



Training Services

Pressure Vessels

Purpose

Introduction of the governing codes and basic considerations and concepts of pressure vessel design, fabrication, inspection, and modification.

Pressure Vessel vs Piping

- **Pressure Vessel** - A container in which an *occurrence* takes place at a different pressure than atmospheric
- **Piping** - A container used for *conveyance* or control (valves)



Outline

- **Process Engineer Responsibilities**
- **Pressure Vessel Geometry and Heads**
- **Codes and Standards**
- **Evaluation Methods (nondestructive examination)**
- **Fabrication and Welding**
- **Testing**
- **Support**
- **Revamps**
- **Stress and Strain**
- **Stress Analysis and Code Rules**
- **Wind and Seismic Loading**

Process/Project Engineer Responsibility

Process Design Conditions

- **Design Pressure**
- **Design Temperature**
- **Vessel Size and Orientation**
- **Metallurgy**
- **Nozzle Sizes and Location**
- **Vessel Elevation**
- **Internal Requirements**

Mechanical Design Features

- **Vessel Thickness**
- **Heads**
- **Shell**
- **Vessel Support**
- **Nozzle and Manway Details**

Mechanical Design Features

(continued)

- **Fireproofing/insulation**
- **Internals, Including:**
 - **Distributors**
 - **Vortex Breakers**
 - **Grids**
 - **Trays**
 - **Centerpipes and Scallops**
 - **Mesh Blankets**

Process Design Considerations

Pressure Nomenclature

- **Normal Operating**
 - **Pressure at which equipment operates**

- **Maximum operating**
 - **Highest operating pressure foreseen for all applicable cases (normal, turndown, startup shutdown)**

- **Design Pressure**
 - **Maximum operating pressure plus a safety margin**

Process Design Conditions

Determine Design Pressure

Maximum Operating Pressure, psig	Design Pressure, psig
Less than 25	50
25 to 250	Oper P + 25
250 to 1000	(Oper P) * (1.1)
More than 1000	(Oper P) * (1.05) (*)
(*) Applicable only if pilot operated relief valves are used, otherwise use a 10 percent margin	

Process Design Conditions

Exchanger Design Pressure

- **Design pressure is normally determined by the preceding guidelines**
- **To avoid the need for an additional relief valve, the low pressure side may be designed for 10/13 of the high pressure side design pressure**

Process Design Conditions When Vacuum Design is Specified

- **Equipment that operates under vacuum (including startup and shutdown)**
- **Equipment is subject to vacuum during drainage**
- **Where loss of reboiler or other heat to a gas with a resultant cooling, even condensation, can result in a vacuum**
- **Operator error normally not considered**
- **Can design equipment for both internal and external pressure**
- **UOP designs for full vacuum if any vacuum is possible**



Process Design Considerations

Effect of Pressure Drop on Mechanical Design

- **Design pressure is at the top of the vessel in its operating position**
- **Mechanical design conditions at the bottom should consider:**
 - **Liquid head**
 - **Upflow or downflow pressure drop**
 - **Hydrostatic test conditions**

Process Design Conditions

Temperature Nomenclature

- **Normal Operating**
 - **Highest temperature expected during the equipment's operating cycle, including start and end of run.**
- **Design Temperature**
 - **Normal operating temperature plus a margin**
- **If operation is cryogenic (cold), the margin is a minus value (usually -25 °F). Alternative margins may be considered where the metallurgy is affected.**

Design Temperature

■ **Maximum**

- **Mean metal temperature based on highest expected operating conditions**

■ **Minimum**

- **Mean metal temperature—considering lowest operating, operational upsets, auto-refrigeration, atmospheric temperature, and many other sources of cooling**

Design Temperature

(continued)

- **Zones with different metal temperatures are allowed.**
- **Based on the minimum temperatures, impact testing may be required.**
- **Consider the effect of elevated design temperature on the allowable design stress. Due to creep considerations, the allowable stress can drop rapidly at elevated temperatures.**

Process Design Considerations

Determine Design Temperature

Normal Operating Temperature, °F	Design Temperature, °F
Less than 200	250 *
More than 200	Operating Temperature + 50

* 150 °F when caustic is present and the operating temperature is 100 °F, or less

Process Design Considerations

Special Cases for Design Temperature

■ Fractionators

- **Design temperature normally constant top to bottom, based upon the highest operating temperature (which is generally at the bottom)**
- **Graduated for large delta T's when the higher design temperature is greater than 650°F**

■ Cooler Failure

- **Failure of coolers upstream of equipment could require a greater margin than 50 °F**

Process Design Considerations

Special Cases for Design Temperature (continued)

- **Heat Exchanger Shells**
 - **Use higher of the inlet or outlet**
 - **Graduate if change in metallurgy possible on large exchangers**
- **Cold Wall Design**
 - **Internally insulated vessels allow lower shell design temperature and possibly a lower and less expensive metallurgy**
- **Flange Classes**
 - **Watch the effect on the flange class when setting the design temperature**

Process Design Considerations

Special Cases for Design Temperature (continued)

- **Short Term Elevated Temperature**
 - **Use a reduced margin (or no margin) when the maximum temperature is a short term condition (e.g., end of run (EOR)) only and is in the creep range of the material(s)**
 - **In the creep range, the allowable stress drops rapidly**
 - **Creep is time dependent and not generally significant in the short term**

- **Design codes do not require or give guidelines for temperature or pressure design margins**

Specified Design Conditions

- **The specified design conditions are those resulting in the most severe head/shell requirements**
 - **Generally the greatest temperature and greatest pressure**
- **If the greatest temperature and pressure do not act simultaneously, the governing case may not include either or both**
- **Different portions of the equipment may have different design conditions**
 - **Consider need to accommodate pressure testing**

Overall Geometry

- **The sphere is the most economical shape for pressure retention**
 - **Used for some gas storage vessels, particularly high pressure**
- **For process equipment, the need to fabricate and install internals, distribute and collect process material, and control the process leads to the need for a consistent cross-section rather than the constantly varying cross-section of a sphere**



Overall Geometry

(continued)

- **Plot space restrictions (i.e. “footprint”) also make a sphere less attractive**
- **Fabrication costs may offset sphere’s material thickness savings**
- **Shape of choice for process equipment is a cylinder**
- **Most vessels are oriented vertically unless there is a specific (process) reason to be placed horizontally (e.g., gravity separators)**

Overall Geometry

(continued)

- Vessel dimensions and orientation are controlled by process requirements (e.g., space velocity, fluid distribution, catalyst contact, residence time, tray design and spacing, etc.)
- Cylinder length to inside diameter ratio of 3 or 4 is typically used
 - Provides good mix of inside volume, cross-section area, and vessel cost (e.g., wall thickness)
- Minimum shell thickness, in inches, of $(D+100)/1000$ is provided for structural stability
 - D is the inside diameter, in inches

Overall Geometry

(continued)

- **Corrosion/erosion allowance is usually provided on the thickness**
 - **Determined based upon internal atmosphere**
 - **Is usually 1/16 to 1/8 inch (1.5 to 3 mm)**

- **Inside diameter and length dimensions are set to increments of 6 inches or 100 mm**
 - **Matches commonly available head sizes and “can” lengths for the shell**

