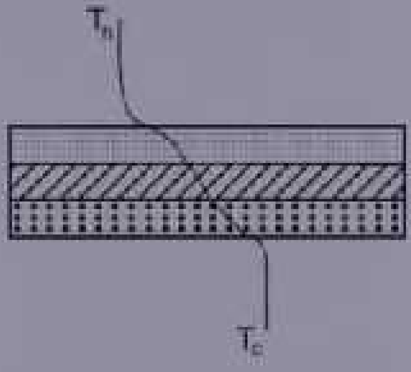
HEAT EXCHANGER DESIGN HANDBOOK



T. KUPPAN

HEAT EXCHANGER DESIGN HANDBOOK

MECHANICAL ENGINEERING

A Series of Textbooks and Reference Books

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L. L. Faulkner

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11
Mechanical Design of Shell and Tube Heat Exchangers

1 STANDARDS AND CODES

Standards and codes were established primarily to ensure safety against failure. The need for safety standards is obvious in a world growing increasingly aware of the hazards posed to people, property, and the environment by complex technology, which may have the potential for doing immense harm [1a,1b]. The codes and standards give guidance and in some cases govern the design, manufacture, construction, operation, and maintenance of heat exchangers and pressure vessels. These codes and standards are themselves based upon research, development, and experience. The present-day codes have their origin in the rules laid down by the insurance companies in the past for the safe operation of boilers and pressure vessels against explosions or accidents and consequential damage to the human lives and property.

1.1 Standards

A standard can be defined as a set of technical definitions and guidelines, or how-to instructions for designers and manufacturers [2]. Standards are mostly voluntary in nature. They serve as guidelines but do not themselves have a force of law. Standards are universally adopted in manufacturing, procurement, and operation of thousands of devices and products, including raw materials, equipment, etc. Many standards have been adopted as a means of satisfying the regulatory or procurement requirements. Standards help to reduce the cost of products and processes in the following manner:

At the design level, rationalization of design procedure, drawings and specifications takes place. This avoids the repetition of detailed design analysis for either identical or similar jobs.

Standards help in complete interchangeability and uniformity of fundamental design, tools, gages, tool accessories, etc.

The standards can be of the following major four types:

1. Company standards
2. Trade or manufacturers association standards
3. National standards
4. International standards

Company Standards

Company standards are followed by individual companies, subcontractors to the companies, and the licence holders.

Trade or Manufacturers Association Standards

Trade or manufacturers association standards are the rules and the recommendations of various manufacturers of common interest, developed based on experience in design, manufacture, installation, and operation. While making the standards, feedback from users is normally included. Manufacturers association standards that are most famous among heat exchanger manufacturers are TEMA [3], HEI [4–6], and API Standards [7]. There are also EJMA Standards [8] for the design of membrane type expansion joints and ANSI (American National Standards Institute) standards for design of fittings, flanges, valves, piping and piping components.

National Standards

National Standards are followed in the country where the standard has been issued and by subcontractors or licence holders in other countries or complied with when the purchasers have so specified. National standards like BSI (Britain) [9], JIS (Japan) [10], and DIN (Germany) [11] are discussed next.

British Standards (BSI), 1993. BSI's major function is to help British industry compete effectively in world markets. Its work in standards, testing, quality assurance and export guidance is geared to enable British companies to meet the quality needs of buyers at home and abroad. BSI is independent, operating under a Royal Charter. BSI was the first national standards body in the world. There are now more than 80 similar organizations that belong to the International Organization for Standardization (ISO) and the International Electrotechnical Commission (IEC). Over 11,000 British Standard publications are listed in British Standards, 1993. Orders for publications should be directed to BSI Publications at Linford Wood, Milton Keynes, United Kingdom MK14 6LE.

Japanese Industrial Standards (JIS). JIS (Japanese Industrial Standards) are national voluntary standards for industrial and mineral products. Various industrial associations also establish voluntary standards for their specific needs. Many companies have a set of company standards like operation manuals some of them adopted from JIS and/or industrial association standards.

DIN—German Standards. The creation of German standards is the task of DIN, Deutsches Institut fur Normung e.V., a self-governing institution of trade and industry. On the basis of its statutes and of DIN 820, the standard that specifies the principles directing its activities, and by virtue of an agreement concluded with the government in 1975, DIN is the institution that is competent for standardization in the Federal Republic of Germany. As the representative of Germany, it fulfills an equivalent function in the European (CEN/CENELEC) and international standards organizations (ISO), while in the field of electrical engineering such activities are coordinated through the Deutsche Elektrotechnischen Kommission von DIN and VDE (DKE).

International Standards. ISO (International Organization for Standardization) is a worldwide federation of national standards bodies, at present comprising 92 members, one in each country. ISO coordinates the exchange of information on international and national standards, technical regulations, and other standards type documents, through an information network called ISO-NET, which links the ISO Information Centre in Geneva with similar national centers in other countries.

International Standards are followed all over the world. International Standards for quality, NDT, materials, heat exchangers, and others are framed by the ISO, Geneva. Relevant international standards for our studies include ISO 1993 [12] and ISO 9000 Series on Quality. Information on ISO 1993 is given next, whereas the ISO 9000 Quality series is covered in Chapter 14, Quality Control and Quality Assurance, Inspection, and NDT.

ISO 1993. The scope of ISO 1993 [12] covers standardization in all fields except electrical and electronic engineering standards, which are the responsibility of IEC, the International Electrotechnical Commission. Together, ISO and IEC form the specialized system for worldwide standardization—the world's largest nongovernmental system for voluntary industrial and technical collaboration at the international level. The results of ISO technical work are published in the form of International Standards. The 1993 ISO Catalogue lists 8651 published international standards. They are available as single documents, in handbook compilations for specific fields, and, in many countries, on microfilms and microfiches as well as on CD-ROM (compact disk–read-only memory).

1.2 Design Standards Used for the Mechanical Design of Heat Exchangers

Some design standards used for the mechanical design of heat exchangers include these: TEMA-USA, HEI-USA, API-USA, BS 3274-UK, and IS:4503-India.

TEMA Standards

Founded in 1939, the Tubular Exchanger Manufacturers Association, Inc., or TEMA, is a group of leading manufacturers of shell and tube heat exchangers who have pioneered the research and development of heat exchangers for over 50 years. TEMA Standards are followed in most countries of the world for design of shell and tube heat exchangers. Standards such as BS 5500 and API 660 incorporate part or all part of the TEMA Standards by reference.

Scope and General Requirements (Section B-1, RCB-1.1). The TEMA mechanical standards are applicable to unfired shell and tube heat exchangers with inside diameters not exceeding 60 in (1524 mm), a maximum product of nominal diameter (inches) and design pressure (psi) of 60,000 lb/in, or a maximum design pressure of 3,000 psi. The intent of these parameters is to limit the maximum shell wall thickness to approximately 2 in (50.8 mm) and the maximum stud diameter to approximately 3 in (76.2 mm). Criteria contained in these standards can be applied to units constructed with larger diameters. For units outside this scope, refer to TEMA Standards Section 10, Recommended Good Practice.

Scope of TEMA Standards. Table 1 shows the scope of the TEMA Standards.

Contents. The contents of TEMA Standards is given here. Each section is identified by an uppercase letter symbol, which precedes the paragraph numbers of the section and identifies the subject matter. TEMA classes R, C, and B have been combined into one section titled class RCB. The differences in design practices among the classes have been to some extent simplified. Section 5 has mechanical standards that apply to three classes of heat exchangers R, C, and B.

Table 1 Scope of TEMA Standards

Parameter	Limit
Inside diameter	60 in (1524 mm)
Nominal diameter × pressure	60,000 lb/in (10,500 N/mm)
Pressure	3000 psi (20,670 kPa)
Shell wall thickness	2 in (50.8 mm)
Stud diameters (approx.)	3 in (76.2 mm)
Construction code	ASME Section VIII, Div. 1
Pressure source	Indirect (unfired units only)

TEMA Standards Contents

Section	Symbol	Paragraph
1	N	Nomenclature
2	F	Fabrication tolerances
3	G	General fabrication and performance information
4	E	Installation, operation, and maintenance
5	RCB	Mechanical standards TEMA class RCB heat exchangers
6	V	Flow-induced vibration
7	T	Thermal relations
8	P	Physical properties of fluids
9	D	General information
10	RGP	Recommended good practice

Construction Code. According to Section RCB-1.13, the construction of heat exchangers shall comply with the ASME Boiler and Pressure Vessel Code, Section VIII, Div. 1.

Differences Among TEMA Classes R, C, and B. Differences among TEMA classes R, C, and B have been summarized by Taborek et al. [13] and are listed in Chapter 5.

TEMA Engineering Software. The Tubular Exchanger Manufacturers Association (TEMA) has made available user-friendly, state-of-the-art engineering software for the IBM PC and compatibles. This software complements the TEMA Standards, seventh edition, in the areas of

1. Flexible shell elements (expansion joints) analysis
2. Flow-induced vibration analysis
3. Fixed tube-sheet design and analysis

The programs handle the complex calculations of their respective sections of the seventh edition TEMA standards.

When Do the TEMA Standards Supplement or Override the ASME Code Specification? ASME Code provides rules for the design of the pressure boundary components like the shell, front and rear heads, flanges and covers, openings, nozzle, and reinforcements, and rules for construction, manufacturer's inspection, and hydrostatic testing. In the 1992 edition, formulas are also included for tube-sheet design, design of membrane-type expansion joints, and flanged and flued type expansion joints (procedures for determination of spring rate and stress analysis are not given). The rest of the information comes from the TEMA Standards. This includes the minimum thickness of shell and end closures, thickness of the pass partition plates and baffles, baffle spacing, tube to baffle hole clearance, drill drift, tolerance on ligaments, shell to baffle clearance, dimensional tolerances, standard clearances and tolerances applying to tube sheets, partitions, covers and flanges, impingement protection, tube-sheet design, stress induced in the shell and the tube bundle, tube-to-tube-sheet joints, design criteria for flat cover deflection, fabrication tolerances, standard tolerances on external dimensions, nozzle and support locations, nozzle extension into the shell and angularity, etc. The seventh edition of TEMA Standards includes formulas to determine the minimum thickness of the tube sheet extended as a flange and a section on flow-induced vibration guidelines. In situations where the specifications are provided both by the codes and the TEMA Standards, the latter generally override the former [14]. However the following points warrants comment here: (1) Maximum allowable stresses in the components designed according to TEMA Standards are limited to ASME Code values, and (2) tube sheets shall be designed as per TEMA only, even though separate procedures are included in the nonmandatory section of ASME Code.

Heat Exchange Institute Standards

The Heat Exchange Institute (HEI), Cleveland, Ohio, is an association of manufacturers of heat transfer equipment used in power generation. The association promotes improved designs by developing equipment design standards. It publishes standards for tubular heat exchangers used in power generation. Such exchangers include surface condensers, feedwater heaters, and other power plant heat exchangers. Among these standards are

1. Standards for Surface Condensers [4]
2. Standards for Power Plant Heat Exchangers [5]
3. Standards for Feedwater Heaters [6]

API Standard 660

API Standard 660, Shell and Tube Heat Exchangers for General Refinery Services [7], covers technical sections that exceed or supplement the TEMA Standards R class heat exchangers. This standards is a purchaser's specification intended for removable bundle floating head or U-tube construction. It does not discuss sections concerned with commercial matters. It requires exchangers to meet the requirements of TEMA Standards, Sections 1, 2, 3, 5 (Class R), and 8, and to be constructed as per ASME Code.

EJMA Standards

The Expansion Joint Manufacturers Association Inc. [8], or EJMA, is a group of leading manufacturers of bellows-type expansion joints. This association issues standards on the design of bellows-type expansion joints known as EJMA Standards. The bellows-type expansion joints are employed primarily in piping systems to absorb differential thermal expansion while containing the system under pressure. Other applications include in pressure vessels and heat exchangers. Typical service conditions range from pressures of 25 μm to 1000 psig and –420 to 1800°F (–251 to 982°C).

1.3 Codes

A code is a system of regulations or a systematic book of law often given statutory force by state or legislative bodies [15]. A code becomes a legal document in a state, a province, or a country if the government concerned passes appropriate legislation making it a legal requirement. Among the codes, the ASME Code for construction of boilers and pressure vessels including heat exchangers is the most widely used and referred to code in the world today. Apart from ASME Code, many other codes are issued by various countries. These codes are shown in Table 2. Basically the codes differ in their legal status in their own countries. Range

Table 2 International Design Codes Used for the Mechanical Design of Heat Exchangers

Code name	Country
ASME Code, Section III [21], Section VIII, Divs. 1 & 2	United States
BS 5500	United Kingdom
CODAP, SNCT [22]	France
A. D. Merkblatter	Germany
ANNC [23]	Italy
Stoomwenzen [24]	Dutch
ISO/DIS-2694 [25]	International
IS:2825-1969 [26]	India
GOST	USSR
The Pressure Vessel Code [27][a]	Japan
Regels Voor Toesellen onder Druck [28]	Netherlands

[a]Dai Isshu Atsuryoku Youki Kousou Kikahu.

of applicability varies with regards to the scope of the codes, which includes basis of design and stress analysis, design pressure and temperature, diameter, volume, materials of constructions, fabrication, inspection, etc. There is no specific code available exclusively for construction of heat exchangers in the world. Generally heat exchanger standards quote certain codes to be followed for construction of the heat exchangers. In the following paragraphs, codes like ASME Code [16,17], BS 500 [18], CODAP [19], and A. D. Merkblatter [20] are discussed.

Addresses of important codes outside the United States and Canada are furnished by Yokell [29].

ASME Codes

What Is the ASME Boiler and Pressure Vessel Code? ASME Code establishes minimum rules of safety governing the design, fabrication, inspection, and testing of boilers, pressure vessels, and nuclear power plant components. It covers new construction and rerating the existing equipment. The existence of the code stamp on a pressure vessel, with the indicated pressure and temperature, establishes the design conditions, new and old. The service conditions such as corrosion, erosion, change in operating pressure, and/or temperature may be reasons to rerate the unit, but the original stamping remains valid. Supplemental stamping is a requirement to address rerating [30].

ASME Code—Historical Background. The steady increase in boiler explosions in the 40 years from 1870 to 1910 excited public feeling to make rules and regulations for safe operation of steam boilers. In 1911, ASME formed a Boiler Committee, now called as Boiler and Pressure Vessel Code (BPVC) Committee, to devise a uniform code to protect life, limb, equipment, and property. With the publication of the ASME Code for Construction of Boiler and Pressure Vessels in 1914, serious boiler explosions steadily decreased despite the fact that the number of boilers in use has increased enormously. Primarily as a result of the ASME Boiler Code, boiler explosions and the consequent loss of life and damage to property are a rarity today [31]. To become familiar with the important aspects of ASME Codes, refer to refs. 1, 29, and 30, Nichols [32], and refs. 33a–35. Readers are advised to refer the latest codes and standards to know the state of the art. Unless otherwise mentioned, the mention of ASME Code throughout this book refers to ASME Code Section VIII, Div. 1, only.

ASME Codes. ASME Codes consist of 11 sections, and Section VIII deals with unfired pressure vessels. The various sections are:

- I Power Boilers
- II Material Specifications
  - Part A—Ferrous Materials
  - Part B—Nonferrous Materials
  - Part C—Welding Rods, Electrodes, and Filler Metals
  - Part D—Material Properties
- III Rules for Construction of Nuclear Power Plant Components
  - Subsection NCA—General Requirements for Division 1 and Division 2
  - Division 1
    - Subsection NB—Class 1 components
    - Subsection NC—Class 2 components
    - Subsection ND—Class 3 components
    - Subsection NE—Class MC components
    - Subsection NF—Component Supports
    - Subsection NG—Core Support Structures
    - Appendices
  - Division 2—Code for Concrete Reactor Vessels and Containments
- IV Heating Boilers
- V Nondestructive Examination
- VI Recommended Rules for Care and Operation of Heating Boilers limited to the operating ranges of heating boilers (Section IV)
- VII Recommended Rules for Care of Power Boilers
- VIII Pressure Vessels, Division 1
  - Pressure Vessels, Division 2 Alternative Rules
- IX Welding and Brazing Qualifications
- X Fiberglass-Reinforced Plastic Pressure Vessels
- XI Rules for In-Service Inspection of Nuclear Power Plant Components
  - Division 1
  - Division 2
  - Division 3

Addenda
Interpretations
Code Cases Boilers and Pressure Vessels

Publication of the SI (Metric) edition of the ASME Boiler and Pressure Vessel Code was discontinued with the 1986 edition.

Address trade enquiries to:

ANSI/ASME—BOILER AND PRESSURE VESSEL CODES
American National Standards Institute
11 West 42nd Street
New York, NY 10036, USA

Sections relevant for the fabrication of heat exchangers other than nuclear power plant units (i.e., unfired pressure vessels) are Section II, Section V, Section VIII, and Section IX. The ASME Code does not dictate what section of the code to use. The law or regulatory body at the point of installation determines what section to use.

Section VIII—Pressure Vessels. This section has two divisions. Division 1 gives the traditional approach for the design of pressure retaining components, and Division 2 presents alternate rules for pressure vessels which permit higher stress level.

Division 1. In terms of scope and limitations, this division covers the minimum safety requirements applicable to the construction, design, fabrication, and certification of unfired pressure vessels under either internal and/or external pressure for operation to pressures exceeding 15 psig and to vessels having an inside diameter, width, height, or cross-section diagonal exceeding 6 in. The pressure may be obtained from an external source, or by the application of heat from a direct or indirect source, or any combination thereof. This division contains both mandatory and nonmandatory appendices detailing supplementary design criteria, nondestructive examination, and inspection acceptance criteria for the fabrication of vessels. Division 1 does not prohibit the application of the U-stamp on any vessel provided the design and construction satisfies all of the code rules in the section. The code calls attention to the fact that above 3000 psi MAWP, variations in the rules and details may be necessary. Salient features of design rules of Section VIII, Divs. 1 and 2, are reviewed by Farr [36].

Except as permitted by UG-10, UG-11, or a Code Case, materials are limited to those listed in the ASME Code, Section II, and also listed in the tables of allowable stresses given in Subsection C.

Division 2 Alternative Rules. These rules provide an alternative to the minimum construction requirements for the design, fabrication, inspection and certification of pressure vessels within the scope of Div. 1. Div. 2 rules cover only vessels to be installed in a fixed location for a specific service where operation and maintenance control is retained during the useful life of the vessel by the user who prepares or causes to be prepared the design specifications. Rules in Div. 1 are not to be mixed with rules in Div. 2. In comparison to Div. 1, Div. 2 requirements on materials and nondestructive examination are stringent, and higher design stress values are permitted. The basic premise in Section VIII, Div. 2, as well as many other sections of the code, is that when more design analysis is made and when more restrictive requirements for materials, fabrication, inspection, and testing are met, the factor of safety may be lowered [36].

ASME Code Section VIII, Divs. 1 and 2—Design Criteria: A Comparison. Section VIII, Div. 1, is based on "design-by-rule" principles. It utilizes formulas, charts, tables, and graphs for sizing and determining the basic thicknesses of pressure retaining components. It is based on a factor of safety of 4 on the ultimate tensile strength (UTS). On the other hand, Div. 2 is based on a factor of safety 3 on the UTS. Division 2 is based on use of some design charts and formulas; however, the basic method is "design-by-analysis," where stresses are arrived at after detailed stress analysis and compared with allowable stresses. The higher permissible stress levels for service temperature below the creep range will permit thinner vessels in Div. 2 than Div. 1. For detailed comparison of Div. 1 and Div. 2, refer to ref. 37.

When should one consider using Div. 2 rules? Some of the conditions that favor Div. 2 rules are [29]:

1. Service is severe enough to indicate the need for intensive analysis; the costs of the necessary engineering analysis and more rigorous construction requirements are economically justified by the anticipated savings in materials and labor.
2. Pressure exceeds 3000 psi (20,670 kPa).

3. The product of pressure times shell diameter exceeds 60,000 lb/in (1.216 N/m).
4. Fatigue may affect the safety and life of the exchanger.
5. The ratio of shell diameter to thickness is less than 16, or shell thickness calculated by Div. 1 rules exceeds 3 in (76.2 mm).

Addenda. Colored sheet Addenda, which include additions and revisions to individual sections of the code, are published annually and will be sent automatically to the purchasers of the applicable sections up to the publication of the next edition/revision.

Interpretations. The ASME issues written replies to inquiries concerning interpretation of technical aspects of the Code. The interpretations for each individual section will be published separately and will be included as part of the update service to that section. They will be issued semiannually (July and December) up to the next publication of the Code. Interpretations of Sections III, Div. 1 and 2, will be included with the update service to Subsection NCA. Interpretations are not part of the Code or the Addenda.

Code Cases, Boilers and Pressure Vessels. This contains provisions that have been adopted by the Boiler and Pressure Vessel Committee that cover all sections of the Code other than Section III, Divs. 1 and 2, Section XI, to provide, when the need is urgent, rules for materials or constructions not covered by existing Code rules. Case revisions in the form of supplements will be sent automatically to purchasers up to the publication of the 1995 Code. Code Cases may be used in the construction of components to be stamped with the ASME Code symbol beginning with the date of their approval by ASME.

Technical Inquiries. When a user of the code has difficulty in understanding a part of the Code, a technical inquiry may be sent to the ASME Code Committee for an interpretation of the provisions or requirements of the Code. The letter should be addressed to the secretary to the following address:

Secretary
Boiler and Pressure Vessel Code Committee
American Society of Mechanical Engineers
345 East Forty Seventh Street
New York, NY 10017

There are several reasons why an inquiry will not be handled by the Committee, and the inquirer is so informed. The reasons for rejection include [37] (1) basis of Code rules, (2) indefinite question with no reference to Code paragraphs, figures, or rules, (3) semicommercial question with doubt as to whether question is related to Code requirements or is asking approval of design, and (4) approval of specific design.

ASME Code Symbol Stamp and Certificate of Authorization Program. To insure compliance with code requirements, ASME has administered a certificate of authorization as a part of its codes and standards program since 1915, and recently it was expanded to include all types of pressure vessels, including nuclear power plant components.

BS 5500

The main design code for pressure vessels as per British Standards is BS 5500 [18]. It covers pressure vessels manufactured from carbon, ferritic alloy, austenitic steels, and aluminum. BS 5500 replaced the earlier standards: (1) BS 1500 fusion-welded pressure vessels for general purposes, and (2) BS 1515 fusion-welded pressure vessels for use in the chemical, petroleum, and allied industries. Annexure AA Supplement to BS 5500 is the requirements for aluminum and aluminum alloys in the design and construction of unfired fusion-welded pressure vessels. BS 5500 contains rules for both "design-by-analysis" and "design-by-rule." Section 3 of BS

5500 provides design rules for specific well-known and established geometries under design pressure and temperature loading only. Salient features of design rules of BS 5500 are given in Houston [38].

BS 5500 Software Solution. ESDU International markets a software package incorporating BS 5500. For further details contact:

ESDU International
27 Corsham Street
London NI 6UA, United Kingdom

BS 3274. This is the specification for tubular heat exchangers for general purposes, design, construction, and inspection testing of the cylindrical shell and plain tube heat exchangers. Shell diameters are 6–42 in, tube lengths 6–16 ft, tube diameters $\frac{1}{2}$–1.5 in. Type 1 is the fixed tube sheet (nonremovable tube bundle); type 2 is the U-tube (removable tube bundle); and type 3 is the floating head (removable tube bundle).

French Codes and Standards on Boiler and Pressure Vessels

Several construction codes and standards are in application in France for the design and construction of pressure components. Steam generators are covered by a set of AFNOR Standards. Unfired pressure vessels are covered by the CODAP Code. Salient features of design rules of French codes and standards on boiler and pressure vessels are reviewed by Thomas et al. [39].

CODAP Code. CODAP Code is applicable to components subjected to pressure whose internal pressure exceeds 0.5 bar or whose external pressure exceeds 0.1 bar [19]. It is not applicable to piping covered by the SNCT industrial piping code, steam generators covered by the AFNOR Code, or storage tanks covered by the CODRES, among others. CODAP Code is updated each year, but there are no formal interpretation or modification request procedures. Requests are sent to the SNCT Union or to the AFIAP Society. CODAP Code rules are based on a "design-by-rule" approach for all categories of construction. The general structure of the CODAP Code is given in Table 3.

Rules for tube sheets are codified in part C7 of the CODAP Code and in NF E 32.104 standard.

German Boiler and Pressure Vessel Codes and Standards

A. D. Merkblatter is the German boiler and pressure vessel codes and standards [20]. It contains rules in the form of data sheets covering different aspects of vessel design and construction, and is produced by a group of associations. Revisions are made from time to time to keep up with technological advancements. Some aspects of vessel and exchanger design are not covered, and the method is agreed upon by the purchaser, inspecting authority, and designer. Design aspects of German boiler and pressure vessel codes and standards are discussed by Hone [40].

Table 3 General Structure of the CODAP Code

Part G	General provisions
Part M	Material
Part C	Design rules
Part F	Fabrication
Part I	Control and inspection
Part S	Protection against overpressure

2 BASICS OF MECHANICAL DESIGN

The structural integrity of pressure vessels and heat exchangers depends on proper mechanical design arrived at after detailed stress analysis keeping in view all the static, dynamic, steady, and transient loads. Heat transfer efficiency and fabrication costs of a tubular exchanger are directly influenced by proper functional and mechanical design. Therefore, an optimum mechanical design of various components of heat exchangers is of paramount importance. Mechanical design of various pressure-retaining components and some non-pressure-retaining components of heat exchangers is discussed in this section. Also discussed in this section are the fundamentals of mechanical design and stress analysis, classification of stresses and stress category concept, allowable stress, weld joint efficiency and joint category, and various design terms. Important source books on mechanical design of heat exchangers and pressure vessels are Singh and Soler [41] and Escoe [42] and on pressure vessels Moss [43], Brownell and Young [44], Bednar [45], Harvey [46], and Chuse [47], among others.

2.1 Fundamentals of Mechanical Design

Mechanical design involves the design of pressure-retaining and non-pressure-retaining components and equipments to withstand the design loads and the deterioration in service so that the equipment will function satisfactorily and reliably throughout its codal life. Mechanical design is done as per the procedure given in the construction codes and standards. Where no guidance is provided by the codes and standards, the procedure may be arrived at by mutual agreement between the purchaser and the fabricator.

Information for Mechanical Design

For mechanical design of shell and tube heat exchangers, certain minimum information is required [14]. The following listing summarizes the minimum information required:

1. Thermohydraulic design details in the form TEMA or an equivalent specification sheet.
2. TEMA class, type of TEMA shell, channels/heads.
3. Shell-side and tube-side passes.
4. Number, type, size, and layout of tubes.
5. Diameter and length of shell, channel/head, and its configuration.
6. Design temperatures and pressures.
7. External pressure if the equipment is under external pressure or is under internal vacuum.
8. Worst-case coincident conditions of temperature and pressure.
9. Nozzle, wind, and seismic loads, impact loads (including water hammer, if any).
10. Superimposed loads due insulation, piping, stacked units, etc.
11. Corrosion properties of the fluids and the environment in which the unit will be installed and the expected service life. This will help to specify corrosion allowances or better material selection to reduce the material loss due to corrosion.
12. Materials of construction except tube material, which is arrived at the thermal design stage.
13. Fouling characteristics of the streams to be handled by the exchanger. This will determine if closures are required for frequent cleaning of internal parts of the exchanger. Many fixed tube-sheet heat exchangers, if not specified otherwise, may be of welded head and shell construction.
14. Flow rate to size the nozzles and to determine whether impingement protection is required.
15. Special restrictions imposed by the purchaser on available space, piping layout, location of supports, type of material, servicing conditions, etc.

16. Construction code and standard to be followed.
17. Installation—vertical or horizontal.

In addition to these points, plant personnel should consider the following factors that influence mechanical design:

Considerations of startup, operating, shutdown, and upset conditions that decide tubesheet thickness [48].
Handling of lethal or toxic fluids, which demand more stringent welding and NDT requirements.
When high-pressure fluid is routed through the tube side, the effect of tube failure in the low-pressure shell. It is essential to provide overpressure protection on the shell side.

Sequence of Decisions to Be Made During Mechanical Design. In addition to the information required at the mechanical design stage as mentioned already, certain decisions are also to be made at the mechanical design stage. Soler [49] summarizes a typical sequence of decisions that must be made at the mechanical design stage of a heat exchanger design. Some of the points are:

1. What kind of connections (welded, flanged, or packed) should be provided at the front head, tube sheet, and rear head?
2. What style of flanged joint should be used—for example, ring type gasket or full face gasket?
3. What kind of closures (hemispherical, ellipsoidal, torispherical, conical, etc.) should be used?
4. What combination of load will govern the pressure part design? (Typical loads are shell-side pressure, tube-side pressure, differential thermal expansion, self-weight, mechanically transmitted vibration and seismic vibration, etc.)
5. Type and style of openings.
6. Type of nozzle connections, such as self-reinforcing forging stock versus pipe schedule.
7. Details of vent and drain design.
8. Minimum bend radii for U-tubes.
9. Whether an expansion joint is required. If so, what is the best type and style of the expansion joint?
10. Whether installation is horizontal or vertical, to decide the type/style of heat exchanger supports.
11. Evaluation of the ability of the exchanger to withstand operational transients, startup, and pressure testing.

Each of these decisions and evaluation steps requires a proper adjudication among various possibilities; many of these considerations require and/or are amenable to mathematical analysis, while others are derived from past experience or experimental data [49].

Content of Mechanical Design of Shell and Tube Heat Exchangers

Mechanical design of shell and tube heat exchangers involves at a minimum the following components design and the determination of stresses induced in that component:

1. Shell thickness.
2. Shell flange and channel flange design.
3. Dished end calculation.
4. Design of openings and nozzles.
5. Tube-sheet thickness. If the differential expansion between shell and tubes is excessive, then an expansion joint is to be designed and thus final tube-sheet thickness is arrived at.

