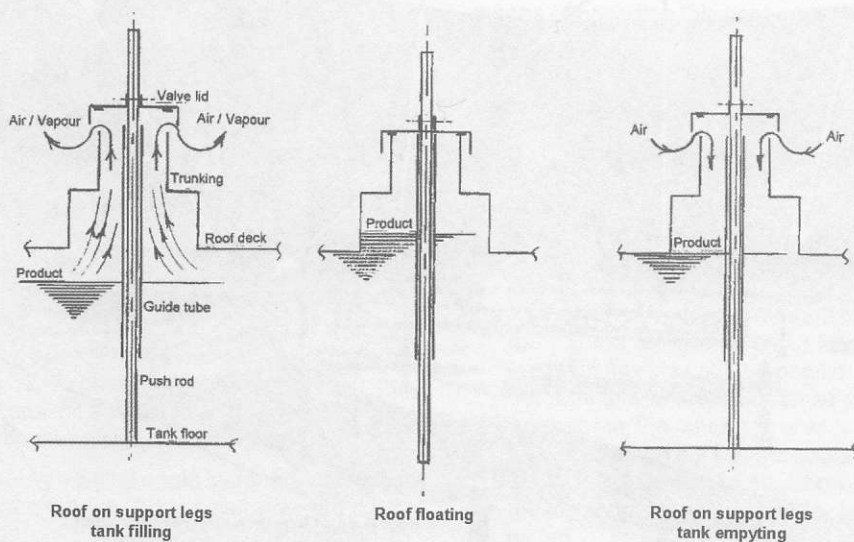


Figure 6.28 Bleeder vents

under the roof to escape when the tank is being refilled, avoiding a pressure under the roof.

The valve is a simple device consisting of a short vertical trunking which forms a valve seating and this is welded to a corresponding aperture in the deck. Through the centre, and supported off of this trunking, passes a vertical guide tube which houses a push rod on to which is attached a disc which forms the valve lid. The length of the push rod is such that as the tank is emptied, the rod contacts the floor plating before the roof support legs land and the valve opens, freely venting the space beneath the deck. Similarly, on refilling the tank the valve closes after all the air beneath the roof has been expelled and the roof floats. The diagrammatic sketch in Figure 6.28 shows the operation of the valve.

However, this type of simple valve is not environmentally friendly because, once open, it remains open, thus allowing vapours to escape when the roof is landed and drained down. The alternative is to use pressure and vacuum valves, which will only open when there is a differential pressure across them and will therefore remain closed after drain down. Also the pressure and vacuum valve will allow the release of vapour from under the roof formed by solar means or imported slugs of vapour from the filling line, whilst in service.

6.5.11 The gaugers platform

The gaugers platform is a relatively small access area of about four square metres, usually elevated about 2 metres above the top curb angle of the shell. The platform overhangs the shell to allow the guide pole to pass through it so that access can be gained to the guide pole, which usually houses the product level indicating equipment or a dip hatch. Also the platform is used as an attachment for the rolling ladder which gives access to the floating roof.

The platform is supported off a stiffened section of the top course of shell plating by a fairly substantial steel structure. The

platform itself is accessed from the grade level via a spiral staircase which follows the external contour of the shell, or from a straight radial staircase, or in some cases from an interconnecting platform from an adjacent tank.

6.5.12 Rolling ladder

The rolling ladder is the means of access on to the floating roof from the gaugers platform. It is shown in Figure 6.29.

The upper end of the ladder is attached to the gaugers platform by hinged brackets. The lower end is provided with an axle with a wheel at each side of the ladder. The wheels run on a steel track mounted on a runway structure supported off the roof so that, as the roof moves up and down, the hinged ladder can take up a varying angle as required.

The first ladders which were produced only had round rungs for treads as these were accessible at whatever angle the ladder



Figure 6.29 Typical rolling ladder with self-levelling treads
Courtesy of McTay Engineering

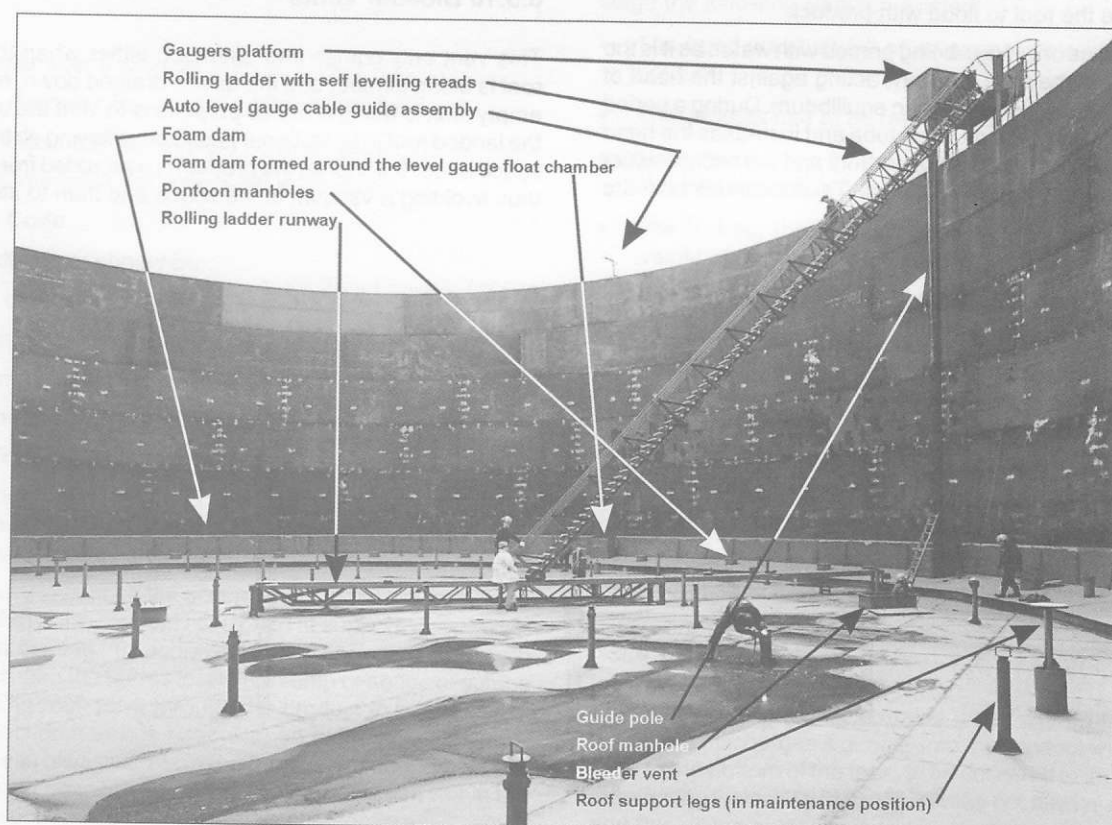


Figure 6.30 The location of some of the common appurtenances found on a floating roof
Courtesy of McTay

happened to be at, but these proved to be unsafe for personnel venturing on to the roof. A much safer system was devised which uses individually hinged stair treads having brackets on their underside which are pinned to a common tie bar linking them all together. This tie bar is fixed to a static bracket at the gaugers platform in such a way that, at whatever angle the ladder may assume, the treads are always level.

Some tank operators now exclude the use of rolling ladders, because there have been reports of accidents to personnel on the roof created by certain products gassing off and causing pools of harmful vapour to collect on the roof. Alternatively, they insist on gas detection being carried out prior to allowing personnel on the roof.

6.5.13 Deck manholes

One or more of these square or circular manholes are provided in the deck of the roof to allow access to the underside of the roof from the top, when maintenance work is required whilst the tank is out of service. Without such access maintenance personnel working on the roof, who were required to work on the underside, would only be able to gain access by the circuitous route involving ascending the steep rolling ladder, descending the external staircase and entering the tank via the shell manhole.

6.5.14 Pontoon manholes

Each pontoon of a floating roof is a separate buoyancy compartment and must be periodically checked to ensure that it is dry and free from leaks. Hence each compartment has its own inspection manhole.

These manholes are generally of light construction consisting of a short circular coaming welded to the top plate of the compartment, the closure being a loose flat lid with a down-turned lip which fits over the coaming to keep out the rain. The lid is fitted with a handle for easy access to the compartment.

Figure 6.30 shows the location of some of the common appurtenances found on a floating roof.

6.5.15 Sample/dip hatch

The sample/dip hatch is fitted either to a nozzle which projects through one of the pontoons or it is fitted to the top of the gauge pole. It is illustrated in Figure 6.31 and may be used as follows:

- To measure the depth of product in the tank using a dip tape. This may be done as a check on the correct functioning of the automatic level gauge.
- To take a sample of the tank contents.
- To take the temperature of the tank contents.

6.5.16 Foam dam

This topic was discussed earlier in Section 6.5.9.3, in conjunction with primary and secondary compression plate type floating roof seals. However, the normal construction for a foam dam consists of a short vertical plate in 3 mm steel, which is welded to the top pontoon plate at a short distance from the seal assembly, see Figure 6.32. To give effective fire protection, the height of the dam plate must be above the tip of the roof seal so that the injected foam will completely cover the seal.

The plate is given rigidity by vertical angle stiffeners at regular intervals around its circumference. Also, small slots are cut in the lower edge of the dam plate at its junction to the pontoon, to



Figure 6.31 Typical dip hatch fitting
Courtesy of Endress+Hauser Systems & Gauging Ltd

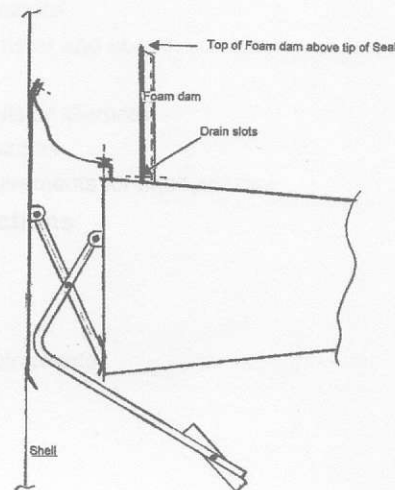


Figure 6.32 Position of foam dam in relation to the seal assembly

give drainage for rainwater which could accumulate in the space between the seal and the dam.

6.5.17 Electrical continuity

In the event of a lightning strike on the tank, or a build-up of static electricity within the tank due to product movements, there needs to be a secure electrical bond between the roof and the tank to make certain that any electrical charge is conducted directly to earth, thus ensuring that a spark can not be created between the roof and the tank which could cause a fire. The means of providing this continuity may be by:

- Providing thin flexible stainless steel shunt strips between the top of the steel sealing ring of a mechanical seal and the seal connection ring on the floating roof.
- A long length of flexible cable attached to the gaugers platform and to the top of the roof pontoons. The length of the cable in this case makes it prone to snagging on other roof fittings so positioning of the attachment points requires careful consideration.
- A variation of the above method is to bond the gaugers platform to the top of the rolling ladder structure with a short length of flexible cable. A position some way down the ladder structure is then chosen as an attachment point for another cable, the other end of which is bonded to the floating roof structure. This second cable is much shorter than that above, and by careful selection of the attachment points, the lay down path of this cable can be fairly accurately predicted.

The first step in the design of a storage tank is the selection of the material of construction. The material must be able to withstand the internal pressure of the liquid being stored, as well as the external pressure of the atmosphere. The material must also be able to resist corrosion from the liquid being stored. The most common materials used for storage tanks are carbon steel, stainless steel, and aluminum. Carbon steel is the most economical, but it is susceptible to corrosion. Stainless steel is more expensive, but it is more resistant to corrosion. Aluminum is the most expensive, but it is the most resistant to corrosion.

The next step in the design of a storage tank is the selection of the shape. The most common shapes are cylindrical, spherical, and conical. Cylindrical tanks are the most common, and they are the easiest to design and construct. Spherical tanks are used for storing gases, and they are the most efficient in terms of volume. Conical tanks are used for storing liquids, and they are the most efficient in terms of volume.

The third step in the design of a storage tank is the selection of the thickness of the walls. The thickness of the walls must be able to withstand the internal pressure of the liquid being stored. The thickness of the walls must also be able to resist corrosion from the liquid being stored. The thickness of the walls must be determined by the design pressure of the tank, the material of construction, and the diameter of the tank.

The fourth step in the design of a storage tank is the selection of the supports. The supports must be able to support the weight of the tank and the liquid being stored. The supports must also be able to resist corrosion from the liquid being stored. The supports must be determined by the weight of the tank, the weight of the liquid, and the material of construction.

The fifth step in the design of a storage tank is the selection of the accessories. The accessories must be able to handle the liquid being stored. The accessories must also be able to resist corrosion from the liquid being stored. The accessories must be determined by the type of liquid being stored, the material of construction, and the design pressure of the tank.

The sixth step in the design of a storage tank is the selection of the painting. The painting must be able to protect the tank from corrosion. The painting must also be able to resist the weather. The painting must be determined by the material of construction, the environment, and the design pressure of the tank.

The seventh step in the design of a storage tank is the selection of the inspection. The inspection must be able to detect any defects in the tank. The inspection must also be able to detect any corrosion. The inspection must be determined by the material of construction, the environment, and the design pressure of the tank.

The eighth step in the design of a storage tank is the selection of the maintenance. The maintenance must be able to keep the tank in good condition. The maintenance must also be able to detect any defects in the tank. The maintenance must be determined by the material of construction, the environment, and the design pressure of the tank.

The ninth step in the design of a storage tank is the selection of the safety. The safety must be able to protect the tank from any hazards. The safety must also be able to detect any defects in the tank. The safety must be determined by the material of construction, the environment, and the design pressure of the tank.

The tenth step in the design of a storage tank is the selection of the documentation. The documentation must be able to provide information about the tank. The documentation must also be able to detect any defects in the tank. The documentation must be determined by the material of construction, the environment, and the design pressure of the tank.

The eleventh step in the design of a storage tank is the selection of the construction. The construction must be able to build the tank. The construction must also be able to detect any defects in the tank. The construction must be determined by the material of construction, the environment, and the design pressure of the tank.

7 Tank fittings and ancillary equipment for ambient temperature tanks

This Chapter deals with the design of the various nozzles, manholes and other appurtenances that are required for the operation of the tank. Also, consideration is given to the access requirements to the tank for the operating personnel, and also to various fire fighting methods.

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7.1 Tank nozzles

7.1.1 BS 2654 requirements for shell nozzles

7.1.1.1 Nozzles 80 mm outside diameter and above

The BS Code requires shell manholes and shell nozzles of 80 mm outside diameter and above to be governed by the following rules:

Minimum wall thickness for various outside diameters shall be as shown in Figure 7.1.

Outside dia. (mm)	Min.wall thickness (mm)
80 to $\leq$ 100	7.5
>100 to $\leq$ 150	8.5
>150 to $\leq$ 200	10.5
>200	12.5

Figure 7.1 Minimum wall thicknesses for various outside diameters
From BS 2654, table 5

With regard to shell manholes, the Code gives details of a standard manhole in Figure 9 of the Code but stipulates that this is only suitable for tank heights up to 25 m. Tank heights are rarely above this height, but if this is the case then the components of the manhole and reinforcement would require analysis to ensure their suitability for the increase in pressure above a 25 m head.

The hole which is cut into the shell to accept the manhole or nozzle obviously weakens the shell in this area and therefore a means of providing reinforcement to compensate for this weakness is required. The Code requires that the cross-sectional area of this reinforcement, measured in the vertical plane containing the axis of the manhole or nozzle shall not be less than:

$$0.75 \times d \times t \quad \text{equ 7.1}$$

where

- d = diameter of the hole cut in the shell plate (mm)
 t = thickness of the shell plate (mm)

Reinforcement is provided by - The area replacement method.

The reinforcement may be provided by any one or any combination of the following three area replacement methods. Note that a corrosion allowance on any surface should be excluded from the computation of reinforcement required.

- a) The addition of a thickened insert plate as in Figures 7.2 and 7.3 or a circular reinforcing plate as in Figure 7.4.

The limit of the reinforcement is such that: 'do', the effective diameter of the reinforcement, is between 1.5.d and 2.d. A non-circular reinforcing plate may be used provided the minimum requirements are complied with. Also, where nozzles are close to the bottom of the tank, a "tombstone"-shaped reinforcing plate shown in Figure 7.3 may be used as long as the Code rules are complied with.

- b) The provision of a thickened nozzle or manhole barrel.

The portion of the barrel which may be considered as reinforcement is that lying within the shell plate thickness and within a distance four times the barrel thickness from the shell plate surface, unless the barrel thickness is reduced within this distance, when the limit is the point at which the reduction begins. Figure 7.5 illustrates this method.

- c) The provision of a shell plate thicker than that required by the shell thickness formula or given in the Table of minimum shell plate thicknesses, (whichever is relevant to the tank under consideration). The additional thickness being used as all or a part of the required reinforcement.

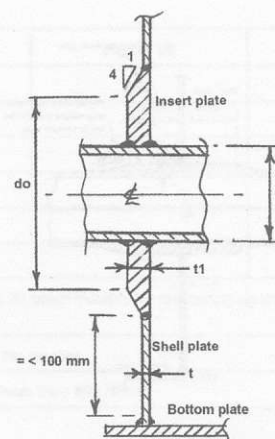


Figure 7.2 Thickened insert plate

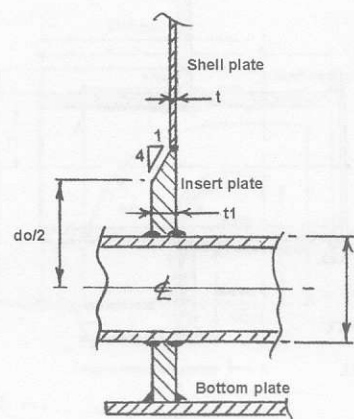


Figure 7.3 Thickened insert plate

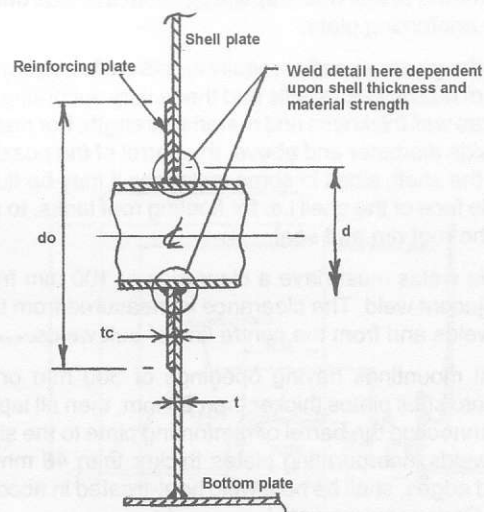


Figure 7.4 A circular reinforcing plate

As an alternative to the area replacement methods, the reinforcement can be made by the provision of a thickened nozzle barrel protruding on both sides of the shell plating as shown in Figure 7.6. This method was devised by R.T. Rose (see Reference 7.1) and was first introduced into the BS Code in the 1973 edition.

The method limits a stress concentration factor 'j' to a maximum value of 2 and this is derived from the graph shown in Figure 7.7 where a replacement factor 'y', based on the ratio of nozzle wall thickness to the mean radius of the nozzle, is plotted against the ratio of the outer to inner radii of the nozzle wall.

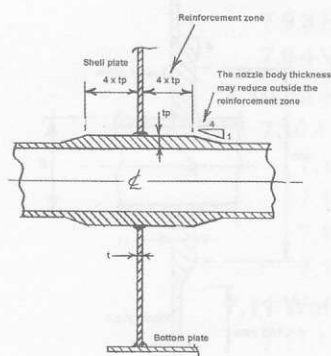


Figure 7.5 Provision of a thickened nozzle or manhole barrel

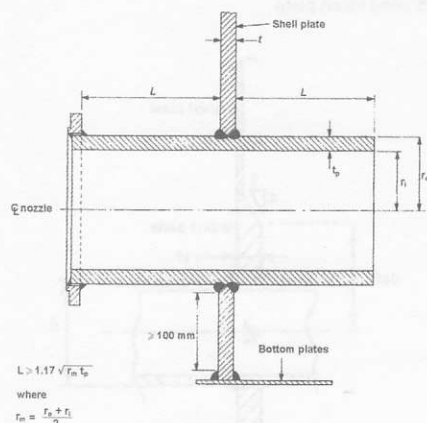


Figure 7.6 Provision of a thickened nozzle barrel protruding on both sides of the shell plating

This method is useful where space beneath a nozzle denies the use of a reinforcing plate.

The Code gives specific requirements with regard to the welding of nozzles into shells and these vary according to shell and nozzle wall thickness and material strength. For nozzles 80 mm outside diameter and above, the barrel of the nozzle is set through the shell, albeit in some instances it may be flush with the inside face of the shell i.e. for floating roof tanks, to prevent fouling the roof rim and seal.

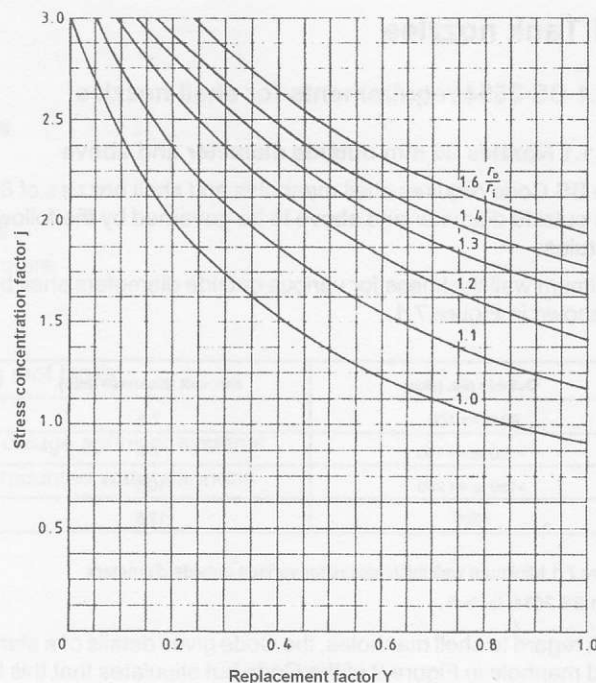
All nozzle welds must have a clearance of 100 mm from any other adjacent weld. The clearance is measured from the toes of fillet welds and from the centre line of butt welds.

For shell mountings having openings of 300 mm or larger, welded into shell plates thicker than 20 mm, then all lap or fillet welds connecting the barrel or reinforcing plate to the shell and all butt welds incorporating plates thicker than 40 mm at the prepared edges, shall be post weld heat-treated in accordance with the Code requirements.

Cautionary note - There have been accidents, especially on older tanks, where cast iron valves have been used on shell nozzles and the bodies of these have failed due to overstressing or freezing. Cast steel valves should always be used in these instances to obviate this problem.

7.1.1.2 Flush type clean-out doors

Some stored products contain entrained sediment, which tends to settle out of suspension during a lengthy storage period. This sediment builds up, generally in an uneven pattern, on the floor of the tank and when landing a floating roof on its support legs it can cause twisting of the deck due to the legs landing on the uneven surface. This is a particular problem with large floating roof tanks storing crude oil coming directly from the field, as these tanks spend many years in service before being taken out



$$y = 1.58 \sqrt{\frac{t_p^2}{r_m^2} + \frac{t_p}{2r_m}}$$

where

t is the shell plate thickness (in mm)

t_p is the nozzle body thickness (in mm)

r_m is the mean radius for branch bodies (in mm)

All dimensions are in millimetres

Figure 7.7 Plot of stress concentration factor v replacement factor

of service for maintenance.

A cautionary tale

A large UK-based refinery was fed by pipeline with oil and gas from the North Sea. The crude oil was stored in a number of 96 m diameter floating roof tanks. Each of these tanks was fitted with three product mixers of the Plenty propeller type, fitted in connections in the bottom course of the tank shell. The function of these mixers was to keep the product stirred up and to prevent the relatively high wax content from settling out of the crude oil and accumulating on the tank bottoms. During the early years of operation of these tanks the mixers were used regularly as envisaged by the tank designers and no problems occurred.

At a certain point in time, the terminal owners decided to institute a review to see if operating costs could be reduced. This task was given to a group fitted with the financial gene, but sadly not its technical equivalent! The collective "beady eye" eventually fell upon the high power consumption and consequent cost of running the tank mixers. It was decided to make savings by the simple expedient of not running the tank mixers at all. All went well for a while.

The roof then began to show an increasing disinclination to behave properly at low product levels. The centre deck would be flat, but the outer perimeter was uneven and at a higher level. This was again overcome by increasing the minimum product level for tank operation. All was again well until the day that oil began to appear from beneath the tank annular plate. This indicated a leak in the tank bottom plating and the flow of oil into the local bund was such that it could not be ignored.

So this meant that the tank would have to be emptied, cleaned and repaired. Sadly the floating roof showed serious signs of distress as the liquid level was lowered and an investigation

	Fig No 28a	Fig No 28b	Fig No 29	Fig No 30
Max UTS of btm. course shell plating (N/mm ²)	460	>460	460	>460
Min: btm. course width to accommodate full door height (mm)	1830	600	1830	600
Max. size of door opening W x H (mm)	915* x 1230	300** x 1230	915* x 1230	300** x 1230
Max. thks. of btm course (mm)	18.5	18.5	37	37
Max. thks. of insert plate (mm)	40	40	37	37
Max. thks. of reinforcing plate (mm)	N/A	N/A	40	40
Figure Nos 28a & 28b are limited to tanks having a bottom shell course no thicker than 18.5 mm, whereas Figure Nos 29 & 30 which incorporate reinforcing plates in their design, can be used on shell plating up to 37 mm thick				
*For Figures 28a & 29 the height of the door opening is: ½ the height of the bottom shell course, or 915 mm, whichever is the smaller				
** For Figures 28b & 30 the height of the door opening is limited to 300mm for shell plate steels having a minimum U.T.S. more than 460 N/mm ²				

Figure 7.8 Principal parameters for each of the four types of door

through roof leg fitting holes revealed an accumulation of waxy material of uneven thickness up to 2.0 m deep in places on the tank bottom. This was of sufficient load bearing capacity to locally support the weight of the floating roof. The original mixers had their propellers embedded in the wax and could not be started. Much time was spent in agitated "navel gazing" until a suitable specialist was found with a solution to the problem.

This involved the connection to the partially-filled tank of a huge pump which re-circulated the oil and eventually forced the wax back into solution so that it could be removed from the tank and disposed of. This process took months to complete and considerable sums of money, many times more than the cost savings so eagerly seized on earlier. The remaining tanks were investigated and all found to be suffering from substantial wax accumulations which required the same expensive and time-consuming treatment!

To assist in the disposal of the sediment once the tank has been taken out of service, the tank may have built into the shell, one or more large clean-out doors.

These flanged doors have can have openings, roughly one metre square, (although there are height limitations — as shown in Figure 7.8) with the bottom edge flush with the tank floor plating thus making for an easier internal cleaning operation.

The large size of the opening being in the highly-stressed bottom course of shell plating causes complicated stress patterns and therefore has to be carefully designed to ensure that the strength of the shell is not compromised.

The tank Codes recognise this and in the BS Code there are fully detailed arrangements for four different types of Flush clean-out doors for the designer to choose from. All of these designs involve the door being fitted into a shell insert plate and all these assemblies have to be post weld heat-treated on completion of fabrication.

The table in Figure 7.8 shows the principal parameters for each of the four types of door, these are identified by the figure numbers as used in BS 2654.

Illustrations of two flush type clean-out doors are shown in Figures 7.9 and 7.10.

A smaller, simpler and less expensive type of clean-out aid is the combined water draw-off and clean-out sump. This fitting is basically formed by a half-section of 610 mm diameter pipe 980 mm long attached beneath a 460 mm x 510 mm hole cut in the outer region of the floor plating. The external opening of the sump is closed with a 'D' shaped flange and cover, see Figure 7.11.

This fitting is used as a water draw-off sump during normal tank operations, with a nozzle and valve fitted at the low point on the cover and as a clean-out opening when removing sludge from

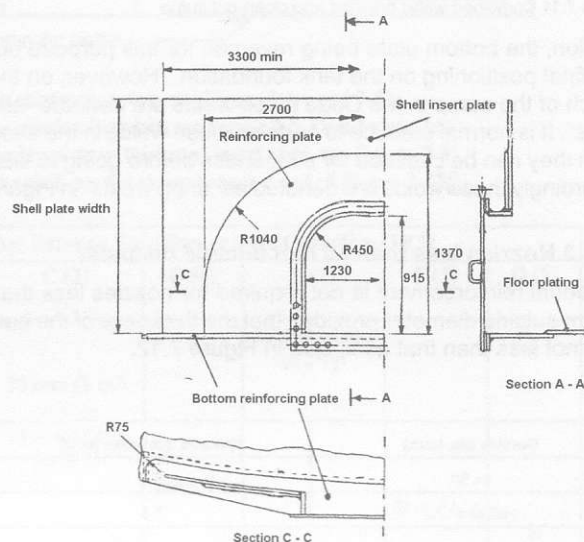


Figure 7.9 Flush type clean-out door with plate reinforcement, size of opening 915 mm x 1230 mm

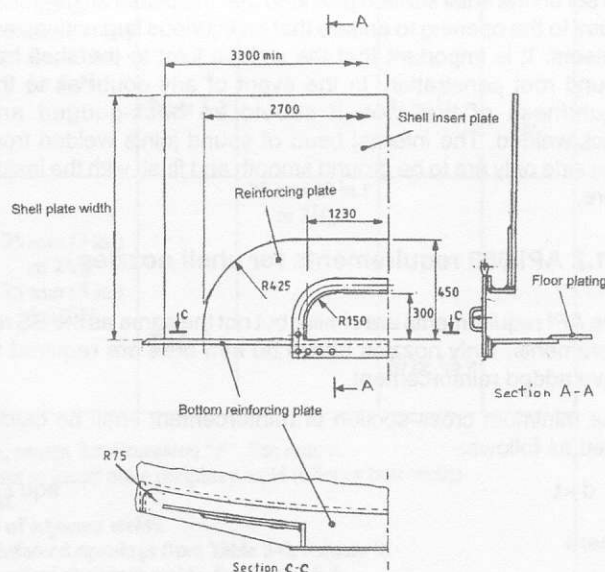


Figure 7.10 Flush type clean-out door with plate reinforcement, size of opening 300 mm x 1230 mm

the tank during maintenance operations. One disadvantage is that this sump can become blocked with excessive sludge and hence, its use as a water draw-off point when in service, is lost.

The Code states that "the fillet weld to the underside of the bottom sketch plate or annular plate shall be deposited in the flat

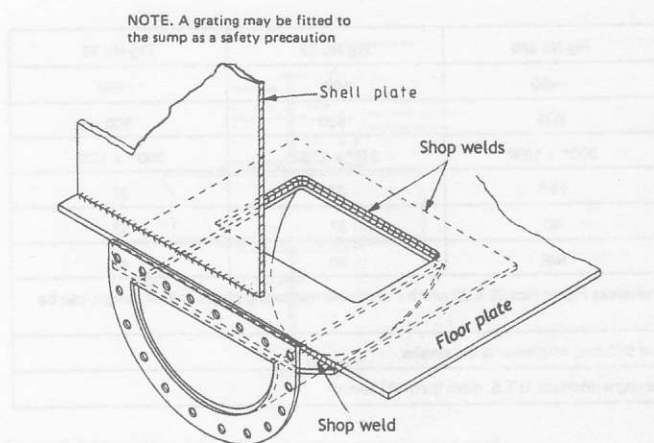


Figure 7.11 Combined water draw-off and clean-out sump

position, the bottom plate being reversed for this purpose before final positioning on the tank foundation." However, on the sketch of the sump in the Code these welds are denoted "site welds". It is normal practice to perform these welds in the shop when they can be checked for soundness before going to site. Accordingly these welds are denoted as "shop welds" in Figure 7.11.

7.1.1.3 Nozzles less than 80 mm outside diameter

Additional reinforcement is not required for nozzles less than 80 mm outside diameter provided that the thickness of the barrel is not less than that as shown in Figure 7.12.

Outside dia. (mm)	Min.wall thickness (mm)
≤ 50	5.0
>50 to ≤ 75	5.5

Figure 7.12 Barrel thicknesses
From BS 2654, table 5

These nozzles do not have to be set through the shell but may be set on the shell surface provided that the plates are checked close to the opening to ensure that no injurious laminations are present. It is important that the welded joint to the shell has sound root penetration. In the event of any doubt as to the soundness of the root, it should be back-gouged and back-welded. The internal bead of sound joints welded from one side only are to be ground smooth and flush with the inside bore.

7.1.2 API 650 requirements for shell nozzles

The API requirements are similar but not the same as the BS requirements. Only nozzles above 50 mm bore are required to have added reinforcement.

The minimum cross-section of reinforcement shall be calculated as follows:

$$d \times t \quad \text{equ 7.2}$$

where

d = diameter of the hole cut in the shell plate (mm)

t = thickness of the shell plate (mm)

Note: Only 75% of this value is required to the BS Code.

However, the calculated minimum required design shell thickness may be used in equation 7.2, instead of the nominal minimum shell plate thickness. (On smaller tanks the calculated design thickness is often less than the nominal shell plate thickness.)

A warning is given with respect to shell nozzles, which are close to the bottom of the tank. Such nozzles can rotate with the vertical bending of the shell under hydrostatic loading and connected piping can cause a restraint on the nozzle giving rise to additional stresses in the nozzle and shell. Attention is drawn to Appendix 'P' of the Code which deals with this problem but it must be remembered that this theory can only be applied to tanks over 36 m in diameter.

There is only an upper limit for the outside diameter of reinforcing plates and this is twice the diameter of the hole cut in the shell. (The BS Code is between 1.5 and 2.0 times the diameter of the hole in the shell plating.)

The means of providing reinforcement together with complete details for the fabrication and welding of nozzles in sizes from 1½" (38 mm) nominal bore, to 48" (1219 mm) nominal bore are given in several tables and diagrams in the Code, together with explanatory clauses.

Similar detailed information is also given for four shell manhole diameters: 500 mm, 600 mm, 750 mm and 900 mm.

There is a proviso in the Code regarding the portion of the barrel which can be considered as acting as reinforcement. In cases where the strength of the barrel material is slightly less than that of the shell plate material, then the portion of the barrel considered as reinforcement is reduced. Where the strength of the barrel material is much less than that of the shell plate material, then the barrel can not be considered as contributing to the reinforcement of the nozzle.

The Code addresses instances where there may be a cluster of nozzles close together in one area of the shell and shows how these should be spaced within one large reinforcing plate.

7.1.3 European Code requirements for shell nozzles

The prEN 14015-1 requirements are the same as given in the BS 2654 Code with the addition of the table of nozzle body thickness requirements that include minimum thickness for stainless steel nozzles and these are given in Figure 7.13.

Outside dia. (mm)	Min.wall thickness (mm)
≤ 50	3.5
>50 to ≤ 75	5.0
>75 to ≤ 100	6.0
>100 to ≤ 150	7.0
>150 to ≤ 200	8.0
>200	9.0

Figure 7.13 Table of nozzle body thickness requirements

7.2 Spacing of welds around connections

7.2.1 BS 2654 requirements

The BS Code requires that the distance between the toes of adjacent fillet welds or between the toes of fillet welds and the centre line of adjacent butt welds or between the centre lines of adjacent butt welds, shall not be less than 100 mm.

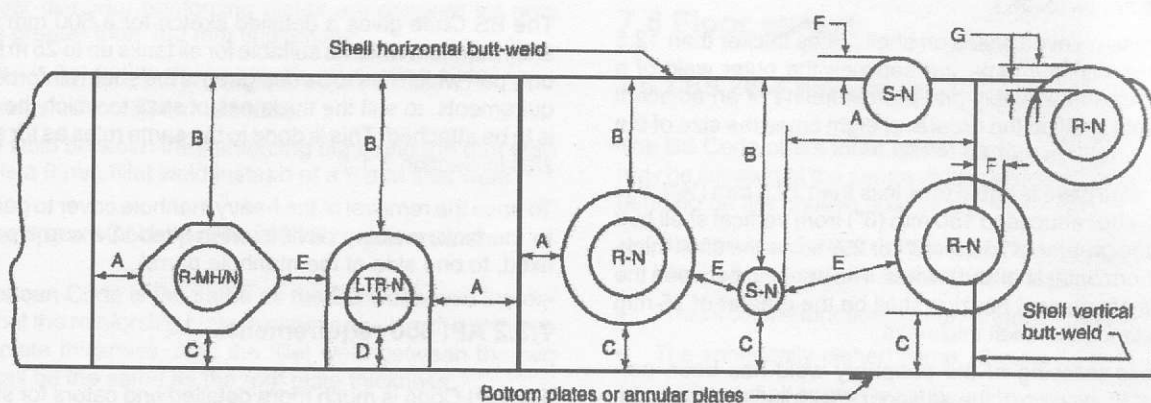
Welds to nozzle bodies shall not be closer to any weld which has been post weld heat-treated than:

$$2.5\sqrt{r_i \cdot t_p} \quad \text{equ 7.3}$$

where

t_p = wall thickness of the nozzle (mm)

r_i = inside radius of the nozzle (mm)



Note:

- R-MH/N = Reinforced Opening (manhole or nozzle with diamond shape reinforcing plate, see Figure 3-4A and 3-5).
 LTR-N = Low Type Reinforced Opening (nozzle with tombstone shape reinforcing plate, see Figure 3-5, Detail a and b).
 R-N = Reinforced Opening (manhole or nozzle with circular reinforcing plate or thickened insert plate, see Figure 3-5).
 S-N = Non-Reinforced Opening (manhole or nozzle inserted into the shell per the alternate neck detail of Figure 3-4B).

Variables		Reference	Minimum Dimension Between Weld Toes or Weld Centerline (1)(3)						
Shell t	Condition	Paragraph Number	A (2)	B (2)	C (2)	D (4)	E (2)	F (5)	G (5)
$t \leq 12.5 \text{ mm}$ ($t \leq 1/2 \text{ in.}$)	As welded or PWHT	3.7.3.2	150 mm (6 in.)	75 mm (3 in.) or $2\frac{1}{2}t$	75 mm (3 in.) or $2\frac{1}{2}t$ 75 mm (3 in.) for S-N	Table 3-6	75 mm (3 in.) or $2\frac{1}{2}t$	$8t$ or $1/2 r$	$8t$
		3.7.3.3							
		3.7.3.3							
		3.7.3.3							
		• 3.7.3.4							
$t > 12.5 \text{ mm}$ ($t > 1/2 \text{ in.}$)	As Welded	3.7.3.1.a	8W or 250 mm (10 in.)	8W or 250 mm (10 in.)	8W or 250 mm (10 in.) 75 mm (3 in.) for S-N	Table 3-6	8W or 150 mm (6 in.)	$8t$ or $1/2 r$	$8t$
		3.7.3.1.b							
		3.7.3.3							
		3.7.3.3							
		• 3.7.3.4							
$t > 12.5 \text{ mm}$ ($t > 1/2 \text{ in.}$)	PWHT	3.7.3.2	150 mm (6 in.)	75 mm (3 in.) or $2\frac{1}{2}t$	75 mm (3 in.) or $2\frac{1}{2}t$ 75 mm (3 in.) for S-N	Table 3-6	75 mm (3 in.) or $2\frac{1}{2}t$	$8t$ or $1/2 r$	$8t$
		3.7.3.3							
		3.7.3.3							
		3.7.3.3							
		• 3.7.3.4							

Notes:

- If two requirements are given, the minimum spacing is the greater value, except for dimension "F". See note 5.
- r = shell thickness. $8W$ = 8 times the largest weld size for reinforcing plate or insert plate periphery weld (fillet or butt-weld) from the toe of the periphery weld to the centerline of the shell butt-weld.
- For tanks designed to Appendix A, see A.5.2. Spacing = $2\frac{1}{2}t$ toe to toe of adjacent welds.
- D = spacing distance established by minimum elevation for low type reinforced openings from Table 3-6, column 9.
- Purchaser option to allow shell openings to be located in horizontal or vertical shell butt-welds. See Figure 3-6.

t = shell thickness, r = radius of opening. Minimum spacing for dimension F is the lesser of $8t$ or $1/2 r$.

Figure 7.14 Minimum weld requirements for openings in shells

From API 650, figure 3-22

7.2.2 API 650 requirements

The API Code is more detailed in its approach and the actual wording in clause 3.7.3. should be consulted. Basically the requirements are as follows:

For non stress-relieved welds on shell plates thicker than 12.5 mm ($\frac{1}{2}$ "), the minimum spacing between the outer weld of a nozzle, or nozzle assembly and the centreline of an adjacent shell butt weld shall be the greater of eight times the size of the outer weld, or 250 mm (10").

Where the shell plate is equal to or less than 12.5 mm ($\frac{1}{2}$ "), this spacing may be reduced to 150 mm (6") from vertical shell butt welds and the greater of 75 mm (3") or $2\frac{1}{2}$ times the shell thickness from horizontal shell butt welds. The spacing between the outer welds of adjacent nozzles shall be the greater of 75 mm (3"), or $2\frac{1}{2}$ times the shell thickness.

Where stress relieving of the periphery weld has been performed prior to welding of the adjacent shell joint, the minimum spacing shall be 150 mm (6") from vertical shell butt welds and the greater of 75 mm (3") or $2\frac{1}{2}$ times the shell thickness from horizontal shell butt welds. The spacing between the outer welds of adjacent nozzles shall be the greater of 75 mm (3"), or $2\frac{1}{2}$ times the shell thickness.

The Code contains a useful reference table in figure 3-22 which gives a pictorial representation of the application of the above rules. This is shown in Figure 7.14.