Tangent and Weld Lines

- **Tangent Line**

- **Point at which the head curvature begins**

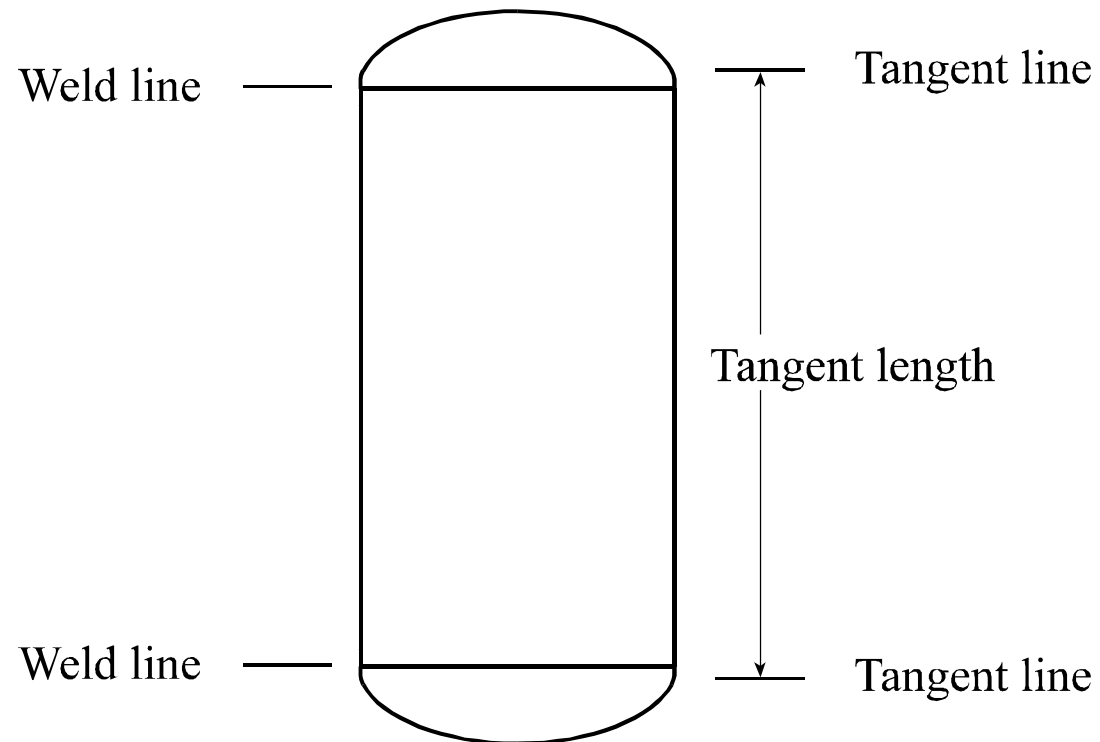
- **Weld Line**

- **Point at which the head and shell are welded together**

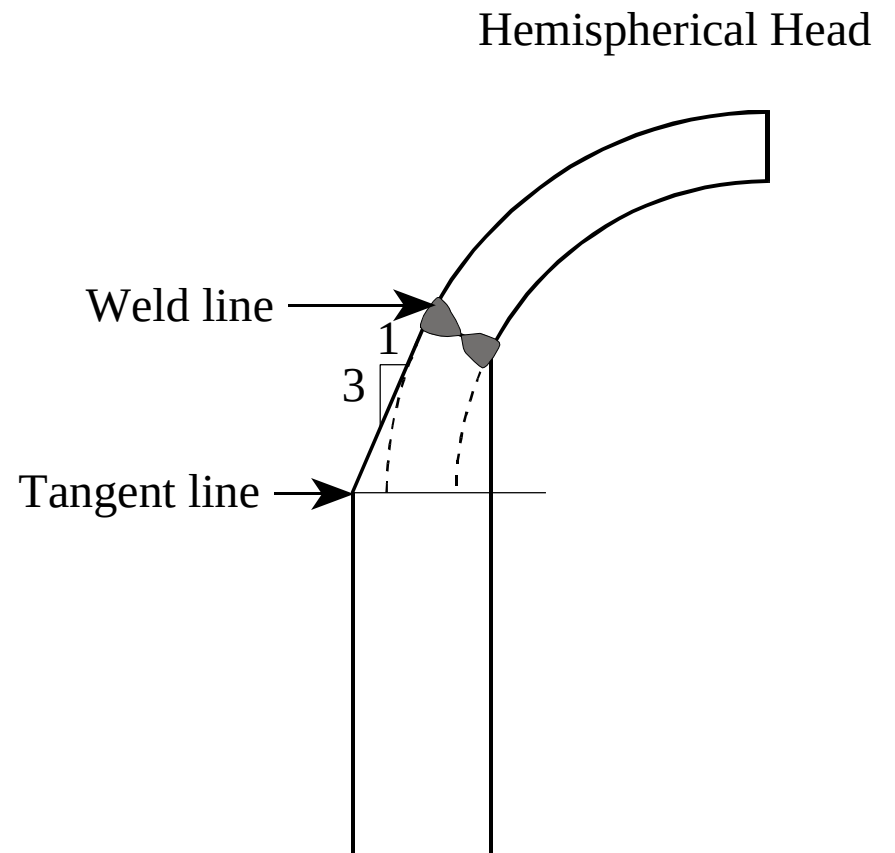
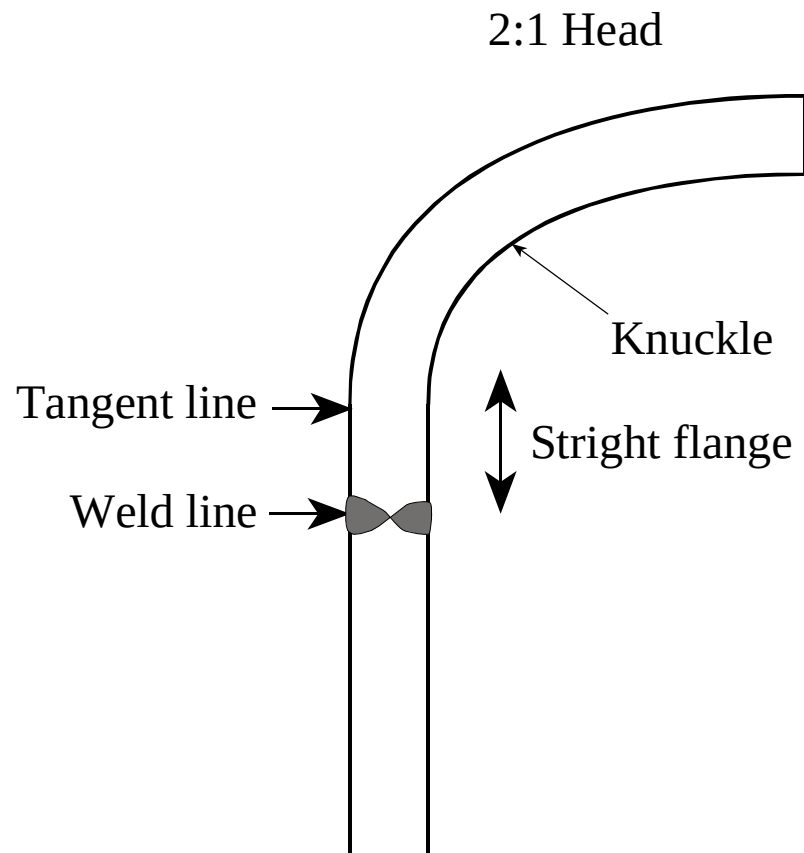
The weld line is very rarely the same point as the tangent line. This moves the weld to a point where fit is easier (e.g., both sections are cylindrical) and away from any stress concentrations present at the geometrical joint.

Tangent and Weld Lines

Overview



Tangent and Weld Lines Detail



Common Head Styles

- **Hemispherical**
- **Elliptical**
- **Conical**
- **Flanged and Dished**
- **Torispherical**
- **Flat**

Hemispherical and 2:1 Elliptical are the most common.