6. Shell longitudinal stress and bending stress.
7. Tube longitudinal stress, both at the tube bundle inside and at the periphery.
8. Channel longitudinal stress and bending stress.
9. Tube-to-tube-sheet joint load.
10. Flat cover thickness.
11. Design of supports.

B-Jac Program Structure for Mechanical Design. One of the pioneers in developing computer software for thermal and mechanical design of shell and tube heat exchanger is M/s B-Jac International, Inc., Midlothian, VA. An insight into the logic of their mechanical design programs for the design of shell and tube exchangers is discussed next.

TEAMS—Mechanical Design Program. TEAMS is the mechanical design program of B-Jac. TEAMS covers a wide range of construction alternatives, including most types of heads, flanges, nozzles, and expansion joints. It can also be used for horizontal pressure vessels. TEAMS conforms with all provisions of the Standards of the Tubular Exchanger Manufacturers Association (TEMA), and appropriate mechanical engineering codes, including ASME and CODAP. The programs are regularly updated as revisions and addenda are issued by TEMA and Code authorities. The contents of TEAMS output are given in Table 4.

Additionally, the following features are available:

1. Cost estimate summary—Cost of materials, tubing, labor, markup, and selling price.
2. Material and labor summary—Material costs for major components (shell, front head, rear head, bundle), assembly costs, labor hours for parts and assembly, major component weights.
3. Final assembly summary—Material and labor costs for final assembly operations (e.g., x-ray, testing, inspection, painting).
4. Bill of materials—Rough (as purchased) and finished dimensions and quantity of all components, material costs and labor hours, material specification, subcontracted labor.
5. Part labor details—Labor hours required for each operation (layout, saw, shear, burn, bevel, drill, machine, mill, form, roll, weld, grind, ream, groove, chamfer) for each component.
6. Assembly labor details—Labor hours to assemble, weld, and grind for the shell components, front head components, rear head components, and bundle components.

Mechanical Design Procedure

A typical sequence of mechanical design procedures is discussed by Singh [50]. They are:

1. Identify applied loadings.
2. Determine applicable codes and standards.
3. Select materials of construction (except for tube material, which is selected during the thermal design stage).
4. Compute pressure part thickness and reinforcements.
5. Select appropriate welding details.
6. Establish that no thermohydraulic conditions are violated.
7. Design nonpressure parts.
8. Design supports.
9. Select appropriate inspection procedure.

Design Loadings

A list of loadings to be considered in designing a heat exchanger or a pressure vessel part is given in UG-22 of ASME Code Section VIII, Div. 1. They include

Table 4 TEAMS Output

Design conditions	Concise summary of design pressures, temperatures, corrosion allowances, radiography, precise weights (empty, full, bundle)
Cylinder and covers	Diameters, Code calculated thickness, TEMA minimum thickness, external pressure minimum thickness, actual thickness, maximum external pressure, maximum length for external pressure, materials of construction
Nozzles	Diameters, Code calculated thickness, actual thickness, reinforcement pad diameter and thickness, materials of construction
Flanges	Flange outer diameter, bolt circle, bolt diameter, bolt number, gasket outer diameter, gasket width, gasket thickness, Code calculated flange thickness, actual flange thickness, lap joint ring dimensions, hub dimensions, materials of construction
Tube sheets	Diameters (front and rear), TEMA bending thickness, shear thickness, flange extension thickness, effective thickness, recesses, actual thickness, clad thickness, tubing details, outer tube limit, materials of construction
Expansion joints	Number of joints, diameter, flexible element thickness, dimensions, spring rates, cycle life, materials of construction
Supports	Support dimensions, gussets, hole dimensions, wear plate thickness, Zick stress analysis, materials of construction
Maximum allowable working pressures (MAWP)	MAWP for all code components at design and ambient conditions, with controlling components flagged
Fitting locations	Elevations and distances from front tube sheet and from front head nozzle for each nozzle, coupling, support, and expansion joint, plus center of gravity
External loadings on nozzles	Local stresses in cylinders with applied loads, design conditions, coefficients, maximum loads and moments, interaction diagram
Overall dimensions	Lengths for head assemblies, tube sheets and shell, unit overall length
Calculation documentation	Formulas used and intermediate results for verification of Code and TEMA calculations
Tubesheet layout details	Number of tubes per row, distances offset from horizontal and vertical center lines for each row, tie rod locations, pass partition locations, balance of tubes per pass, baffle cut dimensions
Drawings	Setting plan (with or without bill of materials and design conditions), sectional drawing, bundle layout, tube-sheet layout
Bill of materials	Rough (as purchased) and finished dimensions and quantity of all components, specification

Source: Courtesy of M/s B-Jac International, Inc., Midlothian, VA 23112, USA.

1. Internal and/or external design pressure (as defined in UG-21).
2. Weight of vessel and normal contents including the static head of liquid.
3. Local loadings on the shell, such as those due to internals, vessel supports, lugs, etc.
4. Cyclic and dynamic reactions due to pressure or thermal variations, mechanical loadings, etc.
5. Wind, seismic, and snow loadings where required.
6. Impact loadings, such as those due to fluid shock.

Topics Covered in the Next Sections

In the next sections, the following topics are covered:

1. Stress analysis, classes, and categories of stress.
2. Calculation or design of (a) shell thickness, (b) dished end thickness, flat cover thickness, (d) flange thickness, (e) tube-sheet thickness, (f) shell longitudinal stress, (g) tube longitudinal stress, (h) tube-to-tube-sheet joint loads at the periphery of the tube bundle, (i) expansion joint, (j) nozzle openings and reinforcement of nozzle openings, and (k) supports.

2 STRESS ANALYSIS, CLASSES, AND CATEGORIES OF STRESS

2.1 Stress Analysis

Stress analysis is the determination of the relationship between external forces applied to a vessel and the corresponding stress. The stress analysis of heat exchangers and pressure vessels is similar to other structural members in that it involves mathematical operations with unknown forces and displacements. In the evaluation of the stress field in heat exchangers and pressure vessels, the problem is considerably simplified due to these reasons [41]: (1) The pressure-retaining components such as shell, heads, and cones are surfaces of revolution; (2) pressure loading—the primary mechanical loading is spatially uniform; (3) the thickness of a pressure vessel is small compared to its characteristic dimensions; and (4) with little accuracy loss, we can assume that the meridian, tangential, and through thickness directions are principal directions.

2.2 Classes and Categories of Stresses

Classes of stress, categories of stress, and allowable stresses as permitted by Codes are based on the type of loading that produced them and on the hazard they cause to the structure.

Stress Categories

The combined stresses due to a combination of loads acting simultaneously are called stress categories.

Stress Classification

The stresses that are present in pressure vessels are separated into various classes in accordance with the types of loads that produced them and the hazard they pose to the vessel. The reason for classifying stresses into various groups is that not all types of stresses require the same safety factors in protection against failure. Limit analysis theory indicates that some stresses may be permitted to a higher level than other stresses. Before discussing stress classification, membrane stress and primary stress are defined.

Membrane Stress

When the thickness is small in comparison with other dimensions ($R_m/t > 10$), vessels are referred to as membranes and the resulting stresses due to contained pressure are called membrane stresses [43]. The membrane is assumed to offer no resistance to bending. When the wall offers resistance to bending, bending stresses occur in addition to membrane stresses.

Primary Stress

Primary stress is a normal stress or a shear stress developed by the imposed loading that is necessary to satisfy the laws of equilibrium. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses that exceed the yield strength will result in plastic deformation, gross distortion, or failure. Thermal stress is not classified as a primary stress. It is classified as a secondary stress only.

Classes of stress and categories of stress are dealt in detail by refs. 41, 43, and 45, among others.

2.3 Stress Classification

In the design codes, stresses are classified into five types. They are:

1. Primary membrane stress, P_m
2. Primary bending stress, P_b
3. Local membrane stress, P_L
4. Secondary stress, Q
5. Peak stress, F

Primary Membrane Stress, P_m

The component of primary stress that is obtained by averaging the stress distribution across the thickness of the pressure vessel is referred to as the primary membrane stress. It is the most significant stress class. An important characteristic of the primary membrane stress is that beyond the yield point, redistribution of stresses in the structure does not take place. It is remote from discontinuities such as head–shell intersections, nozzles, and supports. Design codes limit its value to the allowable stress for the component material. Examples for primary membrane stresses are:

1. Circumferential (hoop) and longitudinal (meridian) stresses due to internal or external pressures
2. Stress due to vessel weight
3. Longitudinal stress due to the bending of the horizontal vessel over the supports
4. Membrane stress in the nozzle wall within the area of reinforcement due to pressure or external loads
5. Stress caused by wind and seismic forces

Primary Bending Stress, P_b

In contrast to a cylindrical shell, certain structural shapes cannot resist external loadings without bending, and the resulting stress is known as primary bending stress. Primary bending stress is capable of causing permanent distortion or collapse of the vessel. Some examples of primary bending stress are

1. Bending stress due to pressure in a flat cover
2. Bending stress in the crown of a torispherical head due to internal pressure
3. Bending stress in the ligaments of closely spaced openings, such as bending stress in the tube sheet averaged across the ligament

Primary general stresses are divided into primary membrane and primary bending stresses, and the reason for such a division so that the calculated value of a primary bending stress may be allowed to go higher than that of a primary membrane stress.

Local Membrane Stress, P_L

Local (primary) membrane stress is produced either by pressure load alone or by other mechanical loads. It has some self-limiting characteristics. Since the loads are localized, once the yield strength of the material is reached, the load is redistributed to stiffer portions of the vessel. Typical examples for local primary membrane stress are stresses at supports and stresses due to internal pressure at structural discontinuities [45].

Secondary Stress

Secondary stress is a normal or shear stress arising because of the constraint of adjacent material or by self-constraint of the structure. These stresses arise solely to satisfy compatibility conditions and are not required to satisfy laws of equilibrium. They are self-limiting in nature. Local yielding can relieve the conditions that lead to the development of these stresses and limit their maximum value. Failure from secondary stress is not to be expected. The concept of primary and secondary stresses is not relevant for brittle materials. Two sources of secondary stresses are (1) temperature and (2) gross structural discontinuity.

Secondary stresses can be subdivided into two major categories: (1) load-actuated secondary stresses and (2) temperature-actuated secondary stresses. Examples for these classes are given next.

Some examples of load actuated secondary stresses are:

1. Bending stress in a shell where it is connected to a head or to a flange
2. Bending stress in a shell or a head due to nozzle loads
3. Bending stress in the knuckle at a head to shell joint

Some examples of temperature-actuated secondary stresses are:

1. Stresses caused by axial temperature variation in a shell.
2. Both membrane and bending stresses due to differential thermal expansion between two adjoining parts of a structure such as nozzle to shell or shell to head.

Moss [43] additionally classifies secondary stresses into two groups: membrane and bending. Examples of secondary membrane stress, Q_m, are:

1. Thermal stresses
2. Membrane stress in the knuckle area of the head

Examples of secondary bending stress, Q_b, are:

1. Bending stress at a gross structural discontinuity due to relenting loads only, such as nozzles and lugs
2. The nonuniform portion of the stress distribution in a thick-walled vessel due to internal pressure

Peak Stress, F

Peak stresses are the additional stresses due to stress concentration in highly localized areas. They are caused by mechanical and thermal loads and they apply to both limiting and self-limiting loads. Peak stresses are added to the primary and secondary stresses to give the total stress at a point. A peak stress does not cause any noticeable distortion. The determination of peak stress is necessary only for fatigue analysis or a source of stress corrosion cracking, or it can be a possible source of brittle fracture [45]. Peak stress applies to membrane, bending, and shear stresses. Examples for peak stresses due to thermal and mechanical loads are given next.

Some examples of peak stresses due to thermal loads are

1. Thermal stress in the cladding or weld overlay of a tube sheet, shell, or vessel head
2. Thermal stresses in a wall caused by a sudden change in the surface temperature (thermal shock)

Some examples of load-actuated peak stresses for specific situations are:

1. Peak stress in a ligament (uniform ligament pattern)
2. Stress at a local structural discontinuity
3. Stress at corner of a discontinuity

4. Stress due to notch effect or stress concentration or small radius fillet, hole, or incomplete penetration [45]
5. Additional stresses developed at the fillet at a nozzle-to-shell junction due to internal pressure or external loads

Discontinuity Stresses

Pressure vessel components and sections usually contain regions of different thickness, material, diameter, and abrupt changes in geometry. The juncture at these locations are known as "discontinuity" areas. Examples include the skirt junction with the shell/vessel and head and shell. The stresses induced in the respective parts at or near discontinuity areas are called discontinuity stresses. Discontinuity stresses are necessary to satisfy compatibility conditions at discontinuity regions. They are not serious under static loads such as internal pressure with ductile materials if they are kept with in limits by the design, but they are important under cyclic loads [45]. The characteristics of discontinuity stresses are [43]:

1. These stresses are local in extent but can be of very high magnitude.
2. Discontinuity stresses are considered as "secondary stresses" and self-limiting, if their extent along the length is limited.
3. In average application, discontinuity stresses will not lead to failure. However, they may be a major consideration for (1) brittle materials and (2) high-pressure applications (>1500 psi).

2.2 Fatigue Analysis

When a vessel is subjected to repeated loading that could cause failure by the development of a progressive fracture, the vessel is said to be in cyclic service.

2.3 Design Methods and Design Criteria

There are two basic design methods used by the codes for the design of heat exchangers and pressure vessels; these are termed, "design-by-rule" and "design-by-analysis." The first is based on experience and does not require a detailed evaluation of all stresses. It gives formulas for sizing the majority of widely used components. The latter is based on criteria requiring detailed stress analysis.

ASME Code Section VIII Design Criteria

In general, pressure vessels conforming to the ASME Code, Section VIII, Div. 1 are designed by "design-by-rules" and do not require a detailed stress analysis. It is recognized that high localized and secondary bending stresses may exist but they are allowed for by use of a higher safety factor. However, as per Code rules, all loadings applied to a vessel or its structural attachments must be considered (UG-22). The Code establishes allowable stresses by stating in Paragraph UG-23 that the maximum primary membrane stress must not exceed the maximum allowable stress value in tension. Further, it states that the maximum primary membrane stress plus primary bending stress may not exceed 1.5 times the allowable stress of the material in tension. It also recognizes that high localized discontinuity stresses may exist, but the design rules have been written to limit such stresses to a safe level consistent with experience.

Design Criteria: Comparison of ASME Code Section VIII Div. 1 Versus Div. 2

Salient features and differences among Code rules between Div. 1 and Div. 2 are discussed in ref. 43 and the differences are given in Table 5.

Table 5 Comparison of Design Criteria of ASME Code Section VIII Divisions 1 and 2

Div. 1	Div. 2
Design is by "design-by-rules" principle and does not require a detailed evaluation of all stresses. It gives formulas for sizing the majority of widely used components.	Design is by "design-by-analysis" and requires a much more rigorous design analysis.
Does not explicitly consider the effects of combined stress and does not give detailed methods on how stresses are combined.	Provides specific guidelines on classes of stresses, stresses categories, and how they are combined.
Stress analysis considers a biaxial state of stress combined in accordance with the maximum principal stress theory.	Considers all stresses in a triaxial state combined in accordance with the maximum shear stress theory.
Does not specifically provide for design of vessels in fatigue.	Criteria for determining when a vessel must be analyzed for fatigue are specified.

2.4 Allowable Stress

Allowable stresses are used in the design of pressure vessels, heat exchangers, structures, machine elements, etc. The Code gives tables for allowable stresses in tension for most structural materials at discrete temperatures. The allowable stresses in compression depend on the slenderness of the pressure components, and are therefore presented in terms of slenderness ratio. The basis of the ASME Code allowable stress values is discussed in ref. 50.

2.5 Combined-Thickness Approach for Clad Plates

In general, the Code does not permit using the clad thickness as additional thickness to resist pressure but rather treats it only as a corrosion allowance. As an exception, for clad material conforming to SA-263, SA-264, and SA-265, the cladding thickness after deducting the corrosion allowance can be used for the thickness calculation purpose. As per Paragraph UCL-23(c), if the nominal thickness of base plate is t_b, then the allowable combined thickness, t_t, that can be used for pressure calculations is given by

$$t_t = t_b + (S_c/S_b)\, t_c$$

where S_c is the maximum allowable stress of cladding at design temperature, S_b the maximum allowable stress of the base material at design temperature, and t_c the nominal thickness of cladding less corrosion allowance. Where S_c is greater than S_b, the multiplier S_c/S_b shall be taken equal to unity.

Example. Given the base plate thickness is 1.0 in, clad plate thickness is 0.25 in, allowable stress for the base plate is 17,500 lb/in^2, and allowable stress for the clad material is 11,200 lb/in^2, what is the allowable combined thickness t_t that can be used for pressure calculations if the corrosion allowance on the cladding is 0.050 in?

$$t_t = 1.0 + (11{,}200/17{,}500)\,(0.25 - 0.05) = 1.128 \text{ in}$$

2.6 Welded Joints

The most common way to fabricate a pressure vessel is by welding. For vessel members with weld joints, all thickness formulas shall contain the weld joint efficiency term E inserted into the equation. The multiplication of the efficiency term E with the code allowable stress, S, gives the effective allowable stress, SE, for the weld seam. If no E term is contained in the formula, the allowable stress may have to be modified by a quality factor of 80% [36]. Special restrictions prevail at the weld joint for the following cases:

1. The vessel contains a lethal substance.
2. The vessel will operate at a temperature lower than −20°F.
3. The vessel is an unfired steam boiler with design pressure exceeding 50 psi.
4. The vessel is subjected to direct firing.

In these cases, all joints are restricted to butt joints and full penetration welds.

Welded Joint Efficiencies

In industry, radiographic examination (RT) is the most common technique to establish soundness of the weld joints. Depending on the type of weld joint (single or double butt, double full fillet lap, single welded butt joint without backing strip, etc.), and also on the extent of RT used to check the soundness of the joint, most of the pressure vessel codes prescribe a "joint efficiency" E to be used in the thickness formulas. The Code recognizes full radiography, spot radiography, and none. As per ASME Code Table UW-12, for a double-welding butt joint, the corresponding efficiencies would then be: fully radiographed 100%, spot radiographed 85%, and none 70%. The decrease in joint efficiency from 100% to 70% when no spot radiographic examinations are made on the welded joints means that a fabricator must provide more thickness.

Joint Categories

As per Paragraph UW-3 of ASME Code Section VIII, Div. 1, the term "category" is used to define the location of a joint in a vessel, but not the type of the joint. Categories are established for the purpose of specifying special requirements regarding joint type and degree of inspections of certain welded pressure vessels. ASME Code categorizes various joint locations into the following four types: Category A locations, Category B locations, Category C locations, and Category D locations. These locations are schematically shown in Fig. 1. Some examples for Category A, B, C, and D are given next. For complete details, refer to ASME Code Section VIII, Div. 1.

Category A Locations. Category A locations are longitudinal welded joints within a main shell, and welded joints within a sphere, within a formed or flat head, or within the side plates

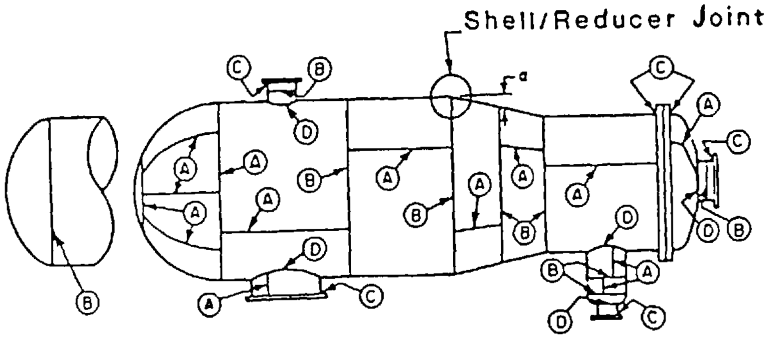


Figure 1 ASME Code joint category designation [16].

of a flat sided vessel; circumferential welded joints connecting hemispherical heads to main shells and several other locations.

Category B Locations. Category B locations are circumferential welded joints within the main shell, and circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers, and several other locations.

Category C Locations. Category C locations are welded joints connecting flanges, tube sheets, or flat heads to the main shell, to formed heads, to transitions in diameter, and to nozzles; any welded joints connecting one side plate to another side plate of a flat sided vessel, and several other locations.

Category D Locations. Category D locations are welded joints connecting communicating chambers or nozzles to main shells, to spheres, to heads or to flat sided vessels, those joints connecting nozzles to communicating chambers, and several other locations.

Weld Joint Types

The category of the weld joint determines permissible joint types, weld examination requirements, and associated weld joint efficiencies used in pressure part thickness calculation. The Code defines six weld joint types (UW-2); their definitions are given in Table 6.

2.7 Key Terms in Heat Exchanger Design

Design Pressure

Design pressure for a pressure vessel or a heat exchanger is the gage pressure at the top of the vessel, and together with the coincident design metal temperature, used in the design calculations of a pressure vessel for the purpose of determining the minimum thickness of the various pressure-retaining components of the vessel. Since a heat exchanger is made of two different pressure zones—tube side and shell side—at least two design pressures shall be defined. ASME code encourages (UG-21) that the design pressure be higher than the normal operating pressure with a suitable safe margin to allow for probable pressure surges in the vessel up to the setting of pressure relief valves (UG-134). When vessels are subjected to inside vacuum and external positive pressure on the outside, then the maximum difference between the inside and outside of the vessel shall be taken into account.

Table 6 Weld Joint Types

Type	Description
Joint type 1	Double-welded butt joint, or by other means that produce the same quality of weld on the inside and outside. Welds using metal backing strips that remain in place are excluded.
Joint type 2	Single-welded butt joints with backing strip.
Joint type 3	Single-welded butt joints with backing strip.
Joint type 4	Double full fillet lap joint.
Joint type 5	Single full fillet lap joints with plug welds.
Joint type 6	Single full fillet lap joint without plug welds.

Design Temperature

This is the temperature stamped on the nameplate along with the design pressure. This temperature shall not be less than the mean metal temperature expected across the thickness, under the operating conditions for the parts under consideration (UG-20). Design temperature can be different for the different pressure parts if the operating conditions ensure a defined temperature variation [50]. For example, in a multipass shell and tube heat exchanger in which there is an appreciable temperature drop or rise on the tube side, the inlet headers and outlet headers can have different design temperatures. In no case shall the design temperature exceed the temperature corresponding to the Code allowable stress for the material used in the thickness calculations nor exceed the allowable working temperature for the material specified in the Code.

Maximum Allowable Working Pressure (MAWP)

The maximum allowable working pressure is the gage pressure for a specified operating temperature that is permitted for the vessel in operation, such that, together with any other likely loadings other than pressure, the stresses computed using Code formulas do not exceed the Code allowable stress values. Metal thickness specified as corrosion allowance is not considered for the calculation of thickness. It is the basis for the pressure setting of the pressure-relieving devices that protect the vessel. The MAWP is normally specified for two conditions—new (uncorroded) and old (corroded).

Operating Temperature or Working Temperature

As per ASME Code, this is defined as the temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel.

Operating Pressure or Working Pressure

As per ASME Code, this is defined as the pressure at the top of the vessel at which it normally operates. It shall not exceed the maximum allowable working pressure, and it is kept at a suitable level below the setting of the pressure-relieving devices to prevent their frequent opening.

3 TUBE-SHEET DESIGN

3.1 Fundamentals

A tube sheet is an important component of a heat exchanger. It is the principal barrier between the shell-side and tube-side pressures. The cost of drilling and reaming the tube holes as well as the overall cost of the tube sheet of a given dimension will have direct bearing on the heat exchanger cost. Additionally, proper design of a tube sheet is important for safe and reliable operation of the heat exchanger. In this section fundamentals of tube-sheet design such as classification of tube sheets, constructional features, etc., are discussed.

Tube-Sheet Connection with the Shell and Channel

Tube sheets are mostly flat circular plates with uniform pattern of drilled holes. Tube sheets of surface condensers are rectangular in shape. The tube sheet is connected to the shell and the channel either by welding (integral) or bolts (gasketed joints) or a combination thereof. The possible tube-sheet connection with the shell and channel of a fixed tube-sheet exchanger can be categorized into two types:

1. Both sides integral construction (Fig. 2a)
2. Shell-side integral and the tube-side gasketed construction (Fig. 2b)

The possible tube-sheet connection with the shell and channel in the case of fixed tube sheet of floating head and U-tube exchanger can be categorized into these three types:

1. Both sides integral construction (Fig. 3a)
2. One side integral and the other side gasketed construction (Fig. 3b)
3. Both sides gasketed construction (Fig. 3c)

Supported Tube Sheet and Unsupported Tube Sheet

Heat exchanger tubes other than a U-tube heat exchanger may be considered to act as stays that support or contribute to the strength of the tube sheets in which they are attached. In the case of fixed and floating head heat exchangers, the tube bundle behaves like an elastic foundation. However, the floating nature of one of the tube sheets makes the staying action partial. This is especially true for an outside packed floating type heat exchanger. In the case of U-tube heat exchangers, tubes provide only a reactive bending moment to the tube-sheet bending. According to the level of support provided by the tubes, TEMA classifies the tube sheets as (1) supported tube sheet, and (2) unsupported tube sheet; examples are:

1. Unsupported tube sheets, e.g., U-tube tube sheets
2. Supported tube sheets, e.g., fixed tube sheets and floating tube sheets

Tube-Sheet Thickness

Being a plate type structure, the tube sheet resists the lateral pressure by bending and the membrane loads are negligible. Hence the limiting stress is the primary bending stress only. The factors that control the tube-sheet thickness are:

1. Tube pitch and layout pattern, which define the ligament efficiency of perforated tube sheets.
2. The manner in which the deformation of tube sheet is influenced by the support being provided by the tube bundle to the tube sheets. For the same process conditions and tube-

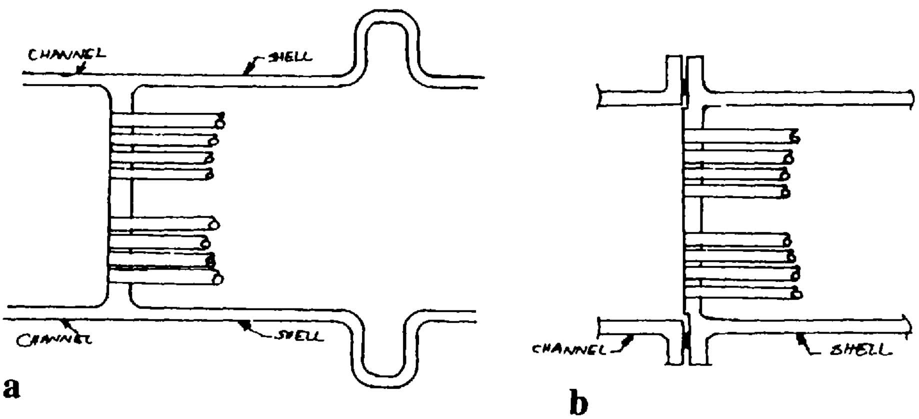


Figure 2 Fixed tubesheet exchanger connection with shell. (a) Both sides integral; and (b) shellside integral and tubeside gasketed.

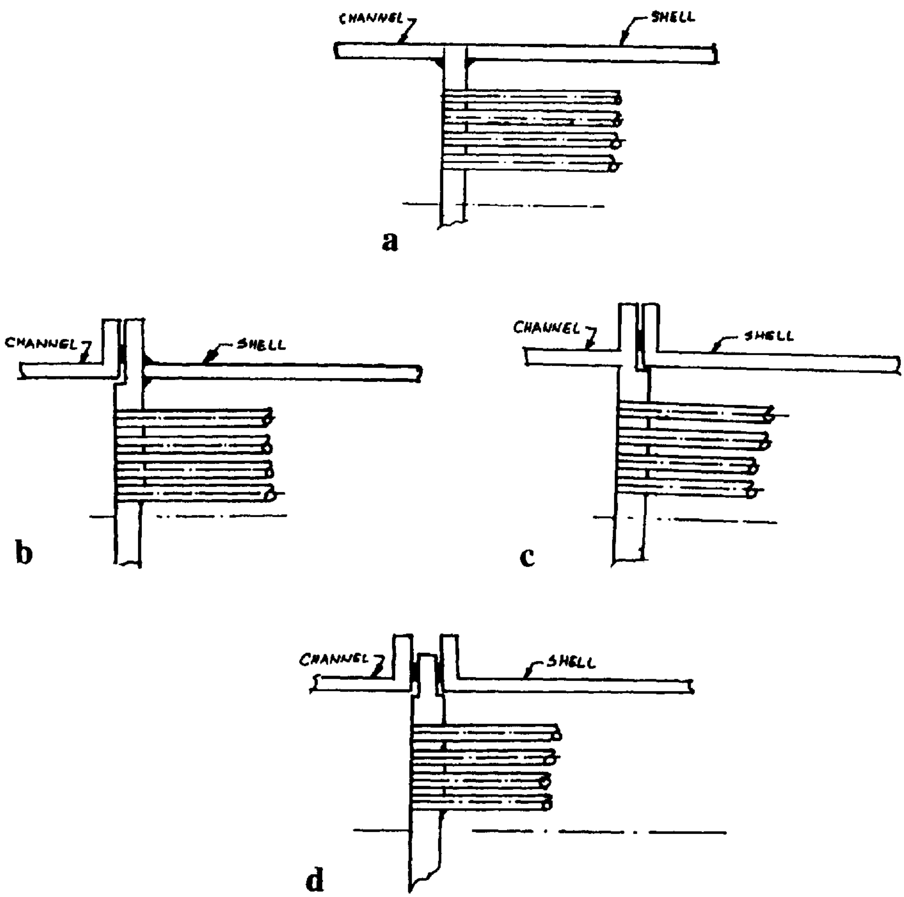


Figure 3 Floating head and U-tube exchanger tubesheet connection with shell. (a) Both sides integral; (b) and (c) one side integral and the other side gasketed; and (d) both sides gasketed.

sheet diameter, tube-sheet thickness decreases in the order of the following exchanger types:

U-tube heat exchanger
Floating head exchanger
Fixed tube-sheet exchanger.