In certain instances it may be found that a nozzle has to be close to or even intersects a shell butt weld and the Code will allow this under rules given in figure 3-6. Where a shell joint is intersected, then 100% radiographic inspection of the weld is required for a distance of 1.5 times the diameter of the opening in the shell, measured each side of the centreline of the opening, except that the part of the shell joint which is being removed need not be radiographed.

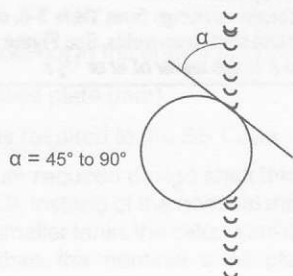
7.2.3 Flush type clean-out doors

The API Code is more flexible in its approach to the design of flush type clean-out doors.

The maximum size for the door opening is dependant on the grade of shell material being used (similar to the BS Code) but has more size options together with tabulated plate thickness and dimensional details. Formulae are given to calculate the required amount of reinforcement above the opening and to determine the thickness of the bottom reinforcing plate. Various methods are given to stiffen or support the bottom reinforcing plate under differing foundation support conditions and the designer is alerted to the requirement to allow for the rotation, due to shell bulging, of these low connections when they have pipework attached to them.

7.2.4 European Code requirements

prEN 14015-1 uses the same requirements as those for the BS Code but includes a further condition for nozzle openings in shell plates which intersect with shell butt welds. Where this condition occurs then the tangent to the opening in the shell at the centre line of the shell butt weld must be between 45° and 90° to the centreline of the butt weld as shown.



7.3 Shell manholes

7.3.1 BS 2654 requirements

The BS Code gives a detailed sketch for a 600 mm diameter shell manhole which is suitable for all tanks up to 25 m high. The only part which has to be designed is the shell reinforcement requirements, to suit the thickness of shell to which the manhole is to be attached. This is done to the same rules as for shell nozzles in Section 7.1.1.

To ease the removal of the heavy manhole cover to gain access to the tank, a swing davit is often fitted in a cup type bracket fixed, to one side of the manhole barrel.

7.3.2 API 650 requirements

The API Code is much more detailed and caters for shell manhole sizes of 500 mm, 600 mm, 750 mm and 900 mm diameter.

A general design sketch is given together with sketches of welded joint options. Tabulated data is also given for the following:

- Cover plate and bolting flange thickness for eight ascending design liquid levels up to a maximum of 23 m.
- Manhole neck thickness based on shell and reinforcing plate thickness ranging from 5 mm to 40 mm.
- Bolt circle and cover plate diameters for the four sizes of manhole.

Instead of a circular reinforcing plate, there is also an option allowing a six sided reinforcing plate the sides of which are at 45° to the horizontal centre line of the manhole.

7.3.3 European Code prEN 14015 requirements

The requirements given in this Code are the same as those in the BS Code.

7.4 Roof nozzles

7.4.1 BS 2654 requirements

The BS Code shows a sketch of a typical roof nozzle together with tabulated dimensions for nozzle sizes from 25 mm to 300 mm diameter.

The duty of roof nozzles is not very arduous and their integrity does not pose a serious threat to the soundness of the tank. Roof nozzles are therefore lighter in construction than shell nozzles. The reinforcement of the aperture in the roof plating for all nozzle sizes is 150 mm larger than the aperture in the roof plating and in all cases is made from 6 mm thick plate. Welding of the nozzle on the underside of the roof is not required. All welded joints on the nozzle are 6 mm fillet welds, regardless of the size of the nozzle.

The Code recommends that the necks of nozzles used for venting should be trimmed flush with the underside of the roof line. This is to ensure that vapour is not trapped by the neck which would otherwise protrude below the roof line. The polar axis of roof nozzles should always be vertical.

Although not mentioned in the Code it is generally thought to be good practice to use flat-faced flanged roof nozzles with full face gaskets for roof vents and other fittings which may be of cast iron or aluminium construction.

7.4.2 API 650 requirements

The requirements to the American Code are very similar to those of the BS Code with the following main exceptions:

- 1) Larger diameter reinforcing plates are required for nozzles greater than 100 mm in diameter.
- 2) There is the option not to provide reinforcing plates for nozzles up to 150 mm diameter.
- 3) The weld between the reinforcing plate and the roof plating is a 5 mm fillet weld instead of a 6 mm fillet weld.

7.4.3 European Code prEN 14015 requirements

The European Code is the same as the BS Code with the exception that the reinforcing plate thickness shall be the same as the roof plate thickness, also the fillet weld between the two plates shall be the same as the roof plate thickness.

7.5 Roof manholes

7.5.1 BS 2654 requirements

The BS Code is very sparse in its guidance on roof manholes, this guidance being as follows:

"The roof manholes shall have a minimum inside diameter of 500 mm. They shall be suitable for attachment by welding to the tank roof sheets.

The manhole covers shall be either as specified by the purchaser or of the multiple-bolt fixed or hinged type."

Because of the vagueness of the requirements, designers generally turn to the more detailed information given in the American Code.

From a practical point of view it is important to avoid the use of ASA 150 lb covers and flanges for roof manholes because of their excessive weight.

7.5.2 API 650 requirements

This Code gives a detailed illustration and tabular information for the design of roof manholes 500 mm and 600 mm in diameter. They are of relatively light construction being in 6 mm plate. The provision of a reinforcing plate is optional.

This Code also gives full details for two types of rectangular roof openings, one with a bolted cover and one with a hinged cover with one locking point. Both types are limited to a maximum opening size of 1800 mm x 900 mm, the provision of reinforcing plates is optional and they are intended for use on fixed steel roofs only (not floating roofs).

Again these rectangular openings are of light construction, the necks and optional reinforcing plates being 6 mm thick, the cover plates 5 mm thick and the flange of the bolted type being 10 mm thick.

The bolted type is limited to tanks having a maximum internal pressure equal to the weight of the roof plates and the hinged type is for use on non-pressure tanks only.

7.5.3 European Code prEN 14015 requirements

This Code follows the BS Code but is more specific as it gives dimensions for 500 mm and 600 mm diameter manholes but does not specify steel thickness.

The illustration in the Code shows the neck and bolting flange as if being rolled from one plate, this is unlikely to be the preferred method of construction and it is more likely that the two

components would be welded together.

The Code also mentions that the manhole covers can be of the multiple bolt type or hinged.

7.6 Floor sumps

7.6.1 BS 2654 requirements

The BS Code offers three types of drain sumps. These sumps may be situated at the centre of the floor or at the periphery, depending on the chosen floor slope. They are:

- The combined water draw-off and clean-out sump (see Figure 7.11, can only be fitted at the periphery.)
- The circular-fabricated sump, (Figure 7.15)
- The spherically-dished sump, (Figure 7.16)

The bottom of all sumps must be adequately supported by the underlying tank foundation to ensure that they do not "hang" off the floor aperture and cause stress in the flange connecting the sump to the floor plating.

The fabricated sump tends to be more popular with tank fabricators because difficulties can be encountered in trying to obtain pressings of the correct dimensions for the spherical type. However, the spherical sump is made out of one piece of plate and therefore has no potential to leak. The fabricated sump welds must be subjected to rigorous inspection to ensure that they are truly sound.

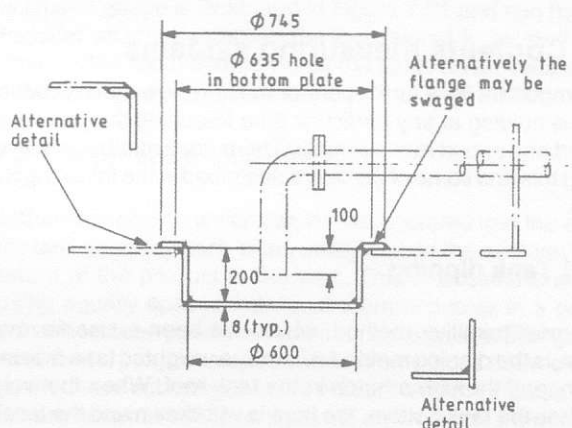


Figure 7.15 Circular-fabricated sump

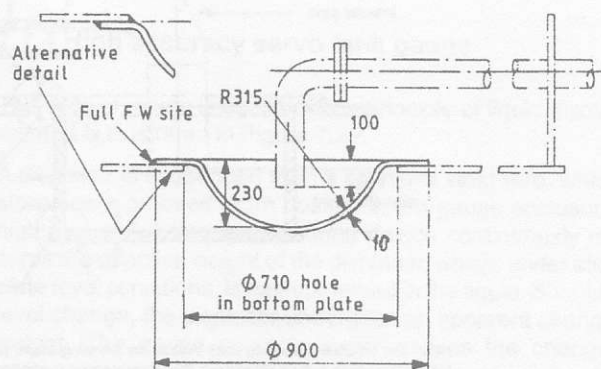


Figure 7.16 Spherically-dished sump

7.6.2 API 650 requirements

The API Code offers details for four sizes of sump each based on the size of the drain line.

Brief details taken from the tabulated data in the Code are shown in Figure 7.17.

Size of drain line Nominal bore (mm)	Diameter of sump (mm)	Depth of sump (mm)
50	610	300
80	910	450
100	1220	600
150	1520	900

Figure 7.17 Details for four sizes of sump based on size of drain line

It can be seen that these sumps are somewhat larger than the BS Code sumps, especially those for the larger sized drain lines.

The fabrication detail for these sumps is shown in Figure 7.18, which is reproduced, from the Code.

The API Code gives positions for the sumps measured from the shell of the tank which indicate that they are close to the shell but, if required, they may be placed anywhere in the floor to suit the floor drainage requirements.

7.6.3 European Code prEN 14015 requirements

The sump requirements here are the same as those for the BS Code.

7.7 Contents measuring systems

It is important for a tank operator to know how much product a tank is holding at any particular time to enable the planning of import and export requirements. There are a number of ways of doing this and some of these are described in the following Sections.

7.7.1 Tank dipping

The most primitive method, which has been in use for many years, is the dipping method whereby a weighted tape measure is dropped through a hatch in the tank roof. When the weight touches the tank bottom, the tape is withdrawn and the level of

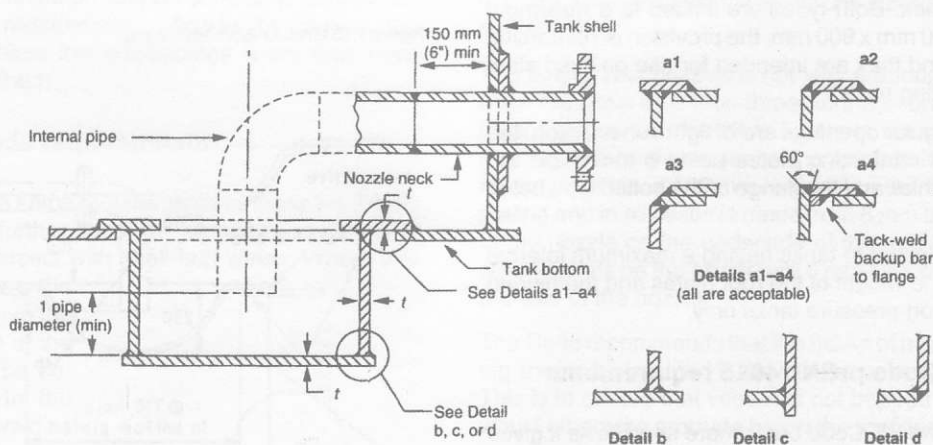
liquid is read from the tape at the point where the tape changes from being dry to wet. There is an art in obtaining a correct dip by this method because of the following factors:

- Care must be taken to ensure that the weight only just touches the tank bottom, as allowing further tape into the tank will give a false increased reading in the dip depth. With experience, tank dipping personnel learn to feel for the tank bottom and can obtain reliable repeatable results.
- Judging the point where the tape changes from dry to wet may be fairly easy when dipping a tank containing, say molasses, but not so easy with light distillate products. Compounds have been developed which can be applied to the tape in the area where the expected level is thought to be and these show more clearly where the dry to wet point is on the tape, hence resulting in a more accurate reading.

There are several types of roof nozzle dip hatches on the market and a selection is shown in Figure 7.19.



Figure 7.19 Different types of roof nozzle dip hatches
Courtesy of Endress+Hauser Systems & Gauging Ltd



Note: The erection procedure shall include the following steps: (a) a hole shall be cut in the bottom plate or a sump shall be placed in the foundation before bottom placement; (b) a neat excavation shall be made to conform to the shape of the drawoff sump, the sump shall be put in place, and the foundation shall be compacted around the sump after placement; and (c) the sump shall be welded to the bottom.

Figure 7.18 API Water draw-off sump

From API 650

7.7.2 Level indicators

There are a number of proprietary mechanisms on the market, which are capable of constantly monitoring the level of product in the tank, and a few of these are as follows:

7.7.2.1 Float, board and target system

This method is not very accurate but it gives a good indication of where the liquid level is in a tank.

A graduated board is attached to the tank shell over the full height of the tank. A float sits on the product and is kept in place by two guide wires which pass through eyes one on each side of the float. The guide wires are stretched taut between the floor and roof of the tank and a flexible stranded wire attached to the float is led over the top of the tank by pulleys. It is led to a target pointer, which is guided to move up and down the graduated board as the level of the product changes. It is important to remember however, that when the target is at the bottom of the graduated board, the tank is full (and not empty, as logic may seem it to be) and vice versa.

The illustration in Figure 7.20 shows the workings of this type of level gauge.

7.7.2.2 Automatic tank gauge

This system is a vast improvement on the above board and target arrangement and operates as follows:

The float is guided between guide wires as in the above example but in this case a flexible tape is attached to the float and this tape is fed through small-bore piping and pulley elbows supported off the roof and shell of the tank and is led to a gauge head near the base of the tank. A spring-loaded mechanism in

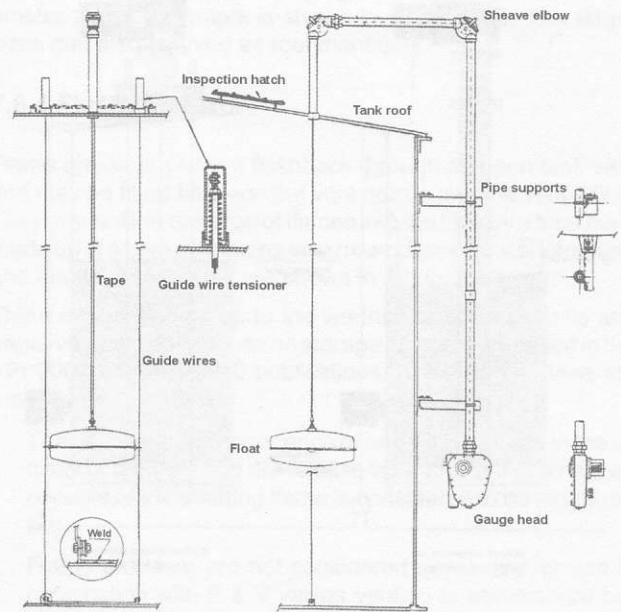


Figure 7.21 Automatic tank gauge

Courtesy of Endress+Hauser Systems & Gauging Ltd

the gauge head allows the tape to coil and uncoil as the product level changes and a series of pulleys and sprockets in the gauge head are connected to a drum which gives a visible read-out in metres and millimetres in a window on the gauge head.

This type of gauge is illustrated in Figure 7.21 and can have a transmitter attached enabling the level signal to be sent to a central control room and hence all the tanks on an installation can be monitored in this way.

7.7.3 Temperature measurement

A further refinement, which can be incorporated into the automatic tank gauge system, is the ability to read the average temperature of the product in the tank. This is accomplished by housing equally spaced individual thermocouples in a perforated vertical tube positioned near the level gauge. The gauge mechanism is programmed to switch in only those thermocouples, which are submerged in the product, and the signals from these are automatically averaged out and read on a monitor in the control room.

This facility is useful to operators as it enables volumetric adjustments to be made to their product inventory to allow for temperature variations.

7.7.4 High accuracy servo tank gauge

This type of gauge is based on the principle of liquid displacement. It is illustrated in Figure 7.22.

A displacer is suspended from a stainless steel wire, which is stored on a grooved drum housed in the gauge enclosure. A highly sensitive torque-measuring device continuously measures the effective weight of the displacer, which, under steady state level conditions, is half-immersed in the liquid. Should the level change, the displacer undergoes an apparent change of weight. The gauge microprocessor senses the change in weight and causes a servomotor to rotate the measuring drum until balance is restored.

Density is determined by measuring the effective weight of the displacer when completely immersed. Sample readings, re-

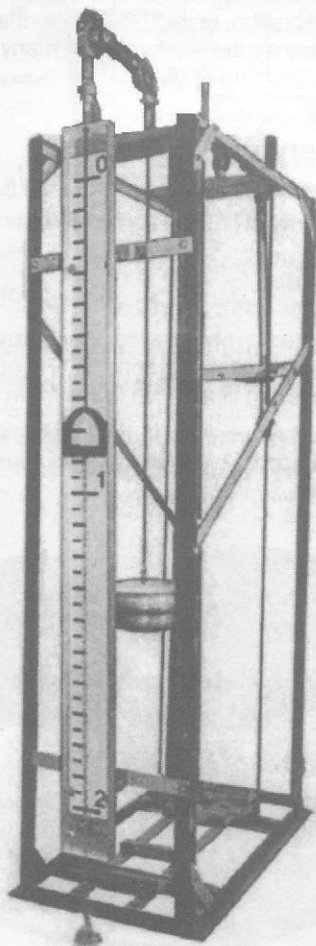


Figure 7.20 Float, board and target level gauge
Courtesy of Motherwell Control Systems Ltd

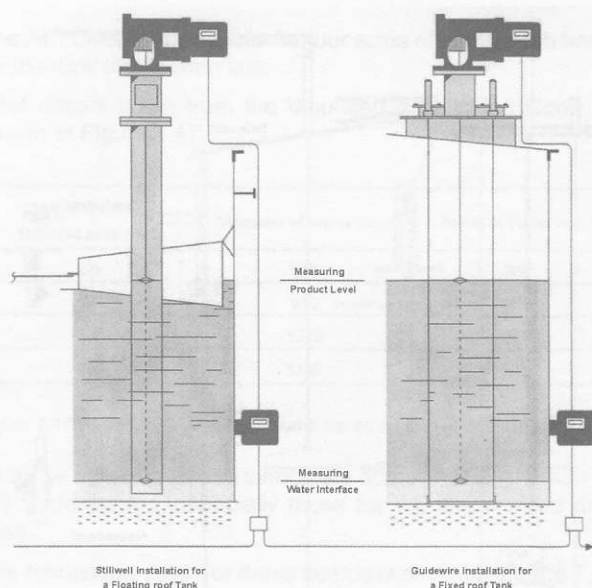


Figure 7.22 High accuracy servo tank gauge
Courtesy of Motherwell Control Systems Ltd

corded at configurable intervals as the displacer, travels down through the liquid, provide density profiling.

Water interface level and tank base measurement are achieved by recording the point at which the gauge recognises the effective displacer weight in water and at the tank base respectively.

7.7.5 High accuracy radar tank gauge

This type of gauge, (see Figure 7.23), achieves level measurement by measuring the time of flight for a radio wave to travel from the radar gauge to the liquid surface and back again.

Normally the gauge is mounted at the top of the tank with its antenna pointing down towards the surface of the stored product.

RADAR TANK GAUGE INSTALLATION DETAILS

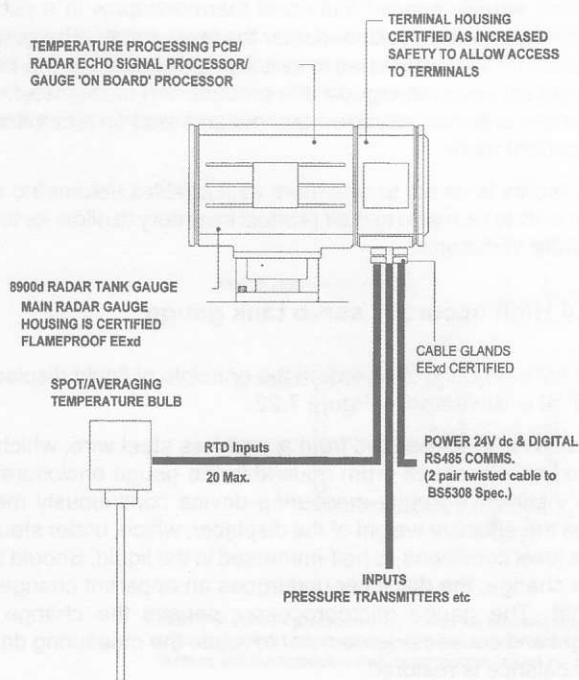


Figure 7.23 High accuracy radar tank gauge
Courtesy of Motherwell Control Systems Ltd

The radio wave signal is emitted from the rod antenna and radiates outwards "seeing" all the tank internals. The reflected radio wave is then collected by the same antenna and the gauge compares the difference in frequency between the outward and return radio waves. The frequency difference is proportional to the distance travelled. This frequency difference then undergoes a number of processes including Fourier transform techniques and peak location algorithms which are then used to digitally locate the peak frequency corresponding to the product level reflection from which the liquid level is then calculated and displayed on a liquid crystal display inside the unit.

Having established the level of product in the tank, this has then to be translated into a capacity and this is done by reference to the tank's calibration table whereby capacities can be read off a table in 1 mm level increments.

Each tank, on completion is calibrated by a specialist company. The earliest form of calibration was by the "strapping method". This method, amongst others, is governed by rules set down by the Institute of Petroleum, see *Reference 7.2*.

This method involved the circumference of the tank being strapped with a measuring tape at many points over its height, enabling the diameter of the tank to be calculated at each level and hence the capacity relating to each measurement, established. The volume at the bottom section of the tank which often contains drain pipework, heating coils etc. (known as deadwood), can be found by filling it with water, which is metered into the tank and recorded against corresponding depths.

More modern laser measuring methods are used nowadays which operate from inside the tank, they are much less labour intensive and give very accurate diameter measurements over the height of the tank.

Her Majesty's Customs officials take a great deal of interest in correct tank calibration, level measurement and the recording of tank capacities as the movement of many petroleum products incurs the payment of duty.

7.8 Tank venting

This subject is dealt with in detail in Chapter 8 so just a brief description of the vent fittings is given here.

7.8.1 Free vents

These are provided on non-pressure tanks and allow the tank to breathe due to product movements in and out of the tank and for diurnal effects. An illustration is shown in Figure 7.24.

Sometimes the free vent fitting incorporates a dip hatch, enabling one roof nozzle to be used for two purposes.

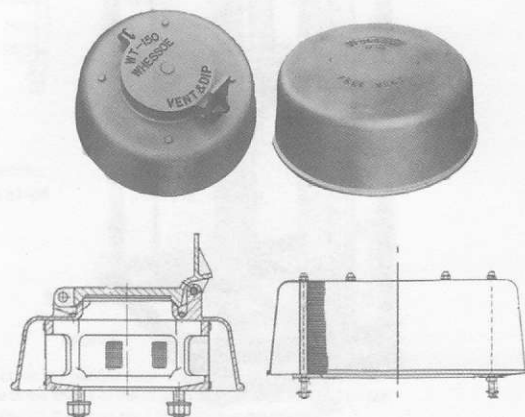


Figure 7.24 Free vent & dip hatch
Courtesy of Whessoe Varc

7.8.2 Pressure and vacuum (P & V) valves

These are used for tanks operating under an internal pressure. The vent opens only when the set internal pressure is exceeded - for instance, when product is imported to the tank. On the vacuum side, the valve opens when the set internal vacuum is exceeded, as is the case when product is exported from the tank.

The illustrations in Figure 7.25, show a valve which uses weighted pallets as the valve operating mechanism, other types of valve use a spring-loaded method.

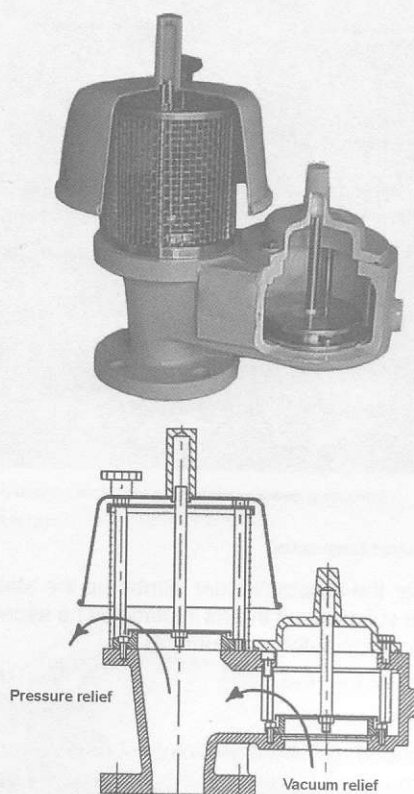


Figure 7.25 Pressure and vacuum relief valve
Courtesy of Tyco Valves & Controls

7.8.3 Emergency vents

The purpose of an emergency vent is to release a sudden rise in internal pressure which is beyond the capacity of the normal vents. Their use is dependant upon the type of product being stored in the tank and whether or not the tank has a frangible roof. A sudden rise in pressure may be caused by events such as an external fire, a burst heating tube or an exothermic reaction in the tank.

The emergency vent consists of a base unit with a weighted hinged cover. The seal between the base unit and the cover is maintained by the knife-edged rim of the base unit acting on a

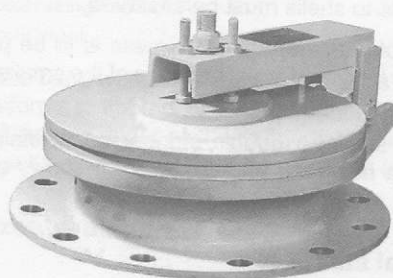


Figure 7.26 Emergency vent
Courtesy of Tyco Valves & Controls

flexible seal ring on the underside of the weighted cover. These units are available in sizes ranging from 250 mm to 600 mm diameter and an example is shown in Figure 7.26. The larger sizes can also be used as roof manholes.

7.8.4 Flame Arrestor

Flame arrestors prevent flashback through an open tank vent and may be fitted between the vent nozzle and the vent fitting. They prevent the passage of flames into the tank by a tube bank made up of a core of numerous narrow passages. A photograph and diagrammatic view are shown in Figure 7.27.

There is some doubt as to the worthiness of these units and negative views on their use on storage tanks is expressed in the API 2000 and API 2210 publications. Some of the views expressed are as follows:

- The simultaneous occurrence of an ignition source in the vicinity of the vent and the release from the vent of a mixture capable of transmitting flame is considered to be highly unlikely.
- Flame arrestors are not considered necessary for use in conjunction with P & V valves venting to atmosphere because flame speeds are less than vapour velocities across the seats of P & V valves.
- Friction loss through the flame arrestor reduces the flow rate through the vent fitting.
- The narrow vapour passages of the flame arrestor can block up and thus cause pressure or vacuum related damage to the tank envelope.

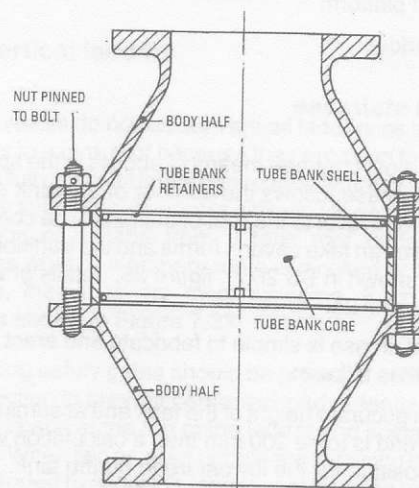


Figure 7.27 A typical flame arrestor
Courtesy of Tyco Valves & Controls

7.9 Tank access

For safety reasons a tank should have two means of egress from the roof. For a single tank, which is not interconnected with another, then the second means of access is usually by a vertical-caged ladder.

The BS 2654, API 650 and prEN 14015-1 Codes all specify similar design requirements for access ways but in using these the designer must also be aware of any local and/or client requirements and safety issues.

Whilst there are some differences between the tank Codes, the principal requirements are as follows:

- Minimum clear width of a staircase, platform or walkway shall be 600 mm.
- Minimum height to the top handrail of a horizontal platform or walkway shall be 1070 mm.
- Minimum depth of a stair tread shall be 200 mm.
- Maximum slope for a staircase 45° (50° in API)
- Handrailing is required on the inside stringer of a spiral staircase where the gap between the stringer and the tank shell exceeds 200 mm.
- The normal "going" and "rise" for treads of a spiral staircases is 200 mm.
- The maximum vertical rise between intermediate platforms of a staircase is 6 m.
- API requires the design to be based on a concentrated moving load of 4450N, whereas the BS Code requirement is for the design to be based on a load of 2400 N/m² plus wind load.
- Handrailing is to be capable of taking a load of 1000 N (890N to API) in any direction.
- Treads which are welded to the shell are prohibited by the BS and European Code for shell thicknesses over 12.5 mm on steel having a UTS greater than 460N/mm² (Yield 275 N/mm²)
- Vertical ladders over 4m high shall be fitted with safety cages. BS 4211 allows a maximum height between intermediate platforms of 9m but it is normal to limit this to 6m on tanks.

Four means of accessing tanks will be considered:

- Spiral staircase
- Radial staircase
- Horizontal platform
- Vertical ladder

7.9.1 Spiral staircase

Probably the most common means of access is the spiral staircase. This staircase follows the contour of the tank shell as it rises from ground level to the roof of the tank. The construction of the staircase can take several forms and the traditional one is that which is shown in BS 2654, figure 25, details of which are shown in Figure 7.28.

This type of staircase is simple to fabricate and erect. Erection on the tank is as follows:

- Obtain an accurate height of the tank and assuming the rise of each tread is to be 200 mm then a calculation will establish the position for the lowest tread on the tank.
- The first eight or so treads can be welded to the shell together with the 25 mm square bar supports (known in the tank business as "dog leg" supports) from ground level.

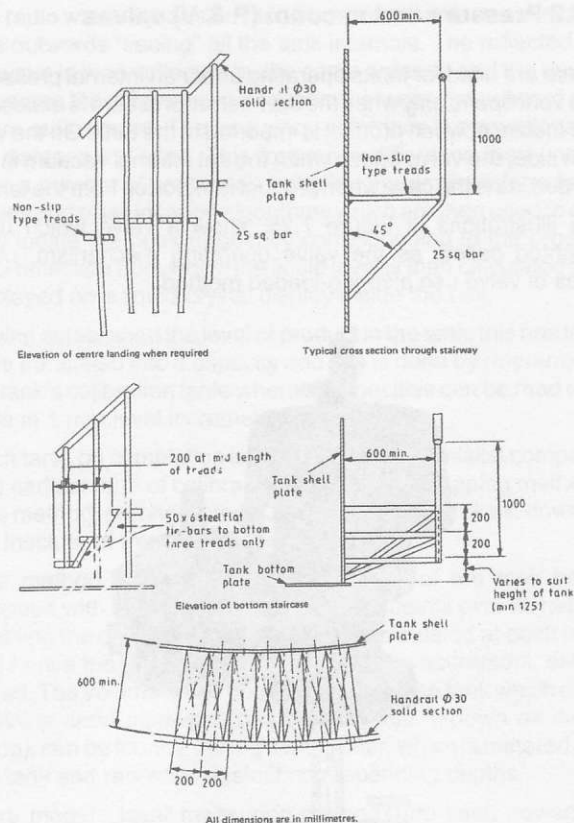


Figure 7.28 Handrail construction

Thereafter the erector/welder climbs up the staircase and welds the subsequent treads in place as he ascends (using the appropriate safety equipment).

There are long-term disadvantages with this type of staircase, and these are:

- Being welded directly to the shell makes corroded treads difficult to replace (galvanised treads cannot be used because of the health risk in welding on to a galvanised surface).
- Where tank shells are thermally insulated, there are numerous penetrations in the cladding where the dogleg supports and treads pass through and offer a path for the rain to get in and cause corrosion on the shell.
- The tread replacement issue can be solved by using bolt on treads where a short length of drilled angle bar is welded toe on to the shell to which the tread is bolted. Similarly at the outer tread support a short length of drilled flat bar is welded to the support to carry the tread.
- Because of the shortcomings of the weld on staircase, most spiral staircases today are constructed with an inner and outer stringer and bolted galvanised treads. The stringers are supported off brackets welded to the shell but the limitations in the Codes regarding the welding of permanent attachments to shells must be observed.
- The double stringer spiral staircase is to be preferred for thermally insulated tanks because of the smaller number of penetrations in the cladding.

Figure 7.29 shows a double stringer spiral staircase being erected on a new tank and Figure 7.30 shows a completed staircase.

7.9.2 Radial staircase

This type of staircase is often used to access large diameter tanks, which have large bundled areas. The staircase com-

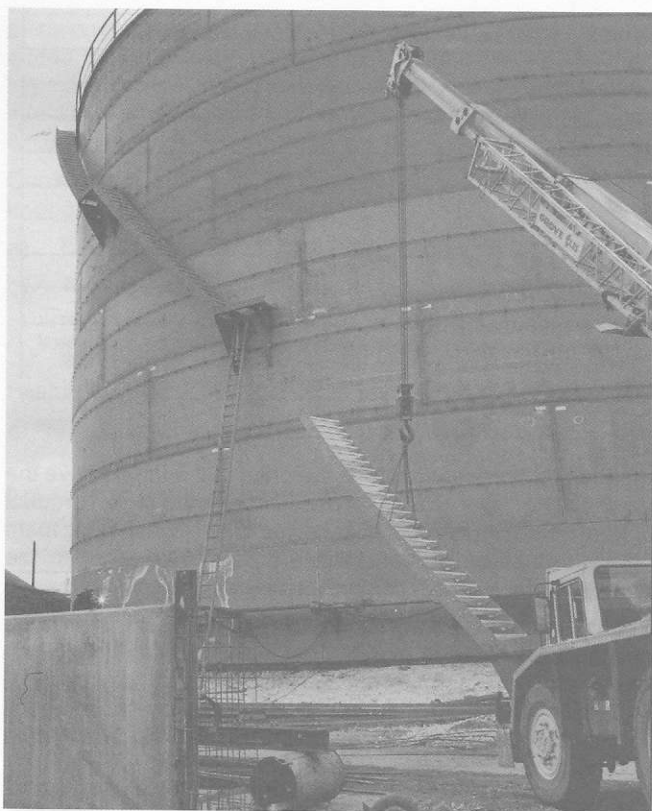


Figure 7.29 A double stringer spiral staircase being erected
Courtesy of McTay

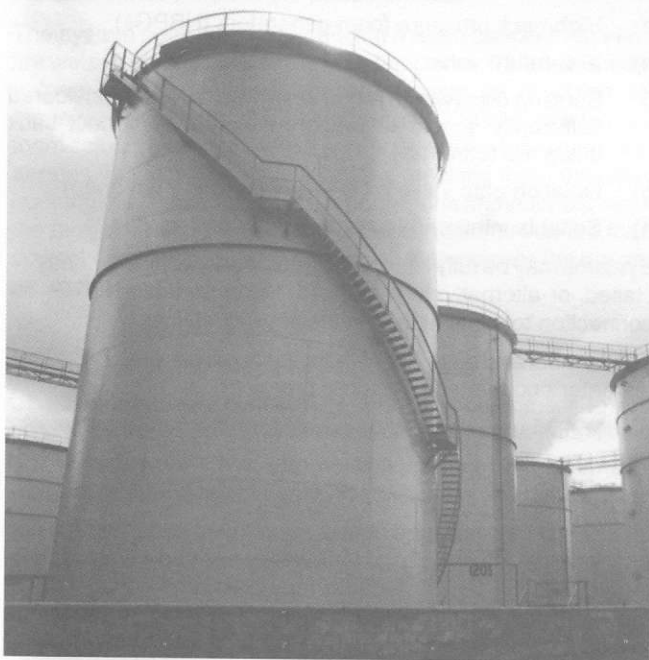


Figure 7.30 Double stringer spiral staircase
Courtesy of Royal Vopak

mences at the bund wall and progressively rises via the intermediate platforms to the tank roof. Support for the staircase is usually by 'A' frames under each intermediate platform. Figure 7.31 shows a typical arrangement on a floating roof tank.

7.9.3 Horizontal platforms

This form of access, shown in Figure 7.32, is favoured on multi-tank installations where the tanks are linked together by platforms and only the extremity tanks each have a spiral or ra-

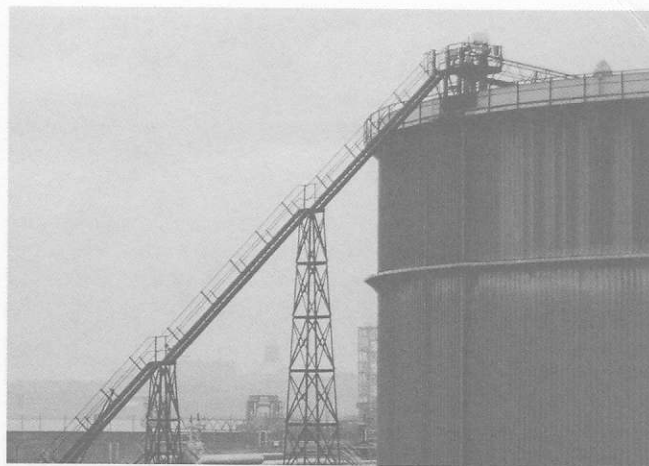


Figure 7.31 Radial staircase on roof tank



Figure 7.32 Horizontal platforms
Courtesy of Royal Vopak

dial staircase for access from the bund area.

The platforms have to allow for movement of the tanks due to product and wind load and foundation settlement. One end of the platform is therefore fixed to hinged brackets on one tank, which allow vertical movement; the other end is restrained laterally but allowed to slide in the horizontal direction to allow for tank movement. Safety chains are connected loosely between this end of the platform and the adjacent tank to prevent the platform falling in the event that there is excessive movement between the tanks.

7.9.4 Vertical ladders

Tank operators do not favour vertical ladders as a main means of access to a tank roof because they are tiring to climb and require the full use of all limbs during the ascent, hence the carrying of any sundry equipment is difficult. However, as a secondary means of escape from a tank roof under emergency conditions when the primary route is blocked or otherwise unavailable, then they are most welcome. Such a means of access is shown in Figure 7.33.

Self-closing safety gates should be provided at the top of each ladder section to prevent personnel inadvertently stepping into the open space at the top of the ladder and sustaining a nasty accident. When two or more people are following each other it is recommended to allow the ladder section to be cleared by one person before the next one starts their ascent or descent. This prevents any boot detritus, equipment or person from falling on to the person below.

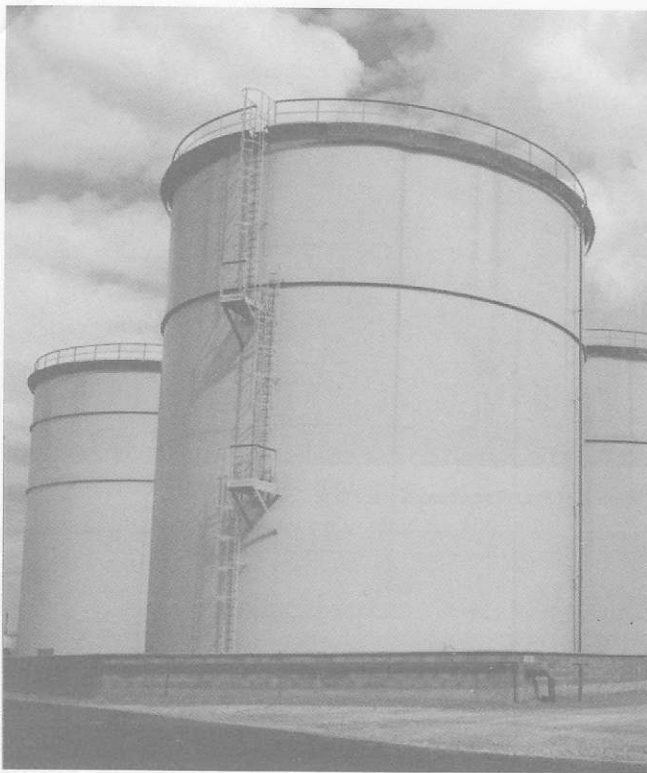


Figure 7.33 Vertical ladder
Courtesy of Royal Vopak

7.10 Fire protection systems

As one can readily understand, the planning for the prevention of fire, especially in petrochemical installations, is high on their managements' priority list, as the consequences of an inferno can have disastrous results, not only to the installation but to the surrounding area and environment.

The subject is well-documented in the National Fire Protection Association, Institute of Petroleum and British Standard Codes. *References 7.3 to 7.6* provide useful information on this important issue.

For the purposes of this Section the protection of storage tanks by the use of foam and water will be considered.

7.10.1 Foam systems

The foam methods considered to be the most widely used and regarded to give an acceptable overall level of protection are referred to in this Section. The design guidelines are to be found in *References 7.3 to 7.6*.

The foam fire fighting system works by introducing a foam making concentrate into the fire fighting water main. This produces a solution, which is fed to a foam generator, and the resulting foam is directed to the fire.

For fixed roof, floating roof and Internal floating roof storage tanks there are three principal foam systems available and these are: base injection, top foam pouring and foam cannons.

These systems are categorised in Figure 7.34.

7.10.1.1 Base injection

Base injection systems (also known as sub-surface foam injection systems) are suitable for use on fixed roof tanks containing liquid hydrocarbons with the exception of Class 1A hydrocarbon liquids or alcohols, esters, ketones, aldehydes, anhydrides, or other products requiring the use of alcohol-resistant foams. In operation, specialised equipment designed to operate against a back pressure introduces aspirated foam at a pre-

Type of Storage tank	Type of Foam system
Fixed roof	Base injection
	Top foam pourers
	Foam cannons *
Floating roof	Rimseal foam pourers
	Top foam pourers
	Foam cannons *
Internal floating roof	Top foam pourers
	Foam cannons *

* Foam cannons are placed external to the tank, in such a position that, in the event of a fire, foam can be sprayed on to the tank from a safe distance. This method is not recommended as the primary form of protection for tanks over 18 m in diameter

Figure 7.34 Principal foam systems

determined application rate at the base of the tank, above the bottom water layer. The foam rises through the stored product to form an extinguishing blanket at the surface. The rising foam causes rotational currents, which carry cold product to the burning surface, which can aid extinction.