Hemispherical versus 2:1 Elliptical Heads

■ Hemispherical

- Optimal pressure containing shape
- Half as thick as the shell
- No sharp radius bends (e.g. knuckles) or stress concentration points
- Minimizes thinning, cracking, and compression concerns
- Entire head is at one smooth, constant, curvature

Hemispherical versus 2:1 Elliptical Heads (continued)

- **Hemispherical (continued)**
 - **Joint with the shell is more complex**
 - **Greater contained volume than 2:1 elliptical**
 - **More surface area than 2:1 elliptical**
 - **More difficult to form or fabricate, fewer potential vendors**
 - **Suitable for thick shells (> 2 inches) (from a cost viewpoint)**
 - **Often fabricated rather than formed in one piece**

Hemispherical vs. 2:1 Elliptical Heads

(continued)

■ **2:1 elliptical**

- **Three dimensional elliptical geometry**
- **Depth equals 1/2 the vessel radius**
- **Same thickness as the shell**
- **Easy butt weld detail at joint with the shell**
- **Commonly available**
- **Less volume and surface area than hemispherical**
- **Knuckles are in hoop compression**
- **Suitable for thin shells (< 2 inches) (from a cost viewpoint)**

Nozzle Details

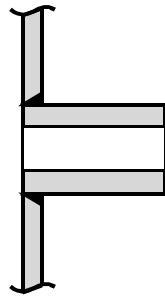
- **Although nearly any orientation is possible, for ease of design and reinforcement, nozzles should be perpendicular to the shell**
- **Although not prohibited by codes, avoid locating nozzles in or near vessel weld seams**
 - **Nozzle and any reinforcement will interfere with the ability to inspect and NDE the vessel weld**

Nozzle Details *(continued)*

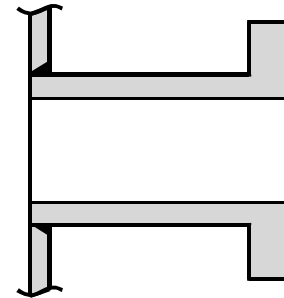
- **Locate nozzles so nozzle and its reinforcement are located within 80% of the head diameter**
- **Nozzle to shell welds are difficult to examine, especially to radiograph, because of the difficulty in accessing welds between two components at a right angle and the interference in the readings caused by the geometrical changes**

Vessel Fabrication Nozzles

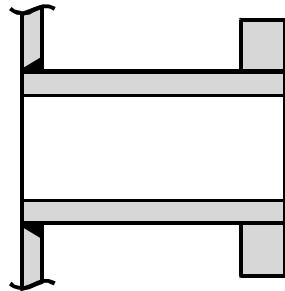
A. Pipe Couplings - Generally Avoided



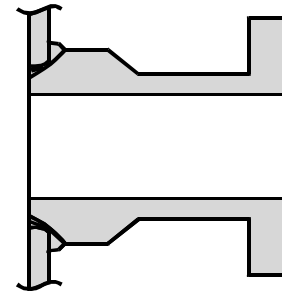
B. Forged Steel Nozzles



C. Built-up Nozzles



D. Integrally Reinforced Nozzles



Nozzle Details

(continued)

- **Nozzle to shell joint geometry (e.g., sharp corners, sudden thickness and geometrical changes) causes stress concentrations**
- **Welding effects (heating, cooling, metallurgical changes, heat affected zones) and geometric constraints also cause residual stress concentrations**

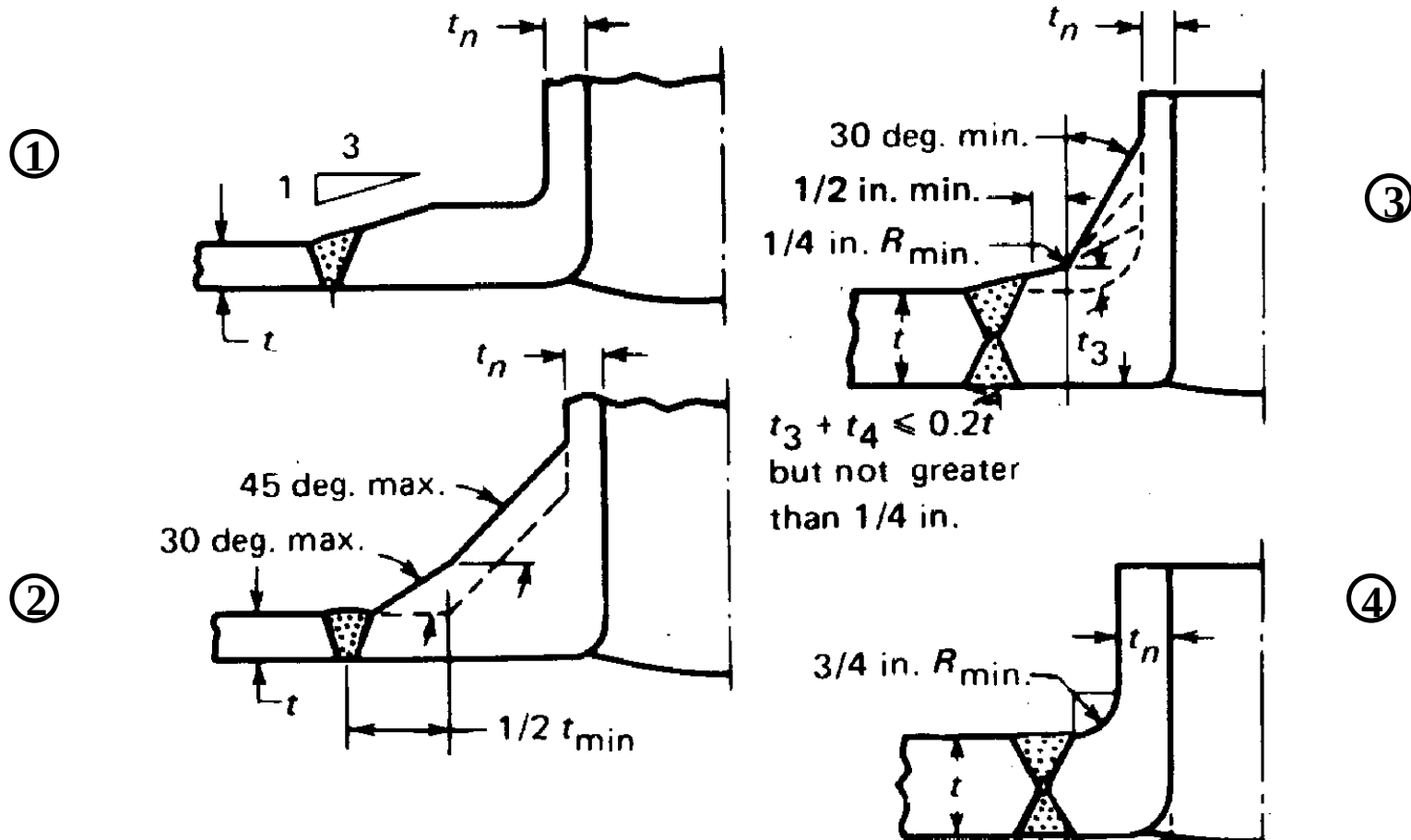
Nozzle Details *(continued)*

- **To minimize effects of stress concentrations and examination difficulty, flared nozzles are sometimes used for high pressure, cyclic, or elevated temperature (creep range) service**
- **This detail moves the weld away from the geometry discontinuities and creates an easier to perform butt weld to the shell, with probable improved weld quality**

Nozzle Details *(continued)*

- Examination of the weld becomes easier and the geometrical stress concentrations are moved from the weld HAZ and are not additive to the stress concentrations/residual stresses due to welding
- A smoothly contoured detail, free of stress concentration points, is more reliably made from a forging than grinding a confined weld
- Flared nozzles are more expensive to produce

Flared Nozzles



Nozzle Details

(continued)

- **Nozzle attachments may be through the shell or butt welded to it**
 - **Through shell**
 - **Welding may be performed and examined from both sides; NDE is easier**
 - **Nozzle ID forms a uniform diameter, smooth, unbroken single metallurgy surface through the shell**

Nozzle Details

(continued)

- **Through shell (continued)**
 - **For thick shells, heat of welding may warp nozzle; may be impractical for small nozzles in thick shells**
 - **Requires weld preparation of the shell plate (e.g., beveling)**
 - **Connection tends to be stronger. Weld is placed into shear by tension, bending, compressive, or torsional loads.**

Nozzle Details

(continued)