3. Mean metal temperatures of tube sheet, tube, and shell
4. Number of tubes
5. Limits of the tube field and the extent of untubed portion
6. Method of joining the tubes to the tube sheets, e.g., rolled, seal welded, or strength welded
7. Shell and the channel connection with the tube sheets, e.g., gasketed and integral
8. Shell and the channel thickness

Tube-Sheet Design Procedure: Historical Background

TEMA set up rules for the design of U-tube and floating head heat exchangers in 1941. These rules were simple but do not ensure an overall safety for all heat exchangers. Hence many researchers [51–75] published papers on tube-sheet design and interpreted Code rules from 1948 onward.

The Codes and Standards periodically updated the procedure for tube-sheet design as and when better methods were published. In recent years, new tube-sheet design procedures have been incorporated in TEMA, British Standards 5500, Stoomwezen—Dutch code, ISO/DIS 2694 Pressure Vessels, ASME Section VIII Div. 1, CODAP, and others. BS 5500 and CODAP have adopted Gardner's method [58] as the basis for the design of tube sheets for floating head and U-tube exchangers. Historical background of tube-sheet analysis is covered in ref. 73.

Among the heat exchanger standards and codes, the tube-sheet design procedures of the Standards of the Tubular Exchanger Manufacturers Association (TEMA) have been used successfully for the last 50 years, partly due to simplicity and partly due to satisfactory performance of the heat exchangers designed as per this standards, and during this period TEMA standards have been modified several times. TEMA with its seventh edition (1988) revised its original formula by including a term for mean ligament efficiency η in the tube-sheet bending formula. The required effective tube-sheet thickness for any type of heat exchanger shall be determined for both tube-side and shell-side conditions with or without the thermal load, using whichever thickness is greater. This procedure is followed in BS 5500. However, in CODAP and ASME Code, the simultaneous action of shell-side and tube-side pressures along with thermal load are considered to arrive at the tube-sheet thickness.

Design procedure for tube sheets varies among the code rules and standards and hence designers get different thickness for the same design condition due to reasons such as:

1. Assumptions that tube sheets are either simply supported or clamped or elastically restrained at its edges.
2. Local stresses developed at the shell/tube-sheet and channel/tube-sheet junction are neither calculated nor required to be limited.
3. Failure to consider the effect of untubed annular rim.

With this background knowledge, tube-sheet design procedure is explained next. First, the assumptions made in various tube-sheet design models are discussed, followed by the basis of fixed tube-sheet design procedure. Subsequently, the tube-sheet design procedure for fixed, floating head, and U-tube-sheet procedure included in TEMA, CODAP, BS 5500, and ASME Code Section VIII Div. 1 is dealt with. Finally, the fixed tube-sheet bending formula of TEMA is compared with that of CODAP and BS 5500. While discussing tube-sheet design procedure, emphasis is on TEMA tube-sheet design procedure. Design aspects of double tube sheets, rectangular tube sheets, and curved tube sheets are covered at the end.

Assumptions in Tube-Sheet Analysis

While analyzing the tube sheets, certain assumptions are made in their models by many researchers. The tube sheets are treated as thin plates compared to their radial dimension, both circumferential and radial stresses vary linearly through the thickness of the tube sheets, and shear stresses vary parabolically from zero at one face to zero at the other face with a maximum at the center. Other assumptions include the following:

1. The tube sheet is uniformly perforated over its whole area; the unperforated annular rim is not considered by some standards; For example, TEMA Standards do not consider the unperforated tube-sheet portion for all classifications of tube sheets; CODAP and BS 5500 neglect it for fixed tube-sheet design only. ASME Code Section VIII, Div. 1, considers the unperforated rim for all types of construction.
2. The membrane loads in the tube sheets are negligible as compared to the bending loads [53].

3. No slip occurs at the junction between the tubes and the tube sheet.
4. The tubes are adequately stayed by baffle plates to enable them to stand up to the calculated loads without sagging.
5. The bending moments in the tubes at their attachment with the tube sheet are neglected.
6. The exchanger is axis symmetrical and symmetric about the plane midway between the tube sheets.
7. Modeling of the tube bundle: The tubes are assumed uniformly distributed over the whole tube sheet and in sufficient number (N_t) so as to act as a uniform elastic foundation of modulus K_w. The expression for K_w is

$$K_w = \frac{N_t K_t}{\pi R^2} \tag{1}$$

where K_t represents the axial rigidity of one tube as given by

$$K_t = \frac{\pi E_t t(d - t)}{L} \tag{2}$$

Note: The elastic modulus for a half bundle, k_w, is equal to $2K_w$, and the axial rigidity of one half tube, k_t, is equal to $2K_t$.
8. Modeling of the tube sheet: The perforated tube sheet is replaced by an equivalent solid plate of effective elastic constants E^* and ν^* (the determination of effective elastic constants is discussed separately). The flexural rigidity of the perforated plate D^*, in terms of the flexural rigidity of unperforated plate D and deflection efficiency η is given by

$$\eta = D^*/D \tag{3}$$

where D^* and D are given by

$$D = \frac{ET^3}{12(1 - \nu^2)} \quad \text{and} \quad D^* = \frac{E^*T^3}{12(1 - \nu^{*2})} \tag{4}$$

One of the drawbacks of the work of Gardner [52] and Miller [53] is the assumption that the Poisson ratio of the perforated tube sheet is same as for the unperforated tube sheet; accordingly, a constant value of $\nu^* = 0.3$ was assumed in their treatment.
9. The maximum stress in the perforated plate will be the maximum stress in the homogeneous plate divided by the ligament efficiency, μ.
10. The analysis is based on the optimum design of tube sheets within their elastic behavior of all components attached to the tube sheet. If the temperatures are high enough, creep becomes of primary importance [56].
11. The deflection of the tube sheet is small, and hence the angular distortion of the tube ends due to the bending of the tube sheet can be neglected.
12. The effect of rotational resistance of the tubes is negligible since it is minor in nature.

Boundary Restraint

Tube sheets are weakened due to drilling holes, whereas they are stiffened by the tube bundle and tube-sheet edge restraint offered by the shell and channel connected with the tube sheet by welding. Based on the tube-sheet connection with the shell and the channel, the edge restraint condition is treated as simply supported, clamped, and an intermediate case. However, the complication of the combined effects of discontinuity stresses due to shell-side and tube-side pressures and differential thermal expansion between the shell and the tube bundle, and the tube sheet and the channel head, makes the determination of boundary restraint an uncertain factor in the estimation of the tube-sheet stresses [52]. In view of these factors, Gardner [52] suggests that the designer be guided by judgment and experience in determining the relative fixity of the tube-sheet periphery as between the simply supported and the clamped. According to Miller [53], the boundary restraints may be treated as (1) a simply supported case, (2) a clamped case, and (3) the intermediate to these two cases. The examples suggested by Miller for these cases are:

1. Simply supported case: a narrow joint face, or trapped ring gasket
2. Clamped case, e.g., full-face gasketed joint

Treatment of Boundary Restraint in TEMA Standards. In the seventh edition of TEMA [3], the weakening effect of the tube hole drilling is taken into account by including the mean ligament efficiency term η in the tube-sheet bending formula. However, the tube-sheet edge restraint offered by the shell and channel connection with the tube sheet is not taken care of adequately. As in the earlier editions, the F factor used to account for the simply supported condition, fixed (clamped) condition, and intermediate condition is retained. Due to this reason, the TEMA formula is not safe for all sizes and all operating pressures [73]. This is especially true for large units with high operating pressures. For certain values of the parameter X (used to represent the relative rigidity of the tube bundle with respect to the tube sheet) the TEMA formula is safe, but for lower values it is not adequate and hence unsafe. The TEMA fixed tube-sheet formula is compared with CODAP by Osweiller [70,73].

Effective Elastic Constants of Perforated Plates

While designing the perforated tube sheets, the weakening effect due to the tube hole perforations has been taken into account by replacing the plate by an equivalent solid plate with new elastic constants known as the effective Young's modulus, E^*, and effective Poisson's ratio, ν^*. The values of E^* and ν^* are such that the equivalent plate has the same deflection as that of the original unperforated plate. This is known as the equivalent solid plate concept. The equivalent solid plate concept has been found to be quite useful in the design and analysis of perforated plates by equating strains in the equivalent solid material to the average strains in the perforated material [76]. These effective elastic constants must be evaluated correctly, especially in fixed tube-sheet heat exchangers [77]. If they are too low, the stresses at the junction with the shell and the head will be lower than in real units. If they are too high, the stress at the center of the plate, which may be a maximum, will be too low.

Determination of Effective Elastic Constants. The effective elastic constants depend on the pattern, size, and pitch of the perforations. During the last two decades many researchers have proposed theoretical and experimental methods to determine the effective elastic constants. However, there are disparities in the values obtained by these methods. Modern pressure vessel codes such as ASME , ISO, BS 5500, CODAP, Stoomweazen, etc. present curves to determine effective elastic constants. TEMA does not determine the effective elastic constants. It assumes a constant value of 0.178 for deflection efficiency. An excellent review of about 60 papers on the elastic constants was done by Osweiller [77]. Osweiller proposed curves for determination of effective elastic constants that have been adopted in CODAP.

Mean Ligament Efficiency

The ligament efficiency is a very useful dimensionless parameter for analysis of perforated plates. The ligament efficiency, defined in terms of the tube layout pattern and pitch ratio in TEMA, is known as mean ligament efficiency, η, and in terms of pitch ratio is known as minimum ligament efficiency in codes such as CODAP (Fig. 4) and BS 5000 and ligament efficiency in ASME Code Section VIII, Div. 1. The general expression for ligament efficiency is:

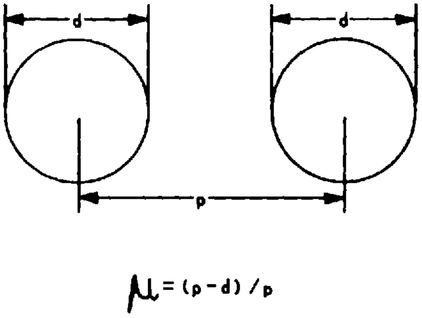


Figure 4 Mean ligament efficiency.

$$\mu = \frac{p - d}{p} \tag{5}$$

where d is the tube outer diameter and p the tube pitch.

A specific expression for ligament efficiency as defined in TEMA, CODAP, and BS 5500 is given while discussing the tube-sheet design as per these standards/codes.

3.2 Basis of Fixed Tube-Sheet Design

Thin Circular Plate on Elastic Foundation

Most tube-sheet design analysis treats the tube sheet as a thin circular plate on an elastic foundation. The elastic foundation is provided by the tube bundle. The basis of tube-sheet design procedure is discussed here. This discussion closely follows the method of Galletly [56], which was further expanded by Osweiller [70,73] for inclusion in CODAP. The heat exchanger is assumed to be of revolution and symmetrical about a plane midway between the tube sheets, so as to analyze a half exchanger as shown in Fig. 5a. Figure 5b shows a circular plate of thickness T resting on an elastic foundation. To minimize the complexity, the untubed tube-sheet portion is neglected and the tube sheet is integral with both the shell and the channel. The tube sheet is disconnected from the remainder of the exchanger, that is, from shell and channel. The plate is elastically restrained against deflection and rotation around its periphery, θ_e by (1) an axial reaction V_E due to the end load acting on the head and to the axial displacement Δ_s of the half shell, and (2) a reactive bending Moment M_E $(-K_\theta\theta_e)$ as shown schematically in Fig. 5c. The plate is subjected to a uniform net effective pressure $q(r)$ given by [73]:

$$q(r) = p_s f_s - p_t f_t - \nu_t \left[\frac{N_t}{R^2} \frac{(p_t - p_s)(d - t)^2}{2} \right] + 2\nu_s p_s Q - k_w \left[w(r) - \frac{\gamma}{2} + \Delta_s \right] \tag{6}$$

In the expression for $q(r)$, the first term takes into account the differential pressure acting on the equivalent plate, which is corrected for the tube hole areas by the shell-side drilling coefficient, f_s, and tube-side drilling coefficient, f_t, respectively; the second and third terms take into account the loads resulting from the axial displacements of tubes and shell by the Poisson effect of shell-side pressure, p_s, and tube-side pressure, p_t, respectively (Q is the ratio of rigidity of tube bundle to the shell); and the fourth term traduces the reactive effect of the elastic foundation. In this, the term $w(r)$ is the deflection of the plate at a distance r from the center axis, and $\gamma/2$ is the differential thermal expansion between the tubes and the shell, which is given for the half exchanger by:

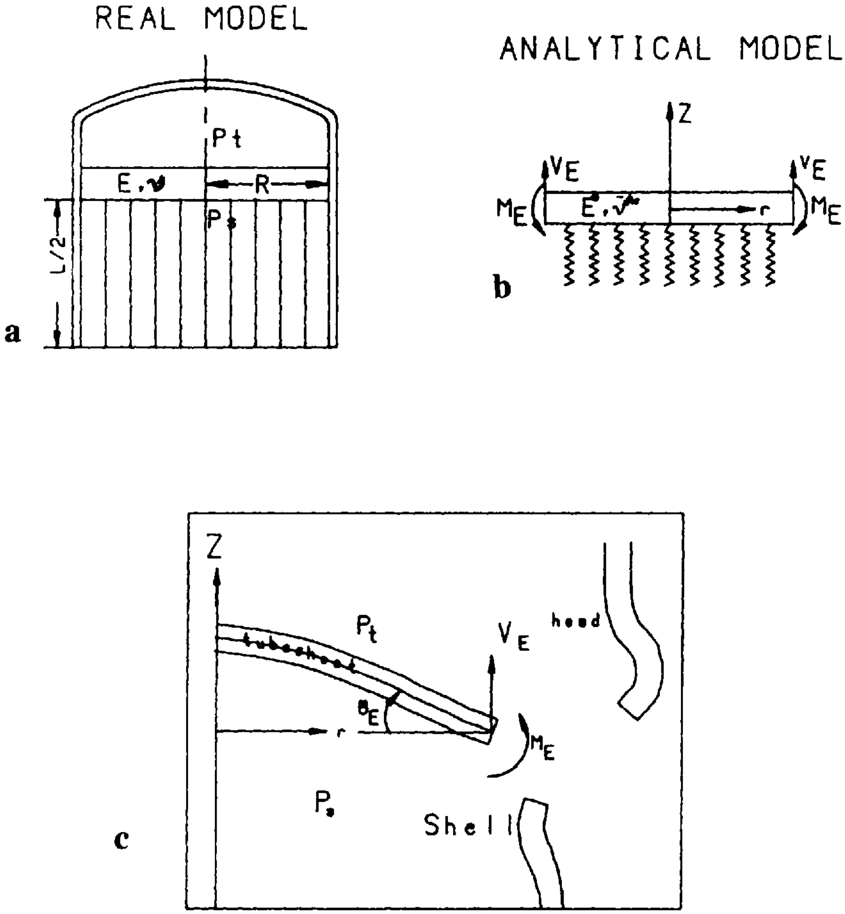


Figure 5 Basis of fixed tube sheet heat exchanger, (a) Half heat exchanger, (b) circular plate on an elastic foundation, and (c) Force and moment at the tubesheet edge. (From Ref. 73.)

$$\frac{\gamma}{2} = [\alpha_t(\theta_t - \theta_{amb}) - \alpha_s(\theta_s - \theta_{amb})]\,\frac{L}{2} \tag{7}$$

The expressions for f_s, f_t, and Q are:

$$f_s = 1 - \frac{N_t d^2}{4R^2} \tag{8}$$

$$f_t = 1 - \frac{N_t(d - 2t)}{4R^2} \tag{9}$$

$$Q = \frac{N_t K_t}{K_s} \tag{10}$$

$$= \frac{1}{K} \tag{11}$$

where K is the ratio of axial rigidity of the shell (K_s) to the axial rigidity of the tube bundle ($N_t K_t$). The parameter K signifies the ratio of the force required to produce a given strain in the shell to the force necessary to produce the same strain in the tube bundle. It is thus the measure of the ability of the shell to resist movement of the two tube sheets relative to the tube bundle [51]. However, when there is a considerable thermal expansion between the shell and the tube bundle, the high rigidity of the shell may induce very high thermal stresses in the tube bundle and the shell. This ability is reduced by the introduction of an expansion joint into the shell. An externally packed floating head exchanger for purposes of tube-sheet design is considered as a perfect expansion joint, and for exchangers so constructed the value of K is zero. The expression for the axial rigidity of the shell K_s is given by:

$$K_s = \frac{\pi t_s (D_o - t_s)\ E_s}{L} \tag{12}$$

From the expression for K_t(Eq. 2) and K_s(Eq. 12), the resulting expression for K is given by

$$K = \frac{E_s t_s (D_o - t_s)}{E_t N_t t(d - t)} \tag{13}$$

where D_o is the shell outside diameter. The expression for axial rigidity of the half shell, k_s, is equal to $2K_s$, and axial rigidity of the half tube, k_t, is equal to $2K_t$.

Deflection, Slope, and Bending Moment

From classical thin-plate theory, the deflection of a solid circular plate of elastic constants E^*, ν^*, and flexural rigidity D^* resting on elastic foundation k_w, subjected to net effective pressure $q(r)$ and elastically restrained at its periphery, is given by:

$$\frac{d^4 w}{dr^4} + \frac{2}{r}\frac{d^3 w}{dr^3} - \frac{1}{r^2}\frac{d^2 w}{dr^2} + \frac{1}{r^3}\frac{dw}{dr} = \frac{q(r)}{D^*} \tag{14}$$

The solution of this equation is of the form

$$w(r) = A\text{Ber}(x) + B\ \text{Bei}(x) + \frac{p^*}{k_w} - \Delta_s + \frac{\gamma}{2} \tag{15}$$

Deflection, slope and bending moment

$$p^* = p_s f_s - p_t f_t - \nu_t \left[\frac{N_t}{R^2} \frac{(p_t - p_s)\ (d - t)^2}{2} \right] + 2\nu_s p_s Q \tag{16}$$

$$x = kr = \sqrt[4]{\frac{k_w}{D^*}}\ r \tag{17}$$

In Eq. 15 Ber(x) and Bei(x) are the modified Bessel functions of the first kind, and A and B are unknown constants.

At the periphery of the tube sheet (i.e., $r = R$), x becomes X. It represents the relative rigidity of the tube bundle with respect to the tube sheet. It may vary from 0 (no tube in the bundle) to above 50 for very stiff tube bundle. The expression for X is:

$$X = k\text{R} = \sqrt[4]{\frac{k_w}{D^*}}\ R \tag{18}$$

$$= \sqrt[4]{\frac{\pi N_t E_t t(d - t)}{\pi R^2 L/2} \frac{12(1 - \nu^{*2})\ R^4}{E^* T^3}} \tag{19}$$

Substituting for $N_t E_t (d - t)$ by $E_s t_s (D_o - t_s)/K$ and for $E^*/(1 - \nu^{*2})$ by $\eta E/(1 - \nu^2)$ and $D_o - t_p \approx 2R$, X becomes

$$X = \sqrt[4]{\frac{24 E_s t_s (D_o - t_s)}{KL} \frac{R^2(1 - \nu^2)}{\eta E}} \tag{20a}$$

$$= \sqrt[4]{\frac{6(1 - \nu^2)}{\eta} \frac{E_s t_s}{KLE} \left(\frac{2R}{T}\right)^3} \tag{20b}$$

From $w(r)$, one may determine the shear forces, the bending moment, and the slope at any point in the tube sheet. By employing the Kirchoffs–Kelvin equation, the expression for slope, dw/dr, and bending moment, M, are given by

$$\theta = \frac{dw}{dr} \tag{21}$$

$$M = D^*\left(\frac{d^2w}{dr^2} + \frac{\nu^*}{r}\frac{dw}{dr}\right) \tag{22}$$

The two constants of integration A and B and the axial displacement of the half shell, Δs, are determined by the three boundary conditions:

1. At $r = R$, the deflection $w(r)$ at the edge of the plate due to bending is zero.
2. At $r = R$, the radial bending moment at the edge of the plate equals the moment exerted by the rotational spring, i.e.,

$$D^*\left(\frac{d^2w}{dr^2} + \frac{\nu^*}{r}\frac{dw}{dr}\right)\Big|_{r=R} = -K_\theta \frac{dw}{dr}\Big|_{r=R} \tag{23}$$

   where K_θ is the spring constant for half shell. Its value is the sum of the bending rigidities of the shell and the channel.
3. The vertical force at the edge, V_E, is given by the net effective force acting on the plate in terms of net effective pressure $q(r)$:

$$2\pi \int_0^R q(r)\, dr = 2\pi R V_E \tag{24a}$$

   where

$$V_E = \frac{\pi R^2 p_t - k_s \Delta_s}{2\pi R} \tag{24b}$$

3.3 Tube-Sheet Design as Per TEMA Standards

Tube-Sheet Formula for Bending

As per RCB-7.131, the formula for minimum tube-sheet thickness to resist bending is given by

$$T = \frac{FG}{3}\sqrt{\frac{P}{\eta S}} \tag{25}$$

where F is the parameter used to account for the elastic restraint at the edge of the tube sheet due to shell and channel connection, G the diameter over which the pressure is acting, P the effective design pressure, S the ASME Code allowable stress, and η the mean ligament efficiency, given in terms of tube layout pattern angle θ and pitch ratio p/d whose expression is given by

$$\eta = 1 - \frac{\pi}{4(\sin\theta)(p/d)^2}$$

$$= 1 - \frac{0.785}{(p/d)^2} \quad \text{for } \theta = 90^\circ \text{ and } 45^\circ$$

$$= 1 - \frac{0.907}{(p/d)^2} \quad \text{for } \theta = 60^\circ \text{ and } 30^\circ \tag{26}$$

The values of F, P, and G differ for supported and unsupported tube sheets. For a fixed tube-sheet exchanger, G shall be the shell inside diameter. For other types of exchangers refer to TEMA for the definition of G. For a fixed tube-sheet exchanger, the effective design pressure, P, shall be calculated as per Sections RCB-7.163, RCB-7.164, and RCB-7.165. The definition of F for supported and unsupported tube sheets is discussed next. For all types of tube sheets, the thickness shall be calculated both for uncorroded and corroded conditions.

Parameter F

Supported Tube Sheet.

1. Gasketed both sides, e.g., stationary tube sheet and floating tube sheet and floating head exchanger:

   $F = 1.0$

2. Integral on both sides or a single side, e.g., stationary tube sheet of fixed tubesheet exchanger and floating head exchanger.

When the tube sheet is integral with both sides or a single side (for fixed tube-sheet exchangers the gasketed side refers to the tube side), F is determined by curve H in Fig. RCB-7.132. The curve is presented in terms of the ratio of wall thickness to internal diameter (ID) of the shell or channel, i.e., (t/ID), whichever yields the smaller value of F. For the shell-side integral condition, use the shell ID to find F. The H curve can be represented by:

$$F = 1.0 \quad \text{for } \frac{t}{\text{ID}} \le 0.02$$

$$= \frac{17 - 100\dfrac{t}{\text{ID}}}{12} \quad \text{for } 0.05 \ge \frac{t}{\text{ID}} > 0.02$$

$$= 0.8 \quad \text{for } \frac{t}{\text{ID}} > 0.05 \tag{27a}$$

As per this condition, the minimum value of F is 0.8 and maximum value is 1.0.

Both the tube sheets of the fixed tube-sheet exchanger are satisfied by this condition, and both the tube sheets shall have the same thickness, unless the provisions of RCB-7.166 are satisfied.

Note: The F value for the tube sheet at the floating head for all configurations is 1.0.

Unsupported Tube Sheet, For Example, U-Tube Sheet.

1. When gasketed at both sides, $F = 1.25$.
2. For tube sheets integral on both sides or a single side, F shall be determined by curve U in Fig. RCB-7.132. The curve is presented in terms of the ratio of wall thickness to internal diameter (ID) of the integral side, i.e., (t/ID). The U curve can be represented by:

$$F = 1.25 \qquad \text{for } \frac{t}{\text{ID}} \le 0.02$$

$$= \frac{17 - 100\frac{t}{\text{ID}}}{15} \qquad \text{for } 0.05 \ge \frac{T}{\text{ID}} > 0.02$$

$$= 1.0 \qquad \text{for } \frac{t}{\text{ID}} > 0.05 \tag{27b}$$

As per this condition, the maximum value of F is 1.25 and the minimum value is 1.0.

The term P is the effective design pressure where $P = p_s + p_b$ or $p_t + p_b$; p_b is defined as equivalent bolting pressure when the tubesheet is extended as a flange. The expression for p_b is given by

$$p_b = \frac{-6.2M^*}{F^2G^3} \tag{28}$$

where M^* is defined in Paragraph RCB-7.1342. Expression for M^* is furnished in the section on 3.6 flanged tubesheet—TEMA design procedure.

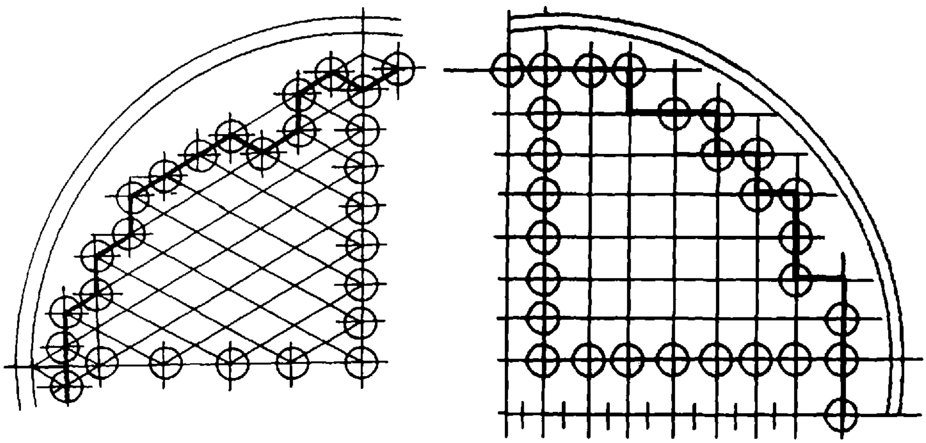
Shear Formula RCB-7.133

The effective tube-sheet thickness T to resist shear is given by

$$T = \frac{0.31 D_e}{(1 - d/p)} \frac{P}{S} \tag{29}$$

where D_e is the equivalent diameter of the perforated tubesheet ($4A_p/C$), C is the perimeter of the tube layout measured stepwise in increments of one tube pitch from center to center of the outermost tubes (Fig. 6 shows the application to typical triangular and square tube patterns; only a portion is shown), and A_p is the total area enclosed by the perimeter C.

The shear stress formula was derived by limiting the maximum allowable shear stress to



(a) (b)

Figure 6 Perimeter of tube layout for shear stress calculation. (a) Triangular layout; and (b) square layout.

0.8 times the Code allowable stress S. The shear formula controls the tube-sheet thickness only in high-pressure and small-diameter cases. Since the quantities C and A_p are available after the tube layout is finalized, TEMA provides a formula to check whether shear stress will be controlling the tube-sheet thickness or not. Shear formula will not control the tube-sheet thickness if

$$\frac{P}{S} < 1.6\left(1 - \frac{d}{p}\right)^2 \tag{30}$$

Stress Category Concept in TEMA Formula

The primary stress in the tube sheet is limited to ASME Code Section VIII, Div. 1, allowable stress S, and the primary stress plus the thermal stress (secondary stress) is limited to $2S$. Accordingly, the expression for effective pressure is halved before being used in the thickness formula.

Determination of Effective Design Pressure, P (RCB-7.16)

The determination of (1) effective shell-side design pressure involves the terms p_s', equivalent differential expansion pressure p_d, and equivalent bolting pressure (p_{bs} and/or p_{bt}), and (2) effective tube-side design pressure involves p_t', p_d, and p_{bs} and/or p_{bt}. The expressions for p_d, equivalent bolting pressure (p_{bs}, p_{bt}), p_s', and p_t' are given next.

Equivalent Differential Expansion Pressure, p_d (*RCB 7.161*). The equivalent differential expansion pressure, p_d, is given by [70]:

$$p_d = \frac{4Jt_sE_s[\alpha_s(\theta_s - \theta_{amb}) - \alpha_t(\theta_t - \theta_{amb})](D_o - t_s)/(D_o - 2t_s)^2}{1 + JKF_q} \tag{31}$$

where D_o is the shell outer diameter, F_q a parameter that is a function of X (G_d of Miller [53] and H_d of Galletly [56]),

$$\theta_t = \theta_{t,m} - \theta_{amb}$$
$$\theta_s = \theta_{s,m} - \theta_{amb}$$

and where J is the expansion joint flexibility parameter (=1.0 for shells without expansion joint).

The expressions for J and F_q and the final expression for p_d follow.

1. The expansion joint flexibility parameter J is given by

$$J = \frac{1}{1 + \dfrac{K_s}{S_j}} \tag{32}$$

Substituting the expression for K_s, J is given by

$$J = \frac{S_jL}{S_jL + \pi(D_o - t_s)\, E_st_s} \tag{33}$$

J can be assumed equal to zero for a shell with expansion joints if the following condition is satisfied:

$$S_j < \frac{(D_o - t_s)t_sE_s}{10L} \tag{34}$$

where S_j is the spring rate of the expansion joint.