The concept of base injection only became possible with the development of fluoro protein type foam concentrates, which have high resistance to product contamination and good fluidity. Additionally the finished foam must have excellent burnback resistance (the ability of a foam blanket to resist direct flame and heat impingement) and stability.

The system requirements are:

- A pressurised supply of fresh or sea water
- Suitable foam concentrate induction equipment to produce a 3% solution of foam concentrate
- Foam concentrate storage facilities
- High back pressure foam generators (HBPGs)
- Non-return valve
- Bursting disc (where a non-return valve is not considered sufficiently secure to prevent leakage of product back along the foam line)
- Isolation gate valve on the tank (normally left open)
- Suitable interconnecting pipe work and valving

Systems may be fully fixed with all components permanently installed, or alternatively semi-fixed, using portable HBPGs for connection to suitable tank inlets or product lines.

The number and diameter of foam inlets will depend on the tank diameter and the type of stored product.

Figure 7.35 may be used as a guide for the number of inlets.

The minimum foam application rate is 4.1 litres/min/m² (0.1 gpm/ft²) and this rate will decide the size of the foam inlets.

Inlets must be positioned above any water layer in the tank and may terminate flush with the tank wall or be fitted with stubs protruding into the tank. The latter may discharge horizontally or may be angled vertically. Discharge downwards should be avoided, particularly if the foam can enter a water bottom or im-

Tank diameter (m)	Flash point < 37.8° C	Flash point = >37.8° C
Up to 24	1	1
>24 to 36	2	1
>36 to 42	3	2
>42 to 48	4	2
>48 to 54	5	2
>54 to 60	6	3
>60 one additional inlet for each additional:	465m ² of exposed product surface	697m ² of exposed product surface

Figure 7.35 Number and diameter of foam inlets

pinge on the base of the tank. Where more than one inlet is required, they should be spaced equally around the tank shell, using either separate inlets, or alternatively a single inlet feeding into an internal manifold with outlet pipes towards the tank circumference.

Correct design will take into account pressure losses in the following areas:

- Friction loss in pipe work, fittings and valves
- The maximum static head of the stored product
- Pressure loss through the foam induction equipment and foam generators

Features of the base injection system include:

- Rapid response with minimum demand on resources, water supply, foam compound and manpower.
- Design application rates of foam are achieved with 100% of the foam reaching the surface of the stored product.
- High resistance of the system components to damage during tank explosion or fire.
- Circulation of cold product dissipates hot product layers near the burning surface and aids extinction.

A schematic of a base injection system is shown in Figure 7.36.

The selection of HBPGs and foam concentrate requirements are by reference to data produced by the manufacturers of the proprietary equipment and foam concentrates.

7.10.1.2 Top foam pourers

Top foam pouring systems are used to protect fixed roof tanks and fixed roof tanks fitted with internal covers. In each case the systems are designed on the basis that the fire risk comprises the total surface area of the stored product.

The system operates by introducing a foam concentrate into a fire water feed line outside the tank bund area. This line is led to a foam generator, foam box and pourer all of which are mounted in line at the top of the tank shell. When initiated, the foam solution is propelled to the tank where the foam generator aerates the solution and delivers the resulting foam through a bursting disc in the foam box. A pourer unit immediately inside the tank shell and connected to the foam box, directs the foam down the shell to form a blanket which extinguishes the burning product

The system requirements are:

- A pressurised supply of fresh or seawater
- Suitable foam concentrate induction equipment to produce the required percentage of foam concentrate in water
- Foam concentrate storage facilities
- Foam generator (immediately under the foam box)
- Foam box with bursting disc (this prevents tank vapours

escaping via the foam pipework)

f) Foam pourer

Normally each of the fixed tank shell units are supplied by individual lines from a safe area outside the tank bund but they can be supplied by one line to the tank which splits at a manifold to feed each unit.

The number of foam inlets is as shown in Figure 7.35 and this, together with a minimum foam application rate of 4.1 litres/min/m² (0.1 gpm/ft²) will determine the size of the foam inlets. The foam solution flow through each inlet should be similar. By dividing the total minimum foam solution application rate by the minimum number of inlets required, the flow rate per pourer unit is established.

Certain low boiling point flammable stored products, gasohols and high viscosity heated liquids may require higher or, in certain circumstances, lower application rates than that stated here. These should, in all instances, be determined by test.

Design notes

If two or more inlets are required they should deliver the foam at the same rate to the surface of the tank and that they are arranged at equal spacing around the shell.

All pipe work, valving and riser systems should be designed to give approximately equal flow rates from each pourer.

Tests have shown that foam will travel effectively across at least 30 m of exposed burning product surface. Thus on very large tanks, it may be necessary to increase the number of pourer units above the minimum recommended number.

The foam inlets to the tank should be 300 mm above the maximum designed product storage level.

Cautionary note

In the event of an explosion in a tank causing ruptures at the roof-to-shell joint and distortion in the upper shell plating, if this is in the area of any of the foam units, these units may be rendered ineffective.

Protection of bitumen storage tanks

For fixed protection on bitumen tanks the only suitable systems are inert gas or steam injection into the vapour space. Water must not be used as this is likely to result in a hazardous, uncontrollable froth-over or a steam explosion owing to the vaporisation of the water at the high storage temperatures used for bitumen. For further information refer to *Reference 7.6*.

Illustrations and examples of top foam pourers are shown in Figures 7.37 to 7.39.

7.10.1.3 Rimseal foam pourers

The basis of this system has already been described in Chapter 6, Section 6.5.6.

The concept of a rimseal protection system is based on the assumption that, in the event of a fire, the fire will be contained in

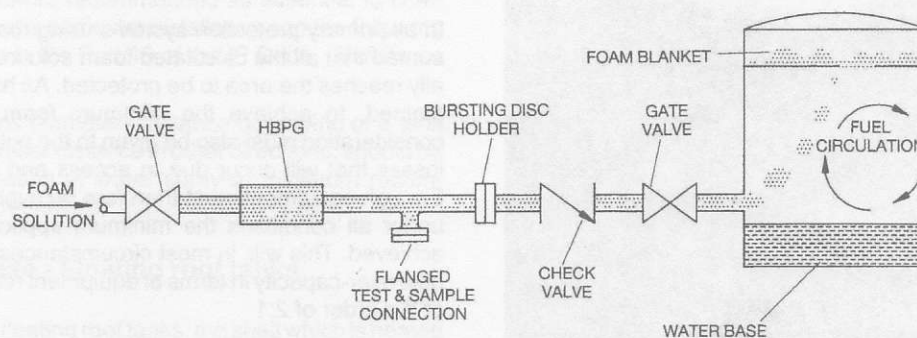


Figure 7.36 Base injection system schematic
Courtesy of Angus Fire

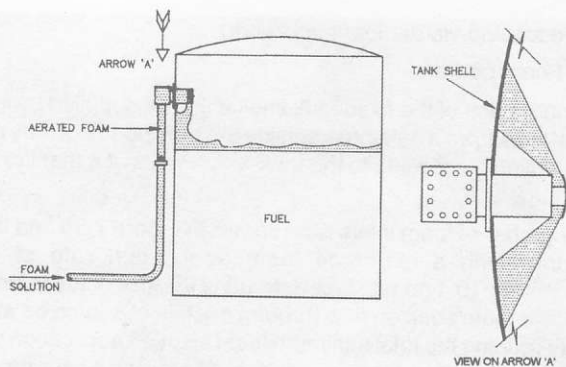


Figure 7.37 Top foam pourer schematic
Courtesy of Angus Fire

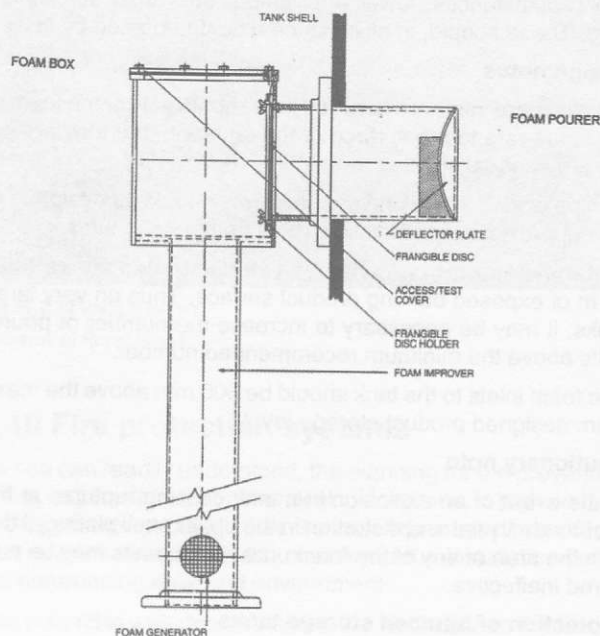


Figure 7.38 Top foam pourer unit
Courtesy of Angus Fire

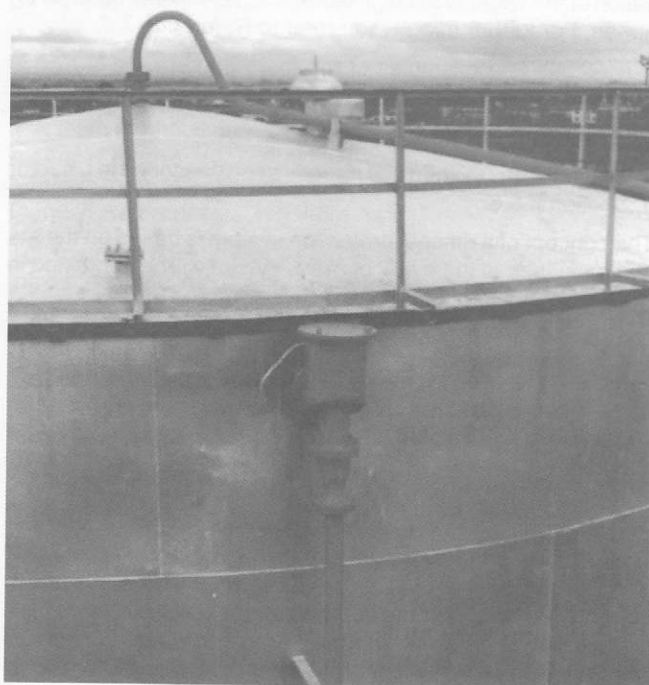


Figure 7.39 Foam pourer and water deluge pipework (at centre of tank)

the seal area between the foam dam and the tank shell and the system design is based on treating only this annular area. This means that if a fire should occur it must be detected early and tackled rapidly before the roof becomes damaged and allows the fire to spread — often to the extent of engulfing the entire surface area. Should a situation arise in which the fire does spread to the whole exposed surface area then a rimseal protection mechanism alone (as dictated by design of the system) is unlikely to achieve extinguishment. If this is perceived as a possibility, then consideration should be given to a top pouring system designed to provide total coverage of the roof area.

The minimum recommended foam solution application rate for rimseal systems is 12.2 litres/min/m².

The minimum number of rimseal foam pourers is dictated by the height of the foam dam and is as follows:

- For a 300 mm high foam dam the maximum spacing between foam pourers should be 12.2 m.
- For 600 mm high foam dams this can be increased to a maximum of 24.4 m.

7.10.1.4 Foam cannons

Fixer and trailer-mounted foam cannons are suitable for protecting all types of vertical storage tanks and though subject to performance limitations they can be used as the primary protection system to protect tanks up to 18 m in diameter. However, they are often better suited and more commonly installed as either a secondary fixed foam system or to tackle spill fires with the added benefit of being able to be used for tank cooling. A foam cannon in operation is shown in Figure 7.40.

The single most important consideration when proposing foam cannons as the primary system is that, to be effective, expanded finished foam must first be delivered to the seat of the fire. As, in most systems, the foam cannons will be close to ground level, the foam produced will first be required to reach up and over the tank shell. This requirement may prove difficult to achieve because of:

- The height of the tank
- The distance between a tank and the cannon position
- The prevailing weather conditions
- The fire updraught
- The high probability that a partial rupture of a fixed roof tank may only leave a small aperture through which the expanded foam can be targeted

A further problem exists in that expanded foam is applied forcefully to the surface of the burning product, which leads to increased contamination of the foam. The effects of this may be reduced by directing the foam stream onto the inside of the tank shell and allowing it to run down onto the surface of the product. However, in a live fire situation this may prove impossible to achieve.

System design criteria

In all primary protection systems using foam cannons it is assumed that all the calculated foam solution requirement actually reaches the area to be protected. As has already been explained, to achieve the minimum foam solution required, consideration must also be given to the potential foam solution losses that will occur due to access and windage problems. Enough equipment must therefore be available to ensure that under all conditions the minimum application rate is being achieved. This will, in most circumstances, result in considerable over-capacity in terms of equipment resource. This is often of the order of 2:1

The minimum specific design requirements can be summarised as:

- The minimum foam solution application rate should be 6.5



Figure 7.40 A foam cannon in operation - 15,000 litre/min of foam solution
Courtesy of Angus Fire

litres/min/m² for all types of foam concentrates on tanks containing liquid hydrocarbons.

- b) The minimum foam solution application rate may have to be increased to tackle special risks i.e. gasohols, Class 1A hydrocarbons, etc.
- c) Greater minimum foam solution application rates may also be required for hot fuels after a prolonged pre-burn.
- d) Foam cannons should not be considered as primary protection mechanisms on vertical fixed roof storage tanks over 18 m diameter.
- e) The minimum foam solution discharge duration time should be:
 - Crude petroleum and hydrocarbons with flash points below 37.8°C - 65 mins.
 - Hydrocarbons with flash points between 37.8°C and 93.3°C - 50 mins.

7.11 Water cooling systems

The individual tank design, layout and piping system for any particular installation will be a function both of the physical factors like terrain, site elevation, drainage, etc. and of the governing Standards regarding permissible tank spacings and position within the installation.

Despite taking all reasonable precautions as demanded by these considerations, a fire in an individual storage tank will generate significant radiated heat, which can damage and/or ignite adjacent tanks which would not otherwise be directly involved. A deep-seated fire in even the smallest diameter tank can create major problems unless cooling water is applied to its close neighbours.

Tank cooling is therefore recommended as essential to complete the protection of a particular installation and the following guidelines are given in the Part 19 of the IP Code, (Reference 7.5).

Tanks within two tank diameters distance downwind of a tank fire, or one tank diameter distance in other directions, should be protected by application of water spray at minimum recommended rate of 2 litres/min/m².

7.11.1 Special case - Floating roof tanks

With rimseal fires in floating roof tanks, the shell which is heated from the fire may be cooled with water whilst attempts are made to achieve and maintain an effective foam blanket, and to avoid re-ignition from hot surfaces. The recommended application

rate of water is 10 litres/min/m² of vertical tank surface in contact with the fire.

For the calculation of water requirements, the area should be assumed to be that based on a nominal half of the vertical height of the tank. Water should not be applied to the tank roof but foam may be used at a rate of 6.5 litres/min/m², based on tank cross-sectional area.

This rate may reduce to 4 litres/min/m² for tanks equipped with fixed foam pourers.

7.11.2 Tank cooling methods

The methods by which tanks may be cooled can be summarised as follows:

7.11.2.1 Water spray and deluge sprinkler systems

This is the most efficient method of delivering water, evenly distributed and at the correct application rate, to the outside roof and shell of the storage tank.

There are two principal ways of accomplishing this:

- 1) Using concentric rings of piping supported about 300 mm above the roof. These rings are fitted with spray nozzles, which give an overlapping spray pattern to cover the whole roof with water. The shell is similarly protected, usually with one spray ring at the top of and about 600 mm clear of the shell. Spray nozzles fitted to this ring and angled down slightly are arranged to spray water over the whole circum-

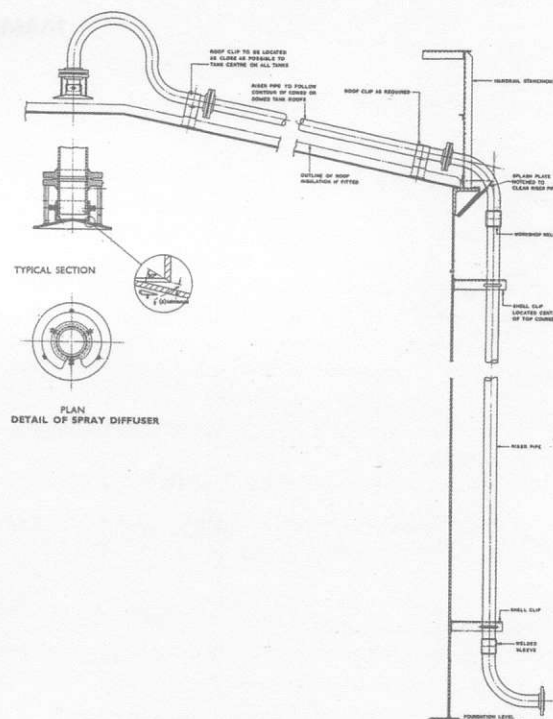


Figure 7.41 Water deluge system with conical diffuser

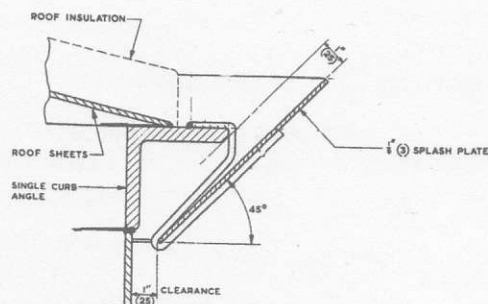


Figure 7.42 Detail of splash plate

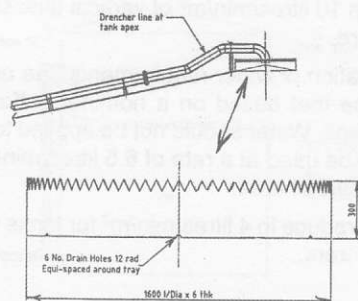


Figure 7.43 Roof deluge system using a coronet

Courtesy of McTay

ference and run down the shell.

- 2) The deluge system consists of a single water main being led to the crown of the tank roof where the water is directed vertically on to the roof and is evenly spread over the roof by a conical nozzle at the end of the outlet pipe or by a coronet attached to the roof plating, (shown schematically in Figures 7.41 and 7.43).

As the water streams down the roof it is directed on to the shell by splash plates fitted to the curb angle at the periphery of the shell. These plates are angled so that as the water hits them it is directed against and runs down the shell. See Figure 7.42.

These systems can be fed from a water deluge valve, which is automatically triggered, by some form of electric, pneumatic or

hydraulic detection system

7.11.2.2 Fixed and trailer-mounted water cannons

Both static and oscillating water cannons are a cost-effective means of delivering water to cool storage tanks and the number, capacity, position and deployment will ultimately depend upon individual site requirements. However, access problems and local water supply considerations must be taken into account when considering their introduction.

7.12 References

- 7.1 *Reinforcement of Manholes*, R. T. Rose, British Welding Journal, October 1961.
- 7.2 *Tank Calibration, Section 1*, The Institute of Petroleum, Petroleum Measurement Manual, Part II.
- 7.3 *NFPA 11 Standard for Low -, Medium -, and High - Expansion Foam*, 2002 Edition.
- 7.4 *NFPA 30 Flammable and Combustible Liquids Code*.
- 7.5 *BS 5306 Section 6.1: 1988 Specification follow expansion Foam systems*.
- 7.6 *IP Model Code of Safe Practice: Part 19, Fire Precautions at Petroleum Refineries and Bulk Storage Installations*.
- 7.7 *Bitumen, Institute of Petroleum Code of Safe Practice, Part 11*.

8 Tank venting of ambient temperature tanks

This Chapter is confined to the venting of ambient tanks. The venting of low temperature tanks is dealt with in Chapter 20.

The requirements of the various tank Codes and of the most influential venting Code API 2000 are discussed and examples of suitable venting devices are provided with information on their installation and relief capacity calculation methods.

Contents:

8.1 Introduction

8.2 The tank design Code requirements

8.2.1 API 650

8.2.2 BS 2654

8.2.3 prEN 14015

8.2.3.1 The evaluation of venting requirements of prEN 14015

8.2.4 API 2000

8.2.4.1 The evaluation of venting requirements of API 2000

8.2.4.2 Means of venting

8.2.4.3 Pressure limitations

8.2.4.4 Relief valve installation

8.3 Typical relief valve equipment

8.4 References

8.1 Introduction

It is probable that tank venting problems have brought more storage tanks to grief than any other single cause. Tales of such failures abound. The draining of the hydrostatic test water whilst failing to allow for any, or at least sufficient air to re-enter the tank is a particular classic. The draining of the test water is often done at the end of the tank test and one of the last activities of the day is to open the tank drain valve before leaving the site and allowing the tank to empty overnight.

The efforts of the tank to express its displeasure at being subjected to unacceptable levels of internal vacuum (or in modern parlance, internal negative pressure) via sundry creaks and groans, followed by early elastic shape changes, are thus played to an absent audience, and the following morning brings a serious surprise. The tank which has been the subject of several months concentrated effort to bring to completion is now in a crumpled heap. Replacement, or repair costs are added to by liquidated damages to further rub the embarrassed contractor's nose in this unfortunate situation which could so easily have been avoided.

The author's experience sadly involves such incidents. In one case the vacuum vent was propped open with a piece of wood which fell out during the night causing the valve to close, resulting in a total roof failure. In another case, a suitable vacuum valve was installed, but complete with its transit packing still in place. This had the effect of jamming the valve closed.

Storage tanks, despite their apparent size and robustness, are in reality quite fragile structures and require to be kept within their design pressure and vacuum envelope. Comparatively small excursions from this safe territory can bring about dramatic consequences.

To ensure that fixed roof tanks are maintained in their safety zone, provision must be made to allow the tank to vent to atmosphere. This is usually achieved by the provision of open vents, pressure relief valves, vacuum relief valves and as an extreme form of pressure relief, a frangible roof arrangement. The design and details of frangible roofs is covered in Chapter 4. Bursting discs are not popular for this service. The performance of bursting discs at the low pressures required by storage tanks is not good. The differences between the maximum and the minimum anticipated bursting pressures is large and would result in unnecessary venting and disc replacement. The performance of bursting discs improves as the design pressure increases, but this is of little use to the tank designer.

Events to which fixed roof tanks can be subjected to require them to need venting provisions include:

- Liquid movement into or out of the tank causing outbreathing or inbreathing of air, product vapours, a mixture of air and product vapours or in some circumstances purge gas.
- Thermal changes to the tank (often diurnal) necessitating inbreathing or outbreathing.
- The rupture of internal heating coils.
- Outbreathing as a result of exposure of the outer surfaces of the tank to fire.
- Process-related events such as the import of warm product, off-specification product liquids or vapours and similar happenings.

8.2 The tank design Code requirements

The protection of fixed roof storage tanks from the harmful effects of excessive levels of internal pressure or vacuum is clearly a matter of considerable importance for both commercial and safety reasons. It is interesting just how the different ambient tank design Codes address this subject.

cial and safety reasons. It is interesting just how the different ambient tank design Codes address this subject.

8.2.1 API 650

This Standard (*Reference 8.1*) is curiously relaxed regarding this issue. It is only in Appendix F (*Design of tanks for small internal pressures*) that there is any mention of the subject.

F.2.1 suggests that vents shall be sized and set so that at their rated capacity, the internal pressure under any normal operating conditions exceeds neither the internal design pressure, nor the maximum design pressure (this latter is the pressure for non-anchored tanks limited by uplift at the base of the tank shell as described in the earlier Chapter on tank design).

F7.7 (which is for anchored tanks with design pressures up to 2.5 lb/in²) states that venting shall be supplied by the purchaser in accordance with API Standard 2000. The manufacturer shall provide a suitable tank connection. The vents shall be checked during or after the testing of the tank.

This suggests that the tank purchaser is responsible for performing the vent sizing calculations, providing the equipment necessary and informing the tank manufacturer as to what connection sizes are required. In the author's view, this is an unsatisfactory situation as many tank purchasers do not have the technical abilities to undertake this responsibility or a clear understanding of the importance of getting it right.

8.2.2 BS 2654

This Standard provides the option for the venting requirements to be specified by the purchaser, or to be determined (presumably by the tank manufacturer) in accordance with a set of rules which are provided. These rules fall into two parts, the general rules which are summarised below and the more specific rules which lead to the calculation of required venting rates for particular tanks and lead to vent sizing. This latter set of rules are basically a metric version of API 2000, and as such do not warrant repetition in this Section.

The general rules include:

- The venting system provided shall cater for the following:
 - a) Normal vacuum relief
 - b) Normal pressure relief
 - c) Emergency pressure relief (this latter shall be specified in accordance with BS 2654 unless disregarded at the purchaser's discretion)
- Where emergency pressure relief is required, it shall be provided by suitable vents or by the provision of a frangible roof joint.
- The number and size of vents shall be based on the venting capacity obtained from Appendix F (i.e. the metric API 2000), and shall be sufficient to prevent any accumulation of pressure or vacuum from exceeding the values given below.
- Valves may be fitted with coarse mesh screens to prevent the ingress of birds. The use of fine mesh screens as anti flash protection is not recommended because of the possibility of blockage, especially under winter conditions. Consideration should be given to the possibility of corrosion when selecting the material for the wire screen.
- The set vacuum plus the accumulation to permit the valves to achieve the required throughput shall not exceed v_a . This is the vacuum to be used for the design of the tank shell section.

ondary wind stiffening which has been the subject of earlier Chapters.

- The set pressure plus the accumulation to permit the valves to achieve the required throughput for normal pressure relief shall not exceed the design pressure.
- No specific rules are provided for the emergency pressure accumulation, but the following shall be considered:
 - a) If it is expected that the design pressure is to be exceeded by the emergency pressure accumulation, then it shall be verified that the strength of the roof-to-shell junction is adequate and whether tank anchorage is required.

Note: This particularly applies to column-supported tank roofs with low roof slopes and to small tanks.

- b) Account shall be taken of the differences which can occur between the opening and closing pressures (blowdown) of vents of different types.
- The Standard does not cater for protection against overpressure caused by explosion within the tank, and where such protection is required special consideration should be given to the design of the tank and the venting devices.

8.2.3 prEN 14015

This draft Standard has departed from the usual practice of following the requirements of API 2000. A subcommittee of European venting specialists was set up to write the requirements for venting systems which appears in Annex L. This Annex describes the sources of the tank venting requirements as follows:

- Normal pressure venting requirements resulting from the maximum anticipated rate of import of product to the tank.
- Normal pressure venting requirements resulting from the maximum anticipated increase in tank surface temperature.
- Normal vacuum venting requirements resulting from the maximum anticipated rate of export of product from the tank.
- Normal vacuum venting requirements resulting from the maximum anticipated decrease in tank surface temperature.
- Emergency pressure venting requirements resulting from the exposure of the tank to an external fire.
- Other emergency conditions. These are listed for both pressure and vacuum relieving systems and include:
 - Malfunction of a gas blanketing system
 - Malfunction of a tank heating system regulation
 - Leakage of a tank heating system
 - Exceeding the maximum allowable pumping capacity due to incorrect connections within the pumping system
 - Chemical reactions
 - Poor pipe cleaning
 - Product transfer by pressurised gas
 - A sudden cool-down due to cold liquid being sprayed into a hot and empty tank
 - Malfunction of a sprinkler system
 - Excessive liquid flow out of the tank

This list is most helpful, but for some reason omits to mention the accidental import of hot liquid. This is a particularly dangerous condition, especially where the tank contents are volatile or have a water heel which may suddenly boil.

It is interesting that venting resulting from changes in barometric pressure is omitted from this list.

Having listed the venting components, this document then goes on to describe how they may be evaluated. This section is completely new, and as such should represent the latest thinking on this subject. For this reason the specific requirements of this document are described in Section 8.2.3.1.

The document does make a number of general points, amongst which are:

- Free vents can be applied to non pressure tanks.
- Pressure and vacuum relief valves must be used for low-pressure, high-pressure and very high-pressure tanks.
- The set pressure plus the accumulation to achieve the desired flow capacity shall not exceed the tank design pressure nor the tank design internal negative pressure.
- If very high emergency outbreathing rates are required, then additional emergency vents shall be supplied or the tank shall meet the requirements of Annex K (frangible roof).
- Flow resistance due to connected pipework or possible back pressures within the system shall be considered.
- The pressure and vacuum settings of emergency relief valves shall be such as to not operate during the normal relief valve operation.
- For the sizing of the emergency relief valve system, the flow capacities of the normal pressure and vacuum relief valves can be taken into account.
- When storing flammable liquids which can lead to an explosive atmosphere within the tank, the venting system shall be capable of preventing the transmission of flame into the tank. This presumably means the use of flame arrestors which are not universally approved of in some circles, due to their tendency to block up with certain products with the passage of time.

8.2.3.1 Evaluation of the venting requirements from prEN 14015

Normal outbreathing and inbreathing

This is otherwise known as the normal pressure and vacuum relief and is made up of liquid import or export and thermal effects.

Liquid movement outbreathing

This falls into three categories dependent upon the liquid storage temperature and the vapour pressure:

- a) For products stored below 40 °C or with a vapour pressure less than 50mbar

$$U_{op} = U_{pf} \quad \text{equ 8.1}$$

where:

U_{op} = outbreathing requirement in normal m³/hr of air

U_{pf} = the maximum filling rate in m³/hr

- b) For spiked products (i.e. with methane) the maximum venting capacity shall be increased by a factor of 1.7 to take into account the gas evolved from spiked products during filling, hence:

$$U_{op} = 1.7U_{pf} \quad \text{equ 8.2}$$

- c) For products stored above 40 °C or with a vapour pressure greater than 50mbar, the outbreathing shall be increased

by the evaporation rate which shall be specified by the purchaser.

Liquid movement inbreathing

In this case:

$$U_{ip} = U_{pe} \quad \text{equ 8.3}$$

where:

U_{ip} = the inbreathing requirement in normal m³/hr

U_{pe} = the maximum liquid export rate in m³/hr

Thermal outbreathing

This falls into two categories:

a) Tanks without thermal insulation

$$U_{ot} = 0.25V_T^{0.9} \left[1 - \frac{\Delta P_{ap}}{140} \right]^{1.6} \quad \text{equ 8.4}$$

where:

ΔP_{ap} = accumulation pressure in mbar gauge

U_{ot} = thermal outbreathing in normal m³/hr of air

V_T = tank volume in m³

Note 1: if $\Delta P_{ap} < 5$ mbarg or is unknown, use the bracketed term = 1.0

Note 2: The 0.25 factor is valid for latitudes between 58° and 43°. North of 58° use 0.20 and south of 43° use 0.32.

b) Tanks with thermal insulation

See below for the reduction factor for insulation or outer containment tanks.

Thermal inbreathing

This falls into two categories:

a) Tanks without thermal insulation

$$U_{IT} = CV_T^{0.7} \left[1 - \frac{\Delta P_{av}}{140 + P_{vp}} \right]^{1.6} \quad \text{equ 8.5}$$

where:

C = 3 for hexane and products with similar vapour pressures and/or stored at temperatures below 25 °C

C = 5 for products with vapour pressures higher than hexane and/or stored at temperatures above 25 °C

P_{vp} = vapour pressure of the liquid at the highest temperature (mbar)

ΔP_{av} = accumulation vacuum (mbar gauge) (internal negative pressure)

U_{IT} = maximum thermal inbreathing requirement (normal m³/hr of air)

Note 1: If the vapour pressure is unknown use $C = 5$

Note 2: The factors $C = 3$ and 5 are valid for latitudes between 58° and 43°. North of 58° use 2.5 and 4 and south of 43° use 4 and 6.5

Note 3: If P_{vp} is unknown the bracketed term becomes 1.0

b) Tanks with thermal insulation or outer containment tanks

The thermal out or inbreathing is reduced when the tank is fully or partially insulated, or fitted with an outer containment tank.

i For fully insulated tanks the reduction factor shall be given by:

$$R_{in} = \frac{1}{1 + \frac{hL_{in}}{\lambda_{in}}} \quad \text{equ 8.6}$$

where:

h = heat transfer coefficient (W/m²K)

L_{in} = thickness of the insulation (m)

λ_{in} = thermal conductivity (W/mK)

Note: As an example, for an insulation thickness of 0.10 m, a thermal conductivity of 0.05 W/mK and an inside heat transfer coefficient of 4W/m²K, the reduction factor is calculated to be 0.11.

ii For a partially insulated tank the reduction factor shall be given by:

$$R_{inp} = \frac{A_{inp}}{A} R_{in} + \left[1 - \frac{A_{inp}}{A} \right] \quad \text{equ 8.7}$$

where:

A = total area of the tank surface area (shell and roof) (m²)

A_{inp} = insulated surface of the tank (m²)

For a tank within an outer containment tank the reduction factor shall be given by:

$$R_c = 0.25 + 0.75 \frac{A_c}{A} \quad \text{equ 8.8}$$

where:

A_c = tank surface area not inside the outer containment tank in m² (probably part of the shell and the tank roof)

Emergency venting

In the case of an external fire or a malfunction of other systems such as a tank blanketing arrangement, outbreathing beyond the capability of the normal venting equipment provided may be required. For this eventuality it is necessary to fit additional emergency venting equipment.

Exposure of the external surfaces of the tank can give rise to an expansion of the gas volume within the tank (within a few minutes) and boiling of the tank contents (after several hours exposure).

Where a frangible roof-to-shell joint is not provided, emergency vents must be supplied to cater for whichever of the following is deemed to be appropriate:

- The flow rate due to gas expansion shall be given by:

$$U_{FE} = 15V_T^{0.7} R_{inf} \quad \text{equ 8.9}$$

$$R_{inf} = \frac{1}{1 + \frac{h_f L_{in}}{\lambda_{in}}} \quad \text{equ 8.10}$$

$$h_f = \frac{40}{A_w^{0.18}} \quad \text{equ 8.11}$$

where:

A_w = surface area of the tank shell heated by the fire (m²)

h_f = heat transfer coefficient (W/m²K)

R_{inf} = reduction factor for insulation if available

Note: Only a tank shell height of up to 9.0m above the bottom corner is to be considered in calculating the surface area.

- The flow rate due to product boiling shall be given by:

$$U_{fb} = 4 \times 10^4 A_w^{0.82} \frac{R_{inf}}{H_v} \sqrt{\frac{T}{M}} \quad \text{equ 8.12}$$

where:

- H_v = heat of vaporisation of the product (kJ/kg)
- M = molar weight of the product (kg/mol)
- T = boiling temperature of the product (°K)

Note 1: For hexane ($M = 86$ kg/mol, $H_v = 335$ kJ/kg, $T = 342$ °K) and similar products where no insulation is fitted (i.e. 1.0), this equation simplifies to:

$$U_{FB} = 238 A_w^{0.82}$$

Note 2: The flow rate calculated for product boiling will always cover the requirement for gas expansion.

8.2.4 API 2000

API 2000 has been around for many years and is undoubtedly the grandfather of tank venting Codes. It covers non-refrigerated tanks (i.e. ambient tanks) and refrigerated tanks up to design pressures of 15 lb/in².

The following covers the Code requirements for non-refrigerated tanks only.

In common with the other tank Codes, API 2000 lists the usual main causes of venting being required as:

- Liquid movement into or out of the tank.
- Tank breathing due to weather changes (e.g. pressure and temperature changes).
- Fire exposure.
- Other circumstances resulting from equipment failure and operating error.

The Standard then lists and describes the "other circumstances" in some detail. In brief these are:

- Pressure transfer blow-off. This can occur at the end of filling from trucks or similar where a surge of vapour enters the tank. A similar situation may occur after connected line pigging.
- Inert pads and purges. Usually related to failure of the pressure regulating system.
- External heat transfer devices. This could be a heated jacketed tank where failure of a control valve or a temperature sensing element has occurred.
- Internal heat transfer devices.
- Vent treatment system. This could be the failure of a system designed to collect and dispose of vented products.
- Utility failure.
- Change in temperature of the input stream to a tank
- Chemical reactions. Usually associated with the inadvertent import of an incompatible material which reacts with the stored product.
- Liquid overfilling
- Atmospheric pressure changes
- Control valve failure

- Steam out. If an un-insulated tank is filled with steam, the condensing rate (particularly aided by rainfall) may exceed the venting capacity provided.
- Un-insulated tanks. A warning about such tanks in rain-storm conditions, especially when the vapour space is hot.

The Standard does not give rules for evaluating the venting requirements caused by these events, but does at least list them and state that they should be considered.

8.2.4.1 The evaluation of the venting requirements of API 2000

API 2000 gives its formulae and tables in both English and metric units. Only the metric versions are given below.

Normal outbreathing (pressure) and inbreathing (vacuum)

As is the case for prEN 14015, these are the venting requirements resulting from liquid movements and thermal effects.

Liquid movement outbreathing

Requirements are given for liquids with flash points above and below 100 °F:

- Liquids with flash points above 100 °F (37.8 °C) or a normal boiling point of 300 °F (148.9 °C): venting equivalent to 1.01 Nm³/hr per cubic metre/hour of the maximum filling rate
- Liquids with flash points below 100 °F (37.8 °C) or a normal boiling point of 300 °F (148.9 °C): venting equivalent to 2.02 Nm³/hr per cubic metre/hour of the maximum filling rate

Note 1: An explanation of the basis of these requirements is given in Appendix A of API 2000.

Note 2: A warning about situations where the liquid is fed into a tank at or near to its boiling point and higher venting rates may be required is given.

Note 3: Table 1B shows these requirements in metric units and is shown in Figure 8.1.

(Nm³/hr of Air per Cubic Meter per Hour of Liquid Flow)
B. Metric Units

Flash Point/Boiling Point ^a	Inbreathing		Outbreathing	
	Liquid Movement Out	Thermal	Liquid Movement In	Thermal
Flash Point ≥ 37.8°C	0.94	See Table 2B	1.01	See Table 2B
Boiling Point ≥ 148.9°C	0.94	" "	1.01	" "
Flash Point < 37.8°C	0.94	" "	2.02	" "
Boiling Point < 149°C	0.94	" "	2.02	" "

^a Data on flash point or boiling point may be used. Where both are available, use flash point (See Appendix A).

Figure 8.1 Normal venting requirements
From API 2000, table 1B

Liquid movement inbreathing

The venting provided should be equivalent to 0.94 Nm³/hr per cubic metre/hour of emptying rate.

Thermal outbreathing

Requirements are given for liquids with high and low flash points and boiling points:

- For liquids with flash points above 100 °F (37.8 °C) or a normal boiling point above 300 °F (148.9 °C): venting at least that shown in column 2 of Table 2B (Figure 8.2).
- For liquids with flash points below 100 °F (37.8 °C) or a normal boiling point below 300 °F (148.9 °C): venting at least that shown in column 4 of Table 2B.

Tank Capacity	Inbreathing (Vacuum)		Outbreathing	
	Column 1 ^d	Column 2 ^a	Column 3 ^b	Column 4 ^c
			Flash Point $\geq 37.8^{\circ}\text{C}$ or Normal Boiling Point $\geq 148.9^{\circ}\text{C}$	Flash Point $< 37.8^{\circ}\text{C}$ or Normal Boiling Point $< 148.9^{\circ}\text{C}$
Cubic Meters		Nm ³ /h	Nm ³ /h	Nm ³ /h
10		1.69	1.01	1.69
20		3.37	2.02	3.37
100		16.9	10.1	16.9
200		33.7	20.2	33.7
300		50.6	30.3	50.6
500		84.3	50.6	84.3
700		118	70.8	118
1,000		169	101	169
1,500		253	152	253
2,000		337	202	337
3,000		506	303	506
3,180		536	388	536
4,000		647	472	647
5,000		787	537	787
6,000		896	602	896
7,000		1,003	646	1,003
8,000		1,077	682	1,077
9,000		1,136	726	1,136
10,000		1,210	807	1,210
12,000		1,345	888	1,345
14,000		1,480	969	1,480
16,000		1,615	1,047	1,615
18,000		1,745	1,126	1,745
20,000		1,877	1,307	1,877
25,000		2,179	1,378	2,179
30,000		2,495	1,497	2,495

Notes:

^a For tanks with a capacity of 20,000 barrels (3,180 cubic meters) or more, the requirements for the vacuum condition are very close to the theoretically computed value of 2 SCFH of air per square foot (0.577 Nm³/h per square meter) of total shell and roof area. For tanks with a capacity of less than 20,000 barrels (3,180 cubic meters), the requirements for the vacuum condition have been based on 1 SCFH of air for each barrel of tank capacity (0.169 Nm³/h per cubic meter). This is substantially equivalent to a mean rate of temperature change of 100°F (37.8°C) per hour in the vapor space (see Appendix A). An engineering review should be conducted for uninsulated tanks where the vapor space temperature is maintained above 120°F (48.9°C) (see 4.2.5.14).

^b For stocks with a flash point of 100°F (37.8°C) or above, the outbreathing requirement has been assumed to be 60 percent of the inbreathing requirement. The roof and shell temperatures of a tank cannot rise as rapidly under any condition as they fall, for example, during a sudden cold rain.

^c For stocks with a flash point below 100°F (37.8°C), the outbreathing requirement has been assumed to be equal to the inbreathing requirement to allow for vaporization at the liquid surface and for the higher specific gravity of the tank vapors.