- **Butt weld to shell surface**
 - **Smaller weld, less distortion possibility**
 - **Shell laminations are a concern, especially if external loads are present**
 - **Access to the weld (for back welding or NDE) from inside the nozzle may be impossible**
 - **Inner surface of the nozzle is broken; shell opening must match nozzle ID**
 - **Connection tends to be weaker because the weld is in tension due to tensile or bending loads.**

Nozzle Neck Thickness

- **Greater of:**
 - **A) Minimum thickness required for the nozzle cylinder by the code design equations for pressure plus external loads, plus corrosion**
 - **B) Smaller of**
 - **Minimum thickness of standard wall pipe plus corrosion**
 - **Vessel shell or head thickness required for pressure, plus corrosion**

Codes and Standards

The rules found in the design codes represent many man-years of experience. If used wisely, the code requirements can:

- **Communicate design requirements**
- **Utilize know-how and technology**
- **Keep equipment costs low**
- **Reduce insurance costs ***
- **Reduce chance of legal entanglements ***

*** Due to the use of standard, recognized, design methods and components.**

Design Codes

- **Provide rules for the design of equipment adequate for design conditions determined by others**
- **Do not provide rules or guidance for the determination of design conditions**
- **Do not provide rules or guidance for the determination of the required material(s) of construction or corrosion allowance**

Design Codes

(continued)

- **Tolerances included in design codes are intended to insure the rules and design methods are applicable (e.g. the vessel is essentially circular)**
 - **They do not insure the equipment is suitable for the desired use or near the specified dimensions**
- **Defined scope of most design codes includes new construction only, not revamps, repairs, or rerates**

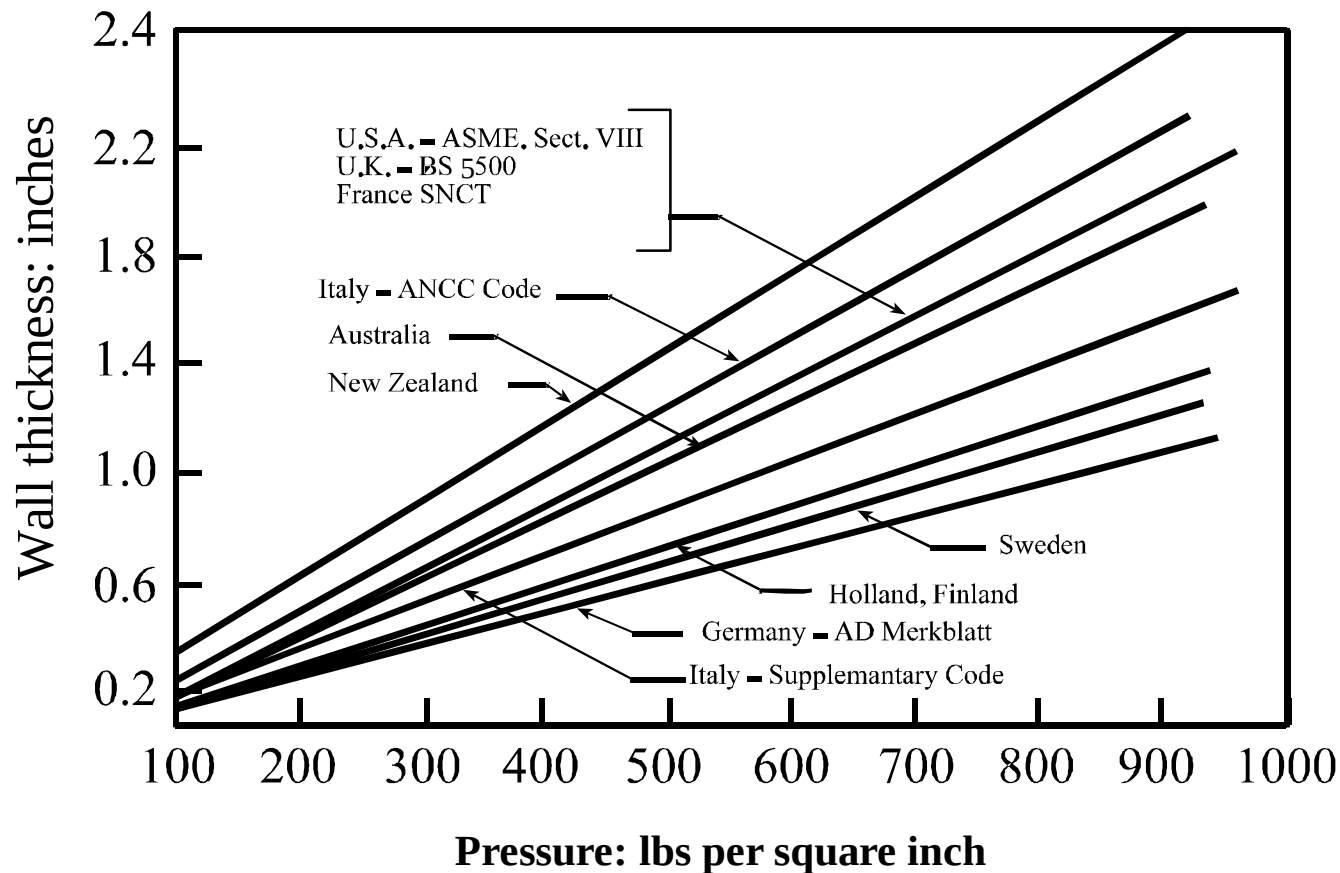
Design Codes *(continued)*

- **Laws and regulations in force at the site determine the Code that must be used.**
- **Laws and regulations may also specify the edition of the Code and could limit use of referenced or auxiliary documents (e.g., Code Cases).**

Code Use

- **Provisions of a design code are an interrelated set of design, fabrication, inspection, and testing requirements. For example, the use of a higher design stress may depend upon use of stringent material, analysis, examination, and testing requirements. Therefore, different codes can arrive at different resulting wall thickness yet have equivalent degrees of reliability (see following slide). Because the provisions are interrelated, any selected code must be used in its entirety. Provisions cannot be mixed from different codes. Use of particular codes is generally written into the national or local laws of the plant site.**

Comparative Wall Thickness Requirements in Various Countries



**Welded Cylindrical Carbon-Steel Shell, 60-inch diameter
100% Radiography**

Codes and Standards

ASME Section I

- **Used for steam generating equipment and certain auxiliary equipment and piping**
- **Often used for power plants that cannot afford to be “down”; therefore, design a little more conservatism into them**
- **Uses factor of safety of 3.5**
- **Maximum joint efficiency of 0.9**
- **More expensive than Section VIII, Division 1**

Codes and Standards

ASME Section VIII, Division 1

- **Used for most unfired refinery equipment**
- **Uses factor of safety of 3.5 against tensile failure and 1.25 for 100,000 hour creep rupture**
- **Limited to 3000 psi (less as a practical matter)**
- **Rigorous evaluations of local, thermal, and fatigue stresses are not usually explicitly performed**

Scope of ASME Section VIII, Division 1

- **Includes most vessels (or portions of vessels) subject to either an internal or external pressure**
 - **Local laws and regulations determine applicability of the Code**
- **Does not include the following vessels within its scope (in some cases they can be constructed and stamped in accordance with the Code if desired)**
 - **Internal and external operating pressures do not exceed 15 psi**
 - **Diameter, width, height, or cross-section diagonal does not exceed 6 inches (no limit on length)**

Scope of ASME Section VIII, Division 1

(continued)

- **Vessels not included in scope of ASME VIII-1, (continued):**
 - **Intended for human occupancy**
 - **Fired heaters**
 - **Equipment within scope of another section of the ASME Code**
 - **Piping systems and components**
 - **Hot and/or pressurized water containment vessels under certain conditions**
 - **Internal parts of rotating or reciprocating devices where design considerations and stresses are derived from the equipment's functional requirements**

Codes and Standards

ASME Section VIII, Division 2

- **Used for high pressure refinery equipment**
- **Uses factor of safety of 3 against tensile failure**
- **Results in thinner vessels (compared to Division 1)**
- **Not permitted in the creep range of materials**
- **Requires additional design analysis (e.g., local and thermal stress, fatigue) and quality control (e.g., full X-ray, stringent material requirements)**