2. From Osweiller [70], the expression for F_q in terms of X is given by

$$F_q = 0.25 + \alpha X \tag{35}$$

where α is a function of the tube-sheet correction factor F. By substituting the expression for α, the resulting expression for F_q is given by

$$\alpha = (F - 0.6)\frac{\sqrt{2}}{0.8} \tag{36}$$

$$\therefore\ F_q = 0.25 + (F - 0.6)\frac{\sqrt{2}}{0.8}\left[\frac{6(1-\nu^2)}{\eta}\frac{E_s t_s}{KLE}\left(\frac{G}{T}\right)^3\right]^{0.25} \tag{37}$$

By assuming that the tube-sheet deflection efficiency η is 0.178 and ν is 0.3, we get an expression for F_q similar to the TEMA form:

$$F_q = 0.25 + (F - 0.6)\sqrt[4]{\frac{300\ E_s t_s}{KLE}\left(\frac{G}{T}\right)^3} \tag{38}$$

where G is the shell inside diameter ($2R$, where R is the shell inside radius). Note: use the calculated value of F_q or 1.0 whichever is greater. By approximating

$$\frac{D_o - t_s}{(D_o - 2t_s)^2} \approx \frac{1}{(D_o - 3t_s)} \tag{39}$$

and replacing $\alpha_s(\theta_s - \theta_{amb})$ and $\alpha_t(\theta_t - \theta_{amb})$ by $\Delta L/L$, the expression for p_d is given by

$$p_d = \frac{4JE_s t_s\left(\dfrac{\Delta L}{L}\right)}{(D_o - 3t_s)(1 + JKF_q)} \tag{40}$$

where ΔL is the differential thermal expansion between the shell and the tube bundle.

Equivalent Bolting Pressure (RCB 7-162). When the tube sheet is extended as a flange for bolting to head or channel with a ring type gasket, the effect of the flange moment acting upon the tube sheet is accounted for in effective design pressure in terms of equivalent tube-side bolting pressure, p_{bt}, and equivalent shell-side bolting pressure, p_{bs}, as given by:

$$p_{bt} = \frac{6.2M_1}{F^2G^3} \tag{41a}$$

$$p_{bs} = \frac{6.2M_2}{F^2G^3} \tag{41b}$$

where M_1 is the bolting moment acting under the operating condition, as defined by the ASME Code as M_o under flange design; M_2 is the bolting moment acting under the gasket seating condition, as defined by the ASME Code as M_o under flange design; and G is the inside diameter of the shell.

Expression for p_s'. The expression for p_s' is given by [70]:

$$p_s' = \frac{p_s\left[JK[2\nu_t + f_s(1 - 2\nu_t)] + 2\nu_s J - \left(\dfrac{1-J}{2}\right)\left(\dfrac{D_J^2}{G^2} - 1\right)\right]}{1 + JKF_q} \tag{42}$$

By substituting $\nu_s = \nu_t = 0.3$, and rearranging the terms, p_s' is given in TEMA form:

$$p_s' = \frac{p_s\left[0.4J[1.5 + K(1.5 + f_s)] - \left(\frac{1 - J}{2}\right)\left(\frac{D_J^2}{G^2} - 1\right)\right]}{1 + JKF_q} \quad \text{for } \nu_s = \nu_t = 0.3 \tag{43}$$

Expression for p_t'. The expression for p_t' is given by [70]:

$$p_t' = \frac{p_t[1 + JK\{f_t + 2_{\nu_t}(1 - f_t)\}]}{1 + JKF_q} \tag{44}$$

$$= \frac{p_t[1 + 0.4\ JK(1.5 + f_t)]}{1 + JKF_q} \quad (\text{for } \nu_t = 0.3) \tag{45}$$

where D_J is the expansion joint diameter, G the inside diameter of the shell, p_{bt} the equivalent bolting pressure when tube-side pressure is acting, p_s the shell-side design pressure, and p_t the tube-side design pressure.

Differential Pressure Design, after Yokell [29]

In differential pressure design, the pressure parts exposed to both fluids are designed for the difference in operating pressures between the two fluids, i.e., simultaneous action of both the shell-side and tube-side pressures. Differential pressure design is seldom used when the simultaneous design pressures are less than 1000 psi (6895 kPa). The operating circumstances that favor differential pressure design are:

1. A common source of supply of the shell-side and tube-side fluids
2. Shell-side and tube-side operating pressures in the range of 1000 psi (6895 kPa) and above, when the pressure in one side is directly controlled by the other
3. High pressure on both sides in which the differential pressure can be measured and controlled

Merits of Differential Pressure Design. Differential pressure design reduces the tube-sheet thickness below what would be required for conventional design. However, this design involves certain safety and protection devices to be incorporated into the system, in the event of failure on one side [29]. To justify this design, the reduction in capital cost due to thin tube sheet and the associated machining cost savings must exceed any operating cost associated with differential pressure design and safety devices to be provided. TEMA and BS 5500 recognize differential pressure design when it is mutually agreed between the purchaser and the seller.

Effective Differential Design Pressure, P (*RCB 7.165*). For differential pressure design, the effective differential design pressure is calculated as per RCB 7.165. For practical use TEMA considers the seven combined design loading cases where pressure acts on the shell side (p_t = 0), on the tube side (p_s = 0), on both sides, and in each case with and without thermal expansion; the seventh case is obtained for thermal expansion acting alone (p_s = 0 and p_t = 0).

Longitudinal Stress Induced in the Shell and Tube Bundle

After arriving at the tube-sheet thickness, it is necessary to determine the stresses induced in the shell, channel, and tubes located at the periphery and interior of the tube bundle in the case of fixed and floating tubesheet exchangers. The check for longitudinal stress induced in the shell is calculated as per RCB-7.22. However, there is no procedure in TEMA to calculate the bending stresses induced in the shell and both the longitudinal stress and bending stress in the channel. The check for tube longitudinal stress both in tension and in compression in the tubes located at the periphery of fixed tube-sheet exchangers are calculated as per RCB-7.23 and RCB-7.24, respectively. This is discussed next.

Shell Longitudinal Stress, $\sigma_{s,l}$ (RCB-7.22). The maximum longitudinal stress induced in the shell, $\sigma_{s,l}$ is given by

$$\sigma_{s,l} = \frac{p_s^* \, (D_o - t_s) \, Z_s}{4t_s} \tag{46}$$

Seven loading cases are examined, which lead to seven values of p_s^*. The value of parameter Z_s depends on the stress category concept. Its value is either 0.5 or 1.0; p_s^* is determined as per TEMA definition.

Longitudinal Stress Induced in the Tubes Located at the Periphery of the Tube Bundle, $\sigma_{t,l}$ (RCB-7.23). The maximum longitudinal stress induced on the tubes located at the periphery of tube bundle is given by

$$\sigma_{t,l} = \frac{F_q G^2 \, p_t^* Z_t}{4N_t t(d - t)} \tag{47}$$

As for the shell longitudinal stress case, seven loading cases are examined, which lead to seven values of p_t^*. The value of parameter Z_t depends on the stress category concept. Its value is either 0.5 or 1.0; p_t^* is determined as per TEMA definition.

Longitudinal Stresses Inside the Tube Bundle. Although the tubes located at the interior of the bundle can become loaded both in tension and compression, longitudinal stresses inside the tube bundle are not calculated in TEMA. Tensile forces are generally not a problem if the requirements of RCB-7.22 are met. However, compressive forces might create unstable conditions for tubes at the interior of the bundle. Typical conditions that can cause this are loading and geometry, as follows:

Loading: Tube-side pressure and/or differential thermal expansion where the shell, if unrestrained, would lengthen more than the tubes (positive P_d per RCB-7.161).

Geometry: Flexible tube-sheet systems. Generally, those that are simply supported at the edge ($F = 1$) and have a value of F_q greater than 2.5.

Methods similar to those provided in the references cited in TEMA, and others such as in refs. 56, 67, 70, and 73 can be used to predict loadings on tubes at the interior of the bundle.

Compressive Stress Induced in the Tubes Located at the Periphery of the Tube Bundle (RCB 7-24). The compressive stress acting on the tubes located at the periphery is limited to the allowable compressive stress S_c based on the Euler critical buckling load for a column as given by

$$\sigma_c = \frac{\pi^2 E_t}{F_s \left(\dfrac{kl}{r_G} \right)^2} \qquad \text{when} \quad \Lambda \le \frac{kl}{r_G} \tag{48}$$

$$= \frac{S_y}{F_s} \left[1 - \frac{kl/r_G}{2\Lambda} \right] \qquad \text{when} \quad \Lambda > \frac{kl}{r_G} \tag{49}$$

where

$$\Lambda = \sqrt{\frac{2\pi^2 E_t}{S_y}} \tag{50}$$

r_G is the radius of gyration of the tubes,

$$r_G = 0.25\sqrt{d^2 + (d - 2t)^2} \tag{51}$$

and kl the equivalent unsupported buckling length of the tube, k a factor that takes into account the tube span end conditions, l the unsupported tube span between two baffles, and F_s the factor of safety

Tube-to-Tube-Sheet Joint Loads (RCB-7.25)

The maximum effective tube-to-tube-sheet joint load, F_j, acting on the tubes located at the periphery of the tube bundle is given by

$$F_j = \frac{\pi F_q p_T^* G^2}{4N_t} \tag{52}$$

where p_T^* is determined as per TEMA definition. This joint load is to be less than the maximum allowable joint load calculated as per ASME Code Section VIII, Div. 1.

Maximum Allowable Joint Loads. In the design of shell and tube heat exchangers other than U-tube construction, the maximum allowable axial load on tube-to-tube-sheet joints shall be determined in accordance with the code formula. The basis for establishing allowable loads for tube-to-tube-sheet joints loads is given in ASME Code Appendix AA. In ASME Code, various joints types are identified by a, b, c, d, e, f, g, h, i, j, and k. The maximum allowable joint load F_{max} is calculated as follows:

1. For joint types a, b, c, d, e:

$$F_{max} = A_t S_a f_r \tag{53}$$

2. For joint types f, g, h, i, j, k:

$$F_{max} = A_t S_a f_e f_r f_y \tag{54}$$

where A_t is the nominal transverse cross-sectional area of tube wall, S_a the Code allowable stresses in tension of tube material at design temperature, f_e a factor for the percentage of tube expansion length, $f_e = 1.0$ for joints made with expanded tubes in grooved tube holes, d the tube outer diameter, f_r the factor for efficiency of joint, and f_y the ratio of tube-sheet material yield stress to tube material yield stress.

Refer to ASME Code Section VIII, Div. 1, for complete details including joint types on the nomenclature.

TEMA Fixed Tube-Sheet Design with Different Thickness

Fixed tube sheets of different thickness in the same unit are designed as per RCB-7.166. Considerable savings may be realized when a process dictates a high-quality material at only one end of a unit where the temperature is very high.

Tubesheet Design as per ASME Code Section VIII, Div. 1

A "design-by-formula" procedure for the design of fixed tube sheets of shell and tube heat exchangers and the connected components is included in the ASME Code, Section VIII, Div. 1, Appendix AA—Nonmandatory. The method follows the approach of Singh and Soler [41] and extended by Soler et al. [67] to present a unified treatment for all fixed tube-sheet styles.

The procedure is based on classical thin plate and shell theory. It includes the effect of unperforated rim of the tube sheet, differential thermal expansion in the radial direction, arbitrary combinations of material properties and arbitrary joint configurations—three different tubesheet, shell, and head configurations, namely, (1) two sides gasketed type construction, (2) two sides integral type construction, (3) one side gasketed and the other side integral type construction—and heat exchanger styles, and the elastic–plastic method, which takes into account the discontinuity stresses at the tube sheet to shell/channel joint and considers the possibility of plastic hinges at the shell–tube sheet and channel joints. As a part of the calculations procedure, charts are provided to evaluate certain parameters. Kuppan [75] provides alternate charts in place of those presented in ASME CODE, numerical values for certain functions used to derive the charts that will help to do design calculations without referring the charts, and extensions of the existing charts for larger values of tubesheet parameter X.

CODAP

The maximum stress, σ, in the tube sheet due to bending is given by [73]:

$$\sigma = \frac{1}{\mu H_1(X,Z)} \left(\frac{R}{T} \right)^2 p^* \tag{55}$$

Inverting this equation, the expression for tube-sheet thickness to resist bending is given by

$$T = \frac{R}{\sqrt{\mu H_1(X,Z)}} \sqrt{\frac{p^*}{\sigma}} \tag{56}$$

where S_f is the CODAP allowable stress, $H_1(X,Z)$ the tube-sheet coefficient depending on parameters X and Z (its expression is given in the Appendix), p^* the equivalent differential pressure, R the outer tube-sheet radius where shell-side pressure is acting, T the tube-sheet thickness, μ the minimum ligament efficiency, and X and Z the parameters of the heat exchanger (an expression for X was given earlier, and Z is given in the Appendix).

Parameters X, Z, *Ligament Efficiency*, μ, *and* p*. The parameter X represents the relative rigidity of the tube bundle with respect to the tube sheet. It may vary from 0 (no tube in the bundle) to above 50 (very stiff tube bundle). The parameter Z represents the elastic rotational restraint at the periphery of the tube sheet due to channel and shell connection. It may vary from 0 for the simply supported case to infinity for the clamped case. The expression for minimum ligament efficiency μ is given by

$$\mu = \frac{p - d}{p} \tag{57}$$

$$= \frac{p - (d - t)}{d} \tag{58}$$

where d is the tube outer diameter, p the tube pitch, and t the tube wall thickness.

Equation 57a is applicable to tubes welded to the tubesheet, and Eq. 58b is applicable to tubes expanded more than 90% of tube-sheet thickness.

Equivalent Design Pressure, p*. The expression for p^* for simultaneous action of shell-side pressure, tube-side pressure and differential thermal expansion (assuming there is an expansion joint on the shell side) is given by [73]:

$$p^* = \frac{p_s}{1 + JKH_4(X,Z)} \tag{59}$$

$$p_x = p_s\left\{[JK\{f_s + 2\nu_t(1 - f_s)\} + 2\nu_s J] - \left[\frac{D_J^2 - G^2}{G^2}\right]\frac{JK_s}{4S_J}\right\}$$

$$- p_t[JK\{f_t + 2\nu_t(1 - f_t)\} + 1] + \frac{JK\gamma k_w}{2} \tag{60a}$$

$$= p_s' - p_t' + p_d \tag{60b}$$

where p_s', p_t', and p_d are the same as in Eq. 42, 44, and 40, respectively.

Stress Category Concept in CODAP Formula. The stress category concept used in CODAP is discussed by Osweiller [70]. The primary membrane stress plus bending stress in the tube sheet is limited to $1.5f$ (f = nominal or allowable design stress in CODAP). When the tube sheet is not extended as a flange, the bending stress due to pressures has been taken up to $2f$. When the tube-sheet rim is extended as a flange, the additional moment at the periphery of the tube sheet is taken into account by limiting the allowable stress to $1.5f$.

Fixed Tube-Sheet Formula as per BS 5500

The fixed tube-sheet design formula of BS 5500 [18] bears certain commonalities with the fixed tube-sheet design procedure of CODAP and TEMA. Similar to CODAP, the fixed tube-sheet formula in BS 5500 takes into account the tube-sheet restraint offered by shell and channel to the tube sheet through the term $H_1(X,Z)$. However, the effective design pressure is calculated like the TEMA procedure. The minimum tube-sheet thickness shall be greater of the values calculated to resist bending and shear given by the following formulas.

Bending:

$$T = \max\left|\frac{G_1}{\sqrt{4H_1(X,Z)}}\sqrt{\frac{p_1}{\Omega\mu S_b}}, \frac{G_2}{\sqrt{4H_1(X,Z)}}\sqrt{\frac{p_2}{\Omega\mu S_b}}\right| \tag{61}$$

Shear:

$$T = \max\left|\frac{0.155 D_{otl} p_1}{\lambda\tau}, \frac{0.155 D_{otl} p_2}{\lambda\tau}\right| \tag{62}$$

Simultaneous action of both the shell-side and tube-side pressure: If agreed, the design shall be based on the simultaneous action of both the shell-side and tube-side pressure (TEMA specifies this condition as effective differential pressure design; RCB-7.165)

Bending:

$$T = \max\left|\frac{G_1}{\sqrt{4H_1(X,Z)}}\sqrt{\frac{p_d}{\Omega\mu S_b}}, \frac{G_2}{\sqrt{4H_1(X,Z)}}\sqrt{\frac{p_d}{\Omega\mu S_b}}\right| \tag{63}$$

Shear:

$$T = \frac{0.155 D_{otl} p_d}{\lambda\tau} \tag{64}$$

where G_1 is the diameter to which shell-side pressure is acting, G_2 the diameter to which tube-side pressure is acting, D_{otl} the diameter of the outer tube limit circle, S_b the allowable stress in BS 5500, p_1 the shell-side effective design pressure, p_2 the tube-side effective design pressure, p_d the effective differential design pressure, Ω the design stress factor (its value is 1.5 for fixed tube sheet and 2 for floating head tube sheet), λ the ligament efficiency of the tube sheet in shear, and μ the ligament efficiency of the tube sheet in bending and allowable shear stress for tubesheet material. Note: The definition of p_1, p_2, and p_d are the same as the effective shell-side design pressure calculated as per TEMA para RCB-7.163, the effective tube-side design pressure calculated as per TEMA para RCB-7.164 and the effective differential design pressure calculated as per TEMA para RCB-7.165, respectively.

The expression for μ and λ are

$$\mu = \lambda = \frac{p - d_h}{P} \tag{65}$$

$$= \frac{p - (d_h - t)}{p} \tag{66}$$

where d_h is the tubehole diameter. Equation 65 applies to:

1. Tubes not expanded into the full depth of the tube sheet
2. Tubes seal-welded to the tube sheet
3. Tubes having significantly lower elastic modulus than the tube-sheet material

Equation 66 applies to:

1. Tubes fully expanded into the tube sheet
2. Explosion-bonded tubes

The determinations of effective design pressure p_1 (Section 4.9.3.4.1), p_2 (Section 3.9.4.3.2), and p_d (Section 4.9.3.4.3) are similar to TEMA procedures RCB-7.163, RCB-7.164, and RCB-7.165, respectively.

Stress Category Concept in BS 5500 Formula. The stress category concept in BS 5500 is similar to that of TEMA but with an additional design stress factor. This factor allows for the fact that the stress calculated using these requirements is the average bending stress across the ligament at the surface of the tube sheet, and the permissible value is higher than the nominal allowable stress f (in ASME Code the allowable stress is designated by S) by a factor Ω, whose value is 1.5 for fixed tube-sheet exchanger and 2 for floating tube-sheet exchanger.

Determination of Joint Load. Determination of the stress induced in the shell and the tube is similar to the TEMA procedure. The determination of compressive stress in the tube due to buckling differs slightly from the TEMA procedure. The determination of the tube joint load is different in BS 5500. It specifies six typical joints types. The joint load formula involves the determination of parameters such as tube joint reliability factor, expansion factor, and material factor.

Comparison of Fixed Tube-Sheet Thickness Formula of TEMA with CODAP and BS 5500

For comparison purposes, assume that (1) the ASME Code allowable stress $S = S_b$ of BS 5500 and (2) for a fixed tube sheet to resist bending, $\sigma = 2S$. The resulting equations for tube-sheet thickness per TEMA, CODAP, and BS 5500 can be written as

$$T_{TEMA} = \frac{FG}{3\sqrt{\eta}}\sqrt{\frac{p}{S}} = \frac{F}{1.5\sqrt{\eta}}\frac{G}{2}\sqrt{\frac{p}{S}} \tag{67}$$

$$T_{CODAP} = \frac{1}{\sqrt{\mu H_1(X, Z)}}\frac{G}{2}\sqrt{\frac{p^*}{2S}}\left(\because R = \frac{G}{2}, \sigma = 2S\right) \tag{68}$$

$$T_{BS5500} = \frac{G}{2\sqrt{H_1(X,Z)}}\sqrt{\frac{P}{\Omega\mu S}} \text{ (for } p_1 = p_2 = p_d = p,\ G_1 = G_2 = G,\ S_b = S) \tag{69}$$

It is seen from these equations that TEMA simplified its formula, replacing $1/\sqrt{H_1(X,Z)}$ by F, with an intention to have a simple hand calculations procedure for the tube-sheet design.

Under the assumptions $p = p^* = p'$ and that the tube sheet is not extended as a flange, and separating the common terms, the F factor in the TEMA, CODAP, and BS 5500 formats can be written as

$$F_{\mathrm{TEMA}} = \frac{F}{1.5\sqrt{\eta}} \tag{70a}$$

$$F_{\mathrm{CODAP}} = \frac{1}{\sqrt{2\mu H_1(X,Z)}} \tag{70b}$$

$$F_{\mathrm{BS5500}} = \frac{1}{\sqrt{1.5\mu H_1(X,Z)}} \tag{70c}$$

The F values of TEMA are compared with CODAP by Osweiller [70,73]. As per the comparison, TEMA formula is not safe for the entire range of X values. It is unsafe for lower values and conservative for higher values of X.

Appendix

The expression for Z [56] and $H_1(X,Z)$ and $H_4(X,Z)$ taken from ref. 70 are given by:

$$Z = \frac{K_\theta}{k\ D^*} = \frac{K_\theta}{\sqrt[4]{k_w\ D^{*3}}} \tag{71}$$

where K_θ is the edge moment coefficient. It depends on the bending rigidities of the shell (δ_s) and the channel (δ_c). It represents the degree of elastic restraint of the tube sheet offered to it by the shell and the channel connected to it. Its value varies between the two extreme values: 0, which corresponds to the simply supported case; and ∞, which corresponds to the clamped case.

$$k_\theta = 2[\delta_s + \delta_c]$$

$$K_\theta = 2\left[\frac{E_s t_s^{2.5}}{[12(1-\nu^2)]^{0.75}(D_s + t_s)^{0.5}} + \frac{E_c t_c^{2.5}}{[12(1-\nu^2)]^{0.75}(D_c + t_c)^{0.5}}\right] \tag{72}$$

$$H_1(X,Z) = \frac{kR}{3\Psi(kr)}\left[bei'(kR)\,\frac{g_1(kR)}{g_2(kR)} - ber'(kR)\right] \tag{73}$$

$$H_4(X,Z) = \frac{kR}{2}\,\frac{\left[ber(kR)\,\dfrac{g_1(kR)}{g_2(kR)} + bei(kR)\right]}{\left[bei'(kR)\,\dfrac{g_1(kR)}{g_2(kR)} - ber'(kR)\right]} \tag{74}$$

where

$$\Psi(kr) = f_1(kr) - \frac{g_1(kR)}{g_2(kR)}\,f_2(kr) \tag{75}$$

The expression for $f_1(kr)$ and $f_2(kr)$, $g_1(kr)$, and $g_2(kr)$ are given by

$$f_1(kr) = ber(kr) - \frac{(1 - \nu^*)bei'(kr)}{kr} \quad (76)$$

$$f_2(kr) = bei(kr) + \frac{(1 - \nu^*)ber'(kr)}{kr} \quad (77)$$

$$g_1(kR) = f_1(kR) + Zbei'(kR) \quad (78)$$

$$g_2(kR) = f_2(kR) - Zber'(kR) \quad (79)$$

3.4 Rules for Tube Sheet of Floating Head Exchangers

BS 5500 and CODAP

The floating head heat exchanger tube-sheet design is comparatively simpler than for a fixed tube-sheet exchanger due to the reason that in the floating head exchanger only one tube sheet is fixed to the shell or channel or both sides, either by welding or by bolting. The second tube sheet is free to move inside the shell. The method of Gardner [58] for the design of floating head exchangers is adopted in BS 5500 and CODAP, and this code procedure is discussed by Osweiller [73]. The tube-sheet thickness to resist bending is given by

$$T = (C_o + \Delta C_o)D_p \sqrt{\frac{|p_s - p_t|}{2\mu S_f}} \quad (80)$$

where C_o is a coefficient that depends on two parameters X_o and R_D (these two parameters are defined later), D_p is the diameter of the perforated region, and f is the nominal design stress.

In CODAP, the tube-sheet thickness to resist bending is expressed in terms of a stress-oriented formula as [73]:

$$\sigma = \frac{(C_o + \Delta C_o)^2(p_s - p_t)}{\mu}\left(\frac{D_p}{T}\right)^2 \quad \text{(where } \sigma \text{ is limited to } 2S) \quad (81)$$

Parameters X_o *and* R_D.

1. X_o is similar to the coefficient X related to the fixed tube sheets with modifications so as to present a direct formula for calculating T.

$$X_o = \frac{\left[1.5\,(1 - \nu^2)(f_t - f_s)\,\frac{E_t}{E^*}\,\frac{D_p}{L}\right]^{1/3}}{\sqrt{\frac{|p_s - p_t|}{2\mu S_f}}} \quad (82)$$

2. $R_D = 2R/D_p$, which accounts for the untubed rim at the periphery of the tubesheet.

Another parameter relevant to the discussion is the mean solidity factor of the tube sheet, f_m, given by

$$f_m = \frac{f_s + f_t}{2} \quad (83)$$

The values of coefficient C_o are given in Fig. 7 for a mean solidity factor of 0.6. The coefficient ΔC_o gives the correction for other values of the solidity factor. It can be seen from Fig. 7 that values for C_o are given only for two limiting cases of tube-sheet edge conditions: (1) simply supported ($Z = 0$), and (2) clamped tube sheet ($Z = \infty$). No credit is given for the other edge

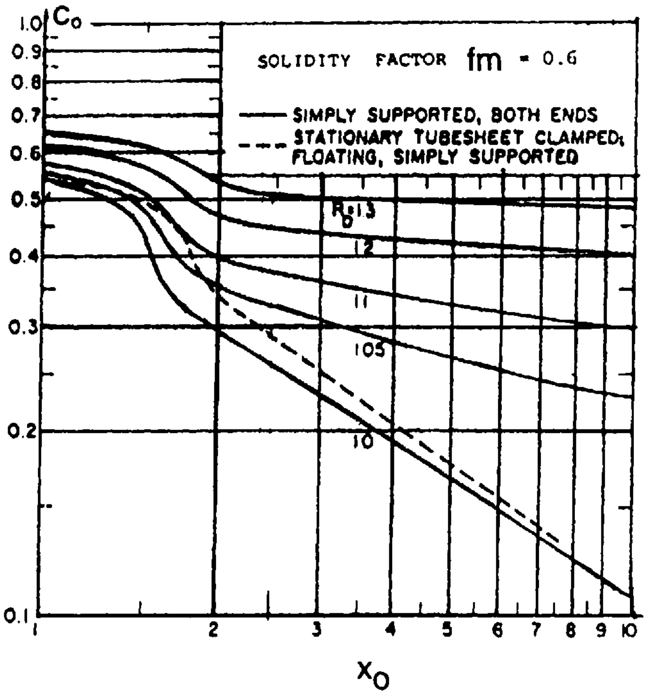


Figure 7 Coefficient C_0 for mean solidity factor $f_m = 0.6$. (From Ref. 58.)

conditions. Therefore for other cases the designer must judge whether the tubesheet is simply supported, clamped or halfway between the extreme cases as defined next [73].

BS 5500.

1. Simply supported condition: tubesheet (either fixed or floating) gasketed at both sides using narrow-faced gaskets.
2. Clamped condition: full-face gaskets on both sides, or full-face gasket on a single side and the tube sheet integral on the other side.

CODAP.

1. Simply supported when bolted on both sides.
2. Halfway between simply supported and clamped in the following cases: gasketed on one side and integral on the other side, or integral on both sides.

Accordingly, the value for C_o for the intermediate case is the average of the simply supported case $C_{o,s}$ and the clamped case $C_{o,c}$ as given by

$$\therefore \quad C_o = \frac{C_{o,s} + C_{o,c}}{2} \tag{84}$$

Maximum Axial Stress in Tubes. The maximum axial stress in the tubes located both at the periphery $\sigma_{t,p}$ of the tube bundle and in interior $\sigma_{t,i}$ is given by [73]:

$$\sigma_{t,p} = \frac{(p_s f_s - p_t f_t) - F_{tp}(p_s - p_t)}{f_t - f_s} \tag{85}$$

$$\sigma_{t,i} = \frac{(p_s f_s - p_t f_t) + F_{ti}(p_s - p_t)}{f_t - f_s} \tag{86}$$

The coefficients F_{tp} and F_{ti} are analogous to coefficient C_o and are given by curves [18,58] that depend on parameters X_o and R_D.

3.5 U-Tube Tube-Sheet Formula

BS 5500

For U-tube exchangers, the design method is simpler since the tubes do not act as an elastic foundation. Hence, the CODAP formula for the floating head exchanger can be used by setting $X_o = 0$ in the tube-sheet formula. This is done in codes/standards such as ISO, BS 5500, and CODAP. The tube-sheet thickness formula to resist bending is given by

$$T = (C_o + \Delta C_o)D_p \sqrt{\frac{|p_s - p_t|}{2\mu S_f}} \tag{87}$$

where $C_o + \Delta C_o$ is given directly as a function of mean solidity factor f_m and ratio $2R/D_p$. The values for C_o and ΔC_o are given in Table 7.

CODAP

The CODAP formula for U-tube exchangers is same as for BS 5500.

ASME Code Section VIII, Div. 1

ASME Code Section VIII, Div. 1, Appendix AA, provides rules for the design of U-tube sheets of (1) simply supported, (2) both sides integral, and (3) integral construction with tube sheet extended as flange.

3.6 Flanged Tube Sheets—TEMA Design Procedure

Formulas are included in the TEMA seventh edition for the calculation of the minimum thickness required on the flanged portion of the tube sheet of fixed, floating, and U-tube exchangers. The calculation procedure is based on the work of Singh et al. [74]. The purpose of this new method is to insure that there is sufficient tubesheet thickness to withstand bending moments transmitted by the adjoining flange.