^d Interpolate for intermediate tank sizes. Tanks with a capacity of more than 180,000 barrels (30,000 cubic meters) require individual study. Refer to Appendix A for additional information about the basis of this table.

Figure 8.2 Requirements for thermal venting capacity, (metric units)
From API 2000, table 2B

Thermal inbreathing

The venting provided should be at least that shown in column 2 of Table 2B in Figure 8.2.

For the case of heated un-insulated tanks where the vapour space is maintained above 120 °F, an engineering review is suggested.

Emergency venting

For tanks where the roof-to-shell joint can be considered frangible according to the rules of API 650, there is no need to provide for emergency venting. Care should be taken to ensure that failure of this joint does not occur during normal service.

For tanks which do not have such a frangible roof-to-shell joint, emergency venting for fire exposure must be provided. The venting requirement is given by:

$$\text{Nm}^3/\text{hr} = 881.5 \frac{QF}{L} \left[\frac{T}{M} \right]^{0.5} \quad \text{equ 8.13}$$

where:

Nm³/hr = venting requirement (normal m³/hr of air)

- Q = heat input from fire exposure (watts)
(see Figure 8.3 for the basis of this)
- A = wetted area of the tank shell (m²)
(see footnotes a and b of Table 3B (Figure 8.4))
- F = environmental factor from Table 4B
(Figure 8.5)
- L = latent heat of vaporisation of the stored liquid
at the relieving pressure and temperature
(kJ/kg)

Wetted Surface Area (square m)	Design Pressure (barg)	Heat Input (Watts)
<18.6	≤1.034	$Q = 63,150A$
≥18.6 and <93	≤1.034	$Q = 224,200A^{0.566}$
≥93 and <260	≤1.034	$Q = 630,400A^{0.338}$
≥260	between 0.07 and 1.034	$Q = 43,200A^{0.82}$
≥260	≤0.07	$Q = 4,129,700$

Figure 8.3 Heat input from fire exposure
From API 2000

Wetted Area ^a (square meters)	Venting Requirement (Nm ³ /h)	Wetted Area ^a (square meters)	Venting Requirement (Nm ³ /h)
2	608	35	8086
3	913	40	8721
4	1,217	45	9322
5	1,521	50	9895
6	1,825	60	10,971
7	2,130	70	11,971
8	2,434	80	12,911
9	2,738	90	13,801
11	3,347	110	15,461
13	3,955	130	15,751
15	4,563	150	16,532
17	5,172	175	17,416
19	5,780	200	18,220
22	6,217	230	19,102
25	6,684	260	19,910
30	7,411	>260 ^b	—

^aThe wetted area of a tank or storage vessel shall be calculated as follows:

Sphere and Spheroids—The wetted area is equal to 55 percent of the total surface area or the surface area to a height of 30 feet (9.14 meters) above grade, whichever is greater.

Horizontal Tanks—The wetted area is equal to 75 percent of the total surface area or the surface area to a height of 30 feet (9.14 meters) above grade, whichever is greater.

Vertical Tanks—The wetted area is equal to the total surface area of the vertical shell to a height of 30 feet (9.14 meters) above grade. For a vertical tank setting on the ground, the area of the ground plates is not to be included as wetted area. For a vertical tank supported above grade, a portion of the area of the bottom is to be included as additional wetted surface. The portion of the bottom area exposed to a fire depends on the diameter and elevation of the tank above grade. Engineering judgment is to be used in evaluating the portion of the area exposed to fire.

^bFor wetted surfaces larger than 2,800 square feet (260 square meters), see Sections 4.3.3.2.2 and 4.3.3.2.3.

Note:

Table 3 and the constants 1107 and 208.2 in Equations 2A and 2B respectively were derived from Equation 1 and Figure B-1 by using the latent heat of vaporization of hexane (144 BTU per pound or 334,900 J/kg) at atmospheric pressure and the molecular weight of hexane (86.17) and assuming a vapor temperature of 60°F (15.6°C). This method will provide results within an acceptable degree of accuracy for many fluids having similar properties (see Appendix B).

Figure 8.4 Emergency venting required for fire exposure versus wetted surface area (metric units)
From API 2000, table 3B

Tank Design/Configuration	Insulation Conductance (Watts/m ² °K)	Insulation Thickness (cm)	F Factor
Bare metal tank	—	0	1.0
Insulated tank ^a	22.7	2.5	0.3 ^b
" "	11.4	5	0.15 ^b
" "	5.7	10	0.075 ^b
" "	3.8	15	0.05 ^b
" "	2.8	20	0.0375 ^b
" "	2.3	25	0.03 ^b
" "	1.9	30	0.025 ^b
Concrete tank or fireproofing	—	—	(see note c)
Water-application facilities ^d	—	—	1.0
Depressuring and emptying facilities ^e	—	—	1.0
Underground storage	—	—	0
Earth-covered storage above grade	—	—	0.03
Impoundment away from tank ^f	—	—	0.5

Notes:

^aThe insulation shall resist dislodgment by fire-fighting equipment, shall be noncombustible, and shall not decompose at temperatures up to 1000°F (537.8°C). The user is responsible to determine if the insulation will resist dislodgment by the available fire-fighting equipment. If the insulation does not meet these criteria, no credit for insulation shall be taken. The conductance values are based on insulation with a thermal conductivity of 4 BTU per hour per square foot per °F per inch of thickness (9 Watts per square meter per °C per centimeter of thickness). The user is responsible for determining the actual conductance value of the insulation used. The conservative value of 4 BTU per hour per square foot per °F per inch of thickness (9 Watts per square meter per °C per centimeter of thickness) for the thermal conductivity is used.

^bThese F factors are based on the thermal conductance values shown and a temperature differential of 1600°F (888.9°C) when using a heat input value of 21,000 BTU per hour per square foot (66,200 Watts per square meter) in accordance with the conditions assumed in API Recommended Practice 521. When these conditions do not exist, engineering judgment should be used to select a different F factor or to provide other means for protecting the tank from fire exposure.

^cUse the F factor for an equivalent conductance value of insulation.

^dUnder ideal conditions, water films covering the metal surfaces can absorb most incident radiation. The reliability of water application depends on many factors. Freezing weather, high winds, clogged systems, undependable water supply, and tank surface conditions can prevent uniform water coverage. Because of these uncertainties, no reduction in environmental factors is recommended; however, as stated previously, properly applied water can be very effective.

^eDepressuring devices may be used, but no credit shall be allowed in sizing the venting device for fire exposure.

^fThe following conditions must be met: A slope of not less than 1 percent away from the tank shall be provided for at least 50 feet (15 meters) toward the impounding area; the impounding area shall have a capacity that is not less than the capacity of the largest tank that can drain into it; the drainage system routes from other tanks to their impounding areas shall not seriously expose the tank; and the impounding area for the tank as well as the impounding areas for the other tanks (whether remote or with dikes around the other tanks) shall be located so that when the area is filled to capacity, its liquid level is no closer than 50 feet (15 meters) to the tank.

Figure 8.5 Environmental factors for non-refrigerated above-ground tanks (metric units)
From API 2000, table 4B

T = temperature of the relieving vapour (°K)

M = the molecular weight of the vapour

An alternative simpler calculation method is given which gives a lesser degree of accuracy.

8.2.4.2 Means of venting

API 2000 provides a considerable amount of sensible advice regarding the types of relieving devices to be used and how these should be installed and maintained. A small part of this advice is repeated here. For those who have a serious interest in this subject, the complete text of this Standard, together with the companion Standards API RP 520 and API RP 521 should be studied in detail (*References 8.5 and 8.6*).

Normal venting

Normal venting for pressure and vacuum shall be accomplished by a pressure/vacuum (PV) valve or an open vent with or without a flame arresting device as described below. Relief devices fitted with a weight and a lever are not recommended.

- PV valves are recommended for petroleum products with a flash point below 100 °F (37.8 °C) and where the fluid temperature exceeds the flash point. A flame arrestor is not considered necessary where PV valves are used as the vapour velocities across the valve seat are considered to exceed the flame speed.
- Open vents with flame arresting devices may be used for the tanks described above.
- Open vents without flame arrestors may be used in the following cases:
 - For tanks in which petroleum or petroleum products with a flash point of 100 °F (37.8 °C) or above are stored, provided the contents are not heated and the fluid remains below the flash point.
 - For heated tanks where the storage temperature of the petroleum or petroleum products is below the flash point.
 - For tanks of capacity less than 9.46 m³ used for any product.
 - For tanks of capacity less than 477 m³ used for crude oil.
- In the case of viscous oils, such as cutback and penetrating grade asphalts, where the danger of pallet sticking or flame arrestor blocking exists, open vents without flame arrestors may be used as an exception to the rules above.
- In areas subject to strict emission regulations, open vents may not be acceptable.

Emergency venting

Tanks with frangible roof joints do not require emergency venting devices. For other tanks the Code offers the following advice:

- Larger or additional open vents may be provided subject to the same provisions as given in Section on Normal venting.
- Larger or additional PV valves.
- A gauge hatch which permits the cover to lift under abnormal internal pressure.
- A manhole cover which lifts when subject to abnormal internal pressure.
- Other forms of construction which can be proved to fulfil the required purpose.
- A rupture disc device (unlikely to be suitable for the low pressures usually associated with ambient tanks).

8.2.4.3 Pressure limitations

For tanks which are designed to API 650 Appendix F (Design of Tanks for Small Internal Pressures) the pressure relief devices shall be sized and set so that at the rated capacity of the device, the internal pressure under any normal operating condition shall not exceed the internal design pressure or the maximum design pressure. Both of these pressures are specifically defined in Appendix F of API 650.

For other API 650 tanks, the pressure relief devices selected should limit the pressure in the tanks to prevent excessive lifting of the tank roof sheeting. For a tank with 3/16" thick roof sheets, this limits the pressure to 3.5 mbar.

8.2.4.4 Relief valve installation

This Code provides much sensible advice on the general details of how relieving devices should be installed. Amongst these are:

- Installation details shall provide direct access to the tank vapour space and not be capable of being sealed off by the liquid contents.
- Where block valves are installed between the relieving devices and the tank (for maintenance purposes), arrangements shall be made to ensure that when one relieving device is isolated, the remaining devices shall provide the full relieving capacity. This in effect means the supply of a spare relieving device and a system to ensure that no more than one relieving device can be isolated at any one time. Block valve interlocking is a commonly used solution to achieve this.
- Inlet and outlet connections and details shall be carefully considered to ensure that any pressure drops occurring do not detract from the ability of the relieving arrangement to provide the full relieving capacity required.
- If discharge pipework is fitted, it shall lead to a safe location, shall not subject the relieving devices to condensation and not discharge vapours into enclosed spaces.
- For tanks located inside buildings, the venting system shall discharge outside the building and frangible roof joints shall not be used.
- If relieving systems from more than one tank discharge into a common header, considerable care shall be exercised to ensure that no problems arise from liquid traps, back pressures, throttling and unforeseen interactions between the relieving systems from different connected tanks.

8.3 Typical relief valve equipment

There are a number of well-known manufacturers of tank relieving equipment around the world. All produce a range of products suitable for use with ambient storage tanks.

Because of the low pressures associated with these tanks, it is usual to use pressure relief valves which are dead weight-operated rather than the pilot-operated types which are more usual at the higher design pressures associated with low temperature tanks. The dead weight pressure relief valves are also much cheaper than their pilot-operated equivalents. A typical dead weight operated valve is shown in Figure 8.6.

For vacuum relief the valves are also dead weight-operated and a typical example is shown in Figure 8.7.

For reasons of economy in terms of reducing the number of tank roof connections and isolation valves (where fitted), it is common to combine the pressure and vacuum valves into a single item and a typical pressure and vacuum relief valve is shown in Figure 8.8.

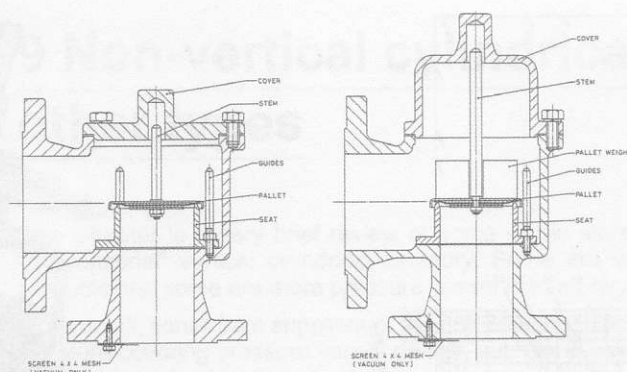


Figure 8.6 Dead weight-operated valve
Courtesy of Tyco Valves & Controls

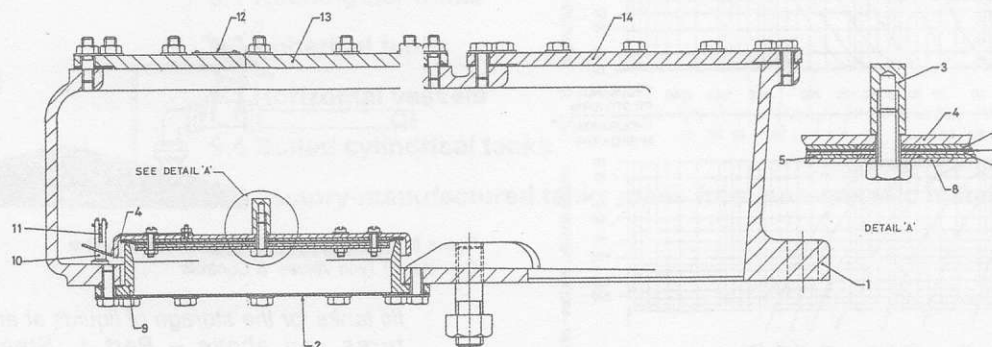


Figure 8.7 Dead weight-operated vacuum relief valve
Courtesy of Tyco Valves & Controls

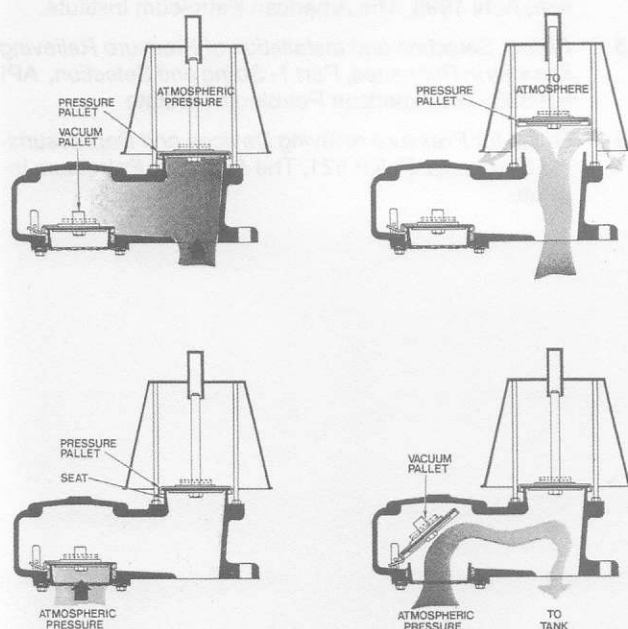


Figure 8.8 Typical pressure and vacuum relief valves
Courtesy of Tyco Valves & Controls

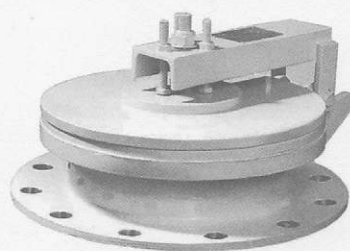


Figure 8.9 Emergency vent and manhole cover
Courtesy of Tyco Valves & Controls

All types of relief valves are manufactured in a range of sizes to suit the flow rates required. These typically range from 2" up to 12" NB.

For emergency relief (i.e. the external fire exposure case) the pressure relief valves described above may not have sufficient capacity for the flow rates involved and valves specifically designed for this higher flow regime are available. One such is shown in Figure 8.9. These valves are commonly supplied in sizes up to 24" NB and some are designed to fulfil a second use as tank roof manways.

It is usual for the valve manufacturers to provide data concerning the pressure/flow characteristics of each valve in their range of products. This enables the tank designer to select the number and sizes of the valves required for relieving duties. Ideally this data should be derived from physical testing of the valves. A typical pressure/flow curve is shown as Figure 8.10. It is usual for these pressure/flow curves to be provided for air.

For pressure relief some adjustment must be made for the characteristics of the product vapour. Some manufacturers provide proprietary software which includes the pressure/flow data and can make appropriate allowances for different product vapours and for suction and exit losses to aid the designer.

For tanks with fixed roofs storing certain products, often with internal floating roofs, it is common to require the space above the liquid or internal roof to be blanketed with nitrogen gas. To control the flow of this purge gas into the tank and ensure minimum wastage, tank blanketing valves are available and an example of these is illustrated in Figure 8.11.

8.4 References

- 8.1 *Welded Steel Tanks for Oil Storage, API 650 Tenth edition*, November 1988, The American Petroleum Institute.

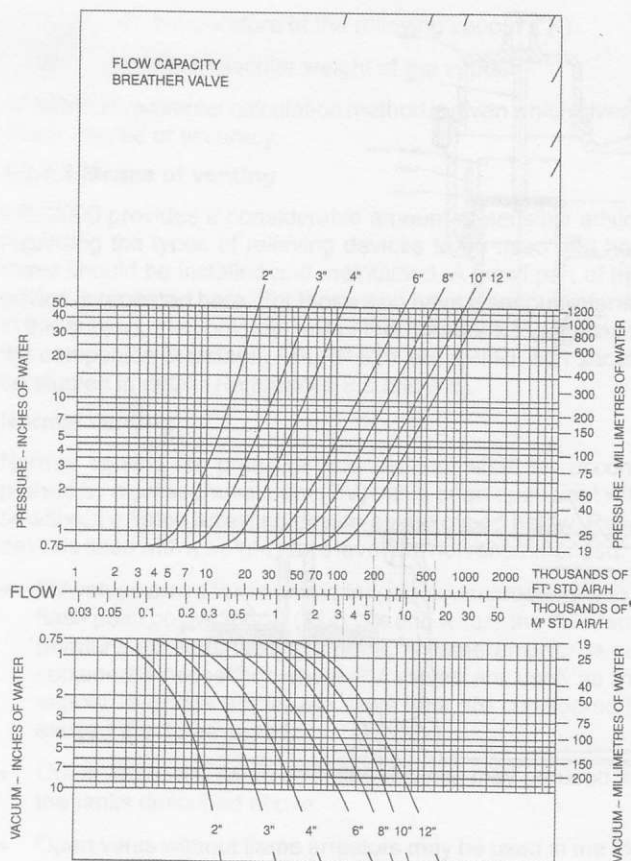
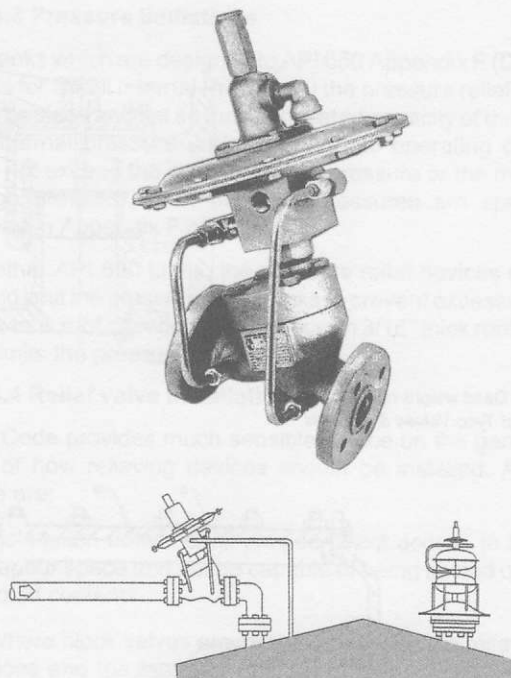


Figure 8.10 A typical pressure/flow curve

- 8.2 *British Standard Specification for Manufacture of vertical steel welded non-refrigerated storage tanks with butt welded shells for the petroleum industry, BS 2654:1989, BSI London*
- 8.3 *Specification for the design and manufacture of site built, vertical, cylindrical, flat-bottomed, welded, metal-*

Figure 8.11 Pilot-operated pressure/vacuum valve
Courtesy of Tyco Valves & Controls

lic tanks for the storage of liquids at ambient temperatures and above – Part 1: Steel tanks, prEN 14015-1:2000

- 8.4 *Venting Atmospheric and Low-Pressure Storage Tanks: Non-refrigerated and Refrigerated, API 2000, Fifth edition, April 1998, The American Petroleum Institute.*
- 8.5 *Sizing, Selection and Installation of Pressure Relieving Devices in Refineries, Part 1- Sizing and Selection, API RP 520, The American Petroleum Institute*
- 8.6 *Guide for Pressure relieving Devices and Depressurising Systems, API RP 521, The American Petroleum Institute*

9 Non-vertical cylindrical tanks and other types

This Chapter is a very brief review of some of the storage tanks which do not fit into the "conventional" vertical cylindrical category. Some are very much proprietary designs and products and some are more pressure vessel than storage tank.

More detail, either from suppliers of the first category, should not be difficult to obtain or from literature covering pressure vessel design, such as *European Pressure Equipment*, which is part of this series of reference books.

Contents:

9.1 Rectangular tanks

9.2 Spherical tanks

9.3 Horizontal vessels

9.4 Bolted cylindrical tanks

9.5 Factory-manufactured tanks made from non-metallic materials

9.6 References

9.1 Rectangular tanks

Rectangular tanks are a common sight in towns, factories and airfields around the UK and elsewhere. They are almost always factory-manufactured in transportable modules to proprietary designs and are commonly called Braithwaite Tanks. They are restricted to quite modest capacities when compared to the vertical cylindrical types. This has much to do with the fundamental unsuitability of the rectangular form to liquid containment. Whilst the conventional tank's shell is stressed by the liquid contents in simple tension, the stressing of a rectangular tank is more complex. The liquid loading on the flat sides requires stiffened panels and often internal bracing. It is usual for the panels to be supplied suitable for bolting together with sealing of these joints. For water storage and for other products where cleanliness is important, the panels may have a factory-applied coating on both inner and outer surfaces. An advantage of these tanks is that they are available "off the shelf" and do not require particularly skilled labour for their erection. They can also be easily dismantled and re-erected elsewhere.

It is usual for such tanks to be supported on elevated steel or masonry structures which must be suitably designed for the loadings.

9.2 Spherical tanks

Spheres fall more correctly into the field of pressure vessels. However, they are such a common sight that they deserve a brief mention. They are designed to pressure vessel Standards such as ASME VIII, BS 5500 and EN 13445. The spherical form is well-suited to resist the internal pressures arising from the product liquid and the vapour. For this reason, spheres were very much in evidence for the land-based storage of products such as LPG and this is discussed further in Chapter 17. Spherical tanks are also a common component of liquid gas carriers and this is also covered in Chapter 17.

The support of spherical tanks is most commonly achieved by the use of legs which attach to the sphere at the equator. It is usual for these legs to be braced together with diagonal tie rods to provide the necessary lateral support to resist wind and seismic loadings. Such a sphere is shown in Figure 9.1 together with the arrangements for access to the top of the vessel where the pressure relief valves and the level instrumentation are located. The liquid inlet and outlet connections are to be found in the bottom cap of the sphere. To ensure that any leakage from the sphere is contained, a local bund is usually provided and an example of this is shown in Figure 9.2.

There have been some spectacular accidents in the past involving spherical vessels storing volatile and inflammable products. Some of these have come about by the ignition of product leakage, possibly coming from the bottom liquid connections, which has not been able to drain away from the vessel and has consequently "cooked" the sphere to the point where the increasing heat input causes the internal pressure to increase at a rate that the pressure relief valve system cannot cope with, leading to an explosive failure of the vessel.

Current thinking is to provide a bunding system from which the leaking liquid can be rapidly removed to a spill containment pit where a foam blanketing system can hopefully prevent or at least minimise the effect of ignition. For reasons which are obvious, the fireproofing of the supporting legs of spheres is a mandatory requirement.

The sphere illustrated in Figure 9.2 has external cladding, suggesting that it is an insulated sphere, possibly for the storage of semi or fully refrigerated LPG. The application, maintenance, longevity and repair of such insulation and associated cladding systems for spherical vessels has caused many problems for the owners of such vessels in the past.



Figure 9.1 A typical spherical tank under construction
Courtesy of Whessoe

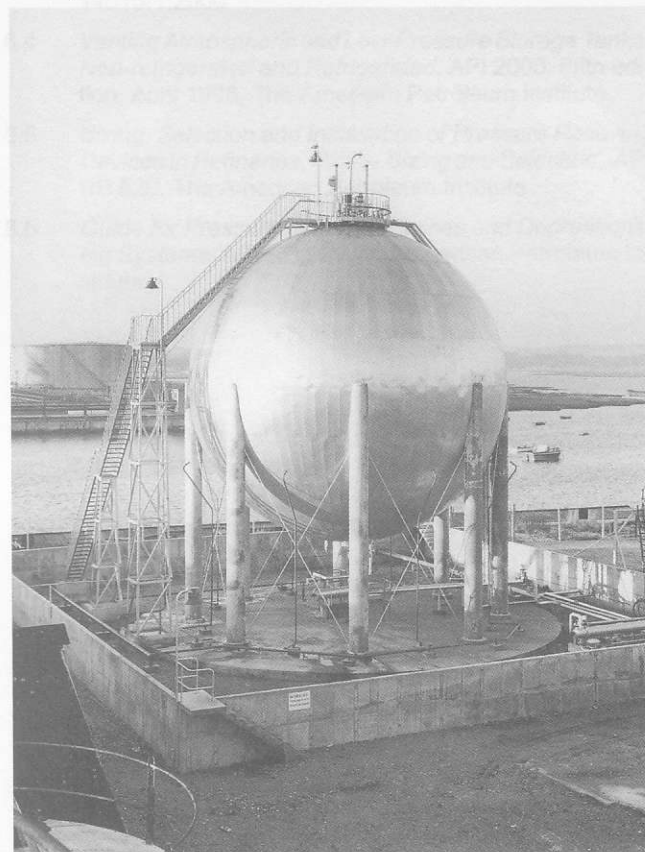


Figure 9.2 Spherical tank with local bund
Courtesy of Whessoe

The safety problems, both real and perceived, which have been associated with spherical vessels has caused them to be less popular choice for certain owners and in certain geographic locations than was the case in times past.

A big sphere would be around 22 m in diameter which would have a gross liquid capacity of some 5575 m³. Above this diam-

eter, problems of plate thickness and site stress-relief tend to provide a size limitation.

A second means of support for spherical vessels is to provide a cylindrical skirt or a cup type of arrangement. This is commonly known as the "Man type" of support and is often considered as a proprietary design, available only from certain designers and suppliers.

9.3 Horizontal vessels

Above ground horizontal vessels have been used for many years for the storage of modest quantities of various products. These range in size from the simple 'gas pigs' for domestic gas supply of around 0.5 m^3 up to vessels for high pressure gas storage or for component parts of mounded storage systems of around 4000 m^3 for each vessel.

The high pressure gas vessels were a common sight at major gas works at one time in the UK. They were an early form of peak shaving for the gas network before the advent of the liquid natural gas tanks at strategic locations around the country for the same purpose. These vessels were built in groups of six or more and were up to 6 m in diameter and 100 m long. They were constructed from factory-built units at the maximum transportable length, which were site-welded together and the closing seams site stress relieved. An example of such a facility during construction is shown in Figure 9.3.

A similar facility for the storage of liquid propane is shown in Figure 9.4. This consists of six vessels, each 12 ft (3.66 m) in diameter and 120 ft (36.6 m) long.



Figure 9.3 Site welding of high pressure gas vessels
Courtesy of Whessoe

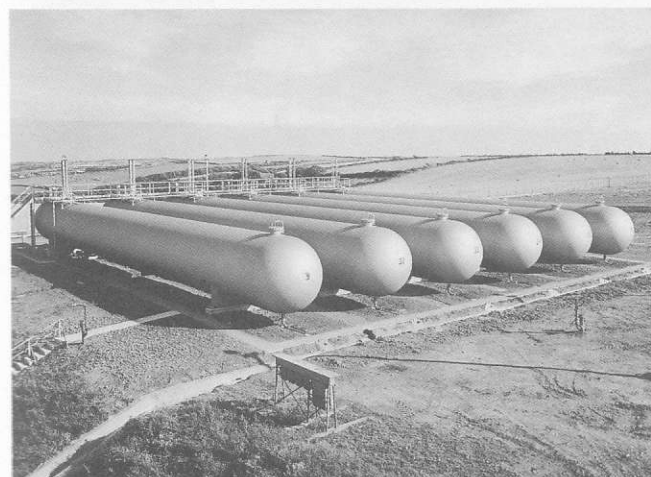


Figure 9.4 Liquid propane storage facility
Courtesy of Whessoe



Figure 9.5 Mounded storage tank system under construction
Courtesy of



Figure 9.6 Mounded storage tank being laid on prepared sand beds
Courtesy of

For safety reasons, such above ground facilities for the storage of products such as LPG have become unpopular. The current trend for the pressure storage of LPG is to use mounded storage systems. Here horizontal pressure vessels are used which are supported on a bed of sand or other suitable soil, and after construction are back-filled and buried. This arrangement provides protection from fire and missile damage. This arrangement also allows for the storage of different products or product mixes in the separate vessels which is convenient for operators of LPG terminals.

Guides to the design of mounded storage facilities are provided by the UK Health and Safety Executive and the Engineering Employers Materials Users Association, (EEMUA), (*References 9.1 and 9.2*). Figures 9.5 and 9.6 show a typical mounded storage tank system under construction. In this instance the vessels were 8 m in diameter and because of the remote location of the site in the Philippines, were constructed in modules from imported edge-prepared flat plate in a temporary workshop on the job site. These were then laid on the prepared sand bed and welded into the complete vessels.

In-ground horizontal cylindrical storage tanks are widely used as garage forecourt tanks for the storage of the various motor fuels. At one time these were simple steel tanks buried in the ground. Problems of corrosion and subsequent leakage of the products into the surrounding soil, and the escalating costs of remedial works and litigation has caused this area of activity to be reconsidered and modern facilities have secondary containment, leak detection and anti-corrosion measures built into them. An excellent book covering the Codes, regulations and design of these tanks from an American perspective is given in *Reference 9.3*.

9.4 Bolted cylindrical tanks

As for the rectangular tanks described in Section 9.1, these are made from factory-manufactured panels which are assembled by bolting at the job site. They are restricted to modest capacities and have the advantage of quick and cheap erection and being re-useable. For water storage, their design and construction in the USA is the subject of the American Water Works Association Code, ANSI/AWWA D103-97, (Reference 9.4).

9.5 Factory-manufactured tanks made from non-metallic materials

There are a number of manufacturers who specialise in the manufacture of tanks made from plastic materials. These are available in capacities up to 70 m³, diameters up to 3.5 m and heights of 10 m. Many are available "off the shelf" and made from plastic materials which are tailored to the corrosive nature of the particular product to be stored.

Some tanks of this type come with built-in bunding arrangements and one such example is shown in Figure 9.7.

9.6 References

- 9.1 *Mounded and buried LPG tanks*, K. W. Blything, J. Gould, B. L. Prescott and R. G. J. Robinson, AEA Technology, Health & Safety Executive, March 1996.
- 9.2 *Guide for the design, construction and use of mounded horizontal cylindrical vessels for pressurised storage of LPG at ambient temperatures*, Publication No. 190 : 2000, EEMUA, London.
- 9.3 *Handbook of storage tank systems*, W. B. Geyer, sponsored by Steel Tank Institute, Lake Zurich, Illinois, Marcel Dekker, New York, ISBN 0 8247 8589 4.
- 9.4 *Standard for factory coated bolted steel tanks for water storage*, ANSI/AWWA D103-97, AWWA Denver, Colorado.

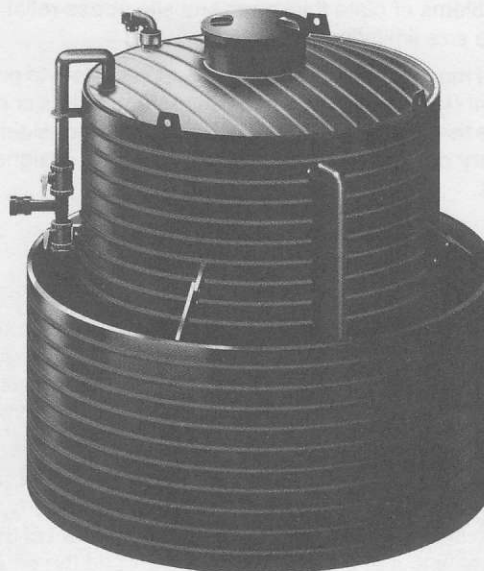


Figure 9.7 Non-metallic tank with built-in bunding
Courtesy of Allibert Buckhorn UK Ltd

10 Material selection criteria for ambient temperature tanks

The basic rules of material selection are covered in this Chapter and a glimpse of a little of the work and experience which lies behind the selection criteria is provided.

This is a big subject and those who wish to practice or study in this area would be well advised to look to the various publications on this topic.

Contents:

10.1 General

10.2 Brittle fracture considerations

10.3 Design metal temperature

10.3.1 Minimum design metal temperature

10.3.2 Maximum design metal temperature

10.4 Requirements of the tank design codes

10.4.1 API 650 requirements

10.4.2 BS 2654 requirements

10.4.3 prEN 14015 requirements

10.5 References

10.1 General

The development of the current material selection criteria for ambient temperature storage tanks is an interesting tale. The move from riveted to welded shells brought brittle fracture onto the scene in much the same way as the various failures of the Liberty Ships focussed attention on the same phenomenon in the ship building world. The paper by Cotton and Denham (*Reference 10.1*) follows the development of the rules for steel selection from the early days of welded tanks up to around 1980.

The first Code to provide rules for welded storage tanks was API 12 C (*Reference 10.2*), first published about 1935. It was this Standard which was the industry Standard until the mid 1950s and formed the basis for the subsequent Standards API 650 (*Reference 10.3*) and BS 2654 (*Reference 10.4*) which are the design Codes for most tanks for ambient temperature service used today. The forthcoming European Code takes a route which has been influenced by both of these Codes, but is probably more BS than API in its final draft form, prEN 14015 (*Reference 10.5*).

The vast majority of ambient tanks are constructed from carbon and carbon manganese steels and the Codes concentrate their attention on these materials. API 650, which it should be remembered is written for tanks for the storage of petrochemical products, does have rules for the design, material selection, fabrication and erection of storage tanks constructed from stainless steels. These are given in Appendix S which is discussed in Section 10.4.1.

BS 2654, which is also restricted to the petrochemical industry products but is frequently used for the storage of products such as water, wine and food related materials where cleanliness and product contamination are important, surprisingly has no rules for stainless steel tanks. This has not stopped the provisions of this Standard from having been used and adapted for this area of activity.

prEN 14015 includes rules for both carbon and carbon manganese steels and for stainless steels. It was the original intention that this Standard would be published in two parts, the first covering steel (C, CMn and SS) tanks and the second covering aluminium alloy tanks. This second part of the Code failed to appear due to a general lack of interest. There is little activity in this area of tank building and it was not possible to assemble a committee with sufficient knowledge and interest to prepare the document. As far as the author is aware, the only set of rules for the design of aluminium alloy storage tanks for service at ambi-

ent temperatures is derived from the USAS B.96.1, now published as ASME B.96.1 :1999, (*Reference 10.6*). Alternatively, those interested could adapt and use the guidance given in API 620 Appendix Q (*Reference 10.7*) for service below temperatures of -60 °F.

10.2 Brittle fracture considerations

At the time that API 12 C was originally written, little or nothing was known about the phenomenon of brittle fracture and the factors which influenced it.

As storage tanks, particularly for oil-based products, increased in size, it was either a fortunate or an inspired decision of API 12 C to limit the maximum shell plate thickness to 1.5" (40 mm), a figure which remains as the limit to this day in BS 2654, prEN 14015 and for many materials in API 650 (in some cases a higher limit of 1.75" (45 mm) is permitted). Plate thickness is an important variable involved in the complex issue of brittle fracture avoidance in welded steel structures. As the knowledge surrounding this subject expanded, it was considered indeed fortunate that this limit had been imposed.

Early storage tanks were built in comparatively modest sizes using steels of low strengths. From the early 1960s onwards, there was an increasing demand for tanks of larger capacities, driven by the increasing volumes of oil-based products being transported and stored around the world. Large tanks mean that greater volumes can be stored on the same area of land, and many existing refineries and terminals were restricted in the amount of space available to them. This required the industry to leave the safe and well understood territory of small tanks, thin shells, weak steels and low joint factors. The appearance of BS 2654 : Part 3 (*Reference 10.8*) was an indication of this change.

The change to the use of stronger and thicker steels, higher joint factors and the increased consequences of a sudden failure in the new, larger tanks meant that the incomplete understanding of the factors surrounding the subject of brittle fracture needed to be addressed.

This was reinforced by the sudden failure whilst under hydrostatic test of a floating roof tank at the Esso Fawley Refinery in 1952 described in detail in *Reference 10.9*. A photograph of this tank after the event is shown in Figure 10.1. The floating roof is intact, but dumped on the ground some one quarter of a



Figure 10.1 The floating roof failure at Fawley in 1952

tank diameter laterally from its starting position, and the tank shell is literally cast around the site in pieces.

In the UK this work involving the Wells Wide Plate Tests, the Pellini Drop Weight Test, the introduction of the CTOD test and the study of the relationship between these and the more economical and convenient Charpy V-notch impact testing for material quality control, which is described in *Reference 10.1*.

Much of this work was sponsored by, and brought into a semblance of order, by the Oil Companies Materials Association Low Temperature committee, which was made up of technical experts from companies such as Shell, ICI and BP together with the Welding Institute. This group took upon itself the task of restructuring the requirements for brittle fracture avoidance and presented its recommendations to BSI. This work gave rise to the current requirements in BS 2654 where the Charpy V-notch impact test temperature is different from the design temperature. This is an essential difference between the BS and API approaches to material selection.

10.3 The design metal temperature

10.3.1 Minimum temperatures

The three design Codes all exclude from their scope the storage of products which are refrigerated below ambient temperatures. Many tanks are insulated and store products which are above ambient temperature, hence they are not fully stressed

at temperatures which are determined by the minimum temperatures to be expected at the particular location where they are to be constructed. Taking some credit for the thermal inertia of the tank and its contents, the design metal temperatures are not based on the absolute minimum temperatures to be statistically expected at the site, but are chosen based on the average minimum daily temperatures conditions to be expected plus an allowance for the thermal inertia of the stored product. When the tank is empty and will respond rapidly to the actual minimum temperatures, the stresses are low and it is argued that they will be insufficient to cause problems of possible brittle fracture.

The Codes describe the minimum design metal temperature as follows:

- **API 650** The design metal temperature shall be assumed to be 8 °C (15 °F) above the lowest one day mean ambient temperature of the locality of the area where the tank is to be installed. For mainland USA these are shown in Figure 10.2. For other areas of the world, suitable equivalent data must be obtained.
- **BS 2654** The design metal temperature shall be specified by the purchaser on the basis of the official weather reports over at least 30 years. The design metal temperature shall be the lower of the lowest daily mean temperature (one half of the daily maximum temperature plus the daily minimum temperature) plus 10 °C or the minimum temperature of the tank contents.

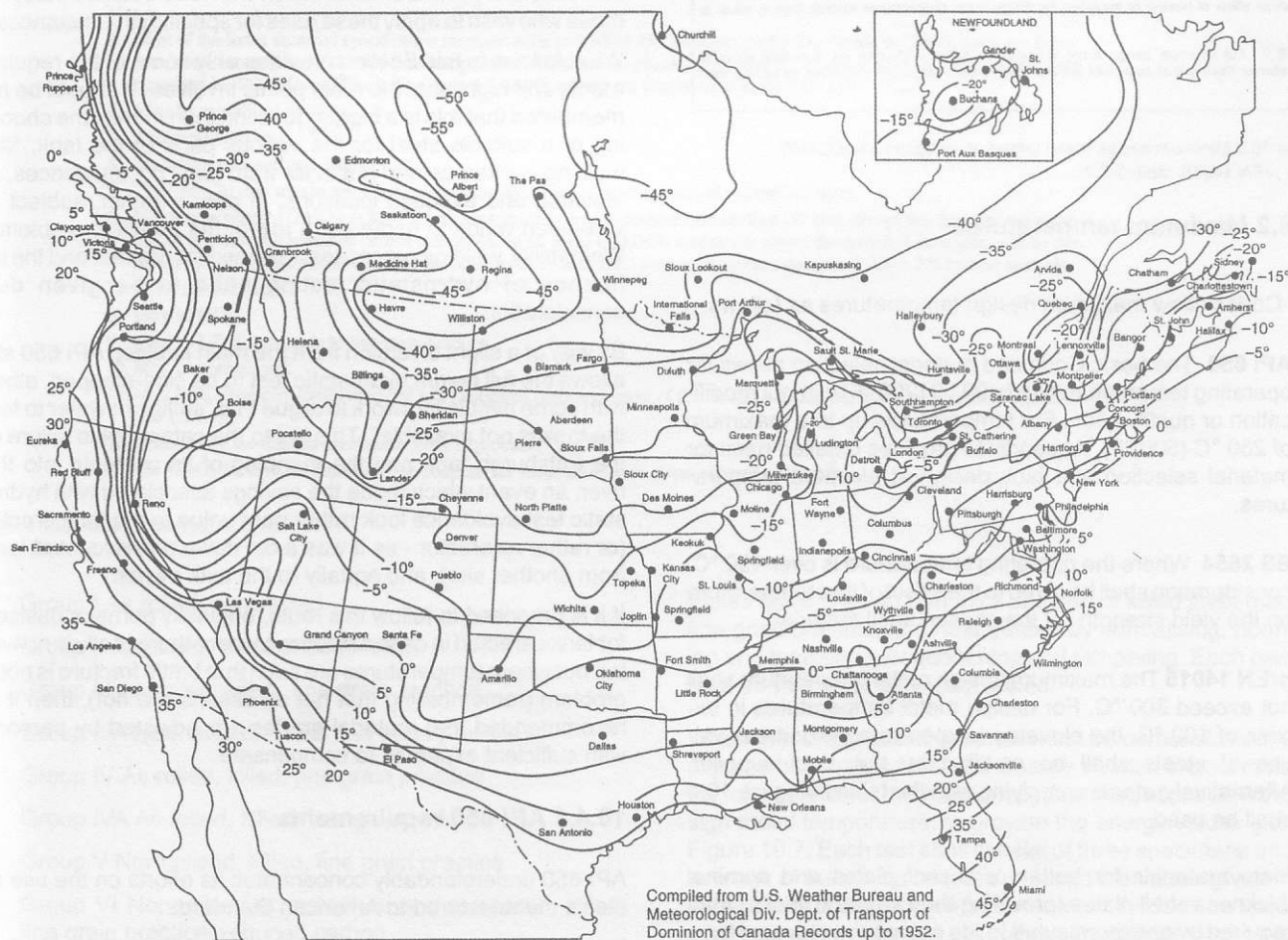


Figure 10.2 Isothermal lines of lowest one-day mean temperatures (°F)
From API 650, figure 2-2

For a storage tank constructed for service in the UK where the shell temperature is controlled by ambient conditions, the minimum design metal temperature shall not exceed 0 °C. For a storage tank constructed for use outside the UK and where no long term data or weather reports are available, the design metal temperature shall be the lower of the lowest daily mean temperature plus 5 °C and the minimum temperature of the contents.