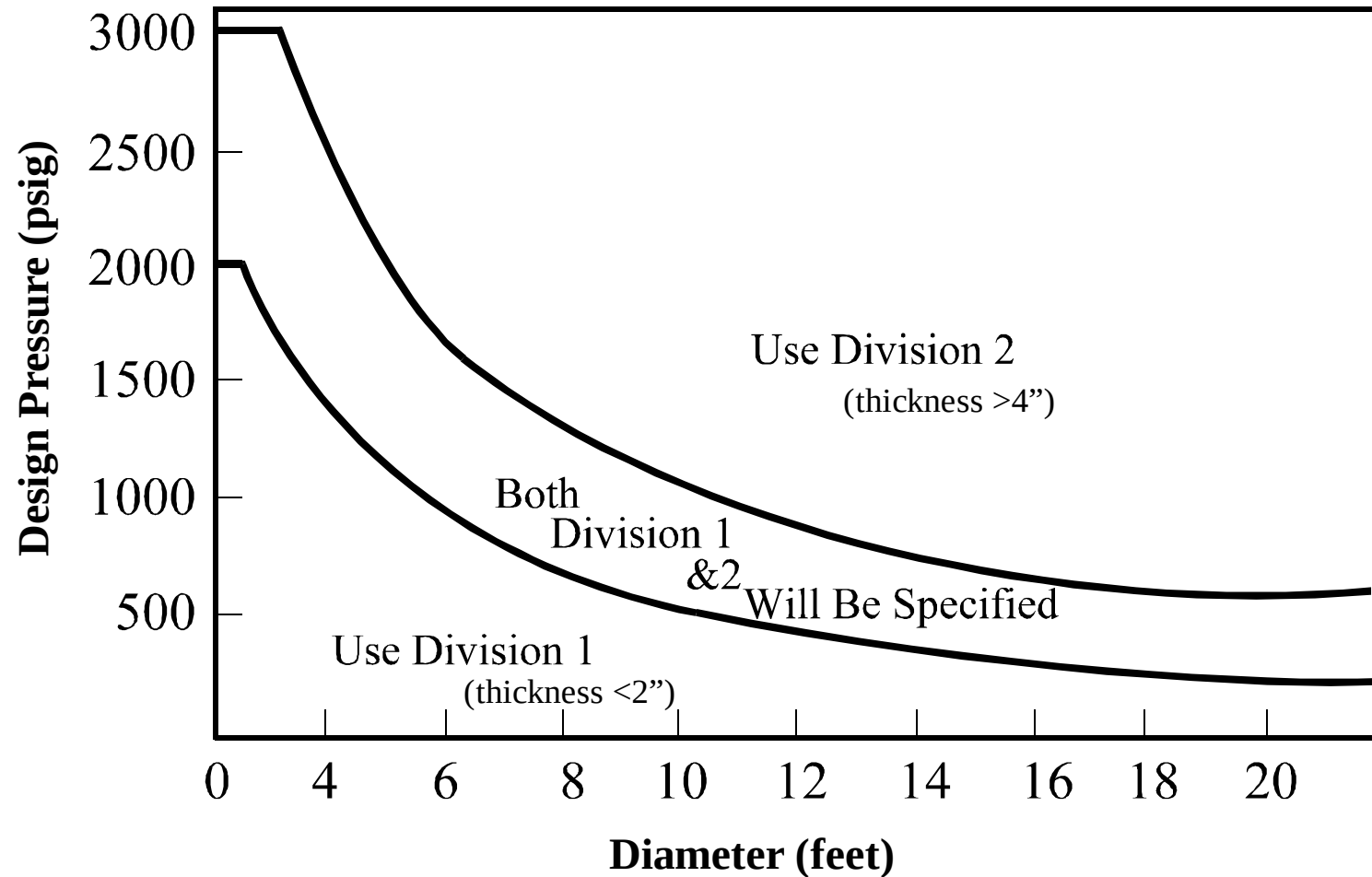
Codes and Standards

ASME Section VIII, Division 2 (continued)

- **More difficult to re-evaluate for future operating condition changes**
- **Limited fabricators**
- **Material and fabrication costs (welding, rolling) are lower, as are transportation, erection, and support costs**
 - **Partly offset by analysis, design, and quality control expenses**