Table 7 Values for Coefficients C_o and ΔC_o [58]

	Clamped		Simply supported			
				ΔC_o		
R	C_o	ΔC_o	C_o	$f_m = 0.45$	$f_m = 0.60$	$f_m = 0.80$
1.00	0.433	0	0.560	0.000	0	0.000
1.05	0.433	0	0.576	−0.002	0	0.002
1.10	0.433	0	0.592	−0.010	0	0.010
1.20	0.433	0	0.625	−0.025	0	0.025
1.30	0.433	0	0.660	−0.040	0	0.040

Fixed Tube Sheet or Floating Tube Sheet

The thickness of the portion of the tube sheet extended as a flange, t_F, is given by [74]:

$$T_f = 0.98\left[\frac{M\{r_f^2 - 1 + 3.72\ \ln(r_f)\}}{S(D_T - G)\ (1.0 + 1.86\ r_f^2)}\right]^{0.5} \tag{88}$$

where M is the bolting moment due either to gasket seating condition or operating condition, whichever is greater; if a joint is integral (welded connection), then the corresponding edge moment is zero; D_T and G are the tubesheet outer diameter and effective gasket diameter as defined in TEMA RCB-7.132, respectively; and S is the Code allowable stress at the design temperature. The quantity r_f is the ratio of D_T to G:

$$r_f = D_T/G \tag{89}$$

U-Tube Tube Sheet

The minimum thickness of the portion of the tube sheet of thickness T extended as a flange, T_f, is given by [87]:

$$T_f = 1.38\left[\frac{M^* + M + 0.39\ PG^2\ W}{(D_T - G)\ S}\right]^{0.5} \tag{90}$$

where

$$M^* = \frac{\left(\dfrac{0.069w}{\eta}\right)F^3PG^3\left(\dfrac{T_f}{T}\right)^3 - MG - 0.39wPG^3}{G + \left(\dfrac{1.37}{\eta}\right)\left(\dfrac{T_f}{T}\right)^3 w} \tag{91}$$

where M is the bolting moment either due to gasket seating condition or operating condition whichever is greater (if a joint is integral, such as welded connection, then the corresponding edge moment is zero); G has different values for different styles of construction (for example, G is the inside diameter of the pressure part for the pressure acting on an integral side of a tube sheet; it is the location of the gasket load reaction as defined in the code for the pressure acting on the gasketed side of a tube sheet, TEMA RCB-7.132); $P = p_s$ or p_t or maximum differential pressure or as defined in TEMA; and $w = 0.5\ (D_T - G)$.

For each pressure loading, the calculation procedure is as follows:

1. Compute M^* from Eq. 91, assuming $T_f = T$ (rim and interior tube-sheet thickness equal).
2. Assume $P = p_s$ or p_t or maximum differential pressure, as applicable.
3. Compute the required tube-sheet rim thickness T_f from Eq. 90 If $T > T_f$ and both are set equal to the computed T, then the calculation may be terminated at this point.
4. If it is desired to reduce the rim thickness below T, the value of T_f arrived at in step 3 is used to calculate a new M^* in Eq. 91. Next, the calculation returns to step 2.

3.7 Rectangular Tube-Sheet Design

The rectangular tube sheet of a surface condenser is designed as per HEI Standards [4]. The basis of the rectangular tube-sheet design as per HEI standards is discussed by Bernstein et al. [78].

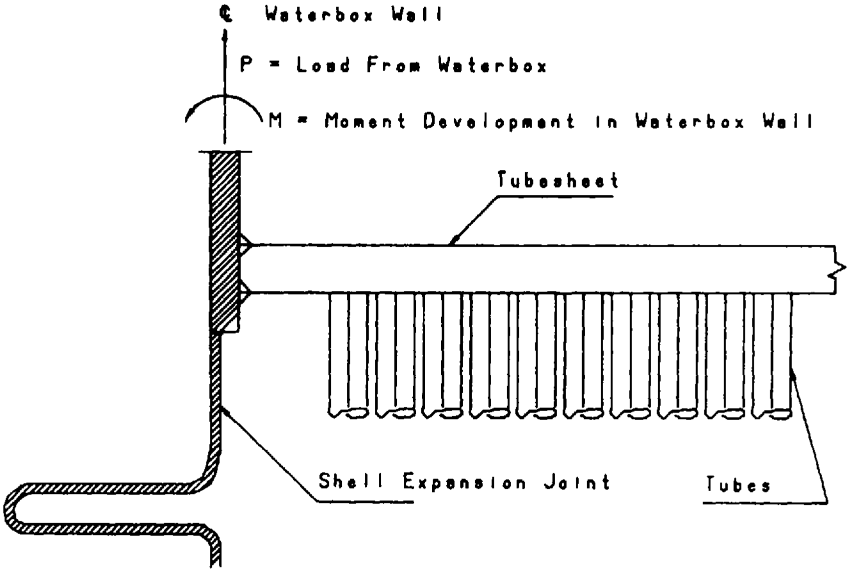


Figure 8 Surface condenser tubesheet connection with waterbox and the shell. (From Ref. 78.)

Methods of Tube-Sheet Analysis

Figure 8 shows the plan section of a surface condenser waterbox and shell end assembly with an expansion joint. Figure 9 shows an idealized representation of the tube sheet and the tube bundle. A condenser tube sheet is seen to be a partially perforated plate on an elastic foundation, with the tubes comprising the foundation. The hydrostatic pull from the water box and surface pressure on the tube sheets are the dominant loads. Irregular tube patterns and variations in edge boundary conditions make solutions of the plate problem difficult. The HEI Standards mention the following four methods by which the structural integrity of the tubesheet and tubes may be demonstrated [78]:

1. Interaction analysis using plate and shell formulas
2. Experimental modeling techniques, or prior services
3. Finite-element analysis (elastic or elastic–plastic)

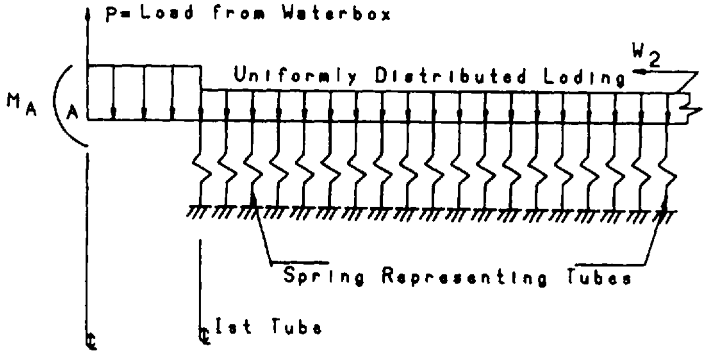


Figure 9 Rectangular tubesheet modelling. (From Ref. 78.)

4. Beam strip on elastic foundation (single or multiple strips)

Of these four methods, the basis of the beam strip method is discussed next.

Beam Strip Method. In the beam strip method, the problem of an elastically restrained perforated rectangular plate on an irregular elastic foundation is replaced by several more readily solved beams on an elastic foundation as shown in Fig. 10. The beams represent narrow strips of tube sheets supported by tubes, loaded by hydrostatic pressure and waterbox pull. The loading on each beam strip must be estimated, taking into account forces and moments acting on the waterbox and the type of end fixity.

The standard explains the steps in the beam strip method, with an example. Briefly, these steps cover the choice of the beam strip models (width, length, reduced stiffness of perforated zones), the loading to be applied, and an estimate of the edge restraint against rotation provided to the beam strip by the water box flange.

The notable features of HEI Standards on tubesheet design include features such as consideration of pressure surges, pump shutoff, head circulating water system characteristics, adequate means for provision of expansion, etc. Allowable stress and loads are to be chosen similar to the design-by-analysis approach of Appendix 4 in the ASME Code, Section VIII, Div. 2.

3.8 Curved Tube Sheets

Traditionally, heat exchangers use thick flat tube sheets. The problem of replacing thick flat tube sheets by thinner curved tube sheets was suggested first by Rachkov and Morozov [79]. They designed a much thinner semi-ellipsoidal curved tube sheet based on membrane theory of shells. The design of shallow spherical curved tube sheets for heat exchangers is discussed by Paliwal et al. [80]. Similar to a flat tube sheet treated as a thin flat plate on an elastic foundation, the curved tube sheet model uses the theory of a thin elastic shallow spherical shell on an elastic foundation, and replaces the curved plate by an equivalent plate having effective elastic constants; the nominal bending and membrane stresses and deflections are determined. A curved tube-sheet exchanger is shown in Fig. 11. So far, no code or standard has included curved tube-sheet design.

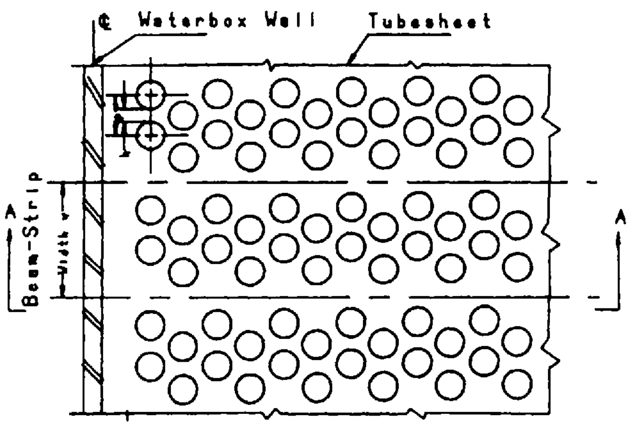


Figure 10 Beam strip for rectangular tubesheet design. (From Ref. 78.)

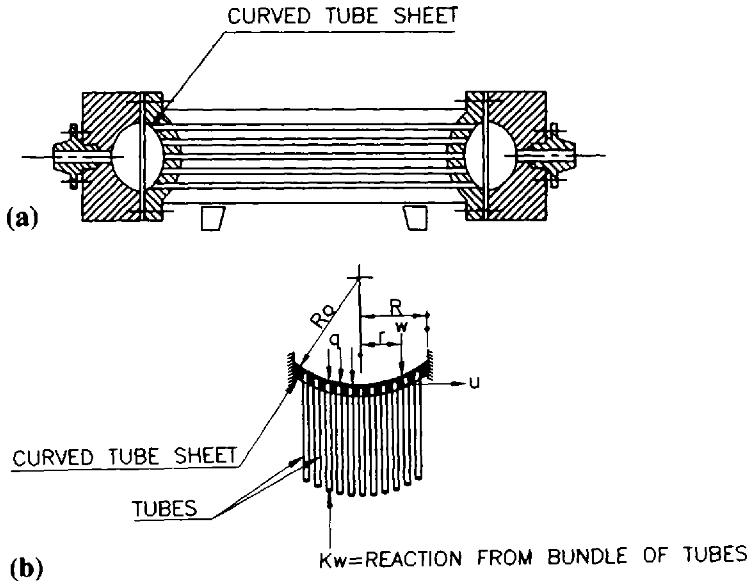


Figure 11 Curved tubesheet exchanger. (From Ref. 80.)

Advantages of Curved Tube Sheets

As compared with thick flat tube sheets, elliptical tube sheets have many advantages [81]:

1. Material savings due to thin tube sheet.
2. Because the sheet is thin, the thermal stresses developed in the tube sheet will be less.
3. Because the elliptical tube sheets are perforated elliptical shells, their stress levels will be far lower than those of flat tube sheets of the same thickness under the same operating conditions.
4. The deformation ability of a thin-wall shell is larger than that of flat plates, so the elliptical tube sheet can compensate for larger thermal deformations produced by temperature differences between the tube side and shell side.

3.9 Conventional Double Tube-Sheet Design (After Singh and Soler [41])

In a conventional double tube-sheet construction as shown in Fig. 12, in addition to the usual thermal loading, a new thermal condition arises due to in-plane differential thermal loading of

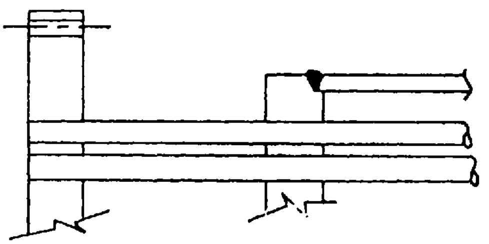


Figure 12 Conventional double tube sheet. (From Ref. 29.)

the two tube sheets. This differential radial expansion of the adjacent tube sheets may be caused by the two tube sheets having different coefficients of thermal expansion and/or differences in mean metal temperatures. The portion of tube bundle between the two tube sheets induces coupling between out-of-plane bending and direct radial expansion of the two tube sheets. Therefore, in contrast to the single tube-sheet analyses, any double tube-sheet analysis must encompass both tube-sheet bending and radial growth. These requirements make the solution procedure amenable to computer analysis only. Design of double tube sheets is discussed by Singh and Soler [41].

4 CYLINDRICAL SHELL, END CLOSURES, AND FORMED HEADS UNDER INTERNAL PRESSURE

The following symbols are used in the formulas required to calculate the minimum thickness of a cylindrical shell and various end closures.

t = minimum required thickness of shell or heads or end closures
P = internal design pressure, psi (see UG-21) (or maximum allowable working pressure, see UG-98)
C = a factor depending upon the method of attachment of head, shell dimensions, and other items as listed in ASME Code
D = inside length of the major axis of an ellipsoidal head; or inside diameter of a torispherical head; or inside diameter of a conical head at the point under consideration measured perpendicular to the longitudinal axis
D_o = outside length of the major axis of an ellipsoidal head; or outside diameter of a conical head at the point under consideration measured perpendicular to the longitudinal axis
d = dimension of the short span for flat heads
G = mean gasket diameter (not the effective gasket diameter as defined in the ASME Code)
h = maximum inside depth of the ellipsoidal head, exclusive of the flange
h_G = gasket moment arm (i.e., radial offset between the circle and bolt circle)
IDD = inside depth of torispherical head
K = factor in the formulas for ellipsoidal heads depending on the head proportion $D/2h$
L = inside spherical radius or crown radius
L_o = outside spherical radius or crown radius
M = factor in the formulas for torispherical heads depending on the head proportion L/r
P = internal design pressure
R = inside radius of the shell course under consideration
R_o = outside radius of the shell course under consideration
r = inside knuckle radius
S = maximum allowable stress value, psi (see UG-23 and the stress limitations specified in UG-24)
E = joint efficiency, or the efficiency of appropriate joint in cylindrical or spherical shells, or the efficiency of ligaments between openings, whichever is less (in decimal form)
W = total bolt load
α = one-half of the included (apex) angle of the cone at the center line of the head

4.1 Cylindrical Shell Under Internal Pressure

Design of cylindrical shell is carried out as per UG-27 or UG-29 of the ASME Code. The design formulas in the code are based on equating the maximum membrane stress to the allowable stress corrected for weld joint efficiency. As per ASME Code procedure, the thick-

Table 8 ASME Code Formulas for Minimum Thickness of Thin Cylindrical Shell to Withstand Internal Pressure [16]

Member	Thickness, t	Maximum internal pressure, P	Limitation
Longitudinal joints	$t = \dfrac{PR}{SE - 0.6P}$	$P = \dfrac{SEt}{R + 0.6t}$	$P \le 0.385SE$ $t \le 0.5R$
Circumferential joints[a]	$t = \dfrac{PR}{2SE + 0.4P}$	$P = \dfrac{2SEt}{R - 0.4t}$	$P \le 1.25SE$ $t \le 0.5R$
In terms of outside radius R_o	$t = \dfrac{PR_o}{SE + 0.4P}$	$P = \dfrac{SEt}{R_o - 0.4t}$	$P \le 0.385SE$ $t \le 0.5R$

[a]Formulas based on stress across circumferential joint will govern only if circumferential joint (Category B) efficiency is less than one-half the longitudinal joint efficiency.

ness of shells under internal pressure shall not be less than that computed by the following formulas. In addition, provision shall be made for any of the other loadings listed in UG-22, when such loadings are expected (see UG-16).

Thin Cylindrical Shells

The minimum thickness or maximum allowable working pressure of cylindrical shell shall be the greater thickness or lesser pressure as given by Eq. 92 and 93.

Circumferential Stress (Longitudinal Joints). When the thickness does not exceed one-half of the inside radius, or P does not exceed $0.385SE$, the following formulas shall apply:

$$t = \frac{PR}{SE - 0.6P} \quad \text{or} \quad P = \frac{SEt}{R + 0.6t} \tag{92}$$

Longitudinal Stress (Circumferential Joints). When the thickness does not exceed one-half of the inside radius, or P does not exceed $1.25SE$, the following formulas shall apply:

$$t = \frac{PR}{2SE + 0.4P} \quad \text{or} \quad P = \frac{2SEt}{R - 0.4t} \tag{93}$$

These formulas are presented in Table 8 along with the formulas based on shell outside dimension; Table 9 gives formulas for thick cylindrical shells.

Design for External Pressure and/or Internal Vacuum

There is no straightforward formula as in the case for the design under internal pressure, because buckling has to be taken into account. The procedure to be followed is given in the ASME Code Section VIII, Div. 1, Paragraph UG-28, and requires the use of various charts given in Appendix 5.

4.2 End Closures and Formed Heads

End closures for heat exchangers and pressure vessels are in the form of either flat covers or formed heads. ASME Code Section VIII, Div. 1, recognizes the following end closures:

1. Flat cover
2. Hemispherical cover
3. Ellipsoidal cover

Table 9 ASME Code Formulas for Minimum Thickness of Thick Cylindrical Shells to Withstand Internal Pressure [16]

Member	Thickness, t	Maximum internal pressure, P	Limitation
Longitudinal joint	$t = R(\sqrt{z} - 1)$ $= R_o \dfrac{(\sqrt{z} - 1)}{\sqrt{z}}$ where $z = \dfrac{SE + P}{SE - P}$	$P = SE \dfrac{(z_1 - 1)}{(z_1 + 1)}$ where $z_1 = \left(\dfrac{R + t}{R}\right)^2$ $= \dfrac{R_o^2}{(R_o - t)^2}$	$P > 0.385SE$ $t > 0.5R$
Circumferential joint[a]	$t = R(\sqrt{z_2} - 1)$ $= R_o \dfrac{(\sqrt{z_2} - 1)}{\sqrt{z_2}}$ where $z_2 = \left(\dfrac{P}{SE} + 1\right)$	$P = SE(z_1 - 1)$ where z_1 is as defined earlier	$P > 1.25SE$ $t > 0.5R$

[a]Formulas based on stress across circumferential joint will govern only if circumferential joint (Category B) efficiency is less than one-half the longitudinal joint efficiency.

4. Torispherical cover (flanged and dished)
5. Conical/toriconical cover

These end closures are shown schematically in Fig. 13. TEMA designates the front and rear covers by B and M, respectively. Closures other than flat heads are normally formed type; sometimes for low-pressure application cast heads are also used. Closures are designed as per UG-32 or UA-4 of the ASME Code. First a brief description of various end closures is presented, and then the determination of minimum thickness to retain internal pressure is presented.

Flat Cover

Flat covers are easy to fabricate in any thickness from plates or forgings. They are widely used from low to high pressure applications. Since a flat cover resists pressure load only by bending, its thickness is significantly greater than that of the cylindrical section to which these are attached as a closure.

Hemispherical

Hemispherical heads are used for high-pressure service since their thickness is about half that of a cylindrical shell. The degree of forming and accompanying costs are greater than any other heads and the available sizes from single plates are limited [44].

Ellipsoidal

These are widely used for low- to intermediate-pressure services. When the minor-to-major axis ratio is 0.5 (most common), the head thickness is almost the same as that of the cylindrical shell. This simplifies the joining of these two and minimizes the discontinuity stresses at the joint. Another popular ellipsoidal head is with minor axis 25% of D.

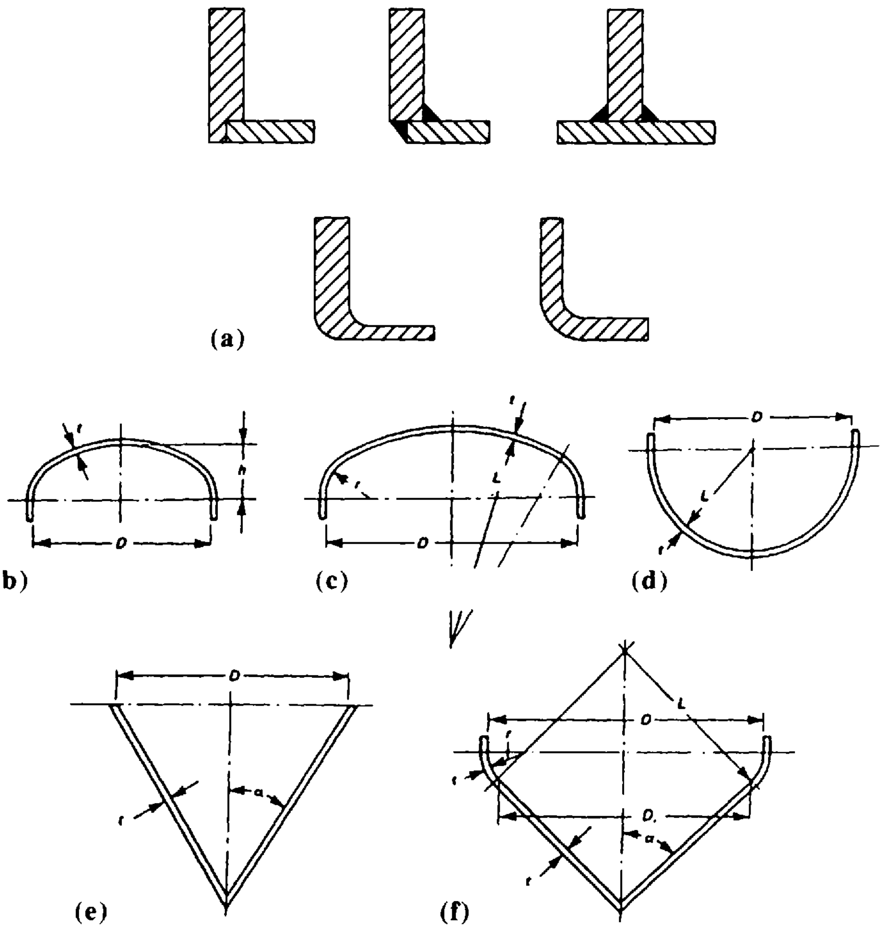


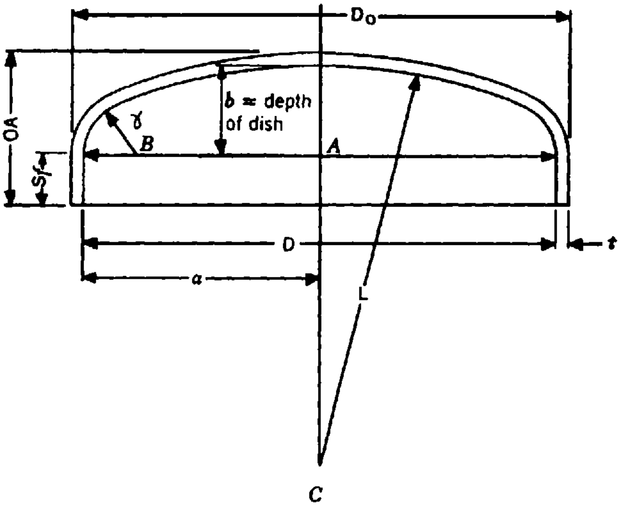
Figure 13 ASME CODE end closures. (a) Flat head (a partial list); (b) ellipsoidal head; (c) torispherical head; (d) hemispherical head; (e) conical head; and (f) toriconical head. (From Ref. 16.)

Torispherical

Among the various types of formed heads, the torispherical head is the most widely used in the industries, particularly for low-pressure service, i.e., up to 200 psi [50]. For pressures over 200 psi gage, ellipsoidal heads are used. The torispherical head is characterized by four geometric parameters: inside head diameter D, crown radius L, knuckle radius r, and head thickness t. Figure 14 shows the details of the torispherical head geometry. In Fig. 14, the depth of dish b is a geometric function of crown radius L and knuckle radius r and the straight cylindrical flange is integral with the dished end. By varying the ratios of L/D and L/r, heads of different shapes can be manufactured. Heads wherein $L \approx D$, $L \approx 16\frac{2}{3}r$, and $r = 0.06D$ are referred to as ASME flanged and dished heads in the pressure vessel industry. Another popular variation is the 80 : 10 head where $L \approx 0.8D$ and $r = 0.1L$.

Conical

These are used for low- and intermediate-pressure service with the half apex angle generally not more than 30°. A knuckle portion is provided to minimize the discontinuity stresses where it joins the shell.



$a = \frac{D}{2}$; $b = L - \sqrt{(BC)^2 - (AB)^2}$; $AB = \frac{D}{2} - \gamma$; $BC = L - \gamma$; $AC = \sqrt{(BC)^2 - (AB)^2}$; and $OA = t + b + S_f$.

Figure 14 Dimensional details of flanged and dished head.

4.3 Minimum Thickness of Heads and Closures

The required thickness at the thinnest point after forming of ellipsoidal, torispherical, hemispherical, conical, and toriconical heads under an internal pressure shall be computed by the appropriate formulas in UG-16 of the ASME Code. In addition, provision shall be made for any of the other loadings given in UG-22. The head design formulas in the code are based on equating the maximum membrane stresses to the allowable stresses corrected for weld joint efficiency.

Flat Cover

1. As per UG-34, the minimum required thickness of flat head, cover, and blind flanges shall be calculated by the formula:

$$t = d\sqrt{\frac{CP}{SE}} \qquad (94a)$$

2. The minimum required thickness of flat head, cover, and blind flange attached by bolts causing an edge moment is given by the formula:

$$t = d\left(\frac{CP}{SE} + \frac{1.9Wh_G}{SEd^3}\right)^{0.5} \qquad (94b)$$

TEMA Standards. The TEMA Standards have a deflection-based formula that seeks to limit the maximum deflection, *d*, of the flat cover of a multitube pass unit to 0.030 in (0.762 mm) [50]. The resulting formula for flat cover thickness, *t*, in inches, is given by [50]:

$$t = \left[\frac{1.425G^4P}{E} + \frac{0.5h_G A_b G \times 10^6}{E d_b^{1/2}} \right]^{1/3} \tag{95}$$

Ellipsoidal Heads

The thickness or the maximum allowable working pressure of a dished head of semi-ellipsoidal form, in which half the minor axis equals one-fourth of the inside diameter of the head skirt, shall be determined by

$$t = \frac{PD}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{D + 0.2t} \tag{96}$$

Torispherical Heads

The required thickness or the maximum allowable working pressure of a torispherical head for the case in which the knuckle radius is 6% of the inside radius and the inside crown radius equals the inside diameter of the skirt (i.e., $L \approx D$, $L_i \approx 16\frac{2}{3}r$, and $r = 0.06D$) shall be determined by

$$t = \frac{0.885PL}{SE - 0.1P} \quad \text{or} \quad P = \frac{SEt}{0.885L + 0.1t} \tag{97}$$

Hemispherical Heads

When the thickness or the maximum allowable working pressure of a hemispherical head does not exceed $0.356L$, or the internal pressure P does not exceed $0.665SE$, the required thickness or the maximum allowable working pressure of a hemispherical head is given by

$$t = \frac{PL}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{L + 0.2t} \tag{98}$$

Conical Heads and Sections (without Transition Knuckle)

The required thickness of conical heads or conical shell sections that have a half apex angle α not greater than 30° shall be determined by

$$t = \frac{PD}{2(\cos\alpha)(SE - 0.6P)} \quad \text{or} \quad P = \frac{2SEt(\cos\alpha)}{D + 1.2t(\cos\alpha)} \tag{99}$$

These formulas and formulas for minimum head thickness referred to outside head dimension are given in Table 10.

4.4 Comparison of Various Heads

1. Compared to a flat cover, which resists pressure load only by bending, a formed head resists pressure by developing membrane stress, and hence the thickness of formed heads will be less than the flat cover [50].
2. The formed head has the drawback of thinning out in areas of sharp curvature and thickening in adjacent regions of moderate curvature. Such a variation is more pronounced in spun heads, and more particularly in hot-formed heads than in cold-formed heads.
3. As far as the cost is concerned, the head that is the lowest in cost and meets the code requirements should be designed. The ellipsoidal and hemispherical heads have the least weight per unit volume.
4. Salient features of fabrication of various heads are discussed in Chapter 15, Heat Exchanger Fabrication.

Table 10 ASME Code Formulas for Minimum Thickness of Heads/End Closures to Withstand Internal Pressure [16]

Heads	Minimum thickness, t	Maximum pressure, p
Ellipsoidal	$t = \dfrac{PD}{2SE - 0.2P}$	$P = \dfrac{2SEt}{D + 0.2t}$
	$t = \dfrac{PDK}{2SE - 0.2P}$	$P = \dfrac{2SEt}{KD + 0.2t}$
	$t = \dfrac{PD_oK}{2SE + 2P(K - 0.1)}$	$P = \dfrac{2SEt}{KD_o - 2t(K - 0.1)}$
		where $K = \dfrac{1}{6}\left[2 + \left(\dfrac{D}{2h}\right)^2\right]$
Torispherical	$t = \dfrac{0.885PL}{SE - 0.1P}$	$P = \dfrac{SEt}{0.885L + 0.1t}$
	$t = \dfrac{PLM}{2SE - 0.2P}$	$P = \dfrac{2SEt}{LM + 0.2t}$
	$t = \dfrac{PL_o M}{2SE + P(M - 0.2)}$	$P = \dfrac{2SEt}{ML_o - t(M - 0.2)}$
		where $M = \dfrac{1}{4}\left[3 + \left(\dfrac{L}{r}\right)^{0.5}\right]$
Hemispherical	$t = \dfrac{PL}{2SE - 0.2P}$	$P = \dfrac{2SEt}{L + 0.2t}$
Conical	$t = \dfrac{PD}{2(\cos\alpha)(SE - 0.6P)}$	$p = \dfrac{2SEt(\cos\alpha)}{D_o + 1.2t(\cos\alpha)}$
	$t = \dfrac{PD_o}{2(\cos\alpha)(SE + 0.4P)}$	$P = \dfrac{2SEt(\cos\alpha)}{D_o - 0.8t(\cos\alpha)}$

5 BOLTED FLANGED JOINT DESIGN

5.1 Construction and Design

Flanges are often employed to connect two sections by bolting them together so that the sections can be assembled and disassembled easily. In heat exchangers, the flange joints are used to connect together the following components:

1. Channel and channel cover
2. Heads or channels with the shell/tube sheets
3. Inlet and outlet nozzles with the pipes carrying the fluids
4. To close various openings such as manholes, peepholes, and their cover plates

The flanged joints play an important role from the standpoint of integrity and reliability of heat exchangers. Improper design of flanges causes leakage of heat exchanger fluids. Therefore, preventing the liquid or gas leaks is one of the most important considerations while designing flanged joints.