In the interests of operational flexibility, the minimum design temperature shall not take into account the beneficial effects of heated or insulated tanks.

- **prEN 14015** The minimum design metal temperature shall be the minimum temperature of the contents or the temperatures given in Figure 10.3. The minimum design metal temperature shall not be lower than -40 °C. Note that this does allow some advantage to be taken of tank insulation or heating.

Lowest one day mean ambient temperature (LODMAT)	Minimum design metal temperature	
	10 years data	30 years data
Warmer than or equal to -10 °C	LODMAT +5 °C	LODMAT +10 °C
Warmer than or equal to -25 °C and below -10 °C	LODMAT	LODMAT +5 °C
Below -25 °C	LODMAT -5 °C	LODMAT

NOTE 1 LODMAT is the lowest recorded average temperature based over any 24 hour period. The average temperature is half (maximum temperature plus minimum temperature).

NOTE 2 The minimum design metal temperature for the tank shall not take into account the beneficial effect of heating or insulation for design metal temperatures warmer than or equal to 0 °C.

NOTE 3 For minimum design metal temperatures below 0 °C, then the beneficial effect of insulation or heating shall be agreed but the design metal temperature should not be warmer than 0 °C.

Figure 10.3 Minimum design metal temperature based on LODMAT
From prEN 14015, table 5.2.2

10.3.2 Maximum temperatures

The Codes allow maximum design temperatures as follows:

- **API 650** The basic Code and material selection allows for operating temperatures up to 90 °C (200 °F) without modification or qualification. For temperatures up to a maximum of 260 °C (500 °F), Appendix M provides detailed rules for material selection and tank design at elevated temperatures.
- **BS 2654** Where the operating temperature is over 150 °C, consideration shall be given to the effect of that temperature on the yield strength (of the chosen shell material).
- **prEN 14015** The maximum design metal temperature shall not exceed 300 °C. For design metal temperatures in excess of 100 °C, the elevated temperature yield stress values of steels shall be certified by the steel supplier. Alternatively, steels complying with the table in Figure 10.4 shall be used.

Plate materials for bottom and roof plates and nominal thickness shell plates (providing they are 20% thicker than required by design calculation) do not require elevated temperature yield stress values to be certified by the steel supplier. When the maximum design metal temperature exceeds 250 °C, steels which are proven to be unaffected by ageing shall be used. The method of proof shall be agreed between the tank contractor and the steel supplier.

Standard	Designation	Steel type as given in Figure 6.1.1	Maximum thickness ¹⁾ mm
EN 10028-2 1993	P235 GH	Type II	30
	P265 GH	Type II	30
	P295 GH	Type VI	40
	P355 GH	Type VI	40
EN 10028-3 1993	P275 N	Type IV	40
	P275 NH	Type IV	40
	P275 NL1	Type IV	40
	P275 NL2	Type IV	40
	P355 N	Type VIII	40
	P355 NH	Type VIII	40
	P355 NL1	Type IX	40
	P355 NL2	Type IX	40

¹⁾ The maximum thickness shall be the lower of that specified in this table and that derived from Figure 6.1.1.

NOTE CEV from ladle analysis ≤ 0.42 for plates thicker than 20 mm.

Figure 10.4 Hot rolled products for use at elevated temperatures (> 100 °C)
From prEN 14015, table 6.1.1-4

10.4 The requirements of the tank design Codes

All of the tank design Codes provide quite specific rules for material selection. Certain Codes, in particular API 650, provide a considerable amount of information on the subject and the various subsidiary requirements which will need detailed study by those who wish to apply these rules for specific circumstances.

What follows in this Section provides only some of the requirements and highlights the main points involved. It should be remembered that this is a bigger question than merely the choosing of a suitable steel for the various parts of the tank. Site welding is often carried out in far from ideal circumstances, at elevated and exposed locations, in poor weather, subject to salt-laden winds to name but a few of the practical problems. Weldability, welding processes, the need for preheat and the influence of hydrostatic testing need to be given due consideration.

By way of a slight diversion from the main subject, API 650 still allows the full height hydrostatic test to be side-stepped, albeit with some nimble footwork to argue that "sufficient water to test the tank is not available". This led to the catastrophic failure of the Pittsburgh tank and the dumping of its contents into the river, an event which made the savings associated with hydrostatic test avoidance look rather poor value to the tank erector (or rather re-erector - as it was a cut down and relocated tank from another site), and equally to the tank owner.

If it is proposed to follow this route, originally perhaps devised for tanks erected in desert locations where there really is no water, but where temperatures are such that brittle fracture is not a problem (remembering that not all deserts are hot), then it is recommended that material grades are adjusted by persons with sufficient expertise to compensate.

10.4.1 API 650 requirements

API 650 understandably concentrates its efforts on the use of steels manufactured to American Standards.

It does provide guidance for the use of steels made to Canadian (CSA) Standards, some ISO Standards and general rules for the use of steels made to other national Standards.

The steels are placed in eight categories in generally ascending order of toughness. These are:

Group I As Rolled, Semikilled		Group II As Rolled, Killed or Semikilled		Group III As Rolled, Killed Fine-Grain Practice		Group IIIA Normalized, Killed Fine-Grain Practice	
Material	Notes	Material	Notes	Material	Notes	Material	Notes
A 283M C	2	A 131M B	7	A 573M-400		A 131M CS	
A 285M C	2	A 36M	2, 6	A 516M-380		A 573M-400	10
A 131M A	2	G40.21M-260W		A 516M-415		A 516M-380	10
A 36M	2, 3	Grade 250	5, 8	G40.21M-260W	9	A 516M-415	10
Grade 235	3, 5			Grade 250	5, 9	G40.21M-260W	9, 10
Grade 250	6					Grade 250	5, 9, 10

Group IV As Rolled, Killed Fine-Grain Practice		Group IVA As Rolled, Killed Fine-Grain Practice		Group V Normalized, Killed Fine-Grain Practice		Group VI Normalized or Quenched and Tempered, Killed Fine-Grain Practice Reduced Carbon	
Material	Notes	Material	Notes	Material	Notes	Material	Notes
A 573M-450		A 662M C		A 573M-485	10	A 131M EH 36	
A 573M-485		A 573M-485	11	A 516M-450	10	A 633M C	
A 516M-450		G40.21M-300W	9, 11	A 516M-485	10	A 633M D	
A 516M-485		G40.21M-350W	9, 11	G40.21M-300W	9, 10	A 537M Class 1	
A 662M B				G40.21M-350W	9, 10	A 537M Class 2	13
G40.21M-300W	9					A 678M A	
G40.21M-350W	9					A 678M B	13
E 275	4, 9					A 737M B	
E 355	9					A 841	12, 13
Grade 275	5, 9						

Notes:

- Most of the listed material specification numbers refer to ASTM specifications (including Grade or Class); there are, however, some exceptions: G40.21M (including Grade) is a CSA specification; Grades E 275 and E 355 (including Quality) are contained in ISO 630; and Grade 37, Grade 41, and Grade 44 are related to national standards (see 2.2.5).
- Must be semikilled or killed.
- Thickness ≤ 20 mm.
- Maximum manganese content of 1.5%.
- Thickness 20 mm maximum when controlled-rolled steel is used in place of normalized steel.
- Manganese content shall be 0.80–1.2% by heat analysis for thicknesses greater than 20 mm, except that for each reduction of 0.01% below the specified carbon maximum, an increase of 0.06% manganese above the specified maximum will be permitted up to the maximum of 1.35%. Thicknesses ≤ 20 mm shall have a manganese content of 0.8–1.2% by heat analysis.
- Thickness ≤ 25 mm.
- Must be killed.
- Must be killed and made to fine-grain practice.
- Must be normalized.
- Must have chemistry (heat) modified to a maximum carbon content of 0.20% and a maximum manganese content of 1.60% (see 2.2.6.4).
- Produced by the thermo-mechanical control process (TMCP).
- See 3.7.4.6 for tests on simulated test coupons for material used in stress-relieved assemblies.

Figure 10.5 Material groups, SI Units
From API 650, table 2-3a

- Group I As rolled, semi-killed
- Group II As rolled, killed or semi-killed
- Group III As rolled, killed, fine grain practice
- Group IIIA Normalised, killed, fine grain practice
- Group IV As rolled, killed, fine grain practice
- Group IVA As rolled, killed, fine grain practice
- Group V Normalised, killed, fine grain practice
- Group VI Normalised or quenched and tempered, killed, fine grain practice, reduced carbon

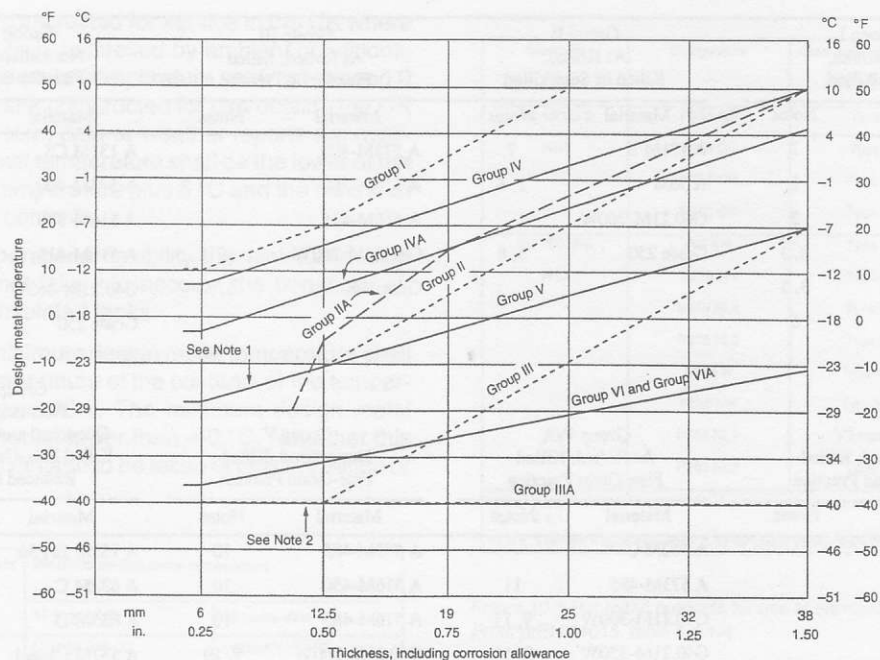
This listing is shown in Figure 10.5.

Plates less than or equal to 40 mm thickness can be used at or above the design metal temperatures indicated by Figure 10.6, without being impact tested.

Plates more than 40 mm thick shall be of killed steel made to fine grain practice, and heat treated by normalising, normalising and tempering, or quenching and tempering. Each plate as heat treated shall be impact tested.

When the toughness of the steel must be demonstrated, each plate as heat treated shall be Charpy V-notch impact tested in the longitudinal (or the transverse) direction, at or below the design metal temperature, to provide the energy values given in Figure 10.7. Each test shall consist of three specimens and the average of these shall equal or exceed the values given in the Table. If any one specimen falls below two thirds of the specified minimum value, a further set of three specimens shall be taken and each must equal or exceed the specified minimum value.

For thin plates where sub-size specimens must be taken, the energy values shall be at least proportional to the values required for full size specimens.



Notes:

1. The Group II and Group V lines coincide at thicknesses less than 12.5 mm ($1/2$ in.).
2. The Group III and Group IIIA lines coincide at thicknesses less than 12.5 mm ($1/2$ in.).
3. The materials in each group are listed in Table 2-3.
4. This figure is not applicable to controlled-rolled plates (see 2.2.7.4).
5. Use the Group IIA and Group VIA curves for pipe and flanges (see 2.5.5.2 and 2.5.5.3).

Figure 10.6 Minimum permissible design metal temperature for materials used in tank shells without impact testing
From API 650, figure 2-1

Plate Material ^a and Thickness (<i>t</i>) in mm (in.)	Thickness		Average Impact Value of Three Specimens ^b			
			Longitudinal		Transverse	
	mm	in	J	ft-lbf	J	ft-lbf
Groups I, II, III, and IIIA <i>t</i> ≤ maximum thicknesses in 2.2.2 through 2.2.5			20	15	18	13
Groups IV, IVA, V, and VI (except quenched and tempered and TMCP)	<i>t</i> ≤ 40	<i>t</i> ≤ 1.5	41	30	27	20
	40 < <i>t</i> ≤ 45	1.5 < <i>t</i> ≤ 1.75	48	35	34	25
	45 < <i>t</i> ≤ 50	1.75 < <i>t</i> ≤ 2	54	40	41	30
	50 < <i>t</i> ≤ 100	2 < <i>t</i> ≤ 4	68	50	54	40
Group VI (quenched and tempered and TMCP)	<i>t</i> ≤ 40	<i>t</i> ≤ 1.5	48	35	34	25
	40 < <i>t</i> ≤ 45	1.5 < <i>t</i> ≤ 1.75	54	40	41	30
	45 < <i>t</i> ≤ 50	1.75 < <i>t</i> ≤ 2	61	45	48	35
	50 < <i>t</i> ≤ 100	2 < <i>t</i> ≤ 4	68	50	54	40

a) See Table 2-3.

b) Interpolation is permitted to the nearest joule (ft-lbf).

Note: For plate ring flanges, the minimum impact test requirements for all thicknesses shall be those for *t* ≤ 40 mm (1.5 in.).

Figure 10.7 Minimum impact test requirements for plates
From API 650, table 2-4

In addition to the requirements for plates, the Code provides details of material selection rules for structural shapes, piping and forgings, flanges and bolting.

In fear of becoming tediously repetitive, it must be remembered that this section of the Code is a minefield of detailed requirements for material selection and the advice of those familiar with its use would be well worth seeking.

The requirements for the mechanical and toughness properties of weld-metal and heat affected zone (HAZ), are quite complex and are probably best left to those familiar with this Code and its various provisions.

In simple terms the following briefly summarises the requirements:

- The welding procedures shall produce weldments with the mechanical properties required by the design

- The materials shall be considered in three groups dependent upon their minimum tensile strength:
 - Less than 485 N/mm² — group 1 requiring 20 J average of three full size specimens
 - Equal to or greater than 485 N/mm² but less than 550 N/mm² — group 2 requiring 27 J average of three full size specimens
 - Greater than 550 N/mm² — group 3 requiring 34 J average of three full size specimens

For plates thicker than 40 mm, enhanced values are required.

API 650 allows plates to be ordered on an edge thickness or a weight basis. The edge thickness ordered shall not be less than the computed design thickness or the minimum permitted thickness. Similarly, the plate weight ordered shall be great enough to provide an edge thickness not less than the computed design

thickness or the minimum permitted thickness. For plates ordered on either basis, an under-run of not more than 0.01" is permitted for both computed and minimum permitted thickness plates.

10.4.2 BS 2654 requirements

It should be remembered that BS 2654 has been the subject of standstill for a number of years now due to the work being carried out in the preparation of the new European Code prEN 14015. This means that it quotes materials to British Standards which have been superseded by European Standards. For example, BS 4360 (Reference 10.10) has been replaced by EN 10025 (Reference 10.11).

Steels shall be made by the open hearth, electric furnace or one of the basic oxygen processes. Semi- and fully-killed steels are permitted, but Bessemer and rimming steel are excluded.

The carbon equivalent based on the ladle analysis shall not exceed 0.43% for plates from 20 mm up to 25 mm thick and 0.42% for plates thicker than 25 mm. The carbon equivalent is calculated using the following formula:

$$CE = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Ni + Cu}{15} \quad \text{equ 10.1}$$

The carbon equivalent based on the check analysis shall not exceed 0.43% calculated using the following formula:

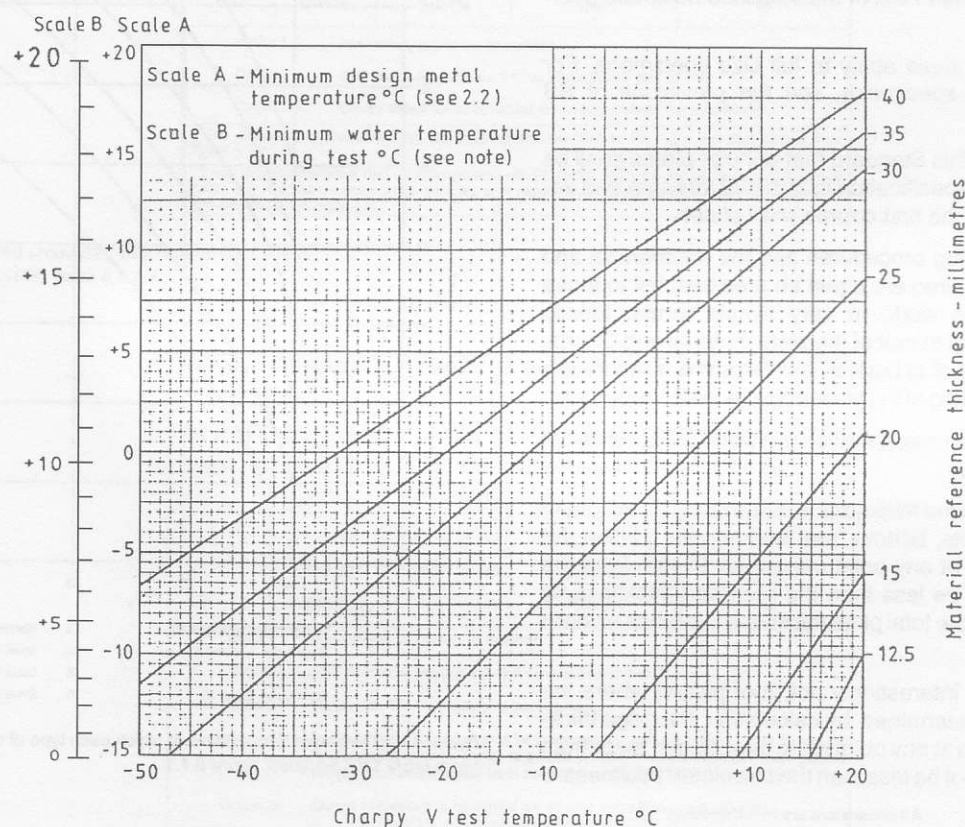
$$CE = C + \frac{Mn}{6} \quad \text{equ 10.2}$$

For steels with a minimum tensile strength greater than 420 N/mm², the phosphorus plus the sulphur shall not exceed 0.08%.

Steels shall be either aluminium treated with a minimum aluminium/nitrogen ratio of 2:1 or have a nitrogen content of less than 0.01%.

The following impact properties are required:

- For plate thicknesses not exceeding 13 mm in materials with specified minimum tensile strengths up to and including 490 N/mm², impact tests are not required
- Materials with specified minimum tensile strengths less than or equal to 430 N/mm², thicker than 13 mm shall be impact tested to show not less than 27 J at +20 °C or at the test temperature indicated in Figure 10.8, whichever is the



(Intermediate values may be determined by interpolation.)

NOTE. Scale A on the ordinate is to be used in determining minimum Charpy V requirements for the thickness and minimum design temperature concerned. For the purposes of this note, conversion of the measured impact value to the 27 J (or 41 J for steels with specified minimum tensile strength greater than 430 N/mm²) value may be made on the basis of 1.35 J per °C, such extrapolation being limited to a maximum range of 20 °C. For example, if the actual value by test is 33.75 J at -20 °C for a steel of specified minimum tensile strength greater than 400 N/mm², the equivalent test temperature for 27 J may be assumed to be -25 °C.

The requirements derived from scale A take into account an improvement in safety which may be anticipated as a result of the hydrostatic test. During the first hydrostatic test the degree of security against brittle fracture may be rather less than on subsequent loading. Attention is drawn to the more conservative requirements of scale B when consideration is to be given to the use of this scale during hydrostatic testing of tank shells constructed of steels with specified minimum tensile strength greater than 430 N/mm². The application of scale B, or any alternative procedure regarding the precautions to be taken during water testing to safeguard the tank from brittle fracture, is the subject of agreement between the purchaser and the manufacturer (see 3.3(b)).

Figure 10.8 Minimum Charpy V-notch impact requirements
From BS 2654 : 1989, Figure 1

lower. Three specimens shall be tested, the value taken being the average of the three results. The minimum individual value shall not be less than 70% of the specified minimum average value.

Note: Provided the design metal temperature is +10 °C or above, it is not necessary to test materials with a specified minimum yield strength not exceeding 300 N/mm², and less than 20 mm thick.

- Materials with specified minimum tensile strengths greater than 430 N/mm² and up to 490 N/mm² thicker than 13 mm shall be impact tested to show not less than 41 J at -5 °C or at the test temperature indicated in Figure 10.8, whichever is the lower. Three specimens shall be tested, the value taken being the average of the three results. The minimum individual value shall not be less than 70% of the specified minimum average value.
- Materials with specified minimum tensile strengths greater than 490 N/mm² and of all thicknesses shall be impact tested to show not less than 41 J at -15 °C or at the test temperature indicated in Figure 10.8, whichever is the lower. Three specimens shall be tested, the value taken being the average of the three results. The minimum individual value shall not be less than 70% of the specified minimum average value.

Note: The energy values apply to full size specimens. For sub-standard specimens, see the provisions of BS 4360.

It is a requirement of this Standard that annular plates shall be of the same material specification in terms of strength and impact requirements as the first course shell plates.

The approval of welding procedures and the mechanical and toughness values required are again an area best left to those experienced with this work. In very simple terms, tensile strengths at least equal to that of the plate material and Charpy V-notch impact values of at least 27J at the same temperature as required for the testing of the plate material will be required.

For thickness requirements, the rules are slightly different from those given in API 650.

For shell plates where the thickness is determined by minimum thickness requirements, bottom, roof and annular plates, the thickness (measured at any point more than 15 mm from the plate edge) shall not be less than the specified thickness by more than one half of the total plate thickness tolerance given in Figure 10.9.

For shell plates (but, interestingly not, roof plates) where the thickness has been determined by calculation, the edge thickness (again measured at any point more than 15 mm away from the plate edge) shall not be less than the calculated thickness.

All dimensions are in millimetres.

Specified thickness	Width of plate				
	Up to and including 2000	Over 2000 up to and including 2500	Over 2500 up to and including 3000	Over 3000 up to and including 3500	Over 3500 up to and including 4000
Under 5	0.80	1.00	1.00		
5 to under 8	0.90	1.20	1.60	1.60	
8 to under 12.5	1.10	1.30	1.60	1.70	1.90
12.5 to under 25	1.10	1.30	1.60	1.70	1.90
25 to under 40	1.10	1.40	1.60	1.70	1.90
40 to under 80	1.20	1.60	1.70	1.90	2.10
80 to under 150	2.20	2.30	2.40	2.50	2.50

NOTE. See 19.3.2 which states that, unless otherwise specified, the thickness tolerance shall be half the total thickness tolerance given in table 8 over and under the specified thickness.

Figure 10.9 Total thickness tolerances for plates
From BS 4360:1979, table 8

10.4.3 prEN 14015 requirements

Rather than present basic requirements for the toughness/temperature/steel strength combinations, prEN 14015 gives specific steel types taken from the various European steel Standards for particular circumstances. The steel Standards are EN 10025, EN 10028 (Reference 10.12), EN 10113 (Reference 10.13) and EN 10210 (Reference 10.14).

Steels shall be selected by the use of Figures 10.10 to 10.14.

Impact testing shall be carried out in accordance with EN 10045-1 (Reference 10.15). The 70% rule again applies to the minimum individual specimen value.

When the material is less than 10 mm thick, 10 mm x 5 mm specimens shall be taken which shall demonstrate 70% of the energy values specified for full sized specimens.

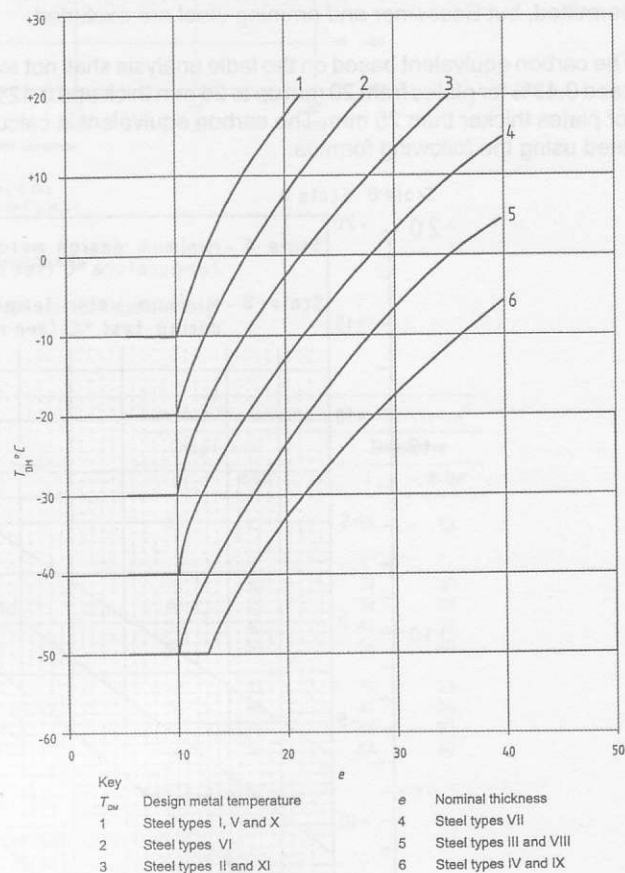


Figure 10.10 Minimum temperature at which each type of steel can be used
From prEN 14015-1:2000, figure 6.1.1

Standard	Designation	Options	Steel type as given in Figure 6.1.1	Maximum thickness ¹⁾ mm
EN 10025 1993	S235 JRG2	1 - 12	Type I	12
	S235 JO	1 - 5 - 12	Type II	30
	S235 J2G3	1 - 5 - 12	Type III	40
	S235 J2G4	1 - 5 - 12	Type III	40
	S275 JR	1 - 12	Type I	12
	S275 JO	1 - 5 - 12	Type II	30
	S275 J2G3	1 - 5 - 12	Type III	40
	S275 J2G4	1 - 5 - 12	Type III	40
Option 1 Steelmaking process to be reported Option 5 CEV from ladle analysis ≤ 0.42 for plates thicker than 20 mm Option 12 Inspection documentation shall be in accordance with EN 10204 Cert. 3.1B except for nominal thickness plates (e.g. roof, bottom, nominal thickness shell plates) where documentation shall be in accordance with EN 10204 Test report 2.2				
Standard	Designation	Options ²⁾	Steel type as given in Figure 6.1.1	Maximum thickness ¹⁾ mm
EN 10113-2 1993	S275 N	1 - 2-19a	Type IV	40
	S275 NL	1 - 2-19a	Type IV	40
EN 10113-3 1993	S275 M	1 - 2-19a	Type IV	40
	S275 ML	1 - 2-19a	Type IV	40
Option 1 Steelmaking process to be reported Option 2 CEV from ladle analysis ≤ 0.42 for plates thicker than 20 mm Option 19a Charpy Impact test to be carried out on each plate thicker than 20 mm				
¹⁾ The maximum thickness shall be the lower of that specified in this table and that derived from Figure 6.1.1. ²⁾ Inspection documentation shall be in accordance with EN 10204 Cert 3.1 B except for nominal thickness plates (e.g. roof, bottom and nominal thickness shell plates) where documentation shall be in accordance with EN 10204 Test report 2.2.				

Figure 10.11 Hot rolled products $\leq 275 \text{ N/mm}^2$ yield stress
From prEN 14015-1:2000, table 6.1.1-1

Standard	Designation	Options	Steel type as given in Figure 6.1.1	Maximum thickness ¹⁾ mm
EN 10025 1993	S355 JR	1 - 6 - 12	Type V	10
	S355 JO	1 - 6 - 12	Type VI	15
	S355 J2G3	1 - 5 - 6 - 12 - 20	Type VII	40
	S355 J2G4	1 - 5 - 6 - 12 - 20	Type VII	40
	S355 K2G3	1 - 5 - 6 - 12 - 20	Type VIII	40
	S355 K2G4	1 - 5 - 6 - 12 - 20	Type VIII	40
Option 1 Steelmaking process to be reported Option 5 CEV from ladle analysis ≤ 0.42 for plates thicker than 20 mm Option 6 Cr, Cu, Mo, Nb, Ni, Ti and V to be recorded Option 12 Inspection documentation shall be in accordance with EN 10204 Cert. 3.1B except for nominal thickness plates (e.g. roof, bottom, nominal thickness shell plates) where documentation shall be in accordance with EN 10204 Test report 2.2 Option 20 Charpy Impact test to be carried out on each plate thicker than 20 mm				
Standard	Designation	Options ²⁾	Steel type as given in Figure 6.1.1	Maximum thickness ¹⁾
EN 10113-2 1993	S355 N	1 - 2 - 19a	Type VIII	40
	S355 NL	1 - 2 - 19a	Type IX	40
EN 10113-3 1993	S355 M	1 - 2 - 19a	Type VIII	40
	S355 ML	1 - 2 - 19a	Type IX	40
Option 1 Steelmaking process to be reported Option 2 CEV from ladle analysis ≤ 0.42 for plates thicker than 20 mm Option 19a Charpy Impact test to be carried out on each plate thicker than 20 mm				
¹⁾ The maximum thickness shall be the lower of that specified in this table and that derived from Figure 6.1.1. ²⁾ Inspection documentation shall be in accordance with EN 10204 Cert 3.1 B except for nominal thickness plates (e.g. roof, bottom and nominal thickness shell plates) where documentation shall be in accordance with EN 10204 Test report 2.2.				

Figure 10.12 Hot rolled products $> 275 \text{ N/mm}^2$ and $\leq 355 \text{ N/mm}^2$ yield stress
From prEN 14015-1:2000, table 6.1.1-2

Standard	Designation	Options ^{a)}	Steel type as given in Figure 6.1.1	Maximum thickness ^{a)} mm
EN 10113-2	S420 N	1 - 2 - 19a	Type X	40
1993	S420 NL	1 - 2 - 19a	Type XI	40
EN 10113-3	S420 M	1 - 2 - 19a	Type X	40
1993	S420 ML	1 - 2 - 19a	Type XI	40
Option 1 Steelmaking process to be reported				
Option 2 CEV from ladle analysis ≤ 0.42 for plates thicker than 20 mm				
Option 19a Charpy Impact test to be carried out on each plate thicker than 20 mm				
^{a)} The maximum thickness shall be the lower of that specified in this table and that derived from Figure 6.1.1.				
^{a)} Inspection documentation shall be in accordance with EN 10204 Cert 3.1 B except for nominal thickness plates (e.g. roof, bottom and nominal thickness shell plates) where documentation shall be in accordance with EN 10204 Test report 2.2.				

Figure 10.13 Hot rolled products > 355 N/mm² yield stress
From prEN 14015-1:2000, table 6.1.1-3

Standard	Designation	Steel type
EN 10210-1	S235 JRH	Type I
1997	S275 JOH	Type II
	S275 J2H	Type III
	S275 NH	Type IV
	S275 NLH	Type IV
	S355 JOH	Type VI
	S355 J2H	Type VIII
	S355 NH	Type VIII
	S355 NLH	Type IX

Figure 10.14 Structural steel products
From prEN 14015-1:2000, table 6.1.2

Minimum design metal temperature °C	Thickness mm
$\geq +10$	≤ 20
≥ 0	≤ 13
≥ -10	≤ 10
< -10	≤ 6

Figure 10.15 Conditions for waiving impact testing
From prEN 14015-1:2000, table 6.1.6

Impact testing is not required for bottom plates other than annular plates.

Impact testing of annular plates is not required when the shell plate attached to them does not require impact testing.

Impact testing of shell plates and items attached to them may be waived according to the conditions provided in Figure 10.15.

For stainless steels a number of general rules are provided and a table of acceptable austenitic steels is given in Figure 10.16. Ferritic steels may be used up to a maximum thickness of 10 mm.

Information is also provided for the material selection of mountings, flanges, structural sections, pipes and welding consumables.

For materials which have been produced to specifications other than the nominated European Standards, Annex F provides detailed requirements for their selection and use.

The requirements for weld-metal and HAZ properties are again subjects requiring detailed study. The basic requirements can be summarised by:

Grade	Steel designation	Number
Austenitic		
X2CrNi18-9		1.4307
X2CrNi19-11		1.4306
X2CrNi18-10		1.4311
X5CrNi18-10		1.4301
X8CrNiS18-9		1.4305
X6CrNiTi18-10		1.4541
X6CrNiNb18-10		1.4550
X1CrNi25-21		1.4335
X2CrNiMo17-12-2		1.4404
X2CrNiMoN17-11-2		1.4406
X5CrNiMo17-12-2		1.4401
X1CrNiMoN25-22-2		1.4466
X6CrNiMoTi17-12-2		1.4571
X6CrNiMoNb17-12-2		1.4580
X2CrNiMo17-12-3		1.4432
X2CrNiMoN17-13-3		1.4429
X2CrNiMo17-13-3		1.4436
X2CrNiMo18-14-3		1.4435
X2CrNiMoN18-12-4		1.4434
X2CrNiMoN18-15-4		1.4438
X2CrNiMoN17-13-5		1.4439
X1NiCrMoCu31-27-4		1.4563
X1NiCrMoCu25-20-5		1.4539
X1CrNiMoCuN25-25-5		1.4537
X1CrNiMoCuN20-18-7		1.4547
X1CrNiMoCuN25-20-7		1.4529
Austenitic-ferritic		
X2CrNiN23-4		1.4362
X2CrNiMoN22-5-3		1.4462
X2CrNiMoCuN25-6-3		1.4507
X2CrNiMoN25-7-4		1.4410
X2CrNiMoCuWN25-7-4		1.4501
Stainless steels selected from EN 10088-1		

Figure 10.16 Stainless steels for tank fabrication
From prEN 14015-1:2000, table 6.2.1

- The approval procedure shall demonstrate that the yield stress and tensile stress of the welded joint shall exceed the minimum required values of the materials being joined.
- Vertical shell welds shall be impact tested at the test temperature required for the plate material and shall show not less than the value required for the thicker plate material being joined.
- Horizontal shell welds shall be impact tested at the test temperature of the thicker plate being joined, or at -10 °C, whichever is the least stringent, and show not less than 27 J.

The thickness requirements are similar to those of BS 2654. Specifically they are:

- The measured thickness at any point more than 25 mm from the edge of any nominal thickness bottom, shell, roof or annular plate shall not be less than the specified thickness less one half of the total thickness specified in EN 10029: Table 1: class D (Reference 10.14 and Figure 10.17)
- The measured thickness at any point more than 25 mm from the edge of shell and roof plates whose thickness has been calculated shall not be less than the calculated minimum thickness (i.e. to EN 10029: Table 1: class C – only positive tolerances).

Dimensions in mm

Nominal thickness	Tolerances on the nominal thickness (see 7.1.1) ^a								Maximum thickness difference within a plate					
	Class A		Class B		Class C		Class D		Nominal plate width					
	Lower	Upper	Lower	Upper	Lower	Upper	Lower	Upper	≥ 600 < 2 000	≥ 2 000 < 2 500	≥ 2 500 < 3 000	≥ 3 000 < 3 500	≥ 3 500 < 4 000	≥ 4 000
≥ 3 < 5	-0,4	+0,8	-0,3	+0,9	-0	+1,2	-0,6	+0,6	0,8	0,9	0,9	-	-	-
≥ 5 < 8	-0,4	+1,1	-0,3	+1,2	-0	+1,5	-0,75	+0,75	0,9	0,9	1,0	1,0	-	-
≥ 8 < 15	-0,5	+1,2	-0,3	+1,4	-0	+1,7	-0,85	+0,85	0,9	1,0	1,0	1,1	1,1	1,2
≥ 15 < 25	-0,6	+1,3	-0,3	+1,6	-0	+1,9	-0,95	+0,95	1,0	1,1	1,2	1,2	1,3	1,4
≥ 25 < 40	-0,8	+1,4	-0,3	+1,9	-0	+2,2	-1,1	+1,1	1,1	1,2	1,2	1,3	1,3	1,4
≥ 40 < 80	-1,0	+1,8	-0,3	+2,5	-0	+2,8	-1,4	+1,4	1,2	1,3	1,4	1,4	1,5	1,6
≥ 80 < 150	-1,0	+2,2	-0,3	+2,9	-0	+3,2	-1,6	+1,6	1,3	1,4	1,5	1,5	1,6	1,7
≥ 150 < 250	-1,2	+2,4	-0,3	+3,3	-0	+3,6	-1,8	+1,8	1,4	1,5	1,6	1,6	1,7	-

^a These thickness tolerances apply outside ground areas (see 7.1.2)

Figure 10.17 Tolerances on thicknesses
From EN 10029:1991, table 6.2.1

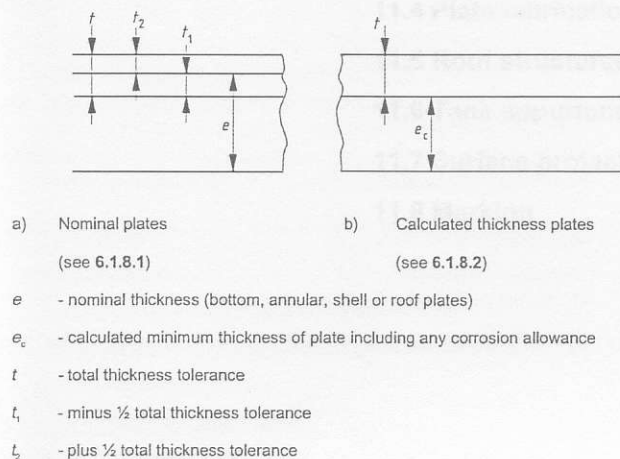


Figure 10.18 Plate thickness tolerances
From prEN 14015-1:2000, table 6.1.8

This is illustrated in Figure 10.18, which it is hoped will clarify this matter. It is curious just how often this apparently simple matter is misunderstood or merely gets into a muddle between the various parties involved, particularly where corrosion allowances are applied.

10.5 References

- 10.1 *A Review of the Development of Fracture Safe Designs and Codes for Oil and LPG Storage Tanks*, H.C.Cotton and J.B.Denham.
- 10.2 *API 12 C Specification for Welded Oil Storage Tanks*, American Petroleum Institute (fifteen editions from 1936 to 1961).
- 10.3 *API 650: Tenth edition, November 1998: Welded Steel Tanks for Oil Storage*, API Washington.
- 10.4 *BS 2654: 1989: British Standard for the manufacture of vertical steel welded non-refrigerated storage tanks with butt welded shells for the petroleum industry*, BSI London.
- 10.5 *prEN 14015-1: October 2000: Specification for the design and manufacture of site built, vertical, cylindrical, flat-bottomed, above ground, welded, metallic tanks for the storage of liquids at ambient temperatures and above – Part 1: Steel tanks*, CEN Brussels.
- 10.6 *ASME B 96.1:1999 – Specification for welded aluminium-alloy field-erected storage tanks*.
- 10.7 *API 620: Tenth edition, February 2002: Design and Construction of Large, Welded, Low-pressure Storage tanks: Appendix Q: Low-pressure Storage Tanks for Liquefied Hydrocarbon Gases*, API Washington.
- 10.8 *BS 2654: Part 3 :1968 – Higher design stresses*, BSI London.
- 10.9 *Why Storage Tanks Fail*, F.J.Feely and M.S.Northup, The Oil and Gas Journal, February 1954.
- 10.10 *BS 4360:1979 – Specification for weldable structural steels*.
- 10.11 *EN 10025 : Hot rolled products of non alloy structural steels – Technical delivery conditions: 1993*.
- 10.12 *EN 10028-2: Flat products made of steels for pressure vessel purposes-Part 2: Non alloy and alloy steels with specific elevated properties – 1993 and EN 10028-3:Flat products made from steel for pressure vessel purposes – Part 3: Weldable fine grain steels normalised – 1993*.
- 10.13 *EN 10113-2: Hot rolled products in weldable fine grain structural steels – Part 2: Delivery conditions for normalised/normalised rolled steels – 1993 and EN 10113-3: Hot rolled products in weldable fine grain structural steels – Part 2 : Delivery conditions for thermo-mechanical rolled steels – 1993*.
- 10.14 *EN 10210-1 Hot finished structural hollow sections of non-alloy and fine grain structural steels - Part 1 : Technical delivery conditions*.
- 10.15 *EN 10029:1991 – Specification for tolerances on dimension, shape and mass for hot rolled steel plates 3 mm thick and above*.