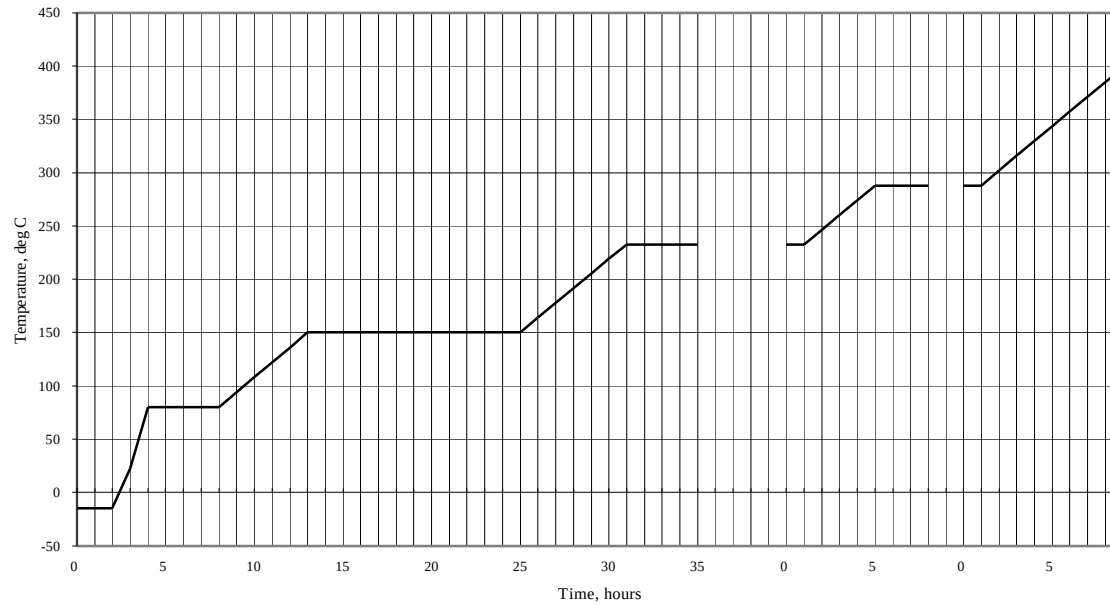
UOP Guidelines

Use of ASME Section VIII Division 2

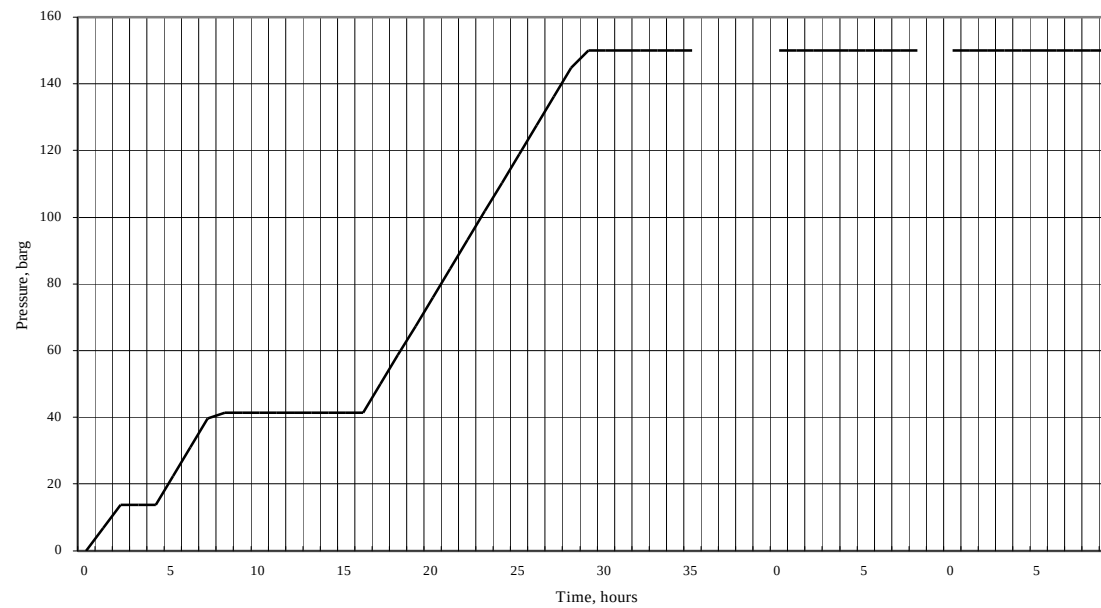


Based upon an allowable stress = 17,000 psi

Normal Fresh Catalyst Startup Reactor, D-2503



Normal Fresh Catalyst Startup Reactor, D-2503



Codes and Standards

ASME Section VIII, Division 3

- **For ultra high pressure equipment ($>10,000$ psi)**
- **High strength materials**
- **Material toughness requirements**
- **Fatigue analysis required**
- **Refinery equipment does not fall within its scope**

Codes and Standards

ASME Code Cases and Interpretations

- **Code Cases are auxiliary to the Pressure Vessel and Nuclear Sections of the ASME Code. If accepted by the local governing body they carry the legal weight and authority of the Code.**
- **Interpretations are committee responses to questions but carry no legal weight. They exist for many Sections of the Code.**

Codes and Standards

Non-Code Vessels

- **Applicable to atmospheric vessels handling water and injection chemicals**
- **Nominal cost savings**
 - **No Code shop**
 - **No Code stamp**
- **Must still be safely constructed—often complies with Code details**

Codes and Standards

Other Related Codes and Standards

- **API Standard 620, Large Low Pressure Storage Tanks, Pressure 0.5 to 15 psig**
- **API Standard 650, Welded Storage Tanks, Pressures up to 0.5 psig**
- **ASME B31.3, Process Piping**
- **ASME B16.5, Pipe Flanges and Flanged Fittings**
- **ASME B16.47, Large Diameter Steel Flanges NPS26 Through NPS60**
- **TEMA for Heat Exchangers**
- **Local codes if more stringent**

Code for Repairs and Alterations

- **Scope of the familiar design codes covers new construction only**
 - **For repairs and alterations (revamps), other documents govern**
- **As with codes for new construction, the applicable document depends upon local laws and regulations**
- **Two common documents are:**
 - **NB23 - National Board Inspection Code**
 - **API 510 - Pressure Vessel Inspection Code, Maintenance, Inspection, Rating, Repair, and Alteration**

UOP Standard Specifications

- **UOP Standard Specifications for pressure vessels augment the codes**
- **Are organized on the basis of the material of construction**
- **Most commonly used are:**
 - **3–11 Pressure Vessels—Carbon Steel**
 - **3–12 Pressure Vessels—Low Alloy Steel**
 - **3–17 Pressure Vessels—ASME Section VIII Division 2**

ASME Versus ASTM Materials

- **ASTM materials are prefaced with “A” (e.g. A387); ASME materials are prefaced with “SA” (e.g. SA387)**
- **Are normally no significant differences between the materials**
 - **Any differences are noted in the ASME listings (Section II of the ASME Code)**

ASME Versus ASTM Materials

(continued)

- **ASME materials (i.e. those designated with “SA”) must be used for fabrication according to the ASME Pressure Vessel Code**
- **ASTM materials are used for most other uses, including piping conforming to ASME B31.3**

SPECIFICATION FOR FORGED OR ROLLED ALLOY-STEEL PIPE FLANGES, FORGED FITTINGS, AND VALVES AND PARTS FOR HIGH-TEMPERATURE SERVICE



SA-182/SA-182M

(Identical with ASTM Specification A 182/A 182M-95c except for the clarified heat treatment requirements by the addition of Note 4 to Table 1.)

A96
A97

1. Scope

1.1 This specification covers forged low alloy and stainless steel piping components for use in pressure systems. Included are flanges, fittings, valves, and similar parts to specified dimensions or to dimensional standards such as the ANSI specifications that are referenced in Section 2.

1.6 The value stated in either inch-pound units or SI units are to be regarded separately as the standard. Within the text, the SI units are shown in brackets. The values stated in each system are not exact equivalents; therefore, each system must be used independently of the other. Combining values from the two systems may result in nonconformance with the specification.

A97 **1.2** For bars and products machined directly from

Low Temperature Requirements

- **At low temperatures, many materials may become brittle**
 - **ASME Code contains additional requirements for these materials depending upon the applicable MDMT**
- **MDMT stands for Minimum Design Metal Temperature**
 - **Is the lowest mean temperature of the metal (not the internal fluid) considering many factors, including operating temperature, low ambient temperature, and auto refrigeration**

Low Temperature Requirements

(continued)

- **Application of additional requirements depends upon the material, MDMT, and thickness**
- **Figure UCS-66 of ASME Section VIII Division 1 is used to determine if Charpy V-notch testing is required**

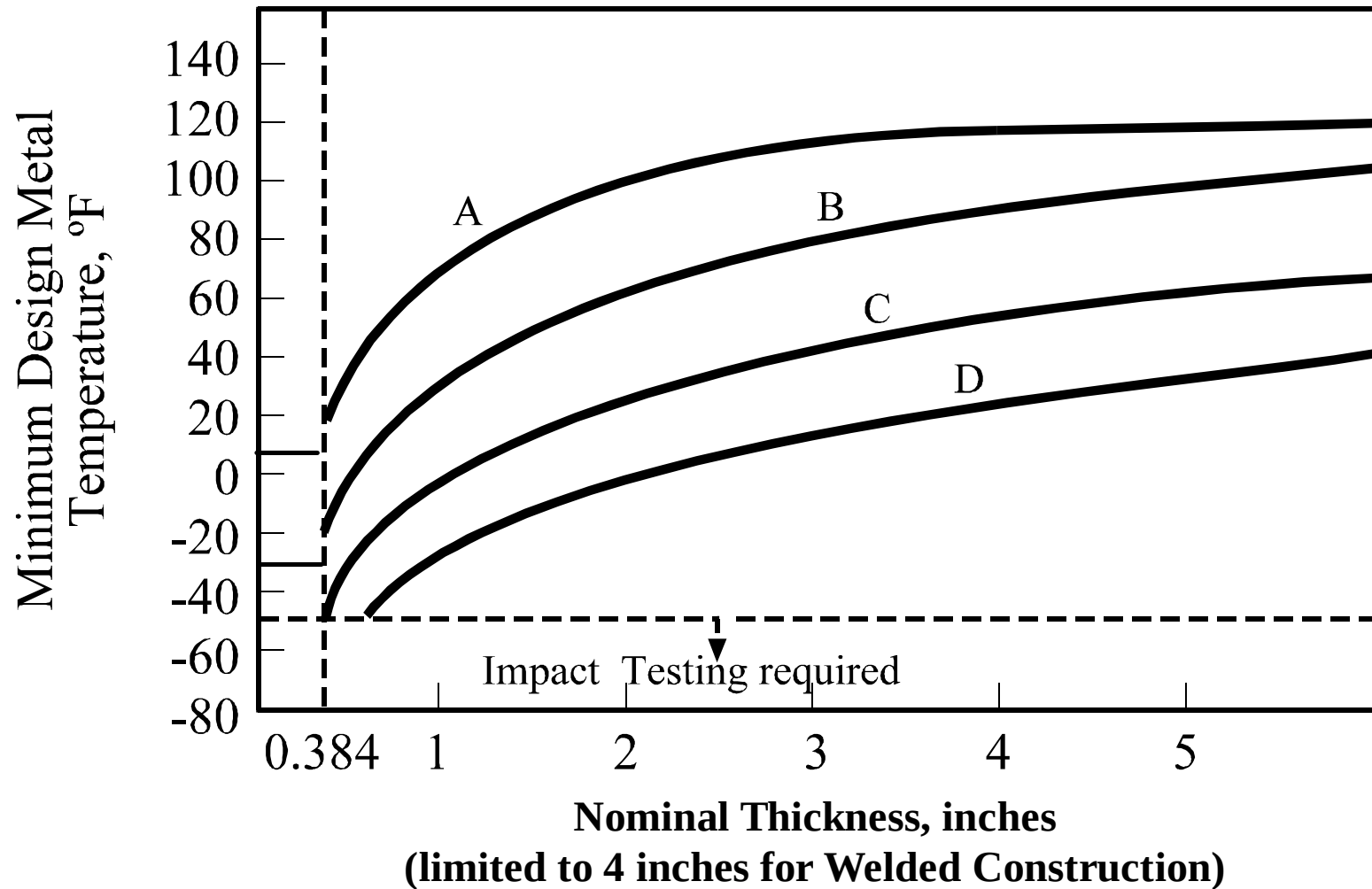
Low Temperature Requirements (continued)