Flanged Joint Types

From a conceptual standpoint, flanged joints may be subdivided into two major categories [41]:

1. Bolted joints
2. Pressure-actuated or self-energizing joints

The bolted joint is by far the most common type. The basic difference between these two joint types lies in the manner by which the pressure load is resisted and leak tightness is achieved. A bolted joint essentially consists of a gasket interposed between two structural members called flanges, which in turn are connected to other structural members like cylindrical shells or pipes, and a set of bolts for joining together the two flanges. To make the joint, the gasket is compressed to a desired value by prestressing the bolts. Pressure-actuated joints exploit the header pressure force to compress and to seal the gasket. Pressure-actuated joints find application in the higher pressure range, typically over 2000 psi [41].

Constructional Details of Bolted Flange Joints

Types of Bolted Flanges. Based on the width of gasket, flange joints are classified as (1) ring type gasket joint and (2) full-face gasket joint. For very-low-pressure applications (100–300 psi), wide gaskets that span the entire flange face may be used. This construction is known as "full-face gasket" design. A design method for flanges utilizing full-face gaskets is presented in ref 41. In general, for high-pressure and medium- to high-pressure applications, only ring type gaskets are used. Sometimes even for low-pressure applications ring type gaskets are used. Based on constructional details, flanged joints are classified as

1. Ring flange
2. Weld neck integral flange or tapered hub flange
3. Lap joint flange
4. Reverse flange

Ring Flange. The ring flange consists of an annular circular plate welded to the end of the cylindrical shell. A number of equidistant bolt holes are drilled on a uniform pitch circle, and the gasket is confined inside the bolt circle. This joint is utilized in low- to moderate-pressure applications. A ring type gasket joint is shown schematically in Fig. 15a.

Weld Neck Integral Flange. The weld neck flange or a tapered hub flange, as shown in Fig. 15b, may be viewed as a structural member consisting of an annular ring and a tapered hub butt welded to the cylindrical shell. This flange has the best characteristics for preventing failure from fatigue and thermal stresses. These flanges have been used at pressures as high as 5000 psi, although the flange becomes massive and heavy as pressure and diameter increase.

Lap Joint Flange. The lap joint flange, shown in Fig. 15c, finds use in low-pressure applications where economy of construction is an important consideration. The backing ring can be made from low-cost but strong structural material (e.g., carbon steel), whereas the "lap ring" may be made from expensive corrosion-resistant material. Furthermore, this flange design facilitates alignment of bolt holes in matching rings in opposing pipe ends. In outside packed floating head heat exchangers, the flange at the rear end is of the lap joint type.

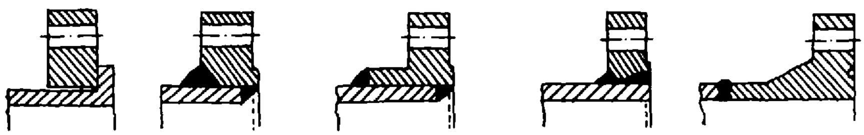


Figure 15 Flange joint construction.

Design of Bolted Flange Joints

The objectives in flange design are to ensure that the residual gasket stress levels and the pressure induced in the flange during bolt preload, as well as under operating conditions, do not exceed allowable stress values in the structural members.

The earliest treatment of the problem of flange design to receive widespread recognition was that of Waters et al. [82], which gave the general basis for the design rules in the ASME Code. Design of flanged joints with ring type gaskets is carried out as per Appendix 2 of ASME Code Section VIII, Div. 1. Appendix S of the Code gives general guidelines for bolting requirements of flanges. The Code method for design of integral type flange and ring flange is briefly described here. Unless otherwise mentioned, the mention of ASME Code throughout this book refers to ASME Code Section VIII, Div. 1 (1992 edition) only [16].

ASME Code Classification of Circular Flanges for Design Purposes. For computation purposes, ASME Code Section VIII, Div. 1, classifies circular flanges with ring type gaskets as

1. Loose type flanges
2. Integral type flanges
3. Optional type flanges
4. Flanges with nut stop
5. Reverse flanges

Salient constructional details, design features, and step-by-step design procedures of various flanged joint types except flanges with nut stop and reverse flanges are described in the following sections.

Loose Type Flanges. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall, and designs where the method of attachment is not considered to give the mechanical strength equivalent of integral attachment. Figures 1 to 4a of the ASME Code conform to loose type flanges [16].

Integral Type Flanges. This type covers those designs in which the flange is cast or forged integrally with the nozzle neck, vessel, or pipe wall, butt welded thereto, or attached by other forms of arc or gas welding of such a nature that the flange and nozzle neck and vessel or pipe wall are considered to be the equivalent of an integral structure. Figures 5 to 7 of the ASME Code [16] conform to integral type flanges.

Optional Type Flanges. This type covers those designs in which the attachment of the flange to the nozzle neck, vessel, or pipe wall is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the construction as a loose type flange provided none of the values given in the ASME Code are exceeded. Figures 12 and 12a of the ASME Code [16] conform to optional type flanges.

Design Procedure. The integrity and reliability of a bolted flanged joint depend to a large extent upon the correct choice of materials, dimensions, and loads on the gasket. The flange design procedure can be summarized as three separate elements:

1. Gasket design
2. Bolting design
3. Flange design

To start with, materials of construction of flange, bolting and gasket, and gasket properties are chosen. Flange inner diameter and shell thickness to which the flange is to be welded are also known. A rough guess of the various dimensions of the flange is made, taking into account the permissible hub slope, and minimum hub length in the case of weld neck flange, and bolting dimensional requirements. Suitable gasket outer diameter and width are also chosen, keeping the minimum width requirements. Flange dimensions shall be such that the calculated stresses in the flange shall not exceed code stress values. In the following paragraphs, the detailed design procedures of the three elements of flange design are described.

Gasket Design.

Gaskets and Their Characteristics. A leak-proof joint with metal-to-metal surfaces without a gasket is difficult to achieve even with use of accurately machined fine finish surfaces. Surface irregularities only a few millionths of an inch will permit the escape of a fluid under pressure [44]. Being a semiplastic material, the gasket deforms under load, which in turn seals the minute surface irregularities and prevents leakage of the fluid.

Selection of Gasket Material. A gasket is essentially an elastoplastic material that is softer than the flange faces. In the gasket seating condition the entire bolt load is borne by the gasket. Hence, the gasket must be strong enough to withstand load due to bolting and operating conditions without crushing or extruding out. Therefore, soft materials like asbestos and organic fibers are precluded for high-pressure applications. Also, the gasket material shall withstand the operating temperature and exhibit corrosion resistance to the fluid contained in the pressure vessel [83].

Gasket Materials. Gaskets are made out of a myriad variety of materials. Good references are available from many gasket manufacturers for the selection of proper gaskets for the intended applications. Table 2-5.1 of the ASME Code gives a list of many commonly used gasket materials (see Table 11). This is a partial list of gasket materials included in Table 2-5.1 of the ASME Code [16]:

Elastomers without fabric or high percentage of asbestos fiber
Asbestos with suitable binder
Elastomers with cotton fabric insertion
Elastomers with asbestos fabric insertion
Spiral-wound asbestos-filled metal
Corrugated metal with asbestos fill
Corrugated metal
Flat metal, jacketed asbestos fill
Grooved metal
Solid flat metal

The choice of the gasket material is often based upon the required gasket width. If the gasket is made too narrow, the unit pressure on it may be excessive, whereas if the gasket is made too wide, the bolt load will be unnecessarily high [44].

Gasket Factors. The basic behavior of the gasket is defined by the gasket factor m and gasket or joint contact surface unit seating load y, which are tabulated in the ASME Code, Section VIII, Div. 1.

Gasket Factor, m. This is the ratio of the residual stress on the gasket under operating pressure to that pressure. In other words, m = (bolt load – hydrostatic end load)/(gasket area × internal pressure).

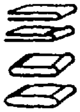
Gasket or Joint Contact Surface Unit Seating Load, y. This is the stress required to make the gasket surface take up the shape of the flange faces, or the gasket stress required to contain zero internal pressure. The factor y is usually expressed as a unit stress in pounds per square inch and is independent of the pressure in the vessel. Table 2-5.1 of the ASME Code gives suggested design values of gasket factor m and minimum design seating stress y.

Table 11 ASME Code List of Gasket Materials [16]

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, psi	Sketches	Facing Sketch and Column in Table 2-5.2
Self-energizing types (O rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0	. . .	. . .
Elastomers without fabric or high percent of asbestos fiber:				
Below 75A Shore Durometer	0.50	0		(1a),(1b),(1c),(1d), (4),(5); Column II
75A or higher Shore Durometer	1.00	200		
Asbestos with suitable binder for operating conditions:				
1/8 in. thick	2.00	1600		(1a),(1b),(1c),(1d), (4),(5); Column II
1/16 in. thick	2.75	3700		
1/32 in. thick	3.50	6500		
Elastomers with cotton fabric insertion	1.25	400		(1a),(1b),(1c),(1d), (4),(5); Column II
Elastomers with asbestos fabric insertion (with or without wire reinforcement):				
3-ply	2.25	2200		(1a),(1b),(1c),(1d), (4),(5); Column II
2-ply	2.50	2900		
1-ply	2.75	3700		
Vegetable fiber	1.75	1100		(1a),(1b),(1c),(1d), (4),(5); Column II
Spiral-wound metal, asbestos filled:				
Carbon	2.50	10,000		(1a),(1b); Column II
Stainless, Monel, and nickel-base alloys	3.00	10,000		
Corrugated metal, asbestos inserted, or corrugated metal, jacketed asbestos filled:				
Soft aluminum	2.50	2900		(1a),(1b); Column II
Soft copper or brass	2.75	3700		
Iron or soft steel	3.00	4500		
Monel or 4%–6% chrome	3.25	5500		
Stainless steels and nickel-base alloys	3.50	6500		

(continued)

Table 11 Continued

Gasket Material	Gasket Factor *m*	Min. Design Seating Stress *y*, psi	Sketches	Facing Sketch and Column in Table 2-5.2
Corrugated metal:				
Soft aluminum	2.75	3700		
Soft copper or brass	3.00	4500		(1a),(1b),(1c),(1d);
Iron or soft steel	3.25	5500		Column II
Monel or 4%–6% chrome	3.50	6500		
Stainless steels and nickel-base alloys	3.75	7600		
Flat metal, jacketed asbestos filled:				
Soft aluminum	3.25	5500		
Soft copper or brass	3.50	6500		(1a),(1b),(1c),[2]
Iron or soft steel	3.75	7600		(1d);[2](2)[2];
Monel	3.50	8000		Column II
4–6% chrome	3.75	9000		
Stainless steels and nickel-base alloys	3.75	9000		
Grooved metal:				
Soft aluminum	3.28	5500		
Soft copper or brass	3.50	6500		(1a),(1b),(1c),(1d),
Iron or soft steel	3.75	7600		(2),(3); Column II
Monel or 4%–6% chrome	3.75	9000		
Stainless steels and nickel-base alloys	4.25	10,100		
Solid flat metal:				
Soft aluminum	4.00	8800		
Soft copper or brass	4.75	13,000		(1a),(1b),(1c),(1d),
Iron or soft steel	5.50	18,000		(2),(3),(4),(5);
Monel or 4%–6% chrome	6.00	21,800		Column I
Stainless steels and nickel-base alloys	6.50	26,000		
Ring joint:				
Iron or soft steel	5.50	18,000		
Monel or 4%–6% chrome	6.00	21,800		(6); Column I
Stainless steels and nickel-base alloys	6.50	26,000		

NOTES:
(1) This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of *m* and *y* that have generally proved satisfactory in actual service when using effective gasket seating width *b* given in Table 2-5.2. The design values and other details given in this Table are suggested only and are not mandatory.
(2) The surface of a gasket having a lap should not be against the nubbin.

Gasket Dimensions. A relationship for making a preliminary estimate of the proportions of the gasket may be derived as follows [41]:

Residual gasket force = gasket seating force − hydrostatic pressure force

The residual gasket force cannot be less than that required to prevent leakage of the internal fluid under operating pressure. This condition results in the following expression:

$$\frac{d_o}{d_i} = \sqrt{\frac{y - pm}{y - p(m + 1)}} \tag{100}$$

where d_o is the gasket outside diameter, m is the gasket factor, and y the minimum design seating stress.

The inside diameter of the gasket (in inches) is normally as follows:

$$d_i = B + 0.01$$

where B equals the shell inside diameter for weld neck flange and shell outside diameter for ring flange.

Gasket Width and Diametral Location of Gasket Load Reaction. The steps involved in arriving at the gasket width and diametral location of gasket load reaction are as follows:

1. Calculate the gasket width, N, given by

$$N = (d_o - d_i)/2 \tag{101}$$

2. Select the gasket width such that it is not less than the minimum specified width of the gasket as specified in Table 2-5.2 of the ASME Code and reproduced in Table 12.
3. Calculate the basic gasket width, b_o, given by Table 2-5.2 of the ASME Code.

Bolting Design. With the size and shape of the gasket established, next determine the bolting required. The bolting should be designed to maintain the required compression on the gasket with the internal pressure acting. Various design aspects of bolting are discussed next.

Determination of Bolt Loads. The bolt loads, W, required under the following conditions could be considered:

1. Gasket seating condition in the absence of internal pressure
2. Operating conditions

The thickness of flanges shall be determined as the greater required either by the operating or by the bolting up conditions, and in all cases both conditions shall be calculated in accordance with the following.

Gasket Seating Conditions. The gasket seating conditions are the conditions existing when the gasket or joint contact surface is seated by applying an initial load on the bolts when assembling the joint, at atmospheric pressure and temperature. The minimum initial load considered to be adequate for proper gasket seating is a function of the gasket material and the effective or contact area to be seated. The minimum initial bolt load required for this purpose, W_{m2}, shall be determined using the following formula

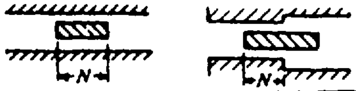
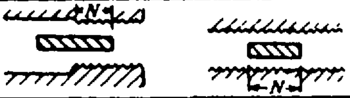
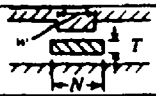
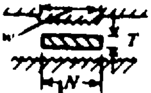
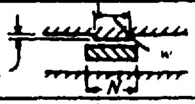
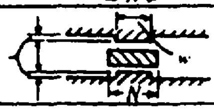
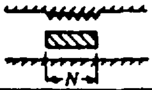
$$W_{m2} = \pi b G y \tag{102}$$

The need for providing sufficient bolt load to seat the gasket or joint contact surfaces in accordance with this formula will prevail on many low-pressure designs and with facings and materials that require a high seating load, and the bolt load calculated for operating conditions is not sufficient to seat the joint. When formula 102 governs, flange dimensions will be a function of the bolting instead of internal pressure.

As per Code formulas, for flange pairs used to contain a tube sheet (both sides gasketed) for a floating head or a U-tube type of heat exchanger, or for any other similar design, and where the flanges and/or gaskets are not the same, W_{m2} shall be the larger of the values obtained from formula 102 as individually calculated for each flange and gasket, and that value shall be used for both flanges.

Operating Conditions. The operating conditions are the conditions required to resist the hydrostatic end force (H) of the design pressure, which tends to part the joint and to maintain on the gasket or joint contact surface sufficient compression force (H_p) to assure a tight joint at all operating conditions. The minimum load is a function of the design pressure, the gasket

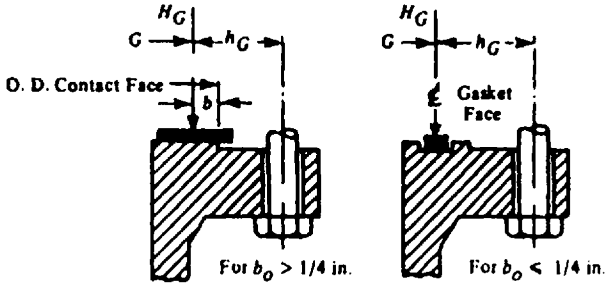
Table 12 Effective Gasket Width

Facing Sketch (Exaggerated)		Basic Gasket Seating Width, b_o Column I	Basic Gasket Seating Width, b_o Column II
(1a)		$\frac{N}{2}$	$\frac{N}{2}$
(1b) See Note (1)			
(1c)	 $w \le N$	$\frac{w+T}{2}; \left(\frac{w+N}{4} \text{ max.}\right)$	$\frac{w+T}{2}; \left(\frac{w+N}{4} \text{ max.}\right)$
(1d) See Note (1)	 $w \le N$		
(2) 1/64 in. Nubbin	 $w \le N/2$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3) 1/64 in. Nubbin	 $w \le N/2$	$\frac{N}{4}$	$\frac{3N}{8}$
(4) See Note (1)		$\frac{3N}{8}$	$\frac{7N}{16}$
(5) See Note (1)		$\frac{N}{4}$	$\frac{3N}{8}$
(6)		$\frac{w}{8}$	...

Effective Gasket Seating Width, b

$b = b_o$, when $b_o \le 1/4$ in.; $b = 0.5\sqrt{b_o}$, when $b_o > 1/4$ in.

Location of Gasket Load Reaction



NOTES:

(1) Where serrations do not exceed 1/64 in. depth and 1/32 in. width spacing, sketches (1b) and (1d) shall be used.

(2) The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

material, and the effective gasket area or the effective contact area to be kept tight under pressure. The required bolt load W_{m1} for the operating condition is given by

$$W_{m1} = H + H_p = \frac{\pi}{4} G^2 P + 2\pi b G m P \quad (103)$$

Various flange forces are shown schematically in Fig. 16 for the ring flange and weld neck integral flange [84].

Bolt Area at the Root of the Threads. The necessary bolt area at the root of the threads, A_m, required for both the gasket seating and operating conditions is the greater of the values W_{m1}/S_b and W_{m2}/S_a as given by the following expression:

$$A_m = \max|W_{m1}/S_b,\ W_{m2}/S_a| \quad (104)$$

From the required bolt area, determine the minimum number of bolts required (generally in multiples of 4) and observe the minimum sizes as recommended by TEMA, RCB-11. From the number of bolts chosen, find out the actual bolt area, A_b. In no case shall A_b be less than A_m. At this point the designer should sketch a tentative layout showing the desired location and size of gasket or contact surface, hub thickness, and diameter of bolts and bolt circle, and from these set the outside diameter of the flange.

Flange Bolt Load W. The bolt loads used in the flange design shall be the values obtained from these formulas:

1. For the gasket seating condition,

$$W = \frac{(A_m + A_b)\ S_a}{2} \quad (105)$$

2. For the operating condition,

$$W = W_{m1} \quad (106)$$

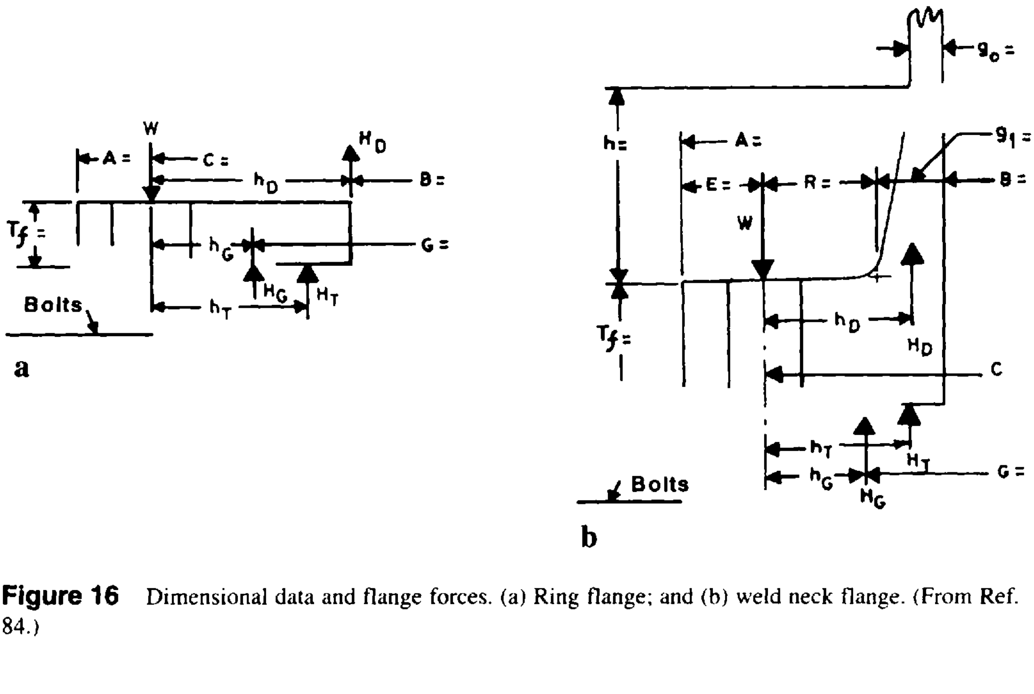


Figure 16 Dimensional data and flange forces. (a) Ring flange; and (b) weld neck flange. (From Ref. 84.)

Pitch Circle Diameter. In general the pitch circle diameter for each particular size of bolt considered should be kept small, to keep the flange bending moment and flange outside diameter small [85].

Minimum Bolt Size. Small bolts should be avoided wherever possible, owing to the case with which they may be over stressed by torsion applied with a wrench. Bolts, studs, nuts, and washers must meet the Code requirements. Appendix 2 of the Code recommends not using bolts and studs smaller than 0.5 in (12.7 mm). If bolts or studs smaller than 0.5 in (12.7 mm), alloy steel bolting material must be used. Precautions must be taken to avoid overstressing small-diameter bolts.

TEMA Standards give guidelines for minimum bolt size in RCB-11. The minimum bolt size is 0.75 in for *R*, 0.5 in for C, and 0.625 in for B class exchangers.

Minimum Recommended Bolt Spacing. Bolts should be spaced far enough apart to permit the clearance necessary for socket wrenches and to insure a uniform compression on the gasket. Likewise, the bolt circle on hubbed or integral flanges should have sufficient diameter to permit a generous fillet between the back of the flange and hub. Waters-Taylor recommends a bolt spacing of at least 2.25 times bolt diameters between centers to avoid high stress concentration.

In the TEMA guidelines, the minimum chordal pitch between adjacent bolts and minimum recommended wrench and nut clearances may be read from TEMA Table D-5.

Maximum Recommended Bolt Spacing. The bolt spacing should not be so great as to result in an appreciable reduction in gasket pressure between bolts. Waters-Taylor recommends a spacing of $3.5d$ (d is the nominal diameter of bolt) between bolt hole centers as a reasonable maximum. An empirical expression given by Taylor Forge and Pipe Works [84] expresses the maximum bolt pitch in the form

$$B_{max} = 2d + \frac{6T_f}{m + 0.5} \tag{107}$$

where B_{max} is the maximum bolt spacing for a tight joint (in), d the nominal bolt diameter (in), T_f the flange thickness (in), and m the gasket factor, which is obtained from ASME Code Table 2-5.1. This is included in TEMA, RCB-11.22.

Load Concentration Factor. As per TEMA RCB-11.23, when the distance between bolt centerlines exceeds the recommended B_{max}, the total flange moment determined shall be multiplied by a load concentration factor equal to [3]:

$$\sqrt{\frac{B}{B_{max}}} \tag{108}$$

where B is the centerline-to-centerline bolt spacing.

Note: To prevent over stressing of bolted flanged connections, the designer should, wherever possible, set the lengths of wrench to be used.

Relaxation of Bolt Stress at Elevated Temperature. A rise in temperature of a flanged joint causes the bolt and flange stresses to diminish, and on maintaining the joint at temperature, further reduction in stresses may occur due to creep as time elapses [85]. A multiphase PVRC elevated temperature program was initiated in 1982 by a task group of the PVRC Subcommittee on Gasket Testing Elevated Temperature Joint Behaviour, and the committee's report is published through WRC Bulletin 391 by Derenne et al. [86].

Flange Design. After the gasket and bolting design, next determine the flange dimensions required to withstand the bolt load without exceeding the allowable stress for the flange material. The outside diameter of the flange must be large enough to seat the bolt with manufacturing tolerance. In addition to bolting data, some more details on flange dimensions can be read from TEMA Table D-5. Since the flange design procedure is iterative in the case of integral weld neck flange and slip-on flange, initially assume a flange thickness. In the case of the ring flange, a closed-form solution for flange thickness is possible. The next step is to determine the moment arm of the various forces and reactions.

Flange Moments. In the calculation of flange stress, the moment of a load acting on the flange is the product of the load and the moment arm.

Various forces acting during the operating condition are the hydrostatic end force on area inside of the flange, H_D, the pressure force on the flange face, H_T, and the gasket load under operating conditions, H_G:

$$H_D = \frac{\pi B^2 P}{4} \tag{109a}$$

$$H_T = H - H_D \tag{109b}$$

$$H_G = W - H \tag{109c}$$

where W is bolt load, W_{m1} or W_{m2}, whichever is greater. For the operating condition, the flange moment M_o is the sum of the three individual moments M_D, M_T, and M_G. Determine the moment arms h_D, h_T, and h_G for flange loads under operating conditions from Table 13 for the three types of flanges. Calculate M_D (moment due to H_D), M_T (moment due to H_T), and M_G (moment due to H_G) as given by

$$M_D = H_D h_D \tag{110a}$$

$$M_T = H_T h_T \tag{110b}$$

$$M_G = H_G h_G \tag{110c}$$

and

$$M_o = M_D + M_T + M_G \tag{111}$$

For the gasket seating condition, the total flange moment M'_o (ASME Code uses the term Mo for moment due to gasket seating condition also), which is opposed only by the gasket load W, is given by

$$M'_o = \frac{W(C - G)}{2} \tag{112}$$

Flange Thickness. For a ring flange, the flange thickness t_f required is the greater of the gasket seating condition or operating condition, given by

$$T_f = \max\left(\sqrt{\frac{M_o Y}{S_{fb} B}}, \sqrt{\frac{M'_o Y}{S_{fa} B}}\right) \tag{113}$$

where Y is the shape constant, defined in Section 5.2, Step 5.

Table 13 Moment Arms for Flange Loads Under Operating Conditions

Type of flange	h_D	h_T	h_G
Integral flange	$R + 0.5g_1$	$0.5(R + g_1 + h_G)$	$0.5(C - G)$
Loose or ring flange	$0.5(C - B)$	$0.5(h_D + h_G)$	$0.5(C - G)$
Lap flange	$0.5(C - B)$	$0.5(C - G)$	$0.5(C - G)$

For the weld neck integral flange, as mentioned earlier, flange thickness is calculated by iterative process until such time as the flange stress falls within allowable stress for the flange material. If not within the limit, increase the flange thickness and continue the steps mentioned earlier. The stresses induced in the flange shall be determined for both the operating condition and gasket seating condition, whichever controls. The procedure to determine flange stresses is listed in the step-by-step procedure given in the Appendix.

Flange Facings. The geometric details of the mating flange surfaces on which the gasket seats are known as flange facings. The ability of a flanged joint to maintain a leak-proof joint depends on a number of parameters, of which the gasket and flange facing details are the most important [41]. Flange facings are prepared to suit the gasket type, the kind of application, and the service conditions. Some of the classifications of the flange facings are:

1. Unconfined and prestressed: flat face and raised face (Fig. 15)
2. Semiconfined and prestressed (male and female joint) (Fig. 17a)

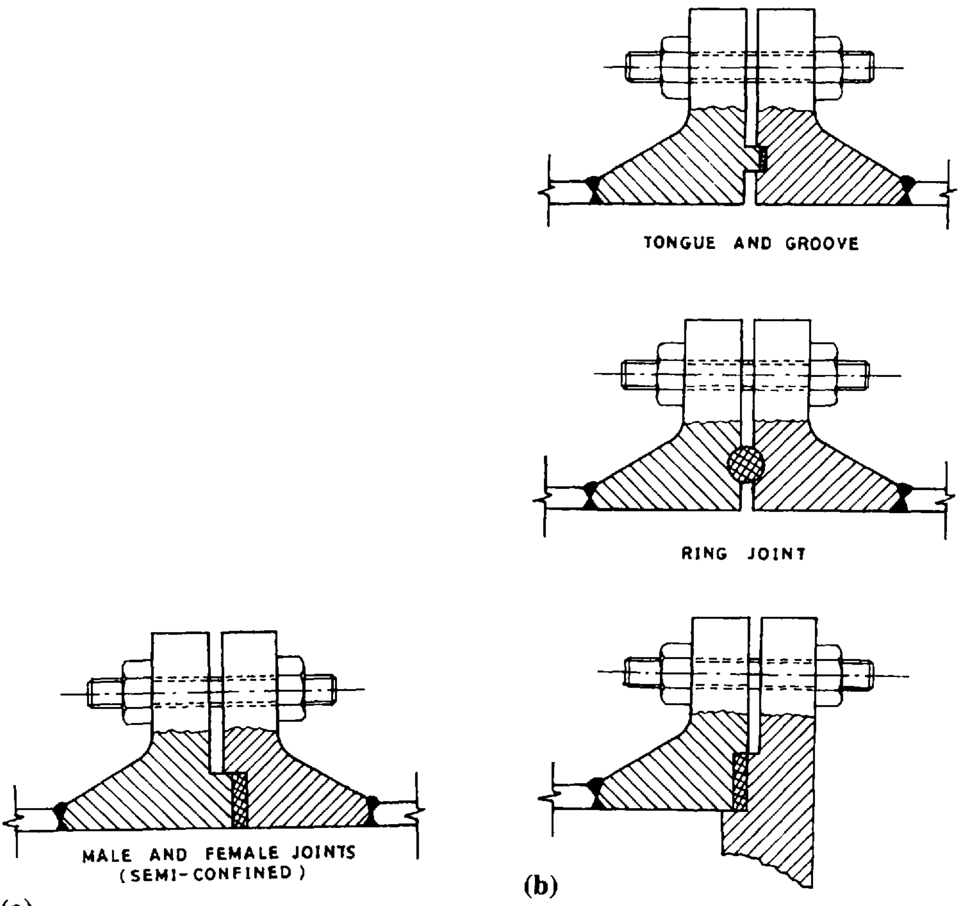


Figure 17 Flange facings. (a) Semiconfined and prestressed; and (b) confined and prestressed. (After Ref. 41.)