11 Fabrication considerations for ambient temperature tanks

In this Chapter some of the more important aspects of tank fabrication are outlined, together with advice on good practices which should be observed.

Contents:

11.1 Material reception

11.2 Stainless steel materials

11.3 Plate thickness tolerances

11.4 Plate fabrication

11.5 Roof structures

11.6 Tank appurtenances

11.7 Surface protection for plates and sections

11.8 Marking

11.1 Material reception

All materials received into the fabrication area or workshop must be checked for conformity with the requirements set out in the purchase order to the supplier in terms of quantity, quality, dimensions, surface finish, appearance, inspection documentation, material certificates and where applicable, installation and maintenance documentation etc.

The steel plates and sections which will form the liquid containing elements of the tank must be carefully checked against the mill certificates provided with the steel to ensure that the physical and chemical properties are in accordance with the steel specification that they were ordered against. It is common practice for the purchaser's inspector (and any third party inspector, as appropriate) to inspect material prior to despatch from the steel mill.

11.2 Stainless steel materials

When fabricating in stainless steel materials within an area where carbon steel materials are also fabricated, it is very important to keep these materials separate from any carbon steel materials in order to prevent any surface contamination of the stainless steel by carbon steel scale, filings, weld or grinding splatter and swarf.

The recommended course of action in such cases is to quarantine an area of the workshop for use exclusively for stainless steel fabrication. The proposed fabrication area should be cleaned of all carbon steel detritus and the floor sealed with a proprietary non-slip concrete sealant. A typical quarantined area is shown in Figure 11.1. Care must be taken especially in handling and placing plates, any plate grabs, handling equipment and lay down cradles should be faced in stainless steel, or in the case of cradle supports, these can be faced with timber. The plates should be covered when not being worked on to prevent contamination by airborne particles.

When rolling shell plates to curvature, the rolls of the machine should be covered with strong template paper to prevent any carbon steel particles from being impressed into the surface of the plate. Failure to do this can result in rust streaking on the plates when they have been erected on site and this is very difficult, if not impossible and very expensive to, completely remove.

Fabrication personnel must be discouraged from walking on the plates as boot marks are also hard to remove and are unsightly on the external surface of the tank. Stainless steel plates are often supplied from the mill on timber pallets and these may

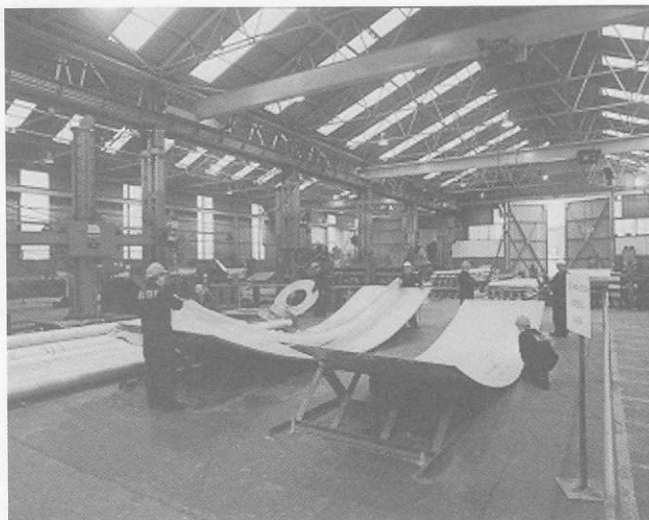


Figure 11.1 Quarantined area for stainless steel fabrication
Courtesy of McTay

be re-used for plate storage between marking, cutting and rolling operations.

Some mills will supply the plate with a plastic film fixed to one or both sides of the plate, this only being removed after erection, welding and weld pickling is completed at site. Care on the selection of the type of film and adhesive is important, as it has been known for the adhesive to be very reluctant in releasing the film, resulting in strips being left on the plate surface. Also, if the adhesive is not completely removed from the steel, a tacky coating is left on the tank surface, which attracts atmospheric grime and dust. There are accepted test methods available, which can detect carbon steel contamination of the stainless steel materials, and use of these can obviate embarrassing blemishes appearing on the tank during or after erection on site.

11.3 Plate thickness tolerances

In determining the allowable plate thickness tolerances the BS 2654 Code groups tank plates into two categories as follows:

- 1) Shell plates whose thickness has been determined by reference to the table of "Minimum specified shell thickness" given in the Code (i.e. shell plates for which the thickness by calculation, is less than the minimum allowed for a given tank diameter).

Annular floor plates, floor plates and roof plates. These plates shall have a minimum thickness not less than the specified thickness less half the total tolerance given in the table of BS EN 10029, class D.

In simple terms these plates are allowed to be thinner than their specified thickness.

- 2) For shell plates whose thickness have been determined by calculation and that are thicker than the "Minimum specified thickness", for a given tank diameter, the thickness of these plates shall not be less than the calculated thickness, i.e. table 1 of BS EN 10029, class C.

This means that these plates can not be thinner than the calculated thickness.

The API 650 Code has a simpler approach stating that all shell, annular floor, floor and roof plates may have an underrun on calculated or minimum permitted thickness of not more than 0.25 mm.

11.4 Plate fabrication

Floor and roof plates (which are generally, but not always, of lapped construction) which are produced in a reversing mill, do not require any edge preparation, as the mill production process gives a square edge to the plates which is suitable for fillet welding. Plates produced by a strip mill will have rounded edges making root penetration difficult during fillet welding and in order to ensure a sound weld there are two alternatives;

- a) Use two runs of weld, the first to ensure root penetration and the second as a capping run.
- b) Trim the plate edges square thus giving a suitable weld preparation.

Rectangular lap-welded roof plates which are laid on to a supporting structure are flat plates, usually in the range of 1.5 m x 4.8 m to 2.0 m x 6.0 m, this is to allow these relatively small plates to form naturally to the curvature of the roof.

Rectangular lap welded floor plates are generally supplied in two size ranges, depending on the tank diameter:

Tanks up to 12.5 m in diameter	1.5 m x 4.8 m
Tanks > 12.5 m in diameter	2.0 m x 7.85 m

Tanks > 12.5 m in diameter have a ring of thicker annular floor plates and the number of annular plates is usually the same as the number of shell plates per course. This is in order to main-

tain a constant spacing between the butt welds in the annular plates and the first shell course vertical butt welds all around the tank. However, larger tanks having shell plates approaching 10 m long, may have two annular plates per shell plate. This is to allow narrower annular plates to be used.

Floor plates larger than those quoted above may be difficult to handle due to the flexibility of large, thin flat plates.

The shell plate length and width shall be cut to a tolerance of ± 2 mm and the diagonal measurements must not differ by more than 3 mm.

The BS Code gives a standard range of tank diameters from 3 m to 114 m, with capacities against tank heights in one metre intervals up to 25 m in height. This is useful for purchasers to judge the size of a tank required for a certain capacity, but very often it is the plot of land that is available for the tank which decides the tank diameter, which can be any size and not necessarily in line with the diameters stated in the table.

Recommended standard shell plate lengths are also given and these are quoted as a function of π and when applied to the standard diameters, give an equal number of plates per shell course. These plate lengths have generally been adopted by tank constructors although slight "tweaking" is sometimes necessary for tanks having out of the ordinary diameters.

The standard BS code plate lengths are stated as follows:

Tank diameter (m)	Shell plate length (m)
3	1.5π
4 to 10	2.0π
12.5 to 30	2.5π
33 to 114	3.0π

There are no recommended standard widths for shell plates but the limiting factor is generally the width which is available from the mill. Common widths are 1.0 m, 1.5 m, 2.0 m, 2.5 m and 3.0 m.

The factors, which have to be borne in mind when selecting shell plate sizes, are:

- The weight of the plate for handling by crane; in the fabrication shop, on site and during transportation.
- The width capacity of the fabrication shop machinery.
- Limitations on maximum width or weight for transport purposes, especially by road or rail.
- Shell courses made in wide plates may require each ring of the erection staging on the tank to be raised from its initial position and re-attached higher up the course to enable completion of the vertical welds.

The API Code does not include guidance on the size of shell plates.

Cutting plates by shearing, which are to be eventually butt-welded is limited to a thickness of 10 mm by the BS and API Codes, except that by agreement with the purchaser, the API code extends this to 16 mm. The limitation is imposed in order to ensure a good clean joint surface for the subsequent butt-welding.

Plates may be also be trimmed to size using oxy-acetylene cutting equipment or by the use of a planning machine.

The weld edge preparation may also be completed using the above methods and there is also a machine available which has serrated clamping rollers allowing it to crawl along the edge of the plate while machining the weld bevels as it progresses along the plate. This machine has the advantage of being able to work on both flat or curved plates.

Rolling of the shell plates to the correct curvature is important in order to obtain a good cylindrically shaped tank. Arguably it is

better to have the plates slightly under-rolled ($>$ tank radius) than over-rolled ($<$ tank radius) because under-rolled plates will generally pull in to the correct diameter whilst over-rolled plates leave the completed course after erection taking a "gull wing" or scalloped appearance which is difficult to get rid of. Care must be exercised to ensure that the plates are entered square-on to the rolls, as any slight offset from square will result in a plate taking on a helical and not cylindrical form, which will make erection of the plate into the tank very difficult, if not impossible.

Plywood templates about 1 to $1\frac{1}{2}$ m long are used to check the radius of the shell plates as they are being rolled to shape. Machines having vertically-mounted rather than horizontally-mounted rolls tend to give a truer radius because the horizontally rolled plate naturally flattens itself due to its own weight and long plates have to have the ends supported by overhead cranes when checking the radius.

Because of the way that most plate rolling machines are built, the extreme ends of the shell plates do not get rolled and are left with "flats" on them. To overcome this, the ends are pressed to a pre-set radius prior to rolling.

The Codes do not insist on pre setting the ends of the shell plates but this is generally known to give a better final shape to the tank (see peaking and banding in Chapter 12).

The API Code does allow the thinner shell plates of the larger diameter tanks to be left flat and for them to be pulled into radius during erection. The allowable limits are shown in the table below (taken from API 650, clause 4.3.1.)

Nominal plate thickness (mm)	Nominal tank diameter (m)
≥ 16	All
From 13 to < 16	≤ 36
From 10 to < 13	≤ 18
From 5 to < 10	≤ 12

However, with the present day demands to produce good quality, good-looking tanks, without flats and wrinkles, most fabricators roll all their shell plates.

Several plate mills have provided themselves with fabrication facilities or they have teamed up with a local fabricator enabling them to offer edge prepared, rolled and surface finished plates ready for direct delivery to site.

Having rolled the shell plates, it is advisable to ensure that they do not loose their shape during storage or transportation and to stack them in purpose-made curved cradles, or if only one-off short journeys by lorry are involved, then they should be chocked with baulks of timber on the bed of the lorry. When transporting by sea, it is worth employing a stevedoring company which is experienced in handling the export of large bundles of steel plates, as the consequences of their unfamiliarity can be disastrous, as is witnessed by the photographs in Figures 11.2 and 11.3.

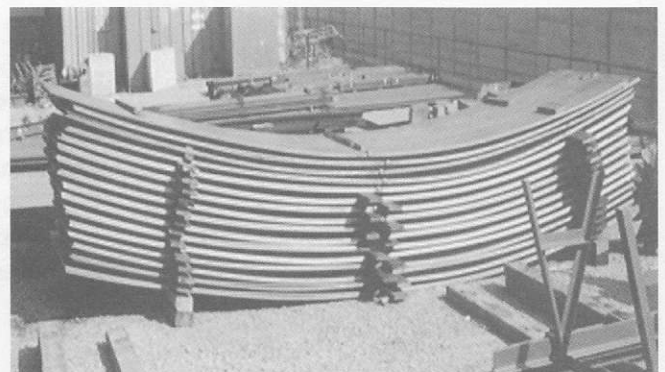


Figure 11.2 Shell plates stacked awaiting shot blasting and priming

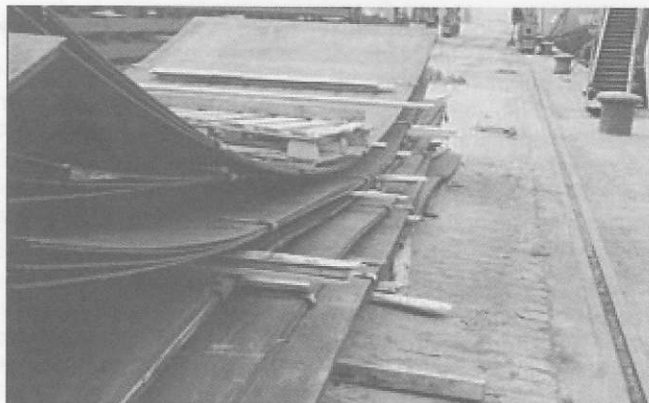


Figure 11.3 The same plates on the quay before loading on board prior to delivery to the docks

These plates had to be returned to the fabrication shop for re-rolling, an expensive and frustrating experience all due to a lack of understanding of materials handling by the shipper.

11.5 Roof structures

After the various structural components comprising the roof structure have been fabricated, the normal procedure is to erect one complete bay of the structure on the shop floor. This is in order to check the radius of the structure, the chord lengths of the purlins and the main shell attachment brackets. Any discrepancies found in the structure are far more easily rectified on the shop floor rather than at site where the structure may be being erected at, say a height of 20 m.

11.6 Tank appurtenances

Nozzles and manholes are normally pre-fabricated in the shop such that the flanges are welded to the barrels and the reinforcing plates rolled to suit the tank radius but supplied loose.

Staircases which have stringers rolled to a helical shape, usually have one section of staircase bolted up with the treads. This is temporarily erected in the fabrication yard, to allow the cylindrical radius and overall lift to be checked and also to ensure that the treads are truly horizontal.

Nozzles which require to be post weld heat-treated (PWHT) are shop-welded into the relevant shell plate (or part shell plate) and sent to the PWHT oven. It is advisable to fit temporary stiffeners to the shell plate so that it keeps its shape and doesn't warp whilst being heat-treated.

If there are a number of nozzles requiring heat treatment then it is advisable, if possible, to keep these together in one shell plate.

Clean-out doors are completely shop-fabricated and PWHT prior to being sent to site.

All fabrications should be dimensionally checked before and after post weld heat treatment.

11.7 Surface protection for plates and sections

It is common practice to protect the surfaces of carbon steel materials by shotblasting or pickling, to remove mill scale and then to prime with a suitable primer to prevent surface deterioration. Pickling is rarely performed nowadays due to Health & Safety requirements and the difficulty of disposing of exhausted pickling fluids. This makes the final painting easier on site as only sweep, or pencil blasting is required prior to applying the final paint system.

Care has to be taken to ensure that the shop-applied system is kept clear of those areas, which will be welded on site, and these must be masked during the priming operation. Alternatively, instead of masking the edges, a weldable primer can be used but this will depend upon whether this suits the final paint system.

11.8 Marking

To enable the various fabricated components to be assembled together correctly on site, each part has to be marked with a unique numbering system which relates to a marking plan made up in the drawing office or template loft. The marking plan shall also identify the position that the markings must occupy on the various components. Hard stamping may be used but the symbols should not be less than 13 mm high and low stress stamps with a minimum nose radius of 0.25 mm should be used. Plates less than 6 mm thick should not be hard-stamped. Where hard stamping is used, the position of the marks is usually ringed in paint to identify where these small markings are on the components.

Markings in paint or ink should be at least 50 mm high and care must be taken to ensure that the composition of the marking materials will be compatible with the materials being marked and the product, which will be eventually stored in the tank.

Shell plate markings should be on the inside surface of the plates.

12 Erection considerations for ambient temperature tanks

Tank constructors are fortunate beings within the construction industry, in that they are not usually responsible for the construction of the tank foundation and accordingly there is a clear demarcation of responsibility between the civil contractor and the tank contractor. Everything below the top finished surface of the foundation is the responsibility of the civil contractor and everything above the responsibility of the tank contractor.

This Chapter discusses the various elements involved in the construction of the tank after handover at the foundation.

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12.1 The foundation

The inspection of the foundation prior to its acceptance by the tank contractor is the first important decision to be made by him before commencement of erection and the following areas should be checked carefully:

- The diameter of the foundation is large enough for the tank. It has been known for a foundation to be constructed exactly as the tank diameter without allowance for the overlap of the floor beyond the shell.
- The civil contractor has clearly marked the cardinal compass points on the periphery and the centre point on the foundation.
- The slope (if any) of the surface of the foundation matches that of the tank floor design.
- The holes or cut-outs for the sump(s) are in the correct place.
- The surface of the foundation meets the allowable level tolerances given in the relevant Code.
- The positions, dimensions and condition of any anchor bolts, straps or pockets should be checked as acceptable.

12.1.1 Foundation tolerances

The part of the foundation which supports the shell receives most attention in the Codes. This is because differentials in level in this area can lead to the erection of a distorted shell.

The Code requirements vary slightly and a summary is given below. Tolerances at the periphery of the foundation under the shell plating are as follows:

12.1.1.1 BS 2654

The maximum differential in level between any two points 10 m apart measured along the periphery shall not be more than ± 6 mm with a maximum between any two points on the periphery of ± 12 mm.

12.1.1.2 API 650

For foundations having a concrete ring wall:

The maximum differential in level between any two points 9 m apart measured along the periphery shall not be more than ± 3 mm with a maximum between any two points on the periphery of ± 6 mm.

For foundations which do not have a concrete ring wall:

The maximum differential in level between any two points 3 m apart measured along the periphery shall not be more than ± 3 mm with a maximum between any two points on the periphery of ± 13 mm.

For foundations formed by a concrete slab:

The area of the foundation measured 300 mm radially inwards from the outside of the tank towards the centre (or the width of the annular ring of floor plates) shall comply with the requirements above for ringwalls. The remainder of the foundation shall be within ± 13 mm of the design shape.

12.1.1.3 The European Code prEN 14015 - 1

The difference in level between any two points 5 m apart around the periphery of the tank shall not be greater than 0.1% of their peripheral distance.

This is not as stringent as the BS and API Codes. Take for example a tank shell having a circumference of 80 m (25.5 m diameter). This gives 16 points around the periphery at 5 m apart. There could be a constant fall between each of eight points (from 0° to 180°) of 5 mm giving a total fall across the base of 40 mm. This presupposes that there will be a identical rise in level over the remaining section (180° to 360°). The maximum differ-

ential across the base of 40 mm is three times that allowed by the BS and API Codes, when considering foundations without concrete ringwalls. Admittedly an extreme case has been sited here but extreme cases do sometimes occur.

The European Code does however go on to say that "The tolerance the erector accepts on the inclination or slope of the foundation shall be such as to enable the final vertical tolerances of the tank to be achieved". If this loose approach to allowable tolerances is not tightened up in the Code, then it will surely lead to heated arguments between the civil and tank contractors on the hand-over of the foundation, as to what is acceptable.

The surface of the foundation, other than the area under the shell plating shall be to the following tolerances:

The sag in the as built surface measured with a 3 m long straight edge shall not exceed 10 mm.

The difference between the design level and the as-built level shall not exceed the following values :

Diameter of tank 'D' (m)	Difference in 'design' to 'as-built' levels
$D \leq 10$	10
$D > 10$ to ≤ 50	$D / 1000$
$D > 50$	50

12.2 Building a tank

As with most construction tasks there is always more than one way of carrying out the various stages of the work to effect a successful completion.

Also method statements, risk assessments, safety procedures and numerous other forms of documentation have to be produced prior to opening up the site but these aspects will not be dealt with here, otherwise *Storage Tanks & Equipment* will consume another tree !

The following sequence for the construction of storage tanks has been used for many years and is offered here to give the reader a reasonable understanding of how a tank is built.

12.2.1 Laying the floor

Taking the case for a standard lap-welded floor, with or without annular plates, the process is as follows:

Using the foundation centre point, the outer radius of the tank floor is scribed onto the surface of the foundation and the floor start mark given on the drawings is orientated from the cardinal points given by the civil contractor.

If the underside of the plates is to be painted (usually with a bitumen solution) this should be applied as they are laid.

Annular plates must have the correct weld gaps and after laying and tack welding in position, each one must be checked to ensure that the outer edges are the correct distance from the centre of the foundation. They should be welded as soon as possible after laying. The annular butt joints should be pre-set by lifting and chocking them about 150 mm above the foundation, this will minimise distortion during welding. They can be left in this position until the completion of the required radiographic inspection.

The centre stake of the rectangular plates is laid, commencing with the centre plate being placed on the line of the floor setting out line. The remaining plates in this stake are then laid from the centre out to the periphery. The stakes either side are laid in a similar way and finally the outer sketch plates are put in place. During the whole of this process, care has to be taken to ensure that the minimum laps are maintained between the plates which is normally $\geq 5 \times$ plate thickness.

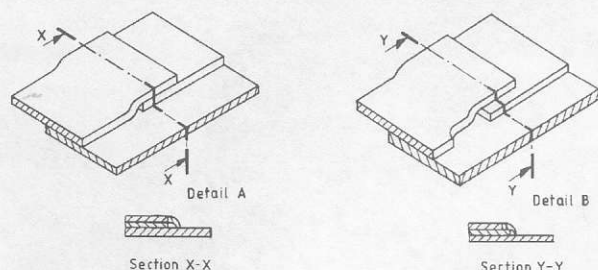


Figure 12.1 Laps in floor plates where three thicknesses occur

To avoid plate distortion, it is important to weld the plates in the following sequence:

First weld the annular plate butt joints.

Then, starting at the centre of the floor, weld the short transverse lap joints working outwards each side of the centre to the periphery of the floor.

Repeat this sequence for the strakes of plate each side of the centre stake.

Similarly, repeat again on the strakes adjacent to those last welded until all the transverse welds are completed.

The longitudinal joints are now welded, starting at the centre of the floor and working outwards to the periphery from each side of the floor centre line which is transverse to the setting out line.

Where three thicknesses occur in the floor lap joints the upper plate is joggled, or cut and joggled as shown in Figure 12.1.

The outer edge of floors which do not have annular plates, are joggled and welded (as illustrated diagrammatically in Figures 12.2 to 12.4) according to the following procedure:

- 1) Tack weld the plates in position and weld a light pass 230 mm long, welding towards the tank centre (Figure 12.2).
- 2) Put a 200 mm wide joggle plate under the joint and hammer the joint to joggle the lower plate (heating the plate will assist the process), (Figure 12.2).
- 3) Complete the welding in the area of the joggle, by welding towards the centre of the tank, (see Figure 12.3).

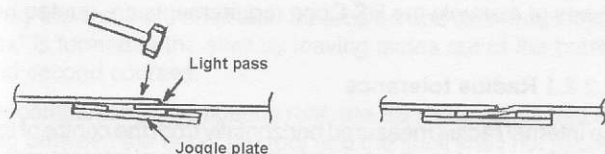


Figure 12.2 Joggling and welding of outer floor edges

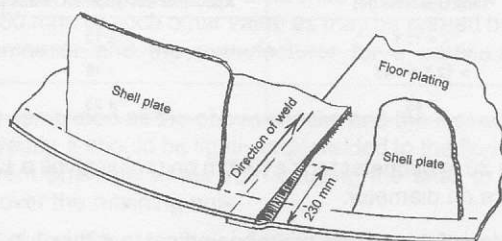


Figure 12.3 Welding in area of joggle

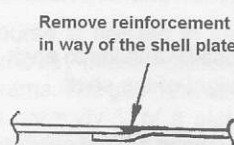


Figure 12.4 Flushing off joint

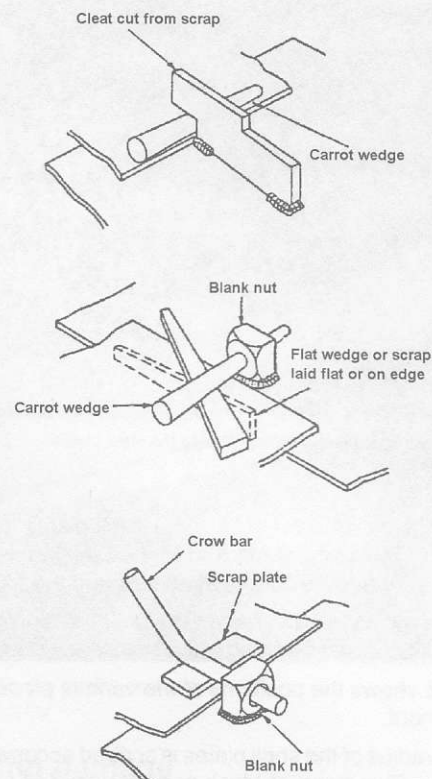


Figure 12.5 Different types of erection equipment

- 4) Flush off the joint with weld metal and grind flush where the shell passes over the joint, (see Figure 12.4).

Care has to be taken when laying rectangular plates on conical shaped foundations because the plate laps will "scissor" giving varying overlaps between adjacent plates and these laps have to be checked to ensure that the minimum lap dimension is complied with.

The plates forming the lap joints have to be kept in close contact while being welded and one way is to use concrete-filled oil drums which can be rolled along the joints while being tack-welded. Other methods using different types of erection equipment are shown in Figure 12.5.

On completion of the welding of the floor, the required number of annular butt welds must be inspected by radiography and all the weld seams vacuum box-tested for leaks by the method given in Section 12.10.2. The erection of the shell plating can now commence.

12.2.2 Erecting the shell by the traditional method

Stacks of shell plates are laid just outside the foundation area. Each stack consists of one plate from each shell course with the inside surface uppermost and the bottom edge of the plates nearest to the foundation. The bottom course plate is on the top of the stack, the second course next and so on, with the top course plate being at the bottom of the stack. Timber chocks are put under each end of the stack to preserve the plate curvature.

Blank erection nuts are accurately positioned and welded to the inside of the plates as they lie on the stacks. Each plate usually has six nuts along each horizontal edge and two on each vertical edge. The nuts are welded on three sides only, but the nuts that are used as lifting points are welded all round. These nuts are used to attach the plates of each course together and to connect each course to the one above using key-plates and carrot wedges.



Figure 12.6 Welding of blank erection nuts to the shell plates
Courtesy of McTay

Welding of blank erection nuts to the shell plates is shown in Figure 12.6. (Plates were stacked in the tank in this case because of a shortage of storage space around the foundation.)

Clips which will be used to mount the tank erection staging on are also positioned and welded to the inside face of the plates.

Figure 12.7 shows the positions of the various pieces of erection equipment.

The inside radius of the shell plates is scribed accurately on the floor plating. Two rings of blank nuts are welded to the floor plates at 600 to 900 mm pitch along the line of the scribed radius, the inside nuts set about 20 mm from the line to allow for wedging and the outside nuts the thickness of the bottom course away from the scribed line. These nuts are welded along one long and one short side only.

The shell start mark, for the bottom course vertical joints, given on the tank drawings, is accurately marked on the floor and the first course of shell is lifted plate by plate into position. Care must be taken to keep each plate of this first course vertical using angled stays welded to the plates and floor.

Each plate is keyed to the adjoining plate using key-plates and carrot wedges as shown in Figure 12.7. The required weld gap between plates, which is usually 3 or 4 mm, is maintained by the

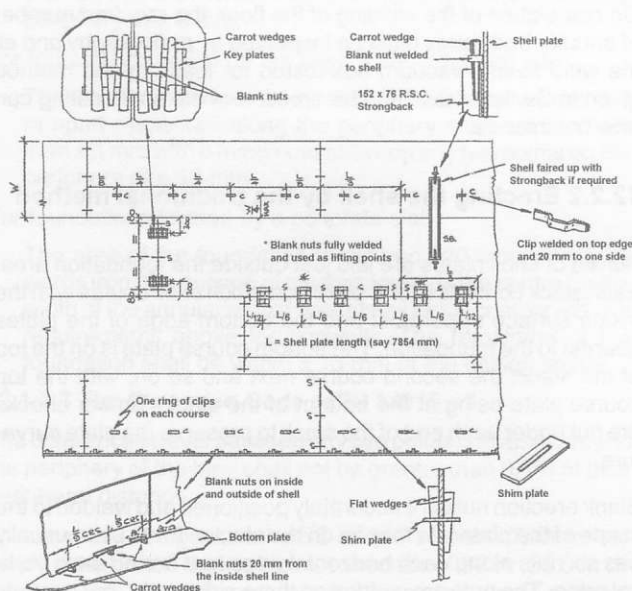


Figure 12.7 Positions of various pieces of erection equipment

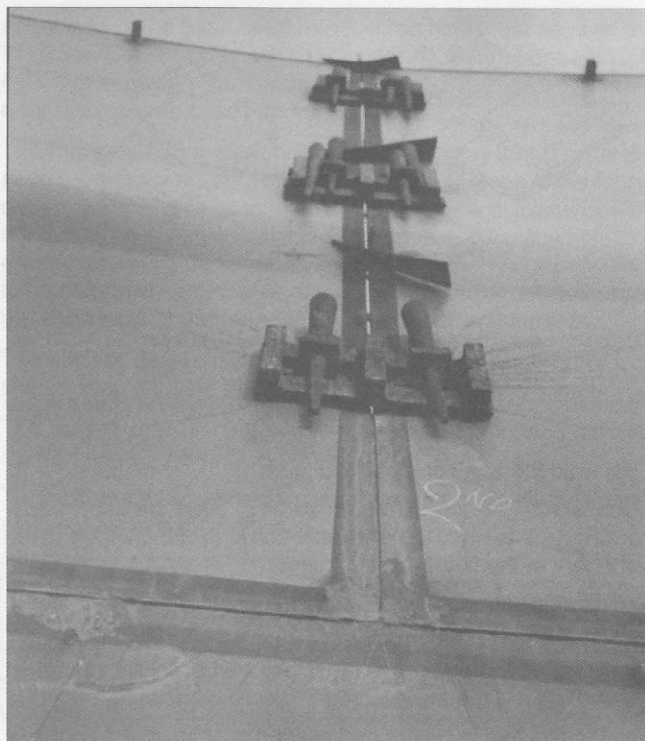


Figure 12.8 Key-plates and shims on a vertical joint
Courtesy of McTay

use of shim plates of that thickness and flat wedges. Key-plates and shims on a vertical joint is shown in Figure 12.8.

12.2.2 Tolerances

After completing the erection of the first course it is checked for compliance with the allowable Code tolerances.

There are slight differences between the Codes regarding the magnitude of allowable erection tolerances and the erection contractor must familiarise himself with those of the Code to which the tank is being built. In particular, the European Code is very detailed in this respect.

By way of example the BS Code requirements are quoted below.

12.2.2.1 Radius tolerance

The internal radius measured horizontally from the centre of the tank at floor level shall not vary from the nominal internal radius by more than:

Tank diameter (m)	Allowable deviation on radius (mm)
≤ 12.5	± 13
$> 12.5 \leq 45$	± 19
> 45	± 25

So for a 30 m diameter tank ± 19 mm on radius gives a ± 38 mm tolerance on diameter.

The plates of the course must be vertical to within 1 in 200.

For, say a 2 m wide course this would allow out of verticality of ± 10 mm.

This standard of verticality applies to each course erected and also to the overall height of the shell.

12.2.2.2 Peaking and banding

There must be no significant change in the shape of the tank at the joints between adjacent shell plates.

For vertical joints any deviation is termed "peaking" and this is measured using a 1 m long horizontal sweep board made to the correct radius of the tank.

For horizontal joints, the deviation is called "banding" and is measured with a 1 m long vertical straight edge sweep board.

The maximum allowable deviation to the BS Code for horizontal and vertical joints is:

Plates ≤ 12.5 mm thick : 10 mm

Plates > 12.5 mm ≤ 25 mm thick : 8 mm

Plates > 25 mm thick : 6 mm

12.2.2.3 Plate misalignment

Plates which are joined by butt welding shall not be misaligned by more than the following:

For completed vertical joints:

Plates ≤ 19 mm thick, 10% of the plate thickness, or 1.5 mm whichever is the larger.

Plates > 19 mm thick, 10% of the plate thickness, or 3 mm whichever is the larger.

For completed horizontal joints:

Plates ≤ 8 mm thick, 20% of the upper plate thickness, or 1.5 mm whichever is the smaller.

Plates > 8 mm thick, 20% of the upper plate thickness, or 3 mm whichever is the smaller.

The above misalignment tolerances assume that the centre lines of all course thicknesses are coincident with each other. That is to say, the step in thickness between courses of different thickness is the same on the inside of the tank as that on the outside. However, for large diameter floating roof tanks it is often a requirement to have the inside face of all courses flush with each other in order to give a smooth surface for the roof seal to act against. In these cases the step due to the difference in thickness is all on the outside of the shell.

12.3 Floating roofs

For ease of construction access, it is common practice for tank erectors to build the floating roof on the floor of the tank after one, or maybe two shell courses have been erected. Alternatively the complete shell may be erected and an access "letter box" is formed in the shell by leaving plates out of the bottom and second courses.

On completion of the floating roof, the BS Code states that the gap between the rim of the roof and the shell shall not exceed ± 13 mm from the nominal gap.

The Code goes on to say that at any other elevation other than that which it was erected, the difference in gap should not exceed ± 50 mm, or such other value as may be agreed between the purchaser and the manufacturer for a particular seal design.

Having completed all the above checks and the first course is set correctly, it should be lightly tack-welded to the floor plates to prevent any high winds from causing the shell to lift and spring over the retaining nuts.

The positions of the manholes in the first course should be orientated on the shell and the openings cut to facilitate the movement of men and materials into and out of the tank.

Each successive course is erected in turn on the preceding course, using the same key-plate and shim method for the vertical and horizontal seams. The gap between the vertical joints in adjacent courses is normally $\frac{1}{3}$ of a plate length, staggered clockwise or anti-clockwise but the minimum gap should not be less than 300 mm. The shell is completed by fitting the curb angle or compression plate to the top course.

Access staging for the erection personnel is erected on the inside of the shell. The staging brackets are attached to the shell plates using clips which must be securely welded to the shell by welding along the top edge and 20 mm down one side, this, to prevent the clips from being levered off the shell when moving the staging brackets.

Normally a three plank width of staging with handrails, stanchions and toe boards is erected and this staging is moved up the tank as each course is erected. Typical access staging is shown in Figure 12.9.

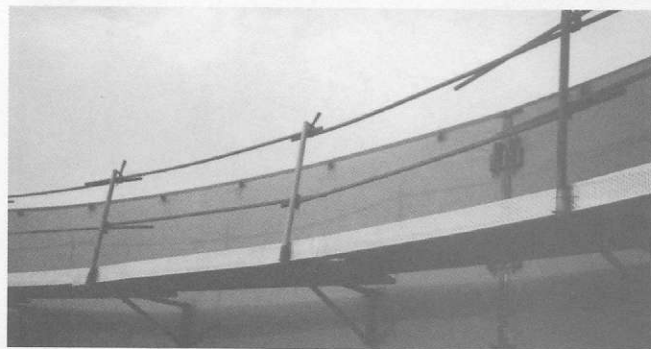


Figure 12.9 Access staging on the tank shell
Courtesy of McTay

12.4 Wind damage

The one thing a tank contractor fears most is high winds, because an uncompleted or partially erected and welded tank is very vulnerable to severe damage from high winds as the sequence of photographs in Figure 12.10 demonstrates.

The tank in question was 22.5 m diameter x 16 m high.

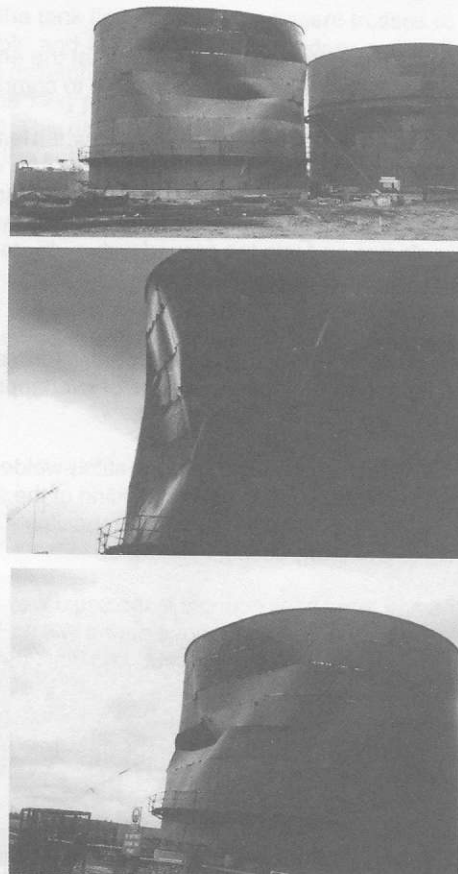


Figure 12.10 Example of severe wind damage to a tank



Figure 12.10 Example of severe wind damage to a tank (continued)

12.4.1 Safety measures against wind damage

- Never leave a uncompleted shell course at the end of the working day even if it means working late to complete it.
- Guy-off the tank during windy weather and when leaving the tank overnight, as illustrated in Figure 12.11. An effective method of guying a tank is by using 'Tirfor' wire tensioners on guy wires which are connected to the shell by welded cleats or clamps and into the ground with multi staked anchor bars, or alternatively large concrete blocks may be used as anchor points.
- The tank erection staging can be adapted to form a temporary wind girder by clamping the ends of overlapping staging boards as shown in Figure 12.12.
- Temporary steel angle wind girders stitch-welded to the shell will greatly assist in resisting buckling of the shell due to high winds. These girders can be repositioned on the shell as erection progresses.
- Erect the first three shell courses in the usual way and take the safety precautions given above during this erection period. At this juncture, cease erection and weld the vertical joints in the first two courses but only 75% of the third course, leaving the upper 25% free for fairing up to the fourth course when it is erected. The first two horizontal joints are then welded. This method makes the shell much stiffer and more able to withstand high winds.

On completion of this partial welding, the shell erection recommences and the procedure is repeated until the whole shell is erected.

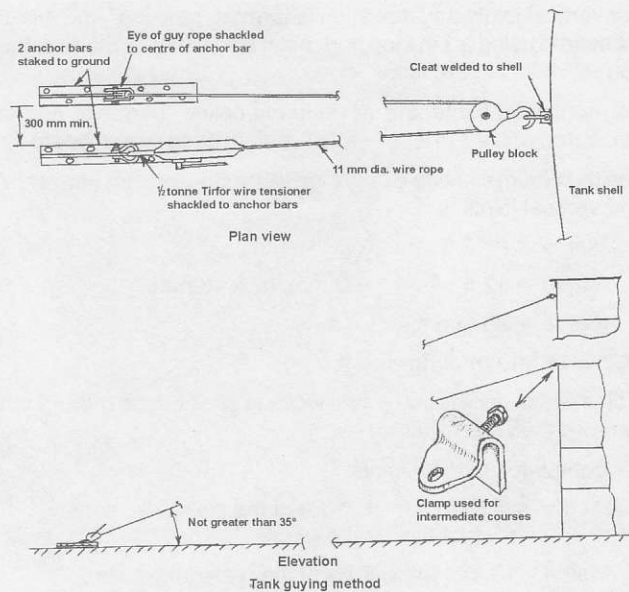


Figure 12.11 An effective method of guying a tank

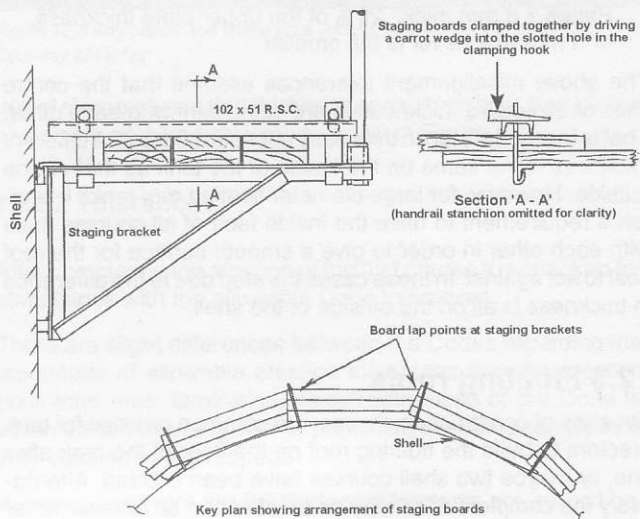


Figure 12.12 Clamping the ends of overlapping staging boards

12.5 Shell welding sequence

The following sequence is based on manual welding although the principles are just the same when using automatic welding machines, except that when welding with the latter, the weld seam is completely welded in one pass.

To ensure the minimum amount of distortion in the welded shell, there is a very simple rule which should be followed and this is:

- Fair up, tack, removing the shims and key plates as this work proceeds and then fully weld the vertical seams on two adjacent courses before fairing, tacking, removing the erection gear and welding the horizontal seam between them.

If this procedure is followed, and assuming the correct welding procedure, electrodes and heat input is adhered to, then a good-shaped shell will be the result.

This sequence can be adhered to when following the "three course" erection procedure described in the preceding paragraph and also when erecting by the "jacking method" described later.

However, where a shell has been completely erected using the conventional erection aids, then for expediency, a variation of the above ideal sequence is often used as follows:

- On completion of the erection of the whole shell, the shell erectors will leave a complete ring of access staging on the top course on the inside of the shell.
- The welders commence welding the shell from the outside using access staging which they erect as they proceed up the tank. The sequence of welding is as described above, i.e. weld two courses of vertical seams and then the horizontal seam between them. However these welds are not welded on the inside at this stage.
- The welders arrive at the top of the shell having completed all the external welding.
- These welds have now to be back-gouged from the inside by pneumatic chipping, grinding or air arcing the root of the welds to sound metal. This commences at the top of the tank, using the access staging already left in place by the erectors. The welds are then cleaned down from the top to the bottom of the tank.
- The welders then complete the welds working from the bottom to the top of the tank.