- **If required by Figure UCS-66, materials must exhibit minimum Charpy V-notch impact test values when tested at the MDMT**
- **Exemptions and exceptions exist for thin carbon steel vessels, low stressed materials, and heat treated items if heat treatment is not otherwise required**

MDMT Determination

- **The MDMT shown by UOP is the lowest of the following temperatures:**
 - **Minimum operating temperature minus 25 °F**
 - **Lowest average ambient temperature for a 24 hour period**
 - **Auto-refrigeration temperature determined by flashing the material to 40 percent of design pressure**
- **This method of determining the MDMT tends to be conservative because the surrounding fluid temperature, not the actual metal temperature, is used.**

Impact Test Exemption Curves ASME Section VIII Division 1



Partial Materials List for Curves

■ **Curve A**

- **All carbon and all low alloy steel not listed for Curves B, C, and D below**
- **SA-216 Grades WCB and WCC; SA-217 Grade WC6 if normalized and tempered**

Partial Materials List for Curves

(continued)

■ **Curve B**

- **SA-216 Grade WCA if normalized and tempered**
- **SA-216 Grades WCB and WCC for thickness not exceeding 2 inches, etc**
- **SA-217 Grade WC9 if normalized and tempered**
- **SA-285 Grades A and B**
- **SA-515 Grade 60**
- **SA-516 Grades 65 and 70 if not normalized**

Partial Materials List for Curves

(continued)

■ **Curve C**

- **SA-182 Grades 21 and 22 if normalized and tempered**
- **SA-336 F21 and F22 if normalized and tempered**
- **SA-387 Grades 21 and 22 if normalized and tempered**
- **SA-516 Grades 55 and 60 if not normalized**

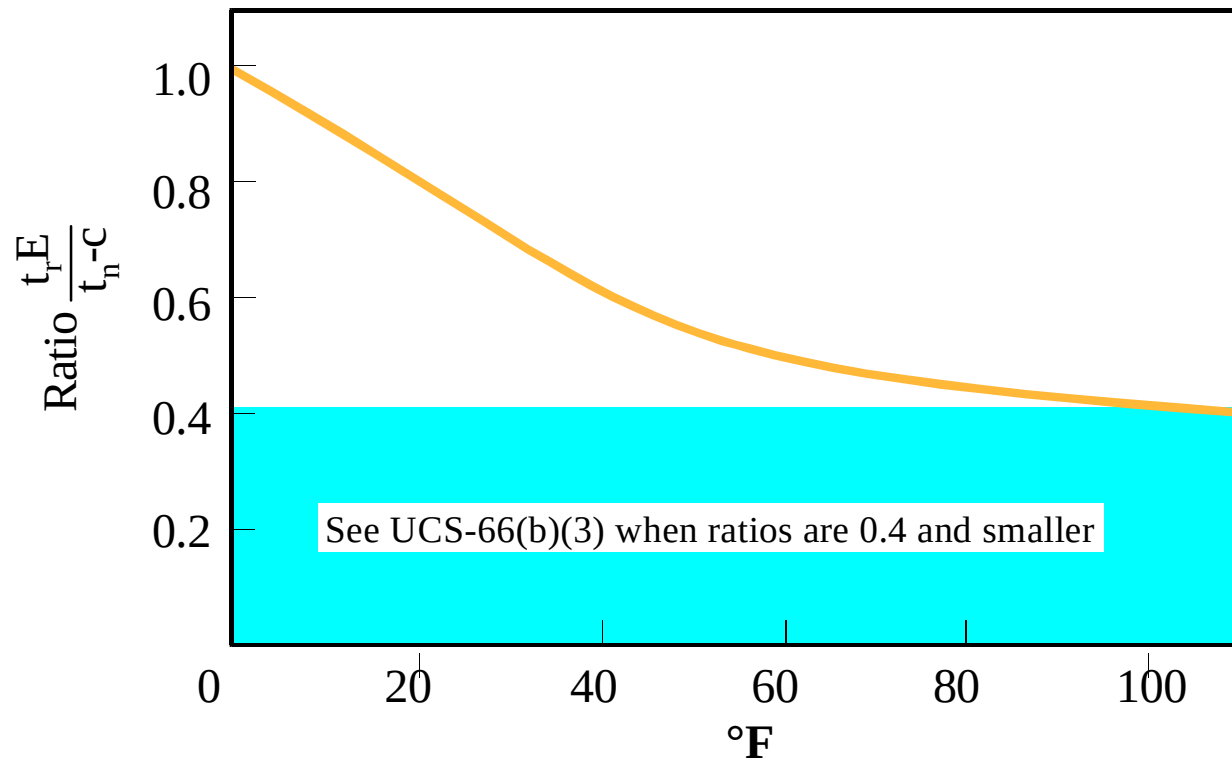
Partial Materials List for Curves

(continued)

■ **Curve D**

- **SA-203**
- **SA-508, Grade 1**
- **SA-516 if normalized**
- **SA-524 Classes 1 and 2**
- **SA-537 Classes 1, 2, and 3**

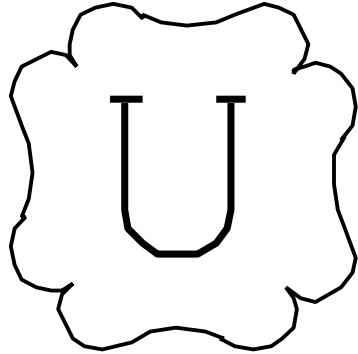
Reduction in Minimum Design Metal Temperature Without Impact Testing



Nomenclature

- t_r = required thickness of the component in corroded condition for all applicable loadings based on the applicable joint efficiency E, inches.
- t_n = nominal thickness of the component under consideration including corrosion allowance, inches.
- c = corrosion allowance, inches.
- E = joint efficiency.