3. Confined and prestressed: tongue and groove, double step joint, ring joint (Fig. 17b)
4. Self-energizing: O-rings, metallic, elastomer, etc.

The applicability of various flange facings is discussed in refs. 41 and 44. Flat face and raised face are used for low-pressure applications; the male and female facings have the advantage of confining the gasket, thereby minimizing the possibility of blowout of gaskets. Since the mating flanges are nonidentical, male and female facings are widely used on heat exchangers and not on pipelines. Also, the tongue-and-groove type facings give protection against deforming soft gaskets into the interior of the vessel. Table 2-5.2 of the ASME Code gives a list of many commonly used contact facings.

Flange Facing Finish. The type and texture of surface finish are important for leak tightness of a flanged joint. There are five distinct styles of surface finish that are commonly used in the industry [41]: rounded nose spiral finish, spiral serrated finish, concentric serrated finish, smooth finish, and cold water finish.

Requirements for Flange Materials. The material used for flanges must meet the ASME Code general requirements for materials for pressure retaining parts. Some specific ASME Code Section VIII, Div. 1 (Appendix 2) requirements for flange materials include the following:

1. Flanges made from ferritic steel must be given a normalizing or full annealing heat treatment when the thickness of the flange section exceeds 3 in (76.2 mm).
2. Material on which welding is to be performed must be proved to be of good weldable quality.
3. Welding shall not be performed on steel that has a carbon content higher than 0.35%.
4. All welding on flange connections must comply with the Code requirements for postweld heat treatment.
5. Fabricated hubbed flanges may be machined from a hot-rolled or forged billet or a forged bar.

Rating of Standard Flanges. Standard flanges are rated as 150, 300, 400, 600, 900, 1500, and 2500 lb flanges. TEMA Table D-3 provides dimensions of ANSI standard flanges and bores of welding neck flanges. Lengths of alloy steel stud bolts for various flange ratings are furnished in TEMA Table D-4.

Drawback of the Existing Flange Design Procedure

The flange design methods given in the ASME Code and several others base the design criteria strictly on the flange stress limits [87]. According to Singh [87], the stress-limit-based design methods do not offer any assurance of sealability; they merely protect the flange from gross plastic deformation. A complete analysis of the bolted joint requires evaluation of the gasket residual pressure, flange stress, and bolt stress. These requirements commanded the attention of many committees on bolted joints study.

Bolted Joint Integrity and Intertube Pass leakage in U-Tube Exchangers

An analysis technique to determine the structural behavior of the both sides gasketed U-tube tube-sheet exchanger is studied in ref. 88. The method also provides procedure to compute the magnitude of the interpass leakage between the channel pass partition lanes. Some of the main conclusions of this study with reference to flange stresses are as follows:

1. The flange for the low pressure chamber may be grossly overstressed if sized using the ASME Code.
2. The stresses in the high-pressure side flange are generally higher than those predicted by the ASME Code.

3. Increasing the bolt prestress increases the stresses in the flanged joint elements. It has, however, a minor effect in reducing the leakage area.

Pressure Vessel Research Council Activities on Bolted Flanged Connections

The PVRC Committee on Bolted Flanged Connections (BFC) was established in 1985 to improve the ASME design rules for bolted full-face flanged connections. It is led by Chairman Dr. K. H. Hsu of Babcock & Wilcox Co. In 1990, the committee developed a 5-year plan and the goals have recently been organized into six committee assignments [89]:

1. Implement PVRC gasket constants and test procedure developments.
2. Issue flange design guidelines considering items such as behavior, tightness, transients, relaxation, etc.
3. Flange rating parameters for standard flanges.
4. Design parameters for ASME joints.
5. Gasket testing for temperature behavior data and test method development.
6. Flanged joint assembly and interaction effects.

Task Group on Flange Parameters Studies (Chairman: J. R. Winter, Jr.). This is a task group to study the parameters and tools for the rating of standard flanges and the design of code flanges consistent with the committee's long-range plans. Flange rotation, leak rate, thermal loads, and gasket performance need to be considered. The emergence of new emissions regulations, new and tighter gaskets, tightness-based gasket constants, and improved analytical tools all support such a new effort.

5.2 Step-by-Step Procedure for Integral/Loose/Optional Flanges Design

The ASME Code procedure for bolted flange joints for integral/loose/optional design is given here. The design procedure for reverse flange design is not covered. Certain steps may not be relevant for any one or two of these varieties. The procedure given here is similar to the ASME Code procedure detailed in the Taylor and Forge Company Bulletin on flange design [84]. The flange design procedure for ring flange and weld neck flange is shown in the Working Sheets furnished in Annexures 1 and 2. The essential steps on bolted flange design are as follows:

1. Selection of material for flange, gasket, bolts
2. Calculation of load for gasket seating condition
3. Calculation of load to withstand hydrostatic pressure known as operating condition
4. Bolting design and number of bolts decided
5. Thickness of flange estimation to withstand governing moment
6. Calculation of stress in the flange and to verify that the calculated stresses are within Code allowable stress

Data Required

Design pressure, P
Design temperature, T
Atmospheric temperature, T_a
Material specification for
  Flange
  Bolting
  Gasket
Code allowable bolt stress
  At design temperature, S_b

At atmospheric temperature, S_a
Code allowable flange stress
At design temperature, S_{fo}
At atmospheric temperature, S_{fa}

Step-by-Step Design Procedure

Step 1.

a. Draw a sketch of the flange with dimensional details, including flange forces, and moment arms. It depicts the flange thickness as t.
b. Select gasket material and choose the gasket factors m and y.
c. Determine the gasket dimensions—internal diameter, d_i, and outside diameter, d_o, as follows: The inside diameter of the gasket (in inches) is normally taken as

$$d_i = B + 0.01$$

where B equals the shell inside diameter for a weld neck flange and shell outside diameter for a ring flange. From d_i, calculate d_o using the following formula:

$$\frac{d_o}{d_i} = \sqrt{\frac{y - pm}{y - p(m + 1)}}$$

where m is the gasket factor and y the minimum design seating stress.
d. Gasket width and gasket load reaction diameter:

Calculate the gasket width, N, given by

$$N = (d_o - d_i)/2$$

Calculate the basic gasket width, b_o.
Calculate gasket load reaction diameter, G:

G = mean diameter of gasket face (if $b_o \leq 0.25$ in)
= OD of gasket contact face-$2b$ (if $b_o > 0.25$ in)

Step 2.

a. Calculate bolt load for gasket seating condition W_{m2} and operating condition W_{m1}:

$$W_{m2} = \pi b G y$$
$$W_{m1} = H + H_p$$

where H is total hydrostatic end force and H_p total joint contact surface compression load. They are given by

$$H = \frac{\pi}{4} G^2 P$$

$$H_p = 2\pi b G m P$$

b. Calculate bolt cross-sectional area, A_m, required to resist the bolt load, which is the greater of

$$\frac{W_{m1}}{S_b} \quad \text{or} \quad \frac{W_{m2}}{S_a}$$

c. From A_m determine the number of bolts required (normally in multiples of 4), keeping the minimum bolt size as recommended in TEMA.

Step 3. Calculation of flange forces and their moments:

a. Calculated various flange forces:

$$H_D - \frac{\Pi B^2 P}{4}$$

$$H_T = H - H_D$$

For operating condition: $H_G = W - H$
For gasket seating condition: $H_G = W$

b. Determine the moment arms h_D, h_T, and h_G for flange loads under operating conditions from Table 13 for the three types of flanges.

c. Calculate M_D (moment due to H_D), M_T (moment due to H_T), and M_G (moment due to H_G):

$$M_D = H_D h_D \qquad M_T = H_T h_T \qquad M_G = H_g h_G$$

Step 4. Calculate flange moments.

a. For operating conditions, the total flange moment M_o is the sum of the three individual moments M_D, M_T, and M_G:

$$M_o = M_D + M_T + M_G$$

b. For gasket seating M_o' (ASME Code uses the term Mo for moment due to gasket seating condition also) as given by

$$M_o' = \frac{W(C - G)}{2}$$

Step 5. Calculate the parameters E and K, and hub factors T, U, Y, and Z as follows, or they may be read from Fig. 2-7.1 of the ASME Code.

$$E = 0.5(A - C) \qquad \text{(shown in Fig. 16.)}$$

$$K = \frac{A}{B}$$

$$U = \frac{K^2\left(1 + 4.6052\frac{1 + \nu}{1 - \nu}\log_{10} K\right) - 1}{1.0472(K^2 - 1)(K - 1)(1 + \nu)}$$

$$T = \frac{(1 - \nu^2)(K^2 - 1)U}{(1 - \nu) + (1 + \nu)K^2}$$

$$Y = (1 - \nu^2)U$$

$$Z = \frac{K^2 + 1}{K^2 - 1}$$

Formulas for factors F, V, F_L, and V_L pertaining to loose flange are given in Table 2-7.1

of the ASME Code. Otherwise, read from ASME Code Figs. 2-7.2, 2-7.3, 2-7.4, and 2-7.5, respectively.

Step 6. Calculate factor h_o:

$$h_o = \sqrt{Bg_o}$$

Step 7. Calculate factors d_1 and e.

$$d_1 \quad \text{(for loose flanges)} = \frac{U\,h_o\,g_o^2}{V_L}$$

$$\text{(for integral flanges)} = \frac{U\,h_o\,g_o^2}{V}$$

$$e \quad \text{(for loose flanges)} = \frac{F_L}{h_o}$$

$$\text{(for integral flanges)} = \frac{F}{h_o}$$

where g_o is the thickness of the hub at the small end.

Step 8. Determine flange thickness:

a. For weld neck integral flange and loose flange, calculate the flange stresses as detailed in step 10 and check if they are within limit (step 11). If not within the limit, increase the flange thickness and continue from Step 9 onward.
b. For ring flange, flange thickness required is the greater of gasket seating condition or operating condition, given by

$$T_f = \sqrt{\frac{M_o Y}{S_{fo} B}} \quad \text{or} \quad T_f = \sqrt{\frac{M_o' Y}{S_{fa} B}}$$

where Y is the ring flange shape factor.

Step 9. Calculate stress formula factors:

$$\alpha = te + 1$$

$$\beta = \frac{4}{3}te + 1$$

$$\delta = \frac{t^3}{d_1}$$

$$\gamma = \frac{\alpha}{T}$$

$$\lambda = \gamma + \delta$$

$$m_o = \frac{M_o}{B}$$

$$m_o' = \frac{M_o'}{B}$$

Step 10. Calculate flange stresses. Flange stresses shall be determined for the governing moment, namely, more severe of the operating or the bolting conditions.

For operating conditions, calculate:

a. Longitudinal hub stress, S_H:

$$S_H = \frac{f m_o}{\lambda g_1^2}$$

b. Radial flange stress, S_R:

$$S_R = \frac{(1.33te + 1)m_o}{\lambda t^2}$$

c. Tangential flange stress, S_T:

$$S_T = \frac{Y m_o}{t^2} - Z S_R$$

For the gasket seating condition, repeat the stress calculations replacing m_o by m_o'.

Step 11. Allowable flange stress: The stresses as calculated earlier are compared with the allowable stresses for the flange material at the design temperature S_{fo} (equation follows), and if required, the thickness can be modified. These stresses shall be calculated separately both for the gasket seating condition and operating condition. For the operating condition, the allowable stresses are given by

$$S_H = 1.5 S_{fo}$$
$$S_R = S_{fo}$$
$$S_T = S_{fo}$$
$$S_{fo} = \max|0.5(S_H + S_R), 0.5(S_H + S_T)|$$

For the gasket seating condition, the flange stresses are compared with the allowable stresses for the flange material at the atmospheric temperature S_{fa} (equation follows), and if required, the thickness can be modified.

$$S_H = 1.5 S_{fa}$$
$$S_R = S_{fa}$$
$$S_T = S_{fa}$$
$$S_{fa} = \max|0.5(S_H + S_R), 0.5(S_H + S_T)|$$

6 EXPANSION JOINTS

Expansion joints are promising for accommodating differential thermal expansion of heat exchanger shells, pressure vessels, and pipelines carrying high-temperature fluids. Differences in the axial expansion of the shell and the tube bundle due to high mean metal temperature differentials warrant incorporation of expansion joints in heat exchangers. This is particularly true for fixed tube-sheet exchangers. For fixed tube-sheet exchangers, when the difference between shell and tube mean metal temperatures becomes large (greater than approximately 50°C for carbon steel), the tube-sheet thickness and tube end loads become excessive [34]. Therefore, an expansion joint is incorporated into the shell. Expansion joints also find applications in floating head exchangers, in the pipe between the floating head cover and the shell cover to cushion the thermal expansion between the tube bundle and the shell. Figures 2a and 8 show expansion joints incorporated into a fixed tube-sheet exchanger.

6.1 Flexibility of Expansion Joints

Expansion joints used as an integral part of heat exchangers or other pressure vessels shall be designed to provide flexibility for thermal expansion and also to function as a pressure-retaining structural element. Hence, an expansion joint must compromise between two contradictory loading conditions [90]: (1) pressure-retaining capacity and (2) flexibility to accommodate the differential thermal expansion. In many cases, the design for a particular application will involve a compromise of normally conflicting requirements. For example, to retain a high pressure, usually a thick-walled bellows is required, whereas high flexibility and high fatigue life require a thin-walled bellows.

6.2 Classification of Expansion Joints

Expansion joints are broadly classified into two types:

1. Formed head or flanged and flued head
2. Bellows or formed membrane

Formed Head or Flanged and Flued Head

Formed head expansion joints, also called thick-walled expansion joints, are characterized by higher spring rates (i.e., force required for unit deflection of a bellow) and usually a lower cycle life than thin-walled bellows. Because of the higher wall thickness, this type of expansion joint is rugged and the most durable from the standpoint of abuse, but it has the disadvantage of very limited flexibility. Construction details of formed head expansion joints are discussed in refs. 41 and 50 and by Singh [91].

Formed head expansion joints are made in two halves from flat annular plates. The outside edges of the plates are formed in one direction (flanged), and the inside edges are formed in the other direction (flued). The two halves are welded together and then welded into the heat exchanger shell as shown in Fig. 18. A flanged and flued head expansion joint consists of the following elements:

1. An outer shell or outer tangent
2. Two outer tori
3. Two annular plates
4. Two inner tori
5. Two inner shells or inner tangents butt welded to the main shell on both sides

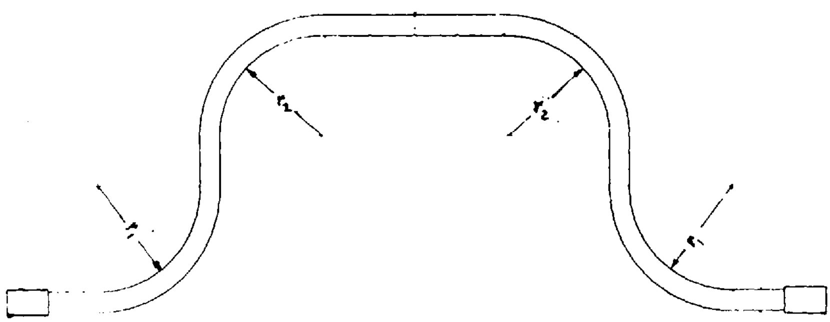


Figure 18 Flanged and flued expansion joint. (After Ref. 41.)

The inner and outer tori serve to mitigate the stress concentration due to geometric discontinuities between the shell and the annular plates. The radii of the tori are seldom less than three times the expansion joint thickness [41]. The annular plate contributes to lower the spring rate of the joint. Where the flexibility requirement is rather feeble, annular plate and the outer shell are eliminated. This type of construction results in the semitorus construction (Fig. 19a) [41,50]. The formed head type, i.e., without the inner tori, will lead to a flanged expansion joint as shown in Fig. 19b.

Bellows or Formed Membrane

According to EJMA Standards [8], a bellows type expansion joint is defined as a device containing one or more bellows used to absorb dimensional changes, such as those caused by thermal expansion or contraction of a pipeline, duct, or vessel. A bellows is defined as a flexible element of an expansion joint, consisting of one or more convolutions and the end tangents, if any. The bellows type expansion joint is also known as a "thin-walled expansion joint." The name "thin-walled expansion joint" is used to mean any form of expansion joint whose thickness is less than the thickness of the heat exchanger shell. A bellows type expansion joint is shown in Fig. 20.

Generally, the thin-walled bellows is formed from a thin plate whose thickness does not exceed $\frac{1}{8}$ in (3.2 mm) of corrosion-resistant material such as austenitic stainless steel or nickel-

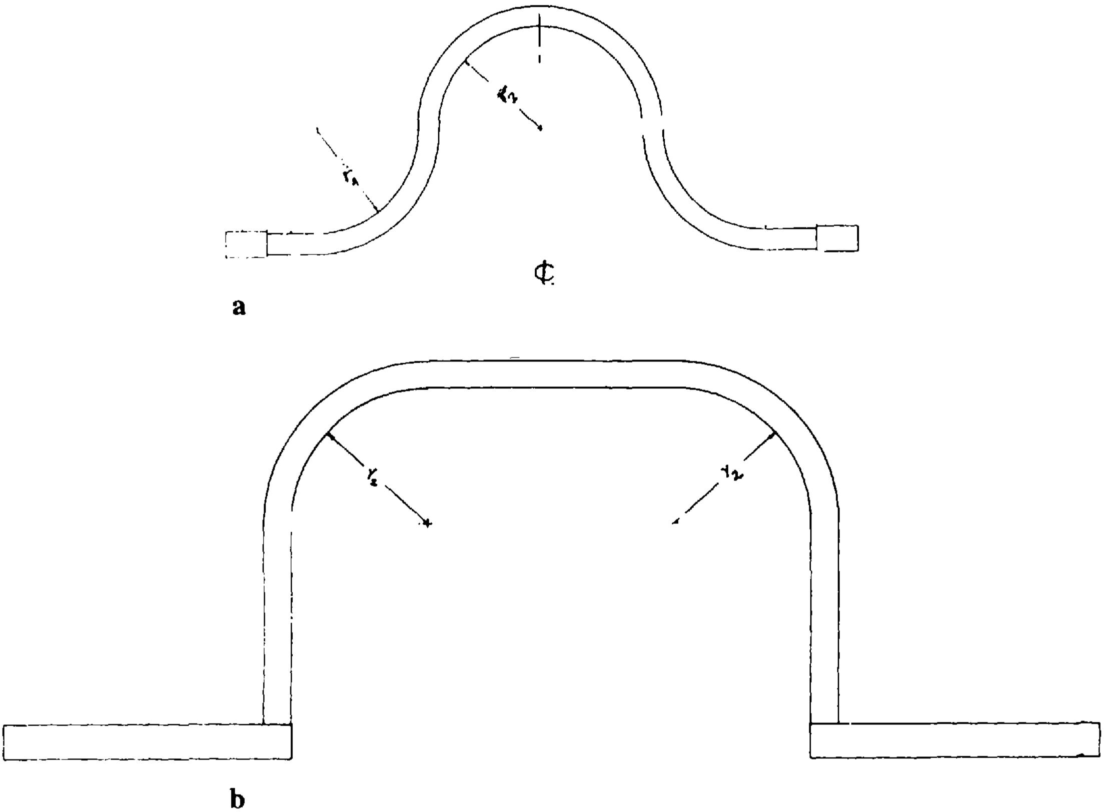


Figure 19 Modified version of flanged and flued expansion joint. (a) Semi-torus expansion joint; and (b) flanged expansion joint. (After Ref. 41.)

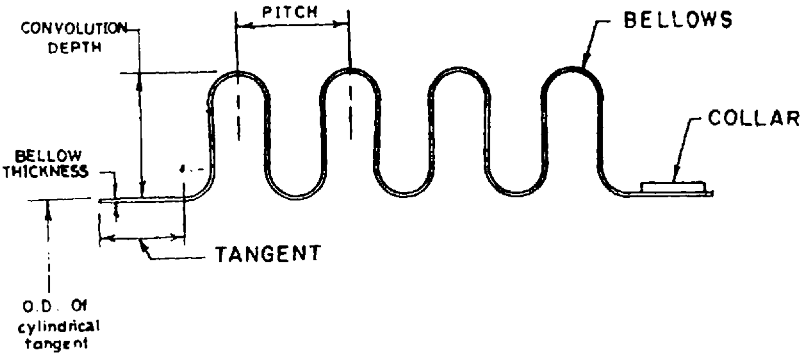


Figure 20 Bellows type expansion joint. (From Ref. 8.)

base alloys or high-alloy material using manufacturing processes like (1) disc or diaphragm forming, (2) elastomeric forming, (3) expansion forming, (4) hydraulic forming, and (5) pneumatic tube forming.

Bellows type expansion joints are made primarily for piping applications. When used in heat exchangers they are costly and delicate [90]. The cost is derived from the fact that such vessels require special sizes and material. They are said to be delicate because a heat exchanger containing a thin-walled joint must be handled and supported carefully in order to avoid damage, puncture, and buckling of the joint.

Deciding Between Thick- and Thin-Walled Expansion Joints

Conditions That Favor Using Thick-Walled Joints. Some of the conditions that favor thick-walled expansion joints are [29]:

1. Shell-side pressure is 300 psi (2070 kPa) or less.
2. Deflections per flexible element are moderate, in the range arbitrarily set at $\frac{1}{8}$ to $\frac{1}{4}$ inch (3.2 to 6.3 mm).
3. Application is noncyclical.
4. The joint must be capable of being vented and drained.

Conditions That Favor Using Thin-Walled Joints. Some of the conditions that favor thin-walled expansion joints are [29]:

1. Shell-side pressure exceeds 300 psi (2070 kPa).
2. Deflections per flexible element are high—arbitrarily greater than $\frac{1}{4}$ in (6.3 mm).
3. Application requires high cyclic life.

6.3 Design of Expansion Joints

Formed Head Expansion Joints

Kopp and Sayre [92] are generally credited for the first comprehensive work to determine analytically the axial stiffness of "flanged only" expansion joints. The method of analysis is based upon replacing the geometric configuration by an equivalent geometry. The outer torus (total length $\pi r/2$) is replaced by an equivalent corner end. One-half of the meridian of the outer torus is assigned to the annular plate and the other half to the outer shell. Figure 21 shows their idealized model. The structural characteristics of the annular plate are modeled by a unit width beam strip, the inner shell (heat exchanger main shell) is modeled using thin-shell

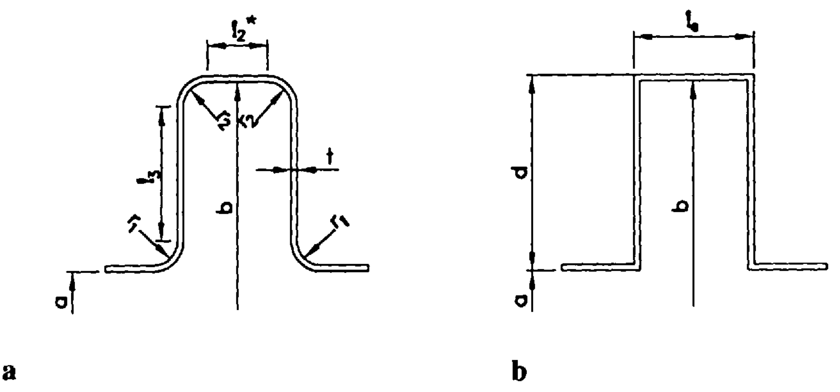


Figure 21 Kopp and Sayre model. (a) Actual model; and (b) idealized model. Note: Dimensions a and b are radii. (From Ref. 91.)

bending equations, and the outer shell is not modeled using classical thin shell equations but instead an approximate relationship is used. Kopp and Sayre also conducted some experimental tests to verify their mathematical model.

Wolf and Mains [90] applied finite-element analysis. The flanged and flued expansion joint is broken into its basic geometric components—a short cylinder, a toroidal segment, a flat annular plate, another toroidal segment, and a semi-infinite cylinder, as shown in Fig. 22. Their method did not attract wide usage, since the acceptance of a purely mathematical method is only a matter of time [41].

Singh and Soler [41] upgraded the Kopp and Sayre solution by using classical plate and shell solutions in place of "beam" solutions. This model suffers from the limitation of considering one standard expansion joint geometry only. In practical applications, myriad variations of the standard flanged and flued configuration are employed. Hence, Singh and Soler [91] present generalized treatment of various forms of flexible shell element (FSE) geometry, while retaining its modeling assumptions, which were directly borrowed from Kopp and Sayre. This model is included in the TEMA seventh edition.

Singh and Soler Model. The basics of the Singh and Soler [91] model are discussed next without the details of stress analysis and determination of spring constant. The most

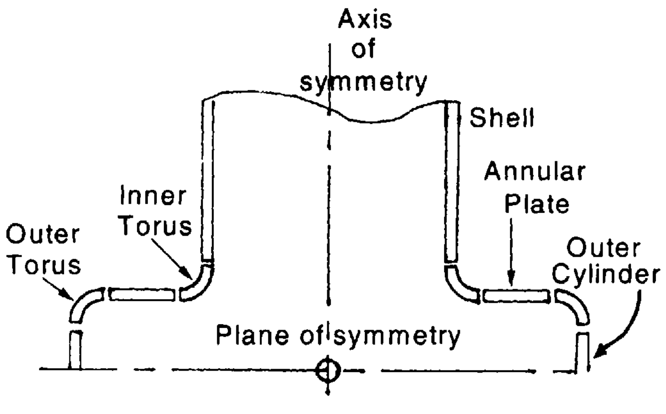


Figure 22 Finite element model for flanged and fluid expansion joint. (From Ref. 90.)

general form of the flexible shell element is shown in Fig. 23. The flexible shell element is one half of a standard expansion joint. Two flexible elements together make a standard expansion joint, such as the one shown in Fig. 18. The first step in the model simplification is to replace the circular segments with straight ones in the manner of Kopp and Sayer. By this step, the composite shells located at radii a and b are replaced by an equivalent shell of thickness t_1 and t_2, respectively, with modified flexural rigidities. Since the chief contribution of the tori lies in reducing in local bending stresses, which are secondary stresses, and since secondary stresses are of less importance, elimination of tori and replacement by sharp corners does not detract from the essence of developing a simple practical solution. The flexible shell element is reduced to two concentric shells of radii a and b connected by an annular plate. The Young's moduli of the three elements can be different. The resulting idealized model has the appearance of Fig. 23b. The three elements of the flexible shell element can be characterized as follows:

1. Inner shell of thickness t_1, equivalent Young's modulus E_1, equivalent length l_1, and radius a.
2. Outer shell of thickness t_2, equivalent Young's modulus E_2, length l_1, and radius b.
3. Annular plate of thickness t_e, with inner and outer radii a and b, respectively.

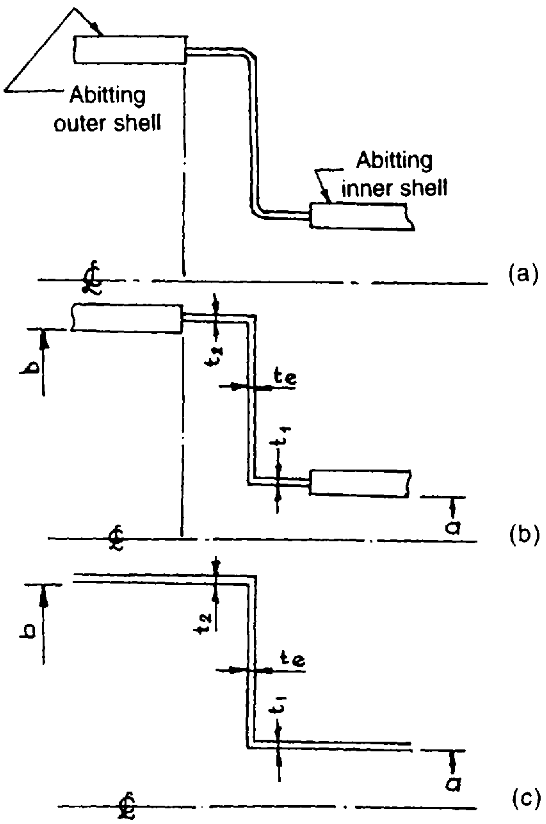


Figure 23 Singh and Soler (final) model for flanged and flued expansion joint. (a) General model; (b) equivalent Kopp and Sayre model; and (c) final idealized model. (From Ref. 91.)