Using this sequence means that at the completion of the shell welding, there are two rings of access staging at the top of the tank, one on the inside and one on the outside. These may now be used by the erectors whilst erecting the roof structure and plating.

For manual metal arc welding, the British and American Codes require that hydrogen-controlled electrodes be used for courses constructed in the range of higher tensile steels and the Code specific requirements should be referred to especially for courses over 12.5 mm thick.

The specific requirements regarding welding are extensively covered in the Codes with regard to: weather conditions, pre-heating, storage of electrodes, cleaning of welds, allowable undercut, back gouging, weld repairs etc., and the reader is advised to refer to the relevant sections of the Code for these details.

12.6 Joints in wind girders

The butt-welded joints between the sections of wind girder should not run into the surface of the shell plating as this can cause undesirable defects in the surface of the shell. To prevent this, "mouseholes" are cut at the joints as shown in Figure 12.13.

12.7 The roof structure

Having completed the erection of the shell the roof structure is now installed. Assume that the structure in this case is a trussed type as described earlier in Chapter 5.

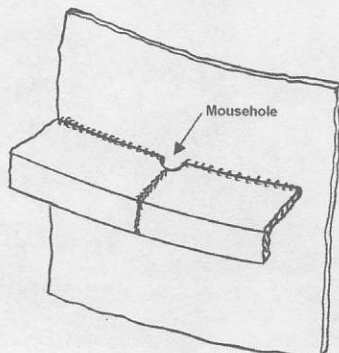


Figure 12.13 Mousehole at joint between wind girders

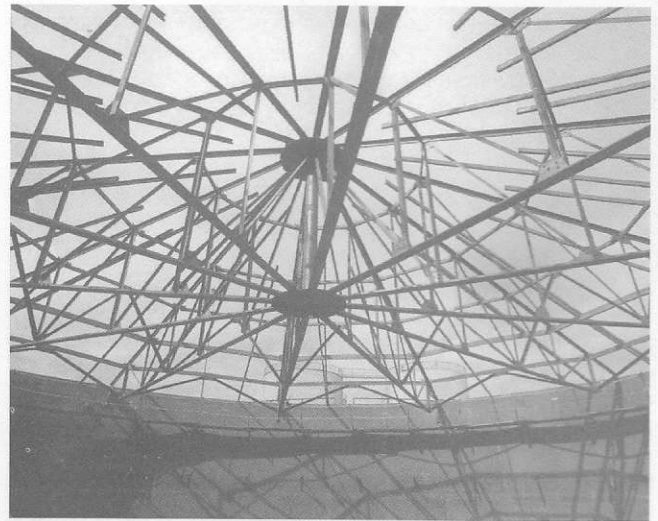


Figure 12.14 Completed structure with king post removed
Courtesy of McTay

A temporary king post is erected on a load spreading grillage at the centre of the tank floor and guyed-off to the periphery of the floor using wires and 'Tirfor' tensioners. Vertical adjustment is provided by two hydraulic jacks placed either side of the post on the grillage which act against lugs welded to the post. A scaffold tower is constructed around the king post to give personnel access to the top of the post.

The centre bobbin of the structure is secured to the top of the post and the roof trusses are lifted and bolted into position, the shell brackets being landed on previously marked positions on the inside of the shell and toggled in place with erection equipment prior to finally welding the brackets to the shell. The completed structure, with the king post removed is shown in Figure 12.14.

Variations of this procedure are as follows:

- On the tank floor, erect two adjacent trusses to the centre bobbin and fit the purlins, secondary and tertiary rafters. This assembly is lifted using a mobile crane and placed on to the king post and the shell brackets connected to the shell. This gives a fairly rigid framework to work off when fitting the subsequent individual trusses etc.
- Dispense with the king post and erect the complete structure on the floor of the tank leaving the shell brackets loose. Using two or more mobile cranes, the complete structure is lifted to the correct level and secured to the top of the shell. This is shown in Figure 12.15. The lift has to be carefully monitored to ensure that all cranes take the same load and that the structure is lifted evenly. The erection supervisor



Figure 12.15 Four cranes lifting a 33 m diameter roof structure
Courtesy of McTay

has to be in radio contact with all the crane drivers in order to pass instructions to them as they cannot see how the lift is progressing from their position outside the tank. Also problems may be encountered in ensuring the lift for this method of erection.

12.7.1 Roof plating

The centre crown plate is laid first followed by the centre strake across the tank diameter. This strake is laid from each side the crown to the curb, all laps being a minimum of 5 x the plate thickness and towards the centre of the tank (opposite to the way tiles lie on the roof of a building). All plates are tack-welded together, but not attached to the roof structure. The two strakes adjacent to the centre strake are then laid in the same sequence and these strakes are also lapped towards the centre of the tank and tack-welded in position. This sequence is repeated until the whole roof is sheeted. The outer roof sketch plates are flame cut to suit the curvature of the curb angle.

Some of these sketch plates may be temporarily removed to allow light into the tank while other operations are being performed inside the tank.

12.7.2 Welding sequence

The short transverse laps of the centre strake are welded first, starting at the crown and working out towards the curb except that the lap to the sketch plates is not welded yet.

This sequence is repeated on the two adjacent strakes to the centre strake and so on until all the short transverse laps are welded, with the exception of the outer sketch plates.

The longitudinal laps between the centre strake and the two adjacent strakes are then welded, starting at the crown and working towards the curb. The welding stops short of the outer sketch plates. This sequence is continued until all longitudinal welds are complete except for the sketch plates and the weld between the roof plating and the curb angle.

The laps of the sketch plates are welded next, starting with those furthest away from the centre strake, and working around clockwise and anti-clockwise to the outer ends of the centre strake. Finally the periphery of the roof plating is welded to the curb angle.

The positions for the roof nozzles and fittings can now be marked off and the roof sheeting flame-cut to allow them to be welded into position. Two tanks nearing completion are shown in Figure 12.16.

12.8 Erecting the shell by the jacking method

This method is gaining in popularity because it keeps the construction activities at a lower elevation and is therefore safer for the construction personnel.

The foundation checks and the erection and welding of the floor is as previously described but the shell is erected in a completely different way. Depending upon the overall height of the jacks being used, the top two, or maybe three shell courses are erected and welded in the conventional way and the roof structure, sheeting and nozzles are completed.

The tank designer will have calculated the number of jacks that are required giving due regard to the overall weight of the tank shell (excluding the bottom course) the roof structure, sheeting and fittings and also taking consideration of the effect of high wind loads on the tank.

The jacks consist of a vertical post which has a specially designed hydraulic jack which climbs up the post carrying the



Figure 12.16 Two 25 m tanks nearing completion
Courtesy of McTay



Figure 12.17 Arrangement of hydraulic climbing jacks
Courtesy of McTay

weight of the tank with it. The tank is lifted in stages until it is high enough for another course of shell to be erected beneath the previous one, this can be between 1.5 to 2.5 metres.

The jacking posts are fixed to the tank floor on a load spreading pad and secured in position by two raking struts set at 45° each side of the post, these also being fixed to the floor plating, as shown in Figure 12.17.

As each course is erected, the vertical joints are welded followed by the horizontal joint between the adjacent courses. It can be seen from Figures 12.18 and 12.19, that all the work is

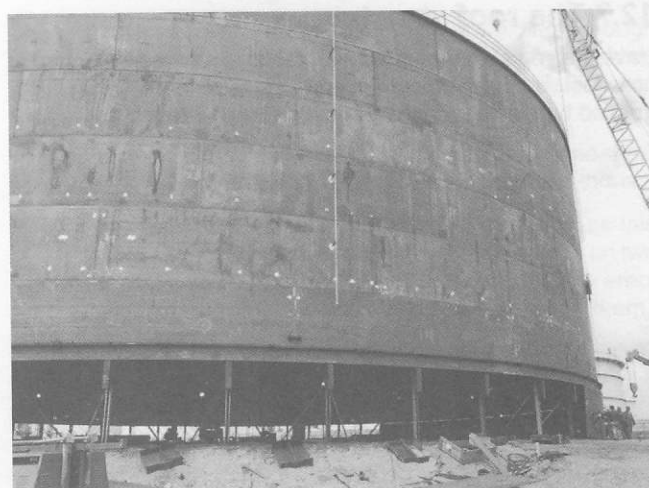


Figure 12.18 Tank being erected by the jacking method
Courtesy of Whessoe



Figure 12.19 Tank being erected by the jacking method
Courtesy of Whessoe

carried out virtually at ground level and is therefore much safer for the construction personnel.

12.9 Other forms of construction

12.9.1 Column-supported roofs

Column-supported roofs have to have the columns guyed-off correctly during erection as the partially erected roof is vulnerable to a spiral type of collapse. Figure 12.20 shows a partially erected column-supported roof.

12.9.2 Pre-fabricated roof section

On smaller tanks it is possible to completely erect the roof on to the top course of the shell and then to lift this section on to the remaining shell. The vertical shell butts in the adjacent courses are only welded for 75% of their length to allow for fairing up when the two sections are joined. An example of a pre-fabricated top section being lifted into place is shown in Figure 12.21.

12.9.3 Air lifting a roof into position

This method is used for large diameter dome roof tanks.

The roof-to-shell compression area has to be of the type which has a conical roof section as shown in Chapter 5, Section 5.4.1.

The roof structure and sheeting is completely constructed on the floor of the tank and a temporary air tight seal is fixed to the



Figure 12.20 A partially erected column-supported roof



Figure 12.21 A pre-fabricated top section being lifted into place
Courtesy of McTay

periphery of the roof, to seal the small gap between the roof and shell. This seal is formed by a thin flexible membrane material.

A number of steel guide cables are fixed to the centre of the floor, led vertically through sealed apertures in the crown of the roof and across the external surface of the roof plating to the periphery of the roof where they are led through pulleys and vertically to anchor frames above the top of the shell.

High efficiency electric fans are connected to the shell man-holes and these pressurise the area under the roof and cause it to lift within the shell. Only about 6 to 10 mbar air pressure is required to move the roof, and as it rises, the friction between the guide cables and the roof plating stabilise the roof and keep it level during the lift. At the top of the tank, the roof comes up against the underside of the compression area and is temporarily toggled into position ready for the final welding of this lap joint.

Figures 12.22 to 12.25 show the sequence of events.

12.9.4 Floating roofs

The floating roof is built at some level above the tank floor and access to build it is gained either over the shell, by restricting the erection of the shell to the bottom and maybe the second



Figure 12.22 31 m diameter dome roof under construction

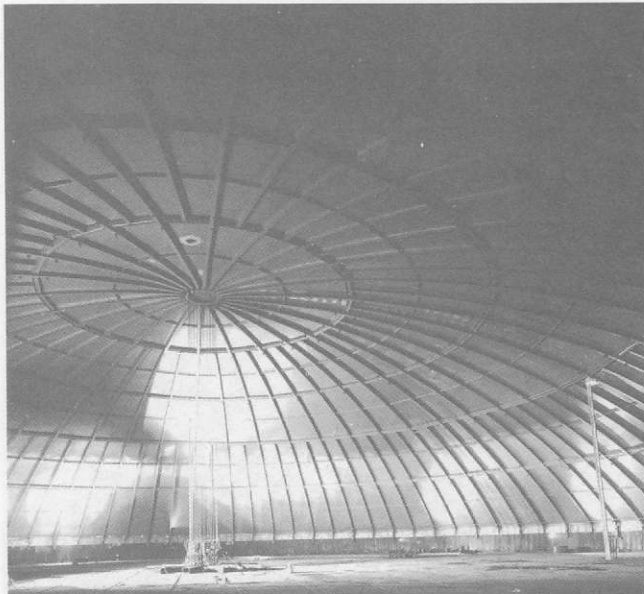


Figure 12.23 31 m diameter dome roof ready for the air lift (guide cables can be seen at the roof centre)



Figure 12.24 31 m diameter dome roof being air-lifted into place



Figure 12.25 The dome roof being secured prior to final welding

course, or by leaving plates out of the bottom two courses of the completed shell thus forming an access "letter box". The former method is to be preferred as this affords easier crane operation and direction by the banksman.

Two erection methods are outlined as follows:

- 1) A set of vertical square or round support pins are welded to the tank floor in a grid formation on which the roof plates are placed. The height of each pin is calculated to allow for any floor slope and the contour of the roof, the minimum height being that amount by which the support leg housings protrude below the underside of the roof plating.

The roof is built on this matrix of pins and when complete, water is pumped into the tank and the roof is floated up to a level whereby the support legs can be dropped into place and pinned (usually in the high, maintenance position).

The water is then drained out and the support pins removed and any drain lines, heating coils etc. can be fitted to the floor area.

A series of illustrations showing parts of the erection sequence are shown in Figures 12.26 to 12.32.

Construction note: There is a variance between the Codes in the requirements for the single side fillet welding of the bulkheads between pontoons to the inner and outer rim plates and to the top and bottom pontoon plating, (see Figure 12.30).

The BS 2654 Code requires single side fillet welding to the inner and outer rim plates and to the bottom pontoon plate but allows the joint between the bulkhead and the top plate to be left unwelded.



Figure 12.26 Laying the bottom deck of a 36 m diameter double deck floating roof
Courtesy of McTay

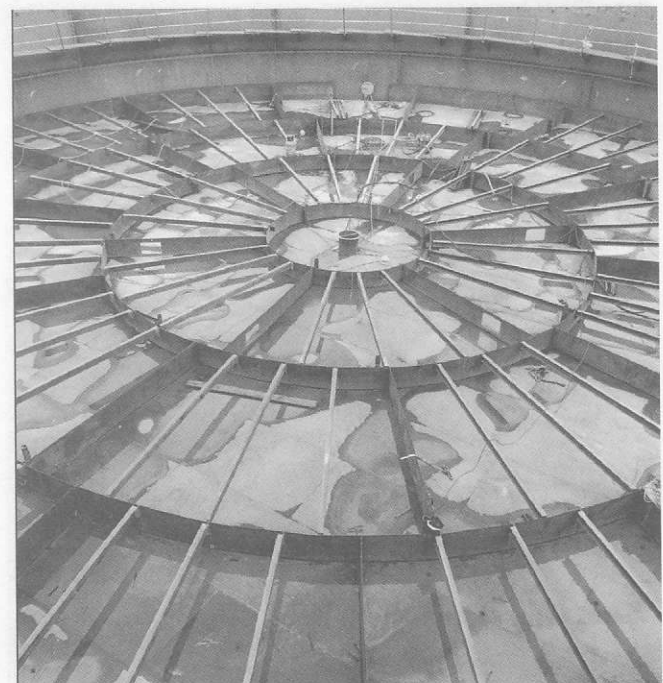


Figure 12.27 Bulkheads and top deck stiffeners of a 36 m double deck floating roof
Courtesy of McTay

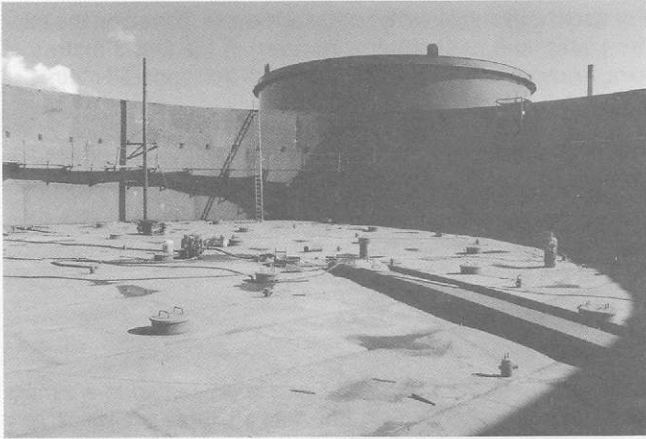


Figure 12.28 Top deck of 36 m diameter double deck floating roof being fitted out
Courtesy of McTay



Figure 12.29 20 m diameter single deck roof pontoons being erected on "pins"
Courtesy of McTay

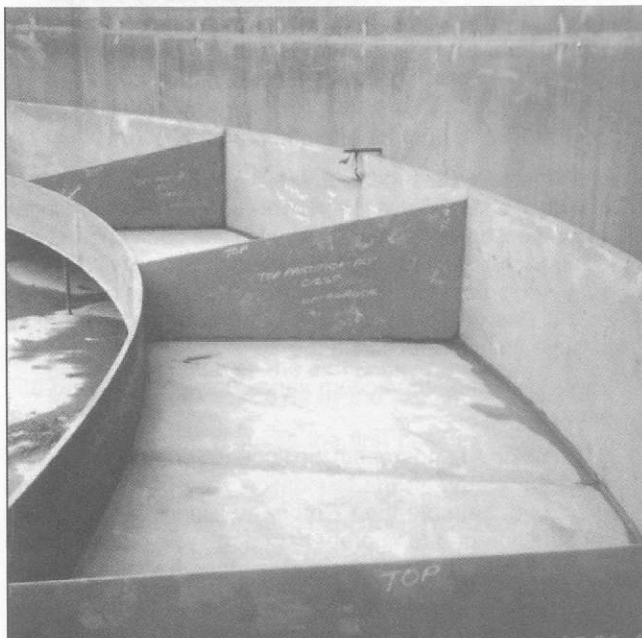


Figure 12.30 20 m diameter single deck roof pontoons being erected on "pins"
Courtesy of McTay

The European prEN Code in addition to the BS requirements requires this top joint to be welded only on alternate bulkheads.

The API 650 Code requires all four edges to be single side fillet-welded.

- 2) A grid formation of vertically adjustable scaffold supports (Acrows) are set to suit the final level of the underside of the roof pontoons and deck. These supports are held se-

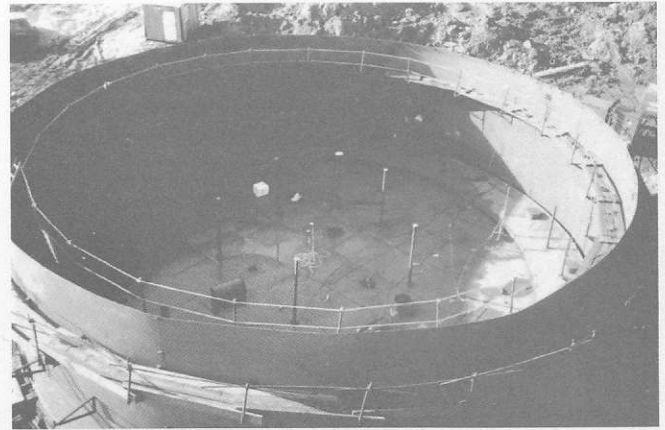


Figure 12.31 20 m diameter single deck floating roof, completed, with support legs in position, ready to be floated up to its correct elevation
Courtesy of McTay

curely in place with scaffold poles and clips. The roof is completely erected and welded on these supports and all the roof support legs, nozzles, manholes etc., are fitted to the roof. Once the legs are in place and pinned in position the supports and scaffolding is removed from the tank through the shell manholes, (see Figures 12.33 and 12.34).

When a single deck roof is constructed using this method, the outer rim of the pontoons is usually supported off temporary brackets welded to the shell.



Figure 12.32 20 m diameter single deck floating roof, at its correct elevation (the original support pins can now be removed)
Courtesy of McTay



Figure 12.33 A 45 m diameter single deck roof supported off scaffolding
Courtesy of McTay



Figure 12.34 Scaffold support system for a single deck type roof of a 45 m diameter tank
Courtesy of McTay

12.10 Inspection and testing the tank

12.10.1 Radiographic inspection

In the interest of brevity and the prevention of boredom, the exact requirements of each of the BS, API and European Codes are not reproduced here. The reader is advised to consult the relevant Code for the complete information as required.

Of the three Codes, the BS Code has the simplest approach and a less demanding quantity of radiography than the other Codes.

12.10.1.1 BS 2654

Shell joints

The requirements are set out as a percentage of the overall length of vertical and horizontal shell joints in three thickness bands.

Annular floor plate joints

The requirement for the annular floor plate butt joints is based on three thickness bands.

For the thickest plates, all the joints require to be radiographed.

For the mid range, half the number of joints require to be radiographed.

For the thinner plates, a quarter of the number of joints require to be radiographed with a minimum of four being required.

12.10.1.2 API 650

The API Code has a different approach but the quantity of radiography is generally more than that required by the BS Code.

Shell joints

The vertical joints are divided into three thickness bands.

For the thickest band, the joints have to be 100% radiographed, plus all 'T' junctions have to be radiographed.

For the mid thickness band, one radiograph is required in the first 3 metres of joint, followed by one radiograph in each additional 30 metres, plus all 'T' joints have to be radiographed.

Also for the bottom course only in this band, two additional radiographs are required, one of them being as close to the bottom as possible.

For the thinnest band, one radiograph is required in the first 3 metres of joint, followed by one radiograph in each additional 30 metres.

For each horizontal joint type and thickness (based on the thinner plate), one radiograph is required in the first 3 metres of joint, followed by one radiograph in each additional 60 metres.

Annular floor plate joints

For joints which have been welded from both sides, one radiograph is required on 10% of the total number of radial joints.

For single-sided butt joints made using a permanent backing bar (the more usual method) then one radiograph is required on 50% of the total number of radial joints.

12.10.1.3 prEN 14015 – 1

Shell joints

Radiography to the European Code is presented in a similar way to that of the BS Code in that there are three shell thickness bands, but the amount of radiography is generally greater than the BS Code within each band.

This Code also differentiates between steel yield strengths. Steels having yield strengths equal to or more than 355 N/mm², require more radiography than those below this value. Also ultrasonic examination of certain welds is called for in this Code.

The Code also gives radiographic and dye penetrant examination requirements for stainless steel shell plates. These are generally not as extensive as for carbon and carbon manganese steels.

Annular floor plate joints

The Code gives an option to radiograph or ultrasonically examine the joints to the following extent:

One full length radiograph (400 mm) from the outer edge of the plate or US examination over the full length of the joint. This shall apply to one joint in four.

However, for annular plates in steels having a yield stress = > 355 N/mm² and > 10 mm, the requirements are as above but shall apply to one joint in two.

12.10.2 Floor plate joint testing

On completion of the tank, the floor joints can be tested for soundness by one or more of a number of methods :

- By the vacuum box method, see Figure 12.35, whereby an open-bottomed box with a seal around the edge is placed over a section of the floor joint which has been painted with a soap solution. A vacuum is drawn in the box which has a toughened glass top and any leak paths in the joint will show as bubbles due to air being sucked from under the floor through the imperfection in the weld.
The recommended vacuum varies between 210 and 350 mbar.
- By pumping air underneath the floor at a pressure sufficient to lift the plates off the foundation. The pressure, which should not be more than 7 mbar maximum is held by the construction of a temporary dam of clay or other suitable material around the periphery of the floor. A soap solution is then applied to the internal floor joints for the detection of leaks.
- By the use of a tracer gas and a suitable compatible detector. The gas is pumped and trapped under the floor in a similar way to the previous method and the detector is passed over the joints and senses the escape of gas through any leaks.
- By the use of dye penetrant or magnetic particle examination methods.

The most common method favoured by most tank contractors is the vacuum box method although this is often supplemented with a dye penetrant or magnetic particle examination.

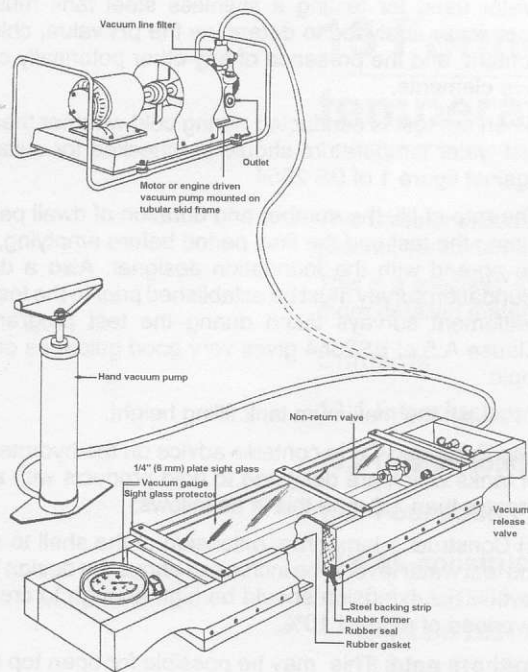


Figure 12.35 Vacuum box and pump

12.10.3 Shell-to-bottom joint testing

This applies to joints formed with a fillet weld both sides of the shell plating and they may be checked by one of the following methods:

- The BS Code is not specific in this area but internal weld is normally tested for leaks using a vacuum box in a similar way to that described above for the floor plating. The box in this case has one side, as well as the bottom missing and it is forced into the corner formed by the floor and shell and seals around the open edges of the box give a air tight seal to the tank. Soapy water applied to the corner weld prior to placing the box shows if there are any leaks in the weld.

The problem with this method is that the contractor has to stock a number of vacuum boxes to cover the range of tank shell diameters.

- By the use of dye penetrant or magnetic particle examination methods.

Contractors usually perform a dye penetrant or magnetic particle examination the first pass of the internal weld followed by an examination by the vacuum box method.

- The API Code requires the first pass internal weld to be thoroughly cleaned and examined both visually and by either the Dye penetrant, Magnetic particle, Vacuum box method, or by applying a penetrating oil to the gap between the shell and the floor. This latter alternative is not recommended because of the difficulty in removing the oil prior to subsequent welding operations. When the weld is found to be sound, the inside and outside welds are completed and visually examined for defects.

- Alternatively, after completing the initial weld passes on the inside and outside, they are thoroughly cleaned and visually examined. After completing the welds, the space between them is pressurised with air to 103 kPa and tested with a soapy solution for leaks.

This method is also included in the European Code for bottom shell plates more than 30 mm thick. The air pressure to be applied to the void between the welds in this case being 30 kPa.

However, the API Code further states that by agreement between the purchaser and the contractor, the above examination methods may be waived if the following examinations are performed on the entire circumference of the welds:

- 1) Visual examination of the initial passes of the inner and outer welds.
- 2) Visual examination of the completed inner and outer welds.
- 3) Examine the completed inner and outer welds by either; liquid penetrant, magnetic particle, or right angle vacuum box and soapy solution.

12.10.4 Fixed roof plate joint testing

The most common and positive method is to pressurise the underside of the roof space when the tank is full of water while under hydrostatic test. For non-pressure tanks, the roof space is pressurised with air to 4 mbar, when working to the API Code and to 7.5 mbar to the BS Code. For low pressure and high pressure tanks to the BS Code, the air test pressure is 3 mbar above the design pressure. A soapy solution is applied to all the welded joints to check for any leakage.

The roof test pressure can be monitored using a simple water manometer 'U' tube made from clear plastic tubing clipped to a vertical wooden board which can be temporarily attached to the roof handrailing near the top roof access platform. The tube is connected to a fitting on the nearest convenient blanked roof nozzle. Note that 1 mbar = 1 cm of water gauge.

The air supply stop valve must be accessible at roof level and if there are no pressure & vacuum valves or emergency vents fitted to the roof then an emergency quick release valve must be fitted to one of the nozzles to enable any excessive build up of air pressure to be released.

Alternatively, the roof joints may be checked by the vacuum box method. This may be the preferred method where large vent openings have been cut in the roof plating of tanks which are to be fitted with internal floating covers. However, in these cases the roof joints can be air pressure-tested prior to cutting the vent apertures. In any event it may be argued that a minute leak path in a roof weld does not matter where large vent openings are present in the roof anyway.

The European Code will accept dye penetrant, vacuum box or an air pressure test as alternative ways of testing roof joints.

12.10.5 Floating roof testing

The centre deck plate, pontoon bottom plate and the rim plate welded joints should be tested as follows:

BS and API Codes — by spraying with penetrating oil on the underside and checking for evidence of leaks on the top side and inside of rim plates.

European Code — by the vacuum box method or by dye penetrant examination.

The fillet welds connecting the bulkheads between pontoons to the inner and outer rim plates and to the pontoon bottom shall be examined for leaks using penetrating oil (or in the European Code, the dye penetrant method) prior to the installation of the pontoon top plates. When continuously welded, the welds connecting the pontoon top plates shall be visually inspected for pinholes or defective welding. In the case of the European Code these latter welds must be inspected by the dye penetrant method.

Compartments which are completely welded can be individually tested with an air pressure of 7 mbar and a soapy solution applied to the welded joints under pressure which have not been previously tested with penetrating oil. The BS and API

Codes offer this test procedure as an alternative to the one outlined in the previous paragraph.

However the European Code requires that both procedures above shall be carried out unless the design of the roof precludes a air pressure test in which case all welds shall be dye penetrant tested.

The primary drain system shall be hydraulically tested prior to the tank hydrotest and the roof drain valves shall be kept open during the hydrotest and observed for leakage.

During the tank hydrotest, the lower deck, the lower pontoon deck and all the submerged roof joints shall be observed for leakage.

Also during the first filling with product the roof decking and pontoon compartments shall be observed for leaks caused by the deeper immersion in the stored product which is likely to have a lower specific gravity than water.

12.10.6 Testing of shell nozzles and apertures

The welds attaching nozzle reinforcing plates to the tank are tested for leaks by pressurising the space between the shell plate and the reinforcing plate with air and applying a soapy solution to the welds to detect leaks. The reinforcing plate has a hole drilled and tapped in it to take the pneumatic connection.

The BS Code states that pneumatic testing of reinforcing plates is not required unless specified by the purchaser but when it is specified it shall be done at a pressure of 1 bar.

Notwithstanding this statement, it is normal practice for a contractor to pneumatically test the reinforcing plates prior to the hydrostatic tank test.

The API and the European Code require the reinforcing plates to be pneumatically tested.

The BS and the European Code also require the nozzle welds to be dye penetrant or magnetic particle tested.

12.10.7 Hydrostatic tank testing

To ensure that the tank is free from leaks, on completion of construction it is filled with water to its design level. What must also be appreciated is that in testing the tank in this way the foundation is also being proved to take the load from the tank. Therefore it is vital that the foundation designer is consulted with regard to the allowable rate of loading for the foundation to prevent excessive settlement or slip failure.

Most tanks in petrochemical service store products with a specific gravity, (s.g.), less than 1.0 and hence the loading that the tank experiences during the hydrotest will not be achieved in service. This effectively assures a factor of safety during the operation of the tank.

Also the initial hydrotest causes plastic yielding in welds where there are localised high stress concentrations.

The following matters have to be considered prior to commencing the hydrostatic test:

- 1) Availability of water source on the tank site.
- 2) Is fresh or salt water to be used (salt water has a s.g. of 1.03).

If salt water is used, then the tank must be thoroughly hosed down with fresh water after being emptied.

A tank fitted with an aluminium or stainless steel internal floating roof must be tested with fresh water.

- 3) Water used for testing a stainless steel tank must be chemically analysed to determine the pH value, chlorine content and the presence of any other potentially corrosive elements.
- 4) When the test is conducted during cold weather then the test water temperature should be checked for suitability against figure 1 of BS 2654.
- 5) The rate of fill, the number and duration of dwell periods during the test and the final period before emptying, is to be agreed with the foundation designer. Also a datum foundation survey must be established prior to the test and settlement surveys taken during the test programme. Clause A.5 of BS2654 gives very good guidance on this topic.
- 6) Establish the maximum tank filling height.
- 7) The European Code contains advice on the hydrotesting of tanks which are designed to hold products with a s.g. greater than 1.0, and this is as follows:

a) Construct a temporary extension of the shell to allow the test water level to be increased above the design liquid level. This extension should be high enough to create a overload of at least 10%.

Authors note: This may be possible for open top tanks but would appear impractical for fixed roof tanks. Also it would seem impractical for products having a high specific gravity. For instance sulphuric acid has a s.g. of 1.84. With the inclusion of a 10% overload this would require a temporary extension equal to the original height of the tank, clearly impractical.

b) The first filling with the high s.g. product should be undertaken under careful supervision, observing the same caution as would apply to the original hydrostatic test. In the case of tanks constructed of carbon and carbon manganese steels, consideration should be given to using materials with enhanced levels of notch ductility, i.e. use a type of steel one or two types higher than would otherwise be required.

- 8) Establish a water disposal point and the maximum allowable rate for the disposal of the water. Also check with the local authority for permission to dispose of rust contaminated water.
- 9) When the tank is filled with water to the maximum height and the roof air test is being performed, the operation of any pressure & vacuum valves and emergency vents can be tested.
- 10) Prior to emptying the tank, all roof nozzles and manholes which were closed off for the test must be opened up to prevent a vacuum forming in the tank which could cause disastrous consequences.

Note: The European Code requires a test for tank stability under negative pressure and the following procedure is adopted:

After the liquid level in the tank has been lowered to one metre above the top of the draw-off nozzle, the tank stability under negative pressure (depressurisation) shall be tested.

All the openings shall be sealed off except for the negative pressure valve (pressure/vacuum) and the water level shall be reduced until the design vacuum is obtained.

Extreme care has to be exercised during this test to ensure that the design vacuum is not exceeded as this could cause a tank collapse.

13 Foundations for ambient temperature storage tanks

This Chapter includes a brief review of various considerations relating to foundations for above ground, vertical cylindrical storage tanks, taken in the main from the tank design Codes.

This is a specialist subject, and those who wish to pursue it in more depth are advised to seek more detailed material for further study.

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13.1 Introduction

This Chapter concentrates its efforts on the foundations for conventional storage tanks, i.e. above ground, vertical cylindrical tanks for the storage of liquids at or above ambient temperatures.

It is clearly important that storage tanks are provided with suitable foundations and there are numerous considerations which must be taken into account where tank foundations are concerned:

- The initial shape of the foundation is important to the tank erector. A level foundation, especially in the area immediately beneath the tank shell, will make the tank erector's task easier and help to ensure that the finished shell is made to good shape tolerances. The various design Codes provide guidance as to acceptable foundation tolerances.
- The behaviour of the foundation in the short term during tank erection and hydrostatic testing, and during service for the life time of the tank is important. Excessive or uneven settlement during erection or testing would clearly be an embarrassment in terms of cost, time and reputation to all concerned. Rectification of foundations which are inconveniently located beneath tanks is an expensive and time consuming business. The tank itself may suffer damage resulting from the settlement which will exacerbate the problems.
- Poor foundations may threaten the integrity of the tank. There have been numerous examples of storage tanks which have suffered sudden bottom failures as a result of foundation shortcomings.
- The initial and ongoing costs of foundations must be given careful scrutiny. A "cheap and cheerful" foundation may appear less attractive when the costs and service outages associated with excessive settlement are made a part of the financial equation.
- The costs associated with ground contamination, particularly by oil-based products are such that leak detection and provisions to prevent ground contamination are now common, and in certain parts of the world mandatory.

13.2 Design loadings

The loading on the foundations of storage tanks divide into three separate areas.

- The central area of the base during operation is subject to uniform loadings from the tank product and non-uniform loadings arising from the influence of the seismic events on the contained liquid which are described in Chapter 15. During tank testing this area of the foundation is subjected to loadings from the hydrostatic head of the test water. For column-supported roofs, there are point loads associated with the column feet which are a combination of the self-weight of the columns plus the relevant parts of the roof loadings.
- The areas of the foundation immediately beneath the tank shell are the subject of line loadings arising from a combination of self-weight, insulation weight, wind, snow, vacuum and seismic loadings.
- Where the tanks are fitted with holding down bolts or straps, the foundation must be designed to resist the calculated uplifts arising from the various loadings. The derivation of these loadings is described in Chapter 4.

13.3 Foundation profiles

It is usual for tanks to be fitted with drains for reasons associated with the removal of unwanted impurities such as water bot-

toms in floating roof tanks, with the need to remove all of the tank contents quickly for tank decommissioning and for tank internal cleaning operations.

For tanks fitted with central drain connections, a slope down to the tank centre sump of a minimum of 1:120 is considered suitable. These tanks usually have a drain line running within the tank, from the central drain to a suitable connection as low as is possible on the tank shell. This is considered a better arrangement than running the drain line beneath the tank bottom to the tank periphery. This has been the cause of leakage and ground contamination problems in the past.

For tanks with one or more peripheral drains and sumps, the tank bottom must be coned up to the tank centre, and a slope of 1:120 is considered suitable. In setting out the as-built slope, consideration must be given to the anticipated edge-to-centre settlement which will occur during hydrostatic testing and operation.

Tanks with a sloping bottom from one side to the other are quite unusual, for reasons connected with the difficulties associated with the cutting and erection of the first course of shell plates. Again a 1:120 minimum slope taking account of anticipated settlement would be normal.

13.4 As-constructed foundation tolerances

To assist in ensuring that a tank is constructed with a shell shape as true as is possible, particularly important for floating roof tanks to prevent roof jamming, it is important that a foundation as close to the design profile as possible, especially around the periphery, is provided. It is quite usual that the foundation contractor and the tank contractor are different companies, either both employed by the owner, or one as a subcontractor of the other. The point in time when the foundation is handed over from one to the other is often a source of a contractual and technical argument, so it is necessary that clear guidelines are provided as to what is required. The various design Codes make efforts to define what is required.

13.4.1 API 650 requirements

API 650 has much to say on this issue in its attempts to provide clear definitions and it is probably worth repeating these in full. The Code divides tanks into those with foundations in a horizontal plane (the vast majority) and those with sloping bases.

For the former:

- Where a concrete ring wall is provided under the shell, the top of the ringwall shall be level within ± 3 mm ($\frac{1}{8}$ ") in any 9 m (30 ft) of the circumference and ± 6 mm ($\frac{1}{4}$ ") in the total circumference measured from the average elevation
- Where a concrete ringwall is not provided, the foundation under the shell shall be level within ± 3 mm ($\frac{1}{8}$ ") in any 3 m (10 ft) of the circumference and within ± 12 mm ($\frac{1}{2}$ ") in the total circumference measured from the average elevation
- Where a concrete slab is provided, the first 0.3 m (1 ft) of the foundation (or width of the annular plate), measured from the outside of the tank shell radially towards the centre, shall comply with the concrete ringwall requirements. The remainder of the foundation shall be within ± 13 mm ($\frac{1}{2}$ ") of the design shape. It is not made clear if this latter requirement is to be applied to the complete perimeter only or to the whole base slab area. If it is the latter, then this seems an onerous requirement for the foundation contractor.

For the sloped foundations the elevations around the circumference shall be calculated from the high point and the actual (measured) elevations shall not deviate from the calculated figures by more than the following:

- Where a concrete ring wall is provided $\pm 3 \text{ mm}$ ($\frac{1}{8}$ "") in any 9 m (30 feet) of the circumference and $\pm 6 \text{ mm}$ ($\frac{1}{4}$ "") in the total circumference
- Where a concrete ringwall is not provided $\pm 3 \text{ mm}$ ($\frac{1}{8}$ "") in any 3 m (10 feet) of the circumference and $\pm 12 \text{ mm}$ ($\frac{1}{2}$ "") in the total circumference

The Code states that the measurements shall be made prior to the water test rather than prior to building the tank. If this includes the foundation tolerances, which it appears to do, then this is unhelpful in sorting out the possible differences between contractors and providing well-defined hand over criteria.

13.4.2 BS 2654 requirements

BS 2654 does specifically address the handover of the foundation from one contractor to another and suggests that it is normal for the owner to provide the foundation to the tank contractor. It states:

The top of the foundation levels shall be checked at a handover stage to the tank erector and the differences in level of the surface of the tank foundation between any two points 10 m apart around the periphery of the tank shall not be greater than $\pm 6 \text{ mm}$ and the envelope of the peripheral surface levels shall lie within 12 mm above to 12 mm below the design levels.

These are locally, and in some cases globally less demanding than the API requirements.

It does suggest that for floating roof tanks, for the reasons mentioned above, that tighter tolerances may be required.

13.4.3 prEN 14015 requirements

This draft Standard also addresses the handover of the foundation to the tank contractor. It requires that, before the erection of the tank, the erector shall ensure that the location, height, shape, geometry, horizontal plane or slope, surface finish and cleanliness of the supporting foundation shall conform to the following:

- Peripheral tolerances
 - The purchaser shall specify the datum height of the foundation and its permissible variation
 - The difference in level between any two points around the foundation shall not be more than 24 mm
 - The difference between any two points 5 m apart around the periphery of the tank shall not be greater than 0.1% of their peripheral distance
 - The tolerance the erector accepts on the inclination or slope of the foundation shall be such as to enable the final vertical tolerances of the tank to be achieved
- Foundation surface tolerances
 - The sag in the as built surface measured with a 3 m long template shall not exceed 10 mm
 - The difference between the design level and as built level shall not exceed the values given in Figure 13.1

This document also has some sensible advice on the provision of detailed information for any holding-down devices which will require accommodating in the foundation and for the dimensional checking of anchor pocket positions and the anchor installation.

13.5 Site investigations

At any site where it is proposed to construct storage tanks, it is necessary to have knowledge of the sub-surface conditions so

Diameter of tank D m	Difference mm
$D \leq 10$	10
$10 < D \leq 50$	$D / 1000$
$50 < D$	50

Figure 13.1 Foundation surface tolerances
From prEN 14015, table 16.2.3

that the ability of the soil to bear the imposed loadings, the necessity for soil improvements and the anticipated settlements can be evaluated. Many storage tanks are constructed at coastal locations on poor estuarine soils with poor load bearing properties. In these situations it is often found necessary to enhance the load bearing properties of the soil, or to modify the tank proportions to decrease the imposed loadings.

Some storage tanks are built at sites where the nature of the sub-soil is well known. In these cases much useful information can be obtained by the study of the performance of similar structures on these sites.

Where this information is not available, a geotechnical site investigation must be carried out. The tank design Codes provide some guidance regarding this matter.

API 650 suggests that the necessary information should be obtained from soil borings, load tests, sampling, laboratory testing and analysis carried out by suitably experienced persons or companies, preferably familiar with similar structures in the same area.

BS 2654 suggests that a site investigation is carried out in accordance with BS 5930 (*Reference 13.1*).

prEN 14015 suggests that wherever possible, storage tanks should be sited in areas where the subsoil conditions are homogeneous, and have good characteristics in respect of load bearing and settlement. Prior to the start of the design and construction of the foundation, a thorough geotechnical investigation should be conducted to determine the stratigraphy and physical properties of the soils underlying the site. Measurements should include soil resistivity, conductivity and local water table depth and variability. In areas subject to seismic excitations, either the local building regulations should be consulted, or if these do not provide sufficient data, then a Seismic Hazard Assessment (SHA) should be conducted by persons or companies suitably experienced and skilled in this type of work.

The Codes are agreed that certain sites should be avoided, or if they must be used, perhaps for economic reasons, then must be subjected to special consideration. API 650 provides the most comprehensive list which is as follows:

- Sites on hillsides, where part of a tank may be on undisturbed ground or rock, and part may be on fill or another construction where the depth of fill is variable
- Sites on swampy or filled ground, where the layers of muck or compressible vegetation are at or below the surface, or where corrosive materials may have been deposited as fill
- Sites underlain by soils, such as layers of plastic clay or organic clays, that may support heavy loads temporarily, but settle excessively over long periods of time
- Sites adjacent to water courses or deep excavations, where lateral stability of the ground is questionable
- Sites immediately adjacent to heavy structures that distribute some of their load to the sub soil under the tank sites,

thereby reducing the sub soils capacity to carry additional loadings without excessive settlement

- Sites where tanks may be exposed to flood waters, possibly resulting in uplift, displacement or scour
- Sited in regions of high seismicity that may be susceptible to liquefaction
- Sited with thin layers of soft clay soils that are directly beneath the tank bottom and can cause lateral ground stability problems

13.6 Soil improvement

If the subsoil is found to be inadequate for the imposed loads without excessive or uneven settlement, and the tank cannot be relocated to another area where the soil conditions are better, then the Codes are agreed that one of a number of means of soil improvement may be used:

- Removal and replacement of unsatisfactory material by suitable compacted fill
- Improvement of the soft or loose material by vibration, dynamic compaction or pre-loading with an overburden of other material
- Sub-soil drainage with or without pre-loading
- Stabilization by chemical grout injection
- Provision of a reinforced concrete raft with or without supporting piles

The design, specification and undertaking of these forms of foundation improvement should be left to those experienced in this type of work.

13.7 Settlement in service

The prime function of the tank foundation designer is to provide a foundation at an economic cost, which will protect the tank from excessive settlements during its construction, hydrostatic test and service life. A conventional storage tank may be subject to a settlement which is made up of a combination of the following:

- Global settlement. This is the uniform downward settlement of the completed structure
- Differential settlements:
 - Tilting of the tank across its diameter
 - Edge-to-centre settlement along a radial line to the tank centre
 - Differential settlement around the tank periphery

Storage tanks have differing tolerances to these various different forms of settlement. The tolerance is also a function of the tank type and geometry. For tanks built on poor but uniform soils where the main settlement is global with little accompanying differential settlement, and the connecting pipework has the necessary flexibility, settlements measured in meters have been recorded without undue detrimental effects. There are sites where this order of settlement is a part of the life cycle of the storage tanks. They are designed with permanent shell jacking brackets, or suitably stiffened for lifting by other means such as air bags. When these tanks have settled by an agreed amount, they are lifted and the foundation is refurbished at the original elevation.

The ability of a tank to accommodate edge-to-centre settlement can be calculated with some degree of confidence. This form of settlement is almost invariably a downward movement of the centre of the bottom relative to the tank shell. Its limiting value is a function of the tensile stresses in the bottom plates and the in-

ward force exerted on the tank bottom corner by the bottom plates. There are rules in the various design Codes to allow these calculations to be made.

Clearly a tank with a coned up to the centre bottom is better suited to cope with this form of settlement as it has to pass from the cone up, through flat to the cone down before serious tensile stresses are imposed on the bottom plates. Some owners have their own rules for situations where this type of settlement is anticipated. In addition to the cone up preset, some of these involve an improved bottom plate joint (perhaps a two pass single-sided fillet, a double-sided fillet or butt welding) and a stiffening of the tank bottom corner.

Tilt, as long as it is pure tilt, is another form of settlement which most tanks can accommodate without undue problems, with the exception of floating roof tanks where some binding may occur.

Differential settlement around the tank periphery is usually problematic. Floating roof tanks change shape giving rise to roof jamming at quite small settlements of this type, and fixed roof tanks can be distressed by their attempts to bridge gaps. It is often difficult to separate the components due to tilt and differential settlement from a set of bottom level readings. The method given in API 653 (Reference 13.2) is useful and Figure 13.2 is taken from that document showing how this is achieved.

Specific guidance as to what represents acceptable limits for the different forms of settlement applied to the different types of tanks is not easy to find. The design Codes are not helpful. The tank maintenance and repair codes are more forthcoming (References 13.2 and 13.3).

The hydrostatic testing of the tank is the point at which the foundation design is first called upon to perform its intended duties. BS 2654 includes some sensible advice regarding tank testing. The testing of the first tank in a new area is critical and should be carried out with caution and comprehensive settlement

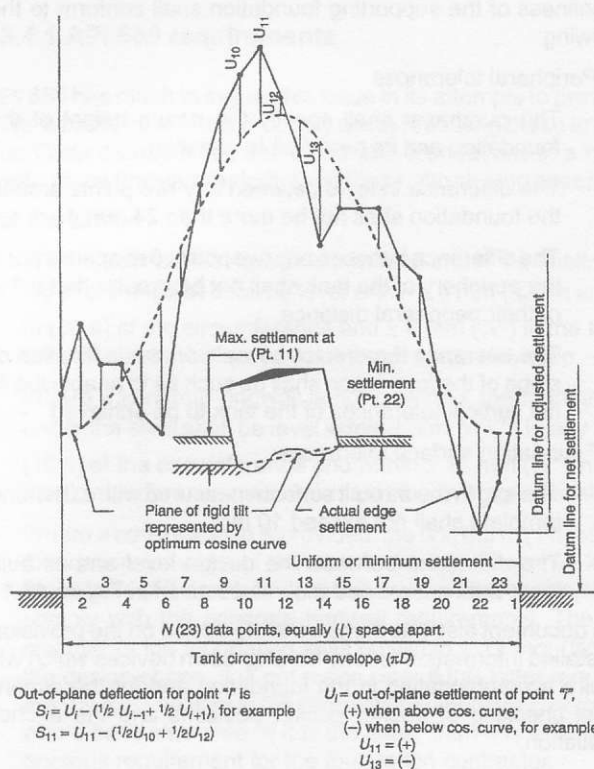


Figure 13.2 Graphical representation of tank shell settlement
 From API 653, figure B-3

measurement provisions. The testing of subsequent tanks in the same area may be adjusted, dependent on the results of this first test.

For tanks where the ground conditions are good and settlements are anticipated to be modest, it is acceptable to half fill the tank as quickly as is practicable before stopping and taking settlement measurements. It should then be filled to three quarters full and then to the full height with pauses for settlements at each point. The full water load should be maintained for 48 hours, and if no significant settlement takes place, the tank can be emptied.

For tanks built on weak ground, a much more cautious test method is proposed with slow filling rates and frequent pauses, some prolonged, for settlement rates to slow or stop. Clearly in these situations, sufficient time must be allowed in the construction programme for the extended test period.

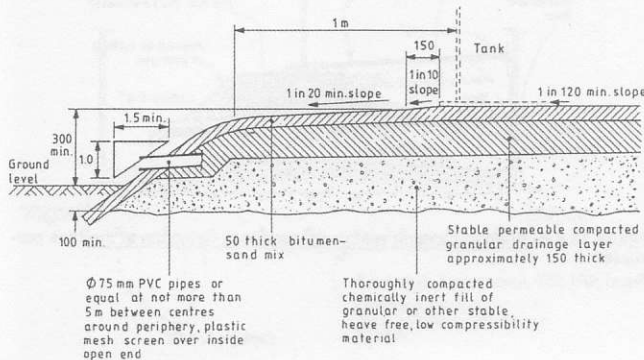
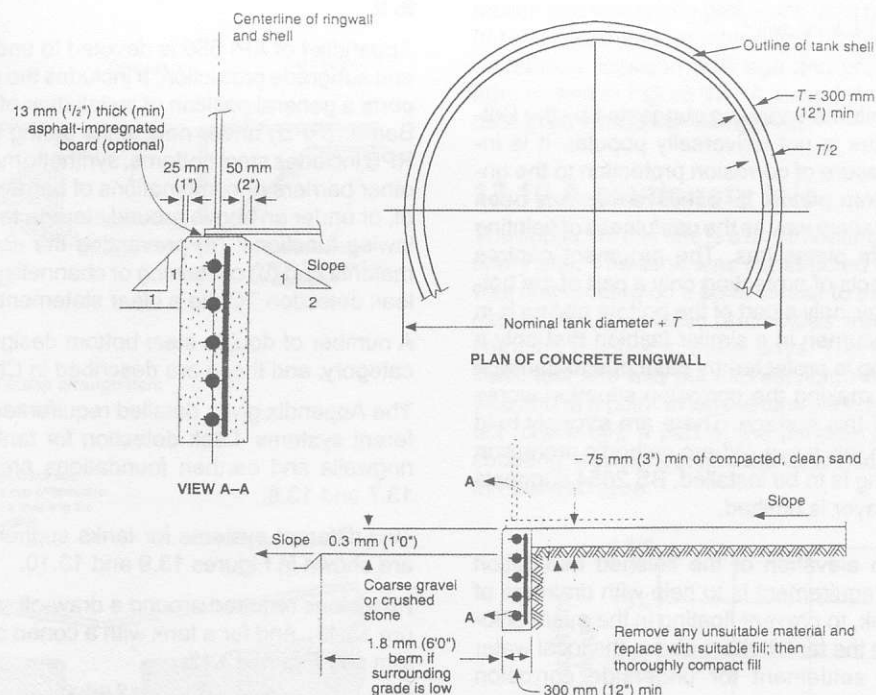


Figure 13.3 Typical tank foundation without a ringwall
From BS 2654, figure 35

13.8 Foundation types

The Codes are in agreement that a number of different types of tank foundation are acceptable. These are:

- Earth foundations without a ringwall. A typical example is shown in Figure 13.3. The capping with sand bitumen is something which both the British and the European Standards are keen, if not insistent on. API 650 makes no such specific requirement. The plastic tubes are for early indication of bottom leakage and to help to prevent foundation washout problems. (See Section 13.10).
- Earth foundations with a concrete ringwall. A typical example is shown in Figure 13.4. The ringwall is of reinforced concrete and details are given in the Standard for the design of this ringwall. Cautionary words are included in all of the Standards regarding the possible problems of differential settlement between the ringwall and the material within the ringwall (usually compacted fill) and its effects on the local support of the tank bottom.
- Earth foundations with a crushed stone or gravel ringwall. See Figure 13.5 for a typical example. It is important that the exposed shoulder is protected from erosion. It should be remembered that heavy rain falling on a storage tank can result in a vigorous waterfall around the periphery of the tank.
- A concrete slab foundation. Figure 13.6 shows a typical example. This particular example indicates a thin slab with a thickened peripheral region. On occasions, the slab diameter is increased to provide additional support to the tank.
- A concrete slab foundation with supporting piles. Where piles are not or cannot have their integrity proven by field testing, it is suggested that the slab is designed to accommodate the failure of an individual pile.



Notes:

1. See B.4.2.3 for requirements for reinforcement.
2. The top of the concrete ringwall shall be smooth and level. The concrete strength shall be at least 20 MPa (3000 lbf/in.²) after 28 days. Reinforcement splices must be staggered and shall be lapped to develop full strength in the bond. If staggering of laps

is not possible, refer to ACI 318 for additional development requirements.

3. Ringwalls that exceed 300 mm (12 in.) in width shall have rebars distributed on both faces.
4. See B.4.2.2 for the position of the tank shell on the ringwall.

Figure 13.4 Example of tank foundation with concrete ringwall
From API 650, Appendix B, figure B-1

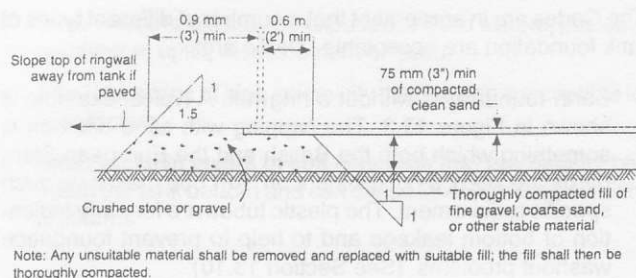
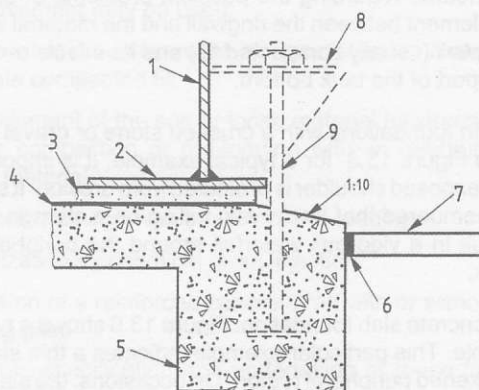


Figure 13.5 Example of foundation with crushed stone ringwall
From API 650, Appendix B, figure B-2



Key		
1 Tank shell	4 Membrane	7 Bund surface
2 Tank bottom	5 Foundation raft	8 Chair (when required)
3 50 mm sand/bitumen	6 Auxiliary seal	9 Holding down bolt (when required)

Figure 13.6 Typical concrete slab foundation
From prEN 14015, figure I.4.1-3

The 50 mm thick sand bitumen capping suggested by the British and European Codes is not universally popular. It is intended to provide a measure of corrosion protection to the underside of the tank bottom plates. Its effectiveness has been challenged, in much the same way as the usefulness of painting the underside of bottom plates has. The argument centres around the possible effects of protecting only a part of the bottom plating. That is to say, only a part of the bottom plating is in contact with the sand bitumen in a similar fashion that only a part of the bottom plating is protected by paint due to damage by welding operations, making the corrosion situation worse than protecting none of this surface. There are strongly held and conflicting views on this issue. Where cathodic protection of the tank bottom plating is to be installed, BS 2654 suggests that the sand bitumen layer is omitted.

The 300 mm minimum elevation of the finished foundation above the local grade requirement is to help with drainage of water away from the tank, to prevent floating in the event of local flooding and to keep the tank bottom above the local water table in the event of settlement for underside corrosion prevention reasons.

For tanks which require holding-down anchors, the foundation will normally be of the concrete ringwall or the slab type. To resist the uplift forces, the dead weight of the ringwall or appropriate portion of the slab can be used. Tee-shaped ringwalls which mobilise part of the local sub grade and ground anchors are also a possibility.

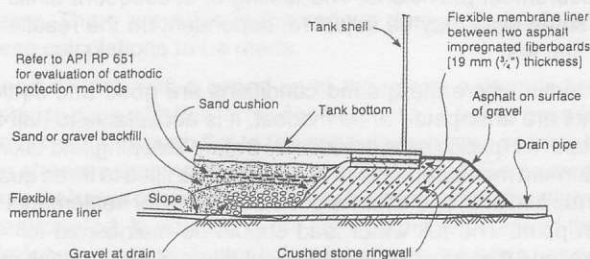


Figure 13.7 Crushed stone ringwall with under-tank leak detection at the tank perimeter
From API 650, Appendix B, figure I-2

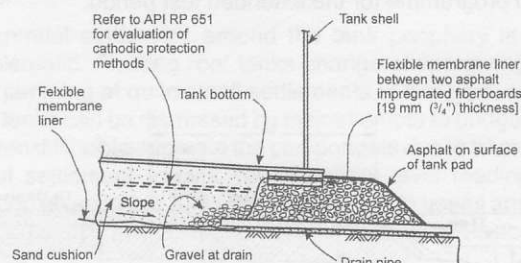


Figure 13.8 Earthen foundation with under-tank leak detection at the tank perimeter
From API 650 Appendix I, figure I-3

13.9 Leak detection and prevention of ground contamination

API 650 has much more to say on this subject than do the British or the European Codes, which only give a passing mention to it.

Appendix I of API 650 is devoted to under-tank leak detection and subgrade protection. It includes the note stating: "API supports a general position of installation of a Release Prevention Barrier (RPB) under new tanks during initial construction. An RPB includes steel bottoms, synthetic materials, clay liners and other barriers or combinations of barriers placed in the bottom of, or under an above ground storage tank, which have the following functions: (a) preventing the escape of contaminated material and (b) containing or channelling released material for leak detection." Quite a clear statement of intent.

A number of double steel bottom designs are included in this category, and these are described in Chapter 3, Section 3.4.

The Appendix gives detailed requirements for a number of different systems. Leak detection for tanks with crushed stone ringwalls and earthen foundations are illustrated in Figures 13.7 and 13.8.

Two different systems for tanks supported by concrete slabs are shown in Figures 13.9 and 13.10.

Provisions required around a draw-off sump are shown in Figure 13.11, and for a tank with a coned down to the centre bottom see Figure 13.12.

This section of the Code also deals with tanks where the bottom is supported by grillages. The use of a grillage allows the tank bottom to be visually inspected for leakage, something which is considered necessary for a small number of products. Typical grillage arrangements using parallel and radial supports are shown in Figure 13.13. Grillage support is restricted to tanks with shell plate thicknesses up to 13 mm and maximum temperatures of 90 °C. By agreement, the shell thickness limit can be

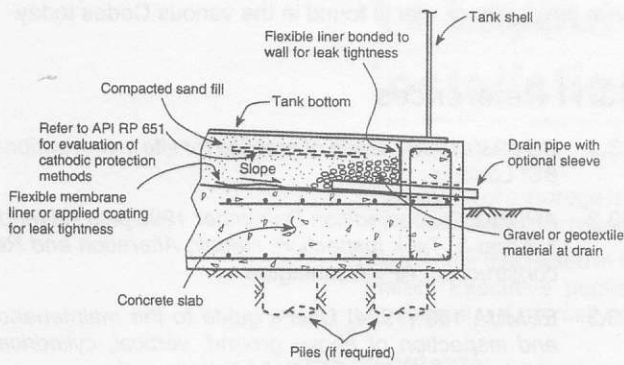


Figure 13.9 Reinforced concrete slab with leak detection at the perimeter
From API 650 Appendix I, figure I-6

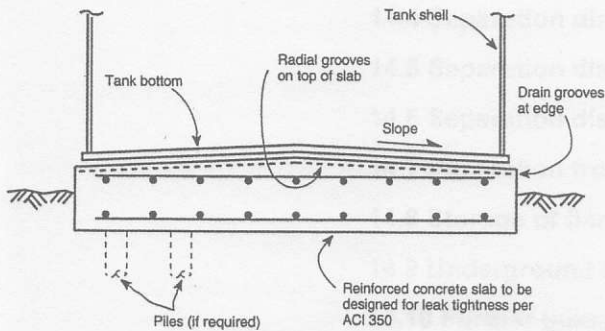


Figure 13.10 Reinforced concrete slab with radial grooves for leak detection
From API 650 Appendix I, figure I-7

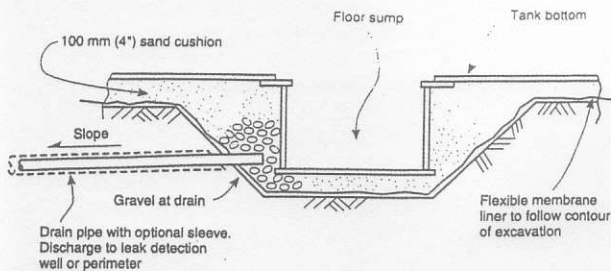


Figure 13.11 Typical draw-off sump arrangement
From API 650 Appendix I, figure I-8

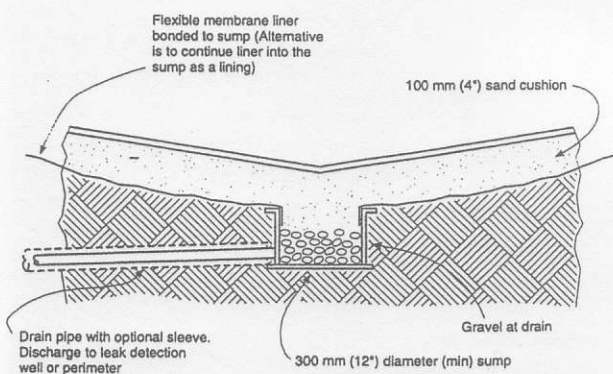


Figure 13.12 Centre sump for downward-sloped bottom
From API 650 Appendix I, figure I-9

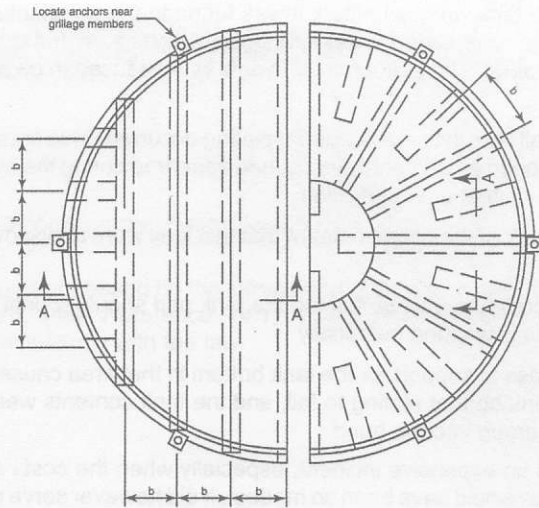
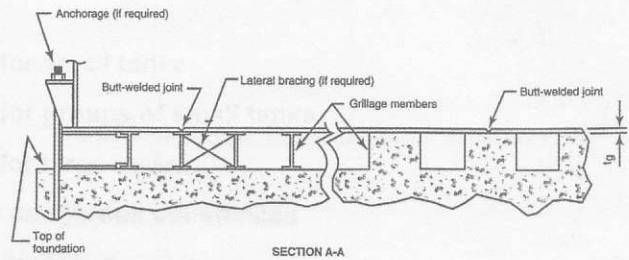


Figure 13.13 Tanks supported by grillage members
From API 650 Appendix I, figure I-11



extended. This section of the Code provides guidance for bottom plate thickness and grillage spacing.

Another useful document for those interested in this subject is EEMUA Publication No. 183, (Reference 13.4). This provides a wealth of sensible information on tank foundations, tank bottom design, corrosion prevention, inspection techniques, leak detection and sub-grade protection from pollution. It includes a list of references and an interesting figure, which gives a simple correlation between tank age and probability of bottom leakage, shown in Figure 13.14, based on a statistical analysis of data from various oil companies.

13.10 A cautionary tale

The subject of this tale is a large floating roof tank on a major refinery site. The tank was constructed in the 1960s. The tank was constructed on a base similar to that shown in Figure 13.3 except that the plastic drain pipes were not fitted, which was common practice in those days. The tank survived its hydrostatic test and was put into service. After a brief period in service and at a point when the tank was close to being full of product (crude oil), a part of the periphery of the foundation pad suddenly washed out and the tank discharged its contents into the bunded area.

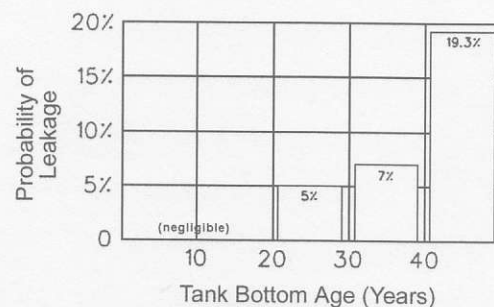


Figure 13.14 Probabilities of leakage from tank bottoms plotted against age
From EEMUA Publication No. 183, figure 1

When the tank was examined, it was found that a substantial failure had occurred in the welded seams of the lap-welded tank bottom plating. The sequence of events was deduced to be as follows:

- A small leak in the tank bottom plating occurred. This could have been an original defect or had appeared during the hydrostatic test or in operation
- The lack of drain pipes meant that this leak went undiscovered
- The pressure built up behind the tank pad shoulder until it suddenly washed out locally
- The loss of support for the tank bottom in that area caused the tank bottom plating to fail, and the tank contents were discharged into the bund.

This was an expensive incident, especially when the costs of prevention would have been so modest. It did however serve to

focus attention on the design of tank foundations and helped to form the guidance that is found in the various Codes today.

13.11 References

- 13.1 BS 5930:1999 – *Code of practice for site investigations*, BSI London
- 13.2 API 653:Second edition December 1995 plus Addenda 1,2 and 3. *Tank Inspection, Repair, Alteration and Reconstruction*, API Washington
- 13.3 EEMUA 159 (1994) *User's guide to the maintenance and inspection of above ground, vertical, cylindrical, steel storage tanks*, EEMUA London
- 13.4 EEMUA 183 (1999) *Guide for the prevention of bottom leakage from vertical, cylindrical, steel storage tanks*, EEMUA London

14 Layout of ambient temperature tank installations

The layout of a storage tank installation must meet with good practice and also the relevant legal and local authority requirements.

The topics discussed in this Chapter are based on the information set out in the UK's Health & Safety Executive publication 176, (see *Reference 14.1*). Following the guidance in this document will normally ensure compliance with the law.

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14.4 Separation distances for small tanks

For the purposes of this guidance, small tanks are defined as those with a capacity of less than 1000 litres. The minimum recommended separation distances are given in Table 14.1. The distances are based on the following assumptions:

Table 14.1	Minimum recommended separation distances for small tanks
Capacity of tank (litres)	Separation distance (metres)
Up to 100	0.5
100 to 250	1.0
250 to 500	1.5
500 to 1000	2.0

Figure 14.1 Minimum recommended separation distances for small tanks



Figure 14.2 Diagram illustrating the layout of small tanks with separation distances

14.1 Introduction

The guidance given in the HSE publication, *Reference 14.1*, generally applies to flammable liquids with a flashpoint of 55°C or below. This includes all highly flammable liquids (as defined by the Highly Flammable Liquids and Liquefied Petroleum Gases Regulations 1972, see *Reference 14.3*) and all petroleum spirit and petroleum mixtures as defined in the Petroleum (Consolidation Act) 1928 (*Reference 14.4*) and the Petroleum (Mixtures) Order 1929, (*Reference 14.5*). It includes all liquids that are classified as flammable, highly flammable or extremely flammable for supply according to CHIP: Chemicals (Hazard Information and Packaging for Supply) Regulations 1996/16-20, *Reference 14.6*

The guidance is also relevant to liquids with a flashpoint above 55 °C which are stored at temperatures above their flashpoint.

The location and layout of a storage installation should be selected with care. The aims are to protect people and property from the effects of a fire at the tank, and to protect the tank from fires which may occur elsewhere on site. As a rule, if the temperature of a steel tank is allowed to rise above 300 °C, then the structure of the storage tank will be adversely affected and it may rupture.

Storage tanks may be located above ground, underground or in mounds. Each location has different advantages and disadvantages.

- Storage at ground level, in the open air, has advantages because leaks are more readily detected and contained, and any vapour produced will normally be dissipated by natural ventilation. Examinations, modifications and repairs are also easier, and corrosion can be more readily identified and controlled.
- Underground or mounded tanks give better fire protection and save space. But leakage, resulting from damage or corrosion, may be difficult to detect. This could lead to ground contamination, environmental problems and possible fire and explosion risks to nearby buildings and basements.

When selecting the location of a single or multi-tank installation, consideration should be given to the distance of the proposed storage from:

- the site boundary
- on-site buildings, particularly those that are occupied
- fixed ignition sources
- storage or processing of other dangerous substances
- road or rail tanker transfer facilities.

Other factors to consider are:

- the position of the tanks (above ground or below ground);
- the size and capacity of the tanks;
- the design of the tanks (fixed roof or floating roof).

Tanks should not be located:

- under buildings
- on the roofs of buildings
- in positions raised high above ground level
- on top of one another
- above tunnels, culverts or sewers.

Tank locations inside buildings should be avoided. (See however Section 14.8.)

14.2 Above ground tanks

Above tanks ground should be sited in a well-ventilated position separated from the site boundary, occupied buildings, sources of ignition, and process areas. Figure 14.1 shows a plan of a typical layout for storage tanks with separation distances. The layout of tanks should always take into account the accessibility needed for the emergency services.

The separation distances will depend on various factors but primarily on the capacity of the tank. Advice on separation distances is given for "small" tanks, generally associated with small to medium chemical processes, and for "large" tanks associated with refinery and other large-scale storage facilities.

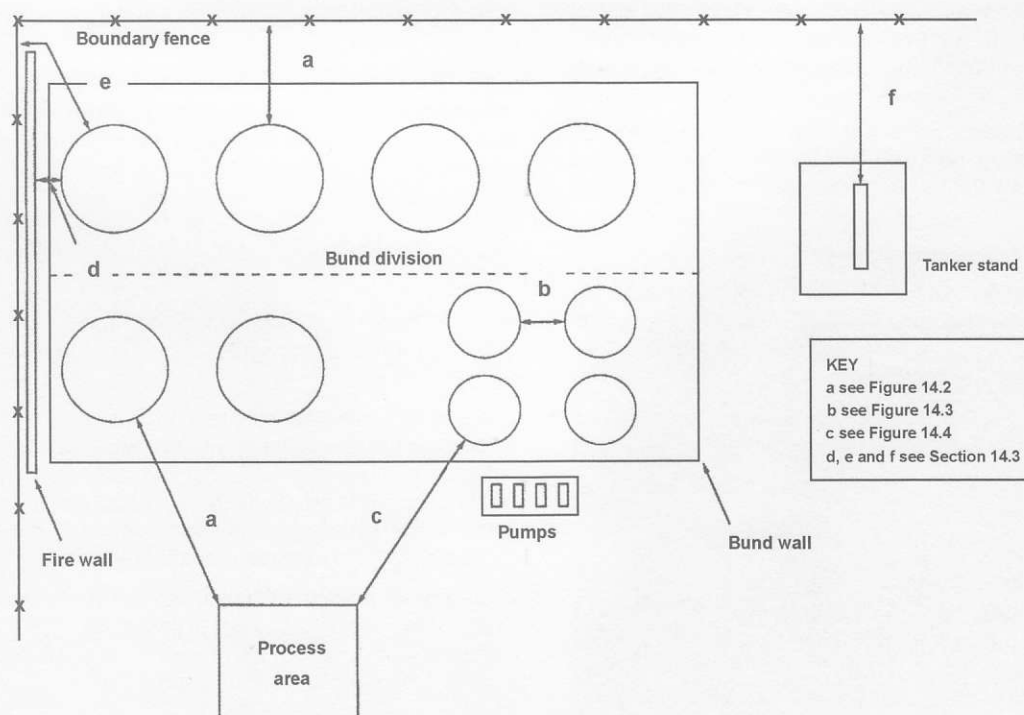


Figure 14.1 Typical storage tanks layout plan

The separation distances given are unlikely to give complete protection in the event of a fire or explosion involving the tank, but should allow sufficient time for people to be evacuated, provided there are good means of escape. They should also allow sufficient time for additional fire-fighting equipment and emergency procedures to be mobilised.

Under certain circumstances, it may be necessary to increase the separation distances or provide additional fire protection. Such circumstances may for example, be where there are problems with:

- the local water supply.
- where the site is remote from external help (such as the fire authority).
- where the tank is close to a heavily populated area.

14.3 Fire walls

A fire wall may be used to give additional protection to small tanks. They are not usually practicable or economic for larger tanks.

Where a fire wall is installed, it should be at least the height of the tank, with a minimum height of 2 m, and should normally be sited between 1 m and 3 m from the tank. It may form part of the bund wall or a building wall. A fire wall should normally be provided on only one side of a tank, to ensure adequate ventilation. The wall should be long enough to ensure that the distance between the tank and a building, boundary, process plant or source of ignition is at least the appropriate distance set out in Figure 14.2, measured around the ends of the wall.

To be effective a fire wall should:

- have no holes in it
- have at least half-hour fire resistance
- be weather-resistant
- be sufficiently robust to withstand foreseeable accidental damage.

A reinforced concrete or masonry construction is recommended.

Loading/unloading bays for road tankers should be located in a safe, well-ventilated position. The minimum recommended distance of a filling point from occupied buildings, the site boundary and fixed sources of ignition is 10 m.

14.4 Separation distances for small tanks

For the purposes of this guidance "small" tanks are considered to be tanks with a diameter of less than 10 m. Figure 14.2 shows the minimum recommended separation distances for single small tanks. The distances are based on what is considered to

Tank capacity (m ³)	Separation distance (m)
Less than or equal to 1	1*
Greater than 1 and less than or equal to 5	4
Greater than 5 and less than or equal to 33	6
Greater than 33 and less than or equal to 100	8
Greater than 100 and less than or equal to 250	10
Greater than 250	15

* But at least 2 m from doors, plain-glazed windows, or other openings or means of escape. Also not below any opening (including building eaves and means of escape) from an upper floor, regardless of vertical distance.

Figure 14.2 Minimum separation distances for small tanks

be good practice and have been widely accepted by industry. The minimum separation distance is the minimum distance between any point on the tank and any building, boundary, process unit, or fixed source of ignition.

14.5 Minimum separation distances for groups of small tanks

Small tanks may be placed together in groups. A tank is considered as part of a group if adjacent tanks are within the separation distances given in Figure 14.2. The aggregate capacity of the group should be no more than 8000 m³ and the tanks should be arranged so that they are all accessible for fire-fighting purposes.

The recommended minimum separation distances between individual tanks in a group are given in Figure 14.3. If a serious fire develops involving one tank in a group then it is unlikely that these between-tank separation distances will prevent damage or even destruction of the adjacent tanks. However, they should allow sufficient time for emergency procedures to be implemented and for people to be evacuated from areas threatened by the incident.

For the purpose of determining separation distances from site boundaries, buildings, process areas and fixed sources of ignition, a group of small tanks may be regarded as one tank. The minimum recommended separation distances for groups of small tanks are given in Figure 14.4. The minimum recommended separation distance between adjacent groups of small tanks is 15 m.

14.6 Separation distances for large tanks

"Large" tanks are considered to be tanks with a diameter larger than 10 m.

The minimum recommended separation distances for large tanks are given in Figure 14.5.

The information is based on the Institute of Petroleum Model Code Of Safe Practice, part 19, (Reference 14.2).

Tank size	Recommended separation distance between tanks
Less than or equal to 100 m ³	The minimum required for safe construction and operation
Greater than 10m ³ but less than 10 m in diameter	Equal to or greater than 2 m

Figure 14.3 Minimum between-tank separation distances for groups of small tanks

Total capacity of the group (m ³)	Separation distance m
Less than or equal to 3	1*
Greater than 3 and less than or equal to 5	4
Greater than 15 and less than or equal to 100	6
Greater than 100 and less than or equal to 300	8
Greater than 300 and less than or equal to 750	10
Greater than 750 and less than or equal to 8000	15

* But at least 2 m from doors, plain-glazed windows, or other openings or means of escape. Also not below any opening (including building eaves and means of escape) from an upper floor, regardless of vertical distance.

Figure 14.4 Minimum recommended separation distances for groups of small tanks, from site boundaries, etc

Factor	Minimum separation from any part of the tank
Between adjacent fixed roof tanks	Equal to the smaller of the following: (a) the diameter of the smaller tank (b) half the diameter of the larger tank (c) 15 m but not less than 10 m
Between adjacent floating roof tanks	10 m for tanks up to and including 45 m diameter 15 m for tanks over 45 m diameter The spacing is determined by the size of the larger tank
Between a floating roof tank and a fixed-roof tank	Equal to the smaller of the following: (a) the diameter of the smaller tank (b) half the diameter of the larger tank (c) 15 m but not less than 10 m
Between a group of small tanks and any tank outside the group	15 m
Between a tank and the site boundary, any designated non-hazardous area, process area or any fixed source of ignition	15 m

Figure 14.5 Minimum separation distances for large tanks

	LPG cylinders (>50 kg total capacity)	LPG vessels (up to 135 m ³)	LPG vessel (over 135 m ³)
Flammable (flashpoint <32°C)	3 m to bund wall	6 m to bund wall	15 m to bund wall
Flammable liquid (flashpoint <32°C - 65°C) Tank size up to 3 m ³	3 m to bund wall	3 m to bund wall	6 m to bund wall
Flammable liquid (flashpoint <32°C - 65°C) Tank size over 3 m ³	3 m to bund wall	3 m to bund wall	15 m to bund wall

Figure 14.6 Minimum recommended separation distance from LPG storage

14.6 Separation from other dangerous substances

Separation may also be used to prevent or delay the spread of fire to and from storage or process areas where other dangerous substances may be present in quantity. Figure 14.6 shows the minimum recommended separation distances from LPG storage.

Figure 14.2 may be used to estimate separation distances from other hazardous substances. If published guidance exists, for the particular hazardous substance concerned, the recommended minimum separation distance is the greater of the distances given in Figure 14.2 and the relevant guidance.

14.8 Storage of flammable liquids in buildings

Flammable liquids should not normally be stored in bulk tanks in buildings. If storage is required in buildings then only the minimum amount should be stored and for the minimum time, preferably no more than that needed for one day or one shift.

Additional safety measures may be needed for the building. These include:

- a single-storey and generally non-combustible construction;

- a lightweight roof or other means of explosion relief. Where this is not reasonably practicable an acceptable alternative is to provide sufficient mechanical ventilation to remove flammable vapour released in the event of an incident;
- a high standard of natural ventilation, using high and low-level openings in the walls (typically 2.5% of the total wall and roof area) leading directly to the open air. Alternatively, permanent mechanical ventilation can be used, equivalent to at least five air changes per hour;
- fire separation (by means of a partition of at least 30 minutes fire resistance) between the part of the building housing the tank and other parts of the building, or other buildings within 4 m; and adequate means of escape.
- adequate means of escape.

The tank should have the following features:

- effective means of preventing the spread of leakage. Where appropriate the building walls may form part of the bund, providing they are impervious, have sufficient strength and doorways are fitted with kerbs, ramps or sills;
- vents which discharge to a safe place in the open air.

Adequate means of cooling the tank surface in the event of fire in the building may be needed. In some cases this may be done by the fire brigade using portable equipment, but in others a fixed water installation may be necessary. Adequate drainage is essential to avoid tank flotation and local flooding.

14.9 Underground tanks

The minimum recommended separation distance from any underground tank to any building line is at least 2 m, to avoid undermining the building foundations. It is advisable to increase this distance to 6 m for a basement or pit, to minimise the risk of vapour accumulation.

14.10 Further guidance

Guidance on the layout of storage tank installations is also contained in the publications listed below, but HSE 176, (*Reference 14.1*) would seem to be the favoured document because of very fact that the Health & Safety Inspectorate will refer to it for guidance and as a basis of good practice.

Refining Safety Code, Model Code of Safe Practice Part 3, The Institute of Petroleum

European Model Code of Safe Practice in the Storage and Handling of Petroleum Products. Part 11: Design, Layout and Construction, European Petroleum Organisations (European Technical Co-operation)

Safety datasheets for substances and preparations dangerous for supply. Guidance on regulation 6 of the Chemicals (Hazard Information and Packaging for Supply) Regulations 1994. Approved Code of Practice, L62 HSE Books 1994, ISBN 0 7176 0859 X.

Approved supply list. Information approved for the classification and labelling of substances and preparations dangerous for supply. CHIP 96 and 97, L76 HSE Books 1997, ISBN 071 761412 3.

Approved guide to the classification and labelling of substances and preparations dangerous for supply. CHIP 971, L100 HSE Books 1997, ISBN 071 760860 3.

CHIP 2 for everyone, HSG126 HS Books 1995, ISBN 0 7176 0857 3.

The storage of LPG at fixed installations, HSG34 ME Books 1987, ISBN 011 883908 X (currently under revision).

Fire precautions at petroleum refineries and bulk storage installations: model code of safe practice part 19, Institute of Petroleum, Wiley 1993, ISBN 047 194328 2.

The keeping of LPG in cylinders and similar containers CSA, HSE Books 1986, ISBN 071 760631 7 (currently under revision).

Code of practice for ventilation principles and designing for natural ventilation, BS 5925: 1991.

14.11 References

- 14.1 *Storage of flammable liquids in tanks*, HSE 176, HSE Books 1998, ISBN 071 761470 0.
- 14.2 *Fire precautions at Petroleum Refineries and Bulk Storage Installations, Model Code of Safe Practice Part 19*, The Institute of Petroleum.
- 14.3 *The Highly Flammable Liquids and Liquefied Petroleum Gases Regulations 1972*, SI 1972/917, HMSO 1972, ISBN 011 020917 6.
- 14.4 *Petroleum (Consolidation) Act 1928 Chapter 32*, HMSO 1928.
- 14.5 *Petroleum (Mixtures) Order 1929*, HMSO 1929, ISBN 011 100031 9.
- 14.6 *The Chemicals (Hazard Information and Packaging for Supply) Regulations 1994*, SI 1994/3247, HMSO 1994 ISBN 011043877 9 as amended by *The Chemicals (Hazard Information and Packaging for Supply) (Amendment) Regulations 1996*, SI 1996/1092, HMSO 1996, ISBN 0 11054570 2 and *The Chemicals (Hazard Information and Packaging for Supply) (Amendment) Regulations 1997*, SI 1997/1460 HMSO 1997, ISBN 011 063750 X.