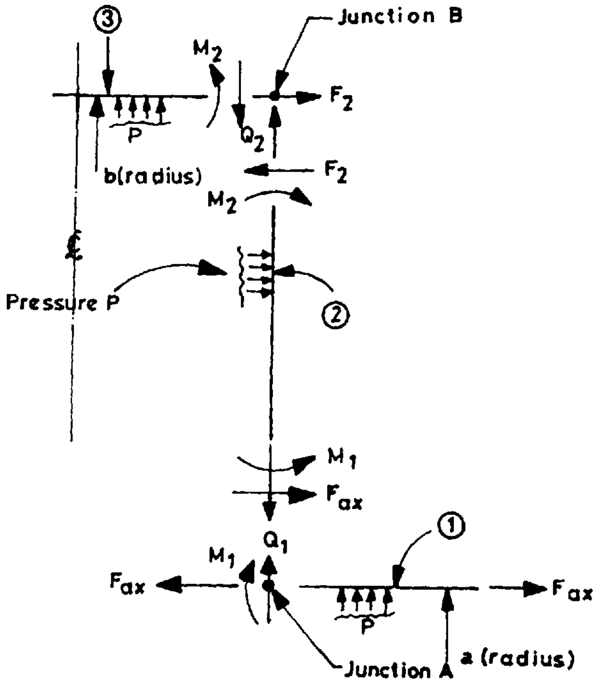


Figure 24 Loading and internal stress resultant acting on the flanged and flued expansion joint. (From Ref. 41.)

The equivalent lengths l_1 and l_2 warrant further comment. For the inner shell l_1 should be taken sufficiently long such that the edge effects (at the annular plate and shell junction) die out. Taking $l_1 = 2.5\ (at)^{0.5}$ will suffice, unless the shell is shorter, in which case the actual length should be used. Similarly, the length l_2 is actually the half-length of the top shell in the expansion joint.

Analysis for Axial Load and Internal Pressure. The resultant loading and internal stress acting on the elements along their inner junction A (the interface between the main shell element and annular plate) and outer junction B (the interface between the annular plate element and the outer shell) are shown in Fig. 24 and listed in Table 14.

Force Due to Internal Shell-Side Pressure. The equilibrium of one-half of the joint, in the axial direction, gives F_2 in terms F_{ax} [41]:

$$2\pi F_2 b = 2\pi F_{ax} a + \pi(b^2 - a^2)p_s \tag{114}$$

Equation 114 can be written as

Table 14 Loading and Internal Stress Resultant Acting on the Expansion Joint [41]

Joint A	Joint B
Moment M_1	Moment M_2
Applied axial load per unit circumference on the exchanger shell, F_{ax}	Applied axial load per unit circumference on the exchanger shell, F_2
Shell edge shear, Q_1	Shell edge shear, Q_2

$$F_2 = F_{ax}\frac{a}{b} + \frac{b^2 - a^2}{2b} p_s \tag{115}$$

The load deflection relations for short shell and annular plate elements to assemble the stiffness equations are derived in their work. They are not repeated here. Most of the formulas that are part of the TEMA procedure are arrived at after substituting $n = 0.3$ in the formulas of Singh and Soler [41] model.

Procedure for Design of Formed Head Expansion Joints

Rules for designing the formed head expansion joint currently exist in TEMA, ASME Code Section VIII, Div. 1, ANCC VSR1P, and A. D. Merkblatter, among others. HEDH [34] summarizes the salient features of flanged and flued type expansion joint design.

TEMA Procedure. The seventh edition (1988) of the TEMA Standards includes a new section RCB-8 on flexible shell elements (not light-gauge bellows type expansion joints), to be used in conjunction with fixed tube-sheet design. The paragraph encompasses several different shapes, such as flanged and flued heads, flanged only heads, and others. Also included is a method to calculate the maximum stress for cycle life calculations. The shell flexible elements shall be analyzed in both corroded and uncorroded conditions and shall be evaluated for hydrostatic test conditions also.

Minimum Thickness. As per TEMA RCB-8.9, the minimum thickness of the flexible shell elements shall be determined by the method of analysis. However, in no case shall the minimum uncorroded thickness be less than 3.2 mm (0.125 in) for nominal diameters up to 18 in, 4.8 mm (3/16 in) for nominal diameters in the range of 19–30 in (482.6–762 mm), or 6.35 mm (0.25 in) for nominal diameter greater than 30 in (762 mm). The industry practice is to set the FSE thickness one gauge less than the shell thickness [41]. When required, use more than one set of formed heads.

Allowable Stress (TEMA RCB-8.8). The allowable stresses in the flexible element, both in the corroded and uncorroded conditions, shall be as defined in the ASME Code using an appropriate stress concentration factor for the geometry.

Design Procedure as Per ASME Code

ASME Code Section VIII, Div. 1, does not give formulas for sizing the formed head expansion joints. However, rules are given for materials of construction, stress limits, cycle life calculation, fabrication, inspection and pressure test, stamping, and reports. The design of expansion joints shall conform to the requirements of Appendix CC of the ASME Code. Design aspects of multilayer, asymmetric geometries or loadings that differ from the basic concepts of Appendix CC are dealt in Paragraph U-2(g). Details of fabrication and inspection of formed type expansion joints are covered in Chapter 15, Heat Exchanger Fabrication. Stamping details are covered here.

Construction Materials and Minimum Thickness. According to ASME Code Section VIII, Div. 1, the materials for pressure retaining components shall conform to the requirements of UG-4. For thick-wall formed heads type expansion joints, in general, the bellows are of the same material as the shell.

Stamping and Reports. Details of stamping and reports are outlined in Section CC-6. As per this section, the expansion joint manufacturer shall have a valid ASME Code U certification of authorization and shall complete a Form U-2 Manufacturer's Partial Data Report, as required by UG-120(c). The Manufacturer's Partial Data Report shall contain data and information like:

1. Maximum allowable working pressure and temperature
2. Spring rate and axial movement
3. Service conditions
4. Design life in cycles
5. A certification that the expansion joint has been constructed as per the rules of Appendix CC
6. Details of the vessel manufacturer

6.4 Design of Bellows or Formed Membranes

Bellow type expansion joint design shall conform to the requirements of EJMA Standards, the ANSI Piping Codes, and the ASME Codes as applicable. The design of structural attachment shall be in accordance with accepted methods, based on elastic theory. In addition to EJMA Standards, design analysis and rules are also included in Appendix BB, ASME Code Section VIII, Div. 1, for circular type bellows with single-ply reinforced and nonreinforced bellows with thickness less than 3.2 mm (0.125 in).

Shapes and Cross Section

The bellows are available both for circular shells and rectangular shells. Rectangular shapes are used for surface condensers.

Bellows Materials

The bellows material shall be specified and must be compatible with the fluid handled, the external environment, and the operating temperature. Particular consideration shall be given to possible corrosion attack.

Bellows Design—Circular Expansion Joints

The design of bellows type expansion joints involves an evaluation of pressure-retaining capacity, stress due to deflection, spring rate, fatigue life, and instability (squirm). The spring rate is a function of the dimensions of the bellows and the bellows material. The determination of an acceptable design further involves the bellows parameters such as material, diameter, thickness, number of convolutions, pitch, height, number of plies, method of reinforcement, manufacturing technique, and heat treatment.

Limitations and Means to Improve the Operational Capability of Bellows

Single-ply bellows are used for low-pressure applications. They are fragile and hence they are easily damaged; external covers to protect personnel against the hazards of bellows blow-out due to failure are necessary. Drainable varieties are expensive, and external supports may be required to maintain alignment of the shell sections welded to the expansion joint [29]. Additionally, single-ply bellows are susceptible to instability. Since a bellows is a thin shell of revolution with repeated U-shaped convolutions, there exists a large number of natural vibration modes. Basically, these vibration modes are classified into three types: axial accordion modes, lateral bending modes, and shell modes, among which the former two are easily excited [93]. Methods and improved designs to overcome various shortcomings are discussed in the EJMA Standards. The following measures are normally adopted by designers to improve the single-ply expansion joint:

1. Use of external reinforcement
2. Use of multi-ply construction and thicker convolutions
3. Pressure balanced expansion joints

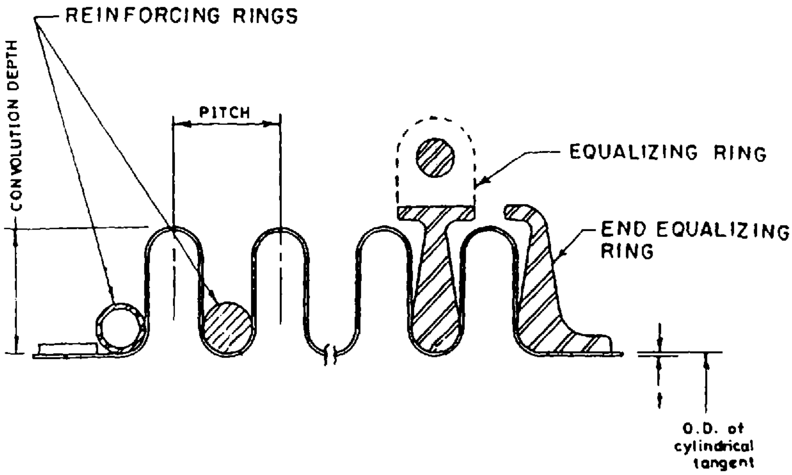


Figure 25 External reinforcement for bellows. (From Ref. 8.)

4. Flow sleeve inside the convolutions

External Reinforcement. A combination of high internal pressure-retaining capacity and large deflection can be achieved by external reinforcement of the U-shaped bellows. The external reinforcement offers circumferential restraint and supports the root radius against collapse from internal pressure loading. Reinforcing rings are also added where instability or squirm of the bellows is a concern. Equalizing and reinforcing ring devices used on some expansion joints fitting snugly in the roots of the convolutions are shown in Fig. 25.

Multi-Ply Construction and Thicker Convolutions. The pressure-retaining capacity of a bellows can be increased by the use of multi-ply construction and by increasing the thickness of the convolutions; however, the latter significantly reduces the bellows flexibility.

Pressure-Balanced Expansion Joints. The pressure balanced expansion joints (Fig. 26) are used for applications where pressure loading upon piping or equipment is considered excessive. The major advantage of the pressure-balanced expansion joint design is its ability to absorb externally imposed axial movement and/or lateral deflection while restraining the pressure thrust by means of tie devices interconnecting the flow bellows with an opposed bellows also subjected to line pressure. Their design should be as per EJMA Standards.

Flow Sleeve Inside the Convolutions. To overcome flow-induced vibration, install a sleeve inside the convolutions as shown in Fig. 27. In this case the bellow is thought to be two coaxial cylinders consisting of the convolutions and the sleeve, and the coupled vibrations through the fluid in the annular region may significantly affect the lateral vibration of the convolutions.

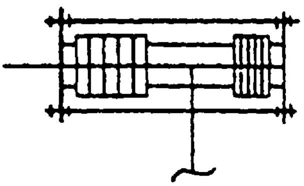


Figure 26 Pressure balanced expansion joint. (From Ref. 8.)

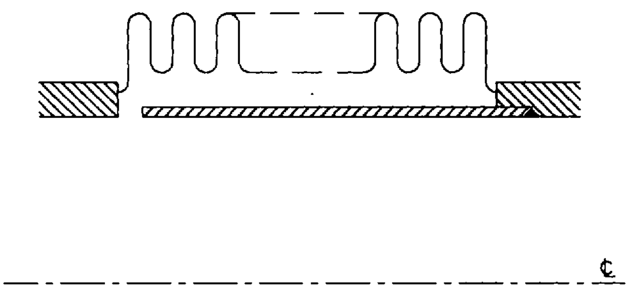


Figure 27 Flow sleeve to overcome FIV of bellow type expansion joint.

Fatigue Life

For a given bellows configuration and material thickness, the fatigue life of the bellows will be proportional to the imposed pressure and deflection. Depending on its material of construction, the suitability of an expansion joint to withstand the required number of cycles shall be determined from equations given in EJMA Standards or Appendix BB of the ASME Code.

7 OPENING AND NOZZLES

7.1 Openings

Openings in pressure vessels and heat exchangers refer to the cuts made in shells, flat covers, channels, and heads for accommodating the nozzles and to provide manholes, peepholes, drains and vents, instrument connections, etc. Openings can be circular, elliptical, or oblong. Whenever an opening is made in the wall of the shell or in the head, the wall is weakened due to the discontinuity in the wall and decrease in cross-sectional area perpendicular to the hoop stress direction. To keep the local stresses within the permissible limits, reinforcements to the openings are made.

Reinforcement Pad

The design of reinforcement is covered in UG-36 to UG-42 of the ASME Code by an area-to-area method. Reinforced pads whenever required as per drawings/codes shall be of the same material or equivalent to the component to which they are welded. Even though a reinforcement pad can be applied on either the outside or inside of the shell, it is the common practice to provide it at the outside due to easiness, and no need to meet the requirement of compatibility of the pad material with the process fluids, except that the pad should be resistant to general corrosion, and of weldable equality. The factors to be kept in mind while considering the reinforcement pad are:

1. The pad should match the contour of the component to which it should be attached.
2. Provide a tell-tale hole to release the entrapped gases during welding and to check the soundness of the welding.

Reinforced Pad and Air–Soap Solution Testing

As per ASME Code UW-15, reinforcing plates and saddles of nozzles attached to the outside vessel shall be provided with at least one tell-tale tapped hole (maximum size NPS 1/4 tap) for compressed air–soap solution test for tightness of welds that seal off the inside of the vessel. Air pressure of 1.25 kg/cm^2 is suggested for these tests. Higher test pressures are not recommended because the soap bubbles have a chance to blow off. Tell-tale holes in the reinforcing pads may be left open or plugged when the vessel is in service.

7.2 Nozzles

Nozzles are incorporated to convey process fluids into the heat exchanger and out of it. Their sizes are arrived after calculating permissible fluid velocity limited by erosion–corrosion, impingement attack, pressure drop, etc. Minimum wall thickness is arrived using the cylindrical shell formula. Good nozzle design involves better distribution of process fluids, ability to withstand operating load and the other loads, and should provide easy accessibility to connect or disconnect the pipes. A well designed nozzle should have a very low pressure drop. Nozzle openings can be circular, elliptical, and oblong. Nozzles are connected by weldment to the shell by

1. Butt welding
2. Through type
3. Reinforcing pads

In addition to the welded type connections, brazed, threaded, studded, and expanded connections are also employed. Nozzle design is carried out as per Code. Considerations in nozzle design should include the inspectability of the nozzle-to-pipe and nozzle-to-vessel welds inspection [94]. Figure 28 shows some design types with reference to inspectibility.

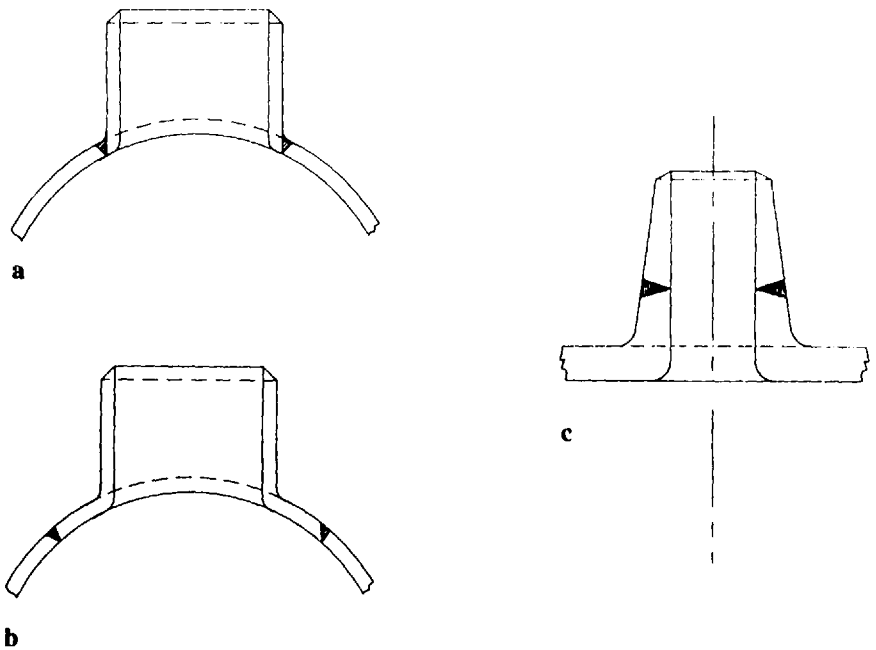


Figure 28 Nozzle design types. (a) Poor inspectiability; (b) better inspectiability; and (c) better than (b), which makes ultrasonic examination easier. (From Ref. 94.)

Reinforcements

Nozzle openings are reinforced by the following means:

1. Using thick forged-blank nozzle (Fig. 29a)
2. Opening compensated by reinforcement pad (Fig. 29b)
3. Welding of thick-walled nozzle pipe (Fig. 29c).

Design aspects of various nozzle reinforcements are discussed by Schoessow et al. [95]. Requirements of reinforcement for openings in shells and formed heads are covered in UG-37 to UG-42, UG-82, and attachment welds in UW-15, and exemption from reinforcement in UG-36. As far as possible, nozzle design should avoid the separate reinforcement plate being welded to the shell, because the weld metal cracks at the interface between the reinforcement pad plate and the shell plate pose additional problems.

7.3 Stacked Units

Stacked units with interconnecting nozzles are a source of many problems for the designer as well as the fabricator. Most of the trouble comes as a result of differential thermal expansion, either radial or longitudinal or both. Several general rules will help avoid trouble [96]:

1. Do not stack one-pass shells more than two deep, without thorough check of differentials.
2. Keep intermediate shell nozzles as near channel nozzles as possible.
3. Avoid ring-type joint intermediate nozzles, if possible.
4. Avoid offset direct interconnecting nozzles.

8 SUPPORTS

All vessels shall be supported and the supporting members shall be attached to the vessel wall. The design of supports shall normally conform to good engineering practice. The supports should be designed to resist internal and external pressure and accommodate the self-weight of the unit and contents, including the flooded weight during hydrostatic test. Based on their installation, the supports differ for horizontal vertical installation. The selection of the type of

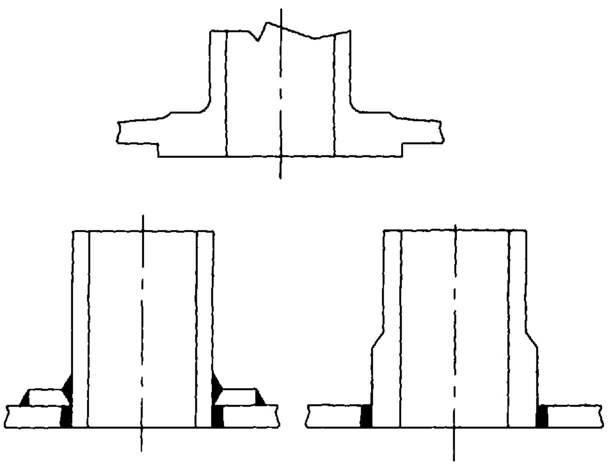


Figure 29 Nozzle opening reinforcement. (a) Thick forged blank nozzle; (b) compensation by reinforcement pad; and (c) thick walled nozzle.

support for a pressure vessel is dependent on parameters such as the elevation of the vessel from the ground level, the materials of construction, and the operating temperature [44].

8.1 Design Loads

While designing the supports of a vessel, care should be taken to include all the external loads likely to be imposed on it. Such external loads include: (1) wind loads, (2) loads due to connected piping, (3) superimposed loads, (4) shock loads due to surging or hydraulic hammer, and (5) seismic vibration. As per TEMA Standards, supports for a removable tube bundle heat exchanger should be designed to withstand a pulling force equal to 1.5 times the weight of the tube bundle, and when additional loads and forces from external nozzle loadings, wind loads, and seismic forces are assumed for the purposes of supports design, the combinations need not be assumed to occur simultaneously. Care should be taken that the thermal stresses in external supports do not exceed those permitted by the code.

8.2 Horizontal Vessel Supports

Horizontal vessels are subject to longitudinal bending moments and local shear forces due to the weight of their contents. They are generally supported by three types of supports: (1) saddle supports, (2) ring supports, and (3) leg supports. Saddle support is used most commonly for heat exchangers. It is shown in Fig. 30. Whenever possible, horizontal vessels shall be supported by two supports only, with holes for anchor bolts. If more than two supports are used, the distribution of the reaction is affected by difference in support level, the straightness and local roundness of the vessel, and the relative stiffness of different parts of the vessel against local deflection [26].

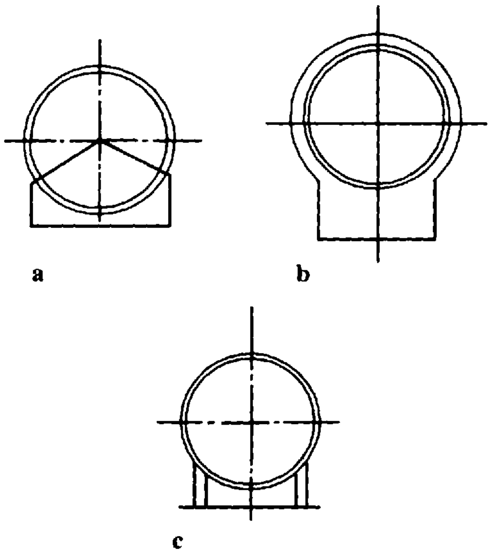


Figure 30 Examples of horizontal supports of pressure vessels. (a) Saddle support; (b) ring support; and (c) lug support.

Saddle Supports

Saddle supports may be used for vessels whose wall is not too thin. Horizontal vessels when supported on saddle supports such as in Fig. 31a behave as beams, and with these kinds of supports, the maximum longitudinal bending stresses occur at the supports and at the mid span of the vessel (Fig. 31c). Hence the location of supports from the mid span of the vessel or head tangent is critical to minimize the bending stresses at the supports. Consideration shall be given to ensure that the saddles should be preferably extended over at least 120° of the

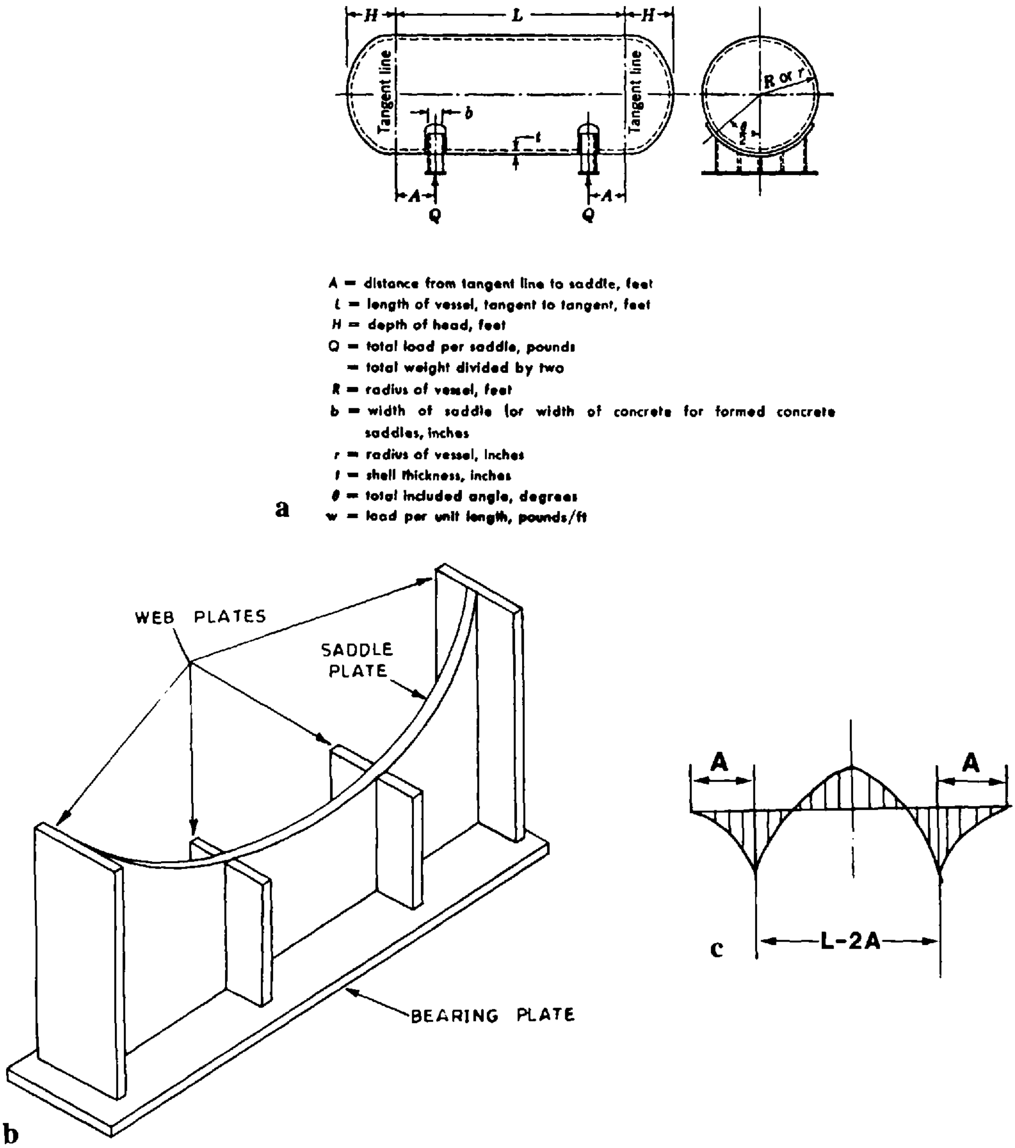


Figure 31 A vessel on horizontal saddle support. (a) Schematic; (b) details of the saddle support; and (c) bending stress distribution. (Item (a) is reproduced from Ref. 44.)

circumference of the vessel. The limitation, which is imposed by most codes of practice is an empirical one based on experience with large vessels [26].

Zick Stress. Zick [97] developed a method for analyzing supports for the horizontal cylindrical shells. The analysis gives a detailed derivation of the equations for longitudinal bending stresses at the supports and at the mid span. These stresses are named as Zick stress. Zick's method is discussed in detail in refs. 41, 42, and 45, among others. Zick's method is adopted in codes such as IS:2825–1969 and BS 5500.

Ring Supports

Ring supports as shown in Fig. 30b are preferred to saddle supports for large thin-walled vessels, vacuum vessels, and in the case of saddles located away from the head. Ring supports are also preferred when to support a vessel at more than two cross sections becomes inevitable. The welds attaching ring supports should have a minimum leg length equal to the thickness of the thinner of the two parts being joined together.

Leg Supports

Leg supports as shown in Fig. 30c are usually permitted for small vessels by the usual code practice because of the severe local stresses that can be set up at the connection of the support to the vessel wall.

8.3 Vertical Vessels

Supports for the vertical units may be skirt supports, ring supports, and lugs (columns). Some of these vertical supports are shown in Fig. 32.

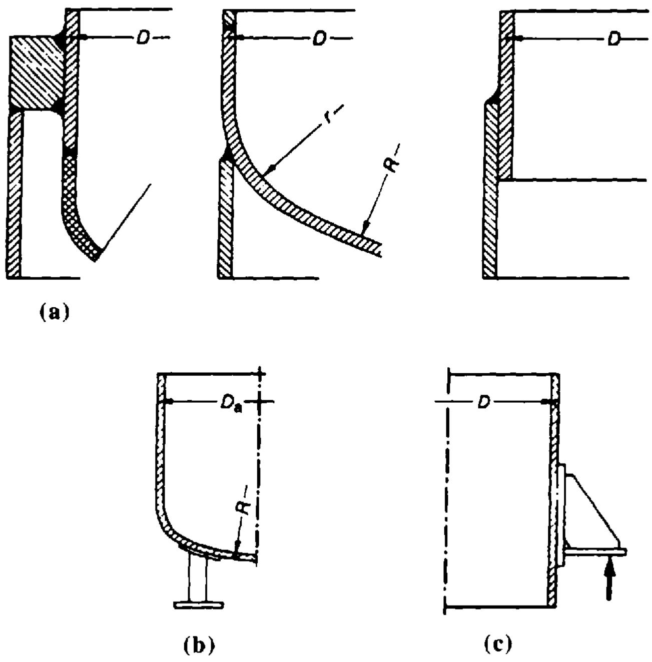


Figure 32 Examples of vertical supports of pressure vessels. (a) Skirt supports, (b) lug support, and (c) ring support. (From Ref. 40.)

Skirt Supports

Skirt supports (Fig. 32a) are recommended for large/tall vertical vessels. Skirt supports are preferred because they do not lead to concentrated local loads on the shell, offer less restraint against differential thermal expansion, and reduce the effect of discontinuity stresses at the junction of the cylindrical shell and the bottom [26]. The skirt supports shall be provided with at least one opening for inspection unless there is a provision to examine the bottom of the vessel accessible from below.

Lug Supports

Vertical vessels may be supported by a number of posts or lugs as shown in Fig. 32b. Lug supports are ideal for thick-walled vessels. For thin-walled vessels, it is not convenient unless proper reinforcements are used or many lugs are welded. Brackets or lugs offer many advantages over other types of vessels [41]: They are inexpensive, can absorb diametrical expansion by sliding over greased or bronze plates, and requirements of welding are minimal.

8.4 Procedure for Support Design

TEMA Rules for Supports Design (G-7.1)

TEMA rules for supports for horizontal units are listed in G-7.11 and for vertical units in G-7.12. For calculating resulting stresses due to the saddle supports, references are suggested under TEMA G-7.13. The "Recommended Good Practice" section of TEMA Standards provides additional information on support design.

ASME Code

ASME Code requirements for supports design are covered in UG-54. Appendix G contains suggested good practices for support design.

8.5 Lifting Devices and Attachments

TEMA rules for the design of lifting devices are given in G-7.2. ASME Code rules for the construction of lifting devices and fitting attachments are covered in UG-82. Some of the TEMA Standards for design of lifting devices are as follows:

1. Channels, bonnets, and covers that weigh more than 60 pounds are to be provided with lifting lugs.
2. Lifting devices are designed to lift the component to which they are directly attached. When lifting lugs are required by the purchaser to lift the complete unit, the device must be adequately designed.
3. The design load shall incorporate an appropriate impact factor.
4. Lifting devices and attachments shall be formed and fitted to conform to the curvature of the component surface to which they are attached.

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