Name Plate



W (if arc or gas welded)
RT (if Radio graphed)
HT (if Postweld heat treated)

Name of Manufacturer

_____ psi at _____°F

Max. Allowable Working Pressure

_____ °F at _____ psi

Min. Design Metal Temperature

Manufacturer's Serial Number

Year Built

ASME Section VIII Division 1 Postweld Heat Treatment Requirements

	Code Reference
Vessels containing lethal substances	UW-2
Carbon-steel vessels for service at temperature below -20°F	UCS-67
Welded vessels	UW-10 UW1-40
Carbon and low-alloy steel vessels t > 1.25 inches	UCS-56 UCS-66
Low alloy steel vessels t > 0.625 inches	UCS-67 UCS-79 U-1
High-alloy steel vessels	UHA-32
Clad-plate vessels	UCL-34
Bolted flange connections	UA-46
Castings	UG-24
Forgings	UF-31
HT under symbol - entire vessel postweld heat-treated	UG-116
PHT under symbol - part of the vessel postweld heat-treated	UG-116

Postweld Heat Treatment Not Required

■ Carbon Steels

- **$t < 1.25$ inches**
- **$t < 1.50$ inches if 200 °F preheat**

■ Low Chrome Steels

- **Circumferential butt welds of pipe or tubes**
- **If pipe < 4 inches outside diameter**
- **$t < 5/8$ inches**
- **Carbon $< 0.15\%$**
- **250 °F preheat, minimum**

Postweld Heat Treatment Requirements for Carbon and Low Alloy Steels

		Minimum Holding Time at Normal Temperature for Nominal Thickness [see UW-40(f)]		
Material	Normal Holding Temperature, °F, min	Up to 2 in.	Over 2 in. to 5 in.	Over 5 in.
P-No.1 Gr. Nos 1,2,3 (carbon steel)	1100	1 hour/inch, 15 minutes, minimum	2 hours plus 15 minutes for each additional inch over 2 inches	2 hours plus 15 minutes for each additional inch over 2 inches
P-No. 4 Gr. 1,2 (low alloy)	1100	1 hour/inch, 15 minutes minimum	1 hour/inch	5 hours plus 15 minutes for each inch over 5 inches

Non-Destructive Examination Methods

Nondestructive examination (NDE) is a quality assurance tool used to check welds for flaws. This results in safer vessels and allows use of higher joint efficiencies; therefore, thinner shells. Methods of NDE include:

- **Visual**

- **Most economical**
- **Most versatile**
- **Requires an experienced inspector**
- **Detects surface imperfections only**

- **Dye Penetrant (PT)**

- **Places a contrasting dye over the weld surface, then wiped clean**
- **Surface imperfections retain the dye**
- **Apply a developer to make dye visible**
- **Detects surface imperfections only**

Non-Destructive Examination Methods

(continued)

■ **Magnetic Particle (MT)**

- **Metallic particles are sprinkled on the surface and magnetic poles are supplied by an electric current, creating a magnetic field**
- **Particles align with the magnetic field**
- **Orientation of the particles indicates surface and very slightly subsurface imperfections**
- **May use fluorescent particles in a liquid suspension to increase visibility and ease of particle movement**
- **Material must be magnetic and surface must be horizontal**
- **Accidental arc strikes possible**

Non-Destructive Examination Methods

(continued)

■ Radiography (RT)

- Detects many types of subsurface imperfections, lack of fusion, slag inclusion, porosity, etc in addition to cracks**
- Dangerous to perform**
 - May require an isolated or roped off area and be done at night or other times when people are not present**
- Requires access to both sides of the examined surface and clearance from obstructions in the immediate vicinity**

Non-Destructive Examination Methods

(continued)

- **Radiography (RT) (continued)**
 - **Generally requires an experienced, specialty contractor**
 - **Can examine the full length or a portion of the length (i.e. spot) of welds**
 - **Provides a permanent record in the form of a film image**
 - **Difficult to perform in the field**
 - **For field inspections, gamma rays are often substituted for X-rays**

ASME Section VIII Division 1

Full Radiographic Requirements

Carbon and Low-Alloy Steels		
P Number and Group Number – Metals		When Thickness Exceeds
P = 1 1 1 Carbon steels	Group 1 2 3	1.25 in
P = 3 3 3 Alloy steels with 0.75 maximum chromium and those with 2.00 maximum total alloy	Group 1 2 3	0.75 in.
P = 4 4 Alloy steels with 0.75 to 2.00 chromium and those with 2.75 maximum total alloy	Group 1 2	0.625 in.
P = 5A 5A Alloy steels with 10.00 maximum total alloy	Group 1 2	0.0 in.
P = 9A 9B Nickel alloy steels	Group 1 1	0.625 in.
P = 10A 10F	Group 1 6	0.75 in.
P = 10B 10C Other alloy steels	Group 2 3	0.625 in.

Non-Destructive Examination Methods

(continued)

■ **Ultrasonic (UT)**

- **Uses reflection of sound waves to detect subsurface flaws**
- **Used to measure thickness**
- **Access required from only one side**
- **Not dangerous**
- **Requires experienced operator to interpret results**
- **Requires smooth, clean surface (including grinding of welds)**
- **Requires frequent calibration and a calibration block for the material being examined**

Non-Destructive Examination Methods

(continued)

- **Ultrasonic (UT) (continued)**
 - **Use of angle beams eliminates some concern with nearby obstructions**
 - **Straight beam is used for thickness determination**
 - **Can be performed while equipment is on stream**
 - **Use of computers allows creation of a permanent record on a disk**
 - **May be difficult to use on thin shells and on austenitic stainless or coarse grained steels**
- **Other specialty methods, including replication and acoustic emission, are available**

Non-Destructive Examination Methods

- **New vessel examination**
 - **Uses all examination methods**
 - **RT and UT detect subsurface fabrication flaws and cracks, allowing for correction**
- **In service examination**
 - **New damage/flaws form at surface, detectable by visual, PT, or MT**
 - **Cracks may grow from existing subsurface defects, detected by RT and UT**
 - **Corrosion detected by visual and UT**

Lethal Services

- **Defined in ASME Section VIII Division 1, Section UW-2.**
- **Lethal is defined as “poisonous gases or liquids of such a nature that a very small amount of the gas or of the vapor of the liquid mixed or unmixed with air is dangerous to life when inhaled.”**
- **API has determined that refinery processes, including HF containing services, do not qualify as lethal services.**

Vessel Fabrication

Methods of Shell Fabrication

- **Shells are formed from a series of cylinders butt welded together**
 - Typically these “cans” are 8 feet (2.5 meters) long

- **Two forming methods are common:**
 - Rolled plate
 - Drum forging

Vessel Fabrication

Methods of Shell Fabrication (continued)

■ **Rolled Plate**

- **Commonly available**
- **Many potential fabricators**
- **Unlimited vessel size**
- **Includes at least one longitudinal weld seam**
- **Longitudinal seams of neighboring sections cannot be aligned**
- **Difficult to form thick shells**

Vessel Fabrication

Methods of Shell Fabrication (continued)

- **Rolled Plate (continued)**
 - **Distortions possible during rolling**
 - **Difficult to maintain a consistent diameter**
 - **May be difficult to match shapes of neighboring sections**
 - **Tends to have a grain alignment in the direction of rolling**
 - **Can be difficult to roll to a small radius of curvature (relative to the thickness)**

Vessel Fabrication

Methods of Shell Fabrication (continued)

■ **Drum Forging**

- **Excellent for thick shells; no thinning or creation of stresses**
- **No longitudinal weld seam**
- **Close ID tolerance; can be machined to very close tolerances**
- **Good thickness and diameter control**

Vessel Fabrication

Methods of Shell Fabrication (continued)

- **Drum Forging (continued)**
 - **Formed directly from ingot**
 - **Due to need to work with a hot ingot, potential fabricators are limited**
 - **Limited diameters possible**
 - **Limited volume of shell section determined by ingot volume**
 - **Material properties vary from surface to center**

Multi-Layer Construction

- **Considered for heavy wall vessels where the thickness makes other methods impractical or expensive**
- **Shell is made of multiple thin layers of material**
 - **Layers may be wound (like a coil) or formed from separate rings and shrink fit onto each other**
 - **Thinner plate is easier to form**
 - **Thin plates have more uniform material properties**

Multi-Layer Construction (continued)

- **Heads remain as single layer construction**
- **Nozzles are solid forgings**
- **Insuring that nozzles are welded to all of the plate layers can be difficult**
- **Vents are provided to detect leakage and, if applicable, hydrogen venting**
 - **Vents extend from the outside through all but the inner layer**

Multi-Layer Construction (continued)

- **Must insure that all layers act together, carrying their share of the load**
- **Attachments (internal or external) can be a concern because they attach to the surface layer**
 - **For significant loads, insure that all layers participate in carrying the load**
- **Cracks do not propagate between layers**
- **Most suited for membrane (uniform) stresses; not well-suited for bending stresses**
- **“Gaps” between layers make NDE nearly impossible**

Multi-Layer Construction

(continued)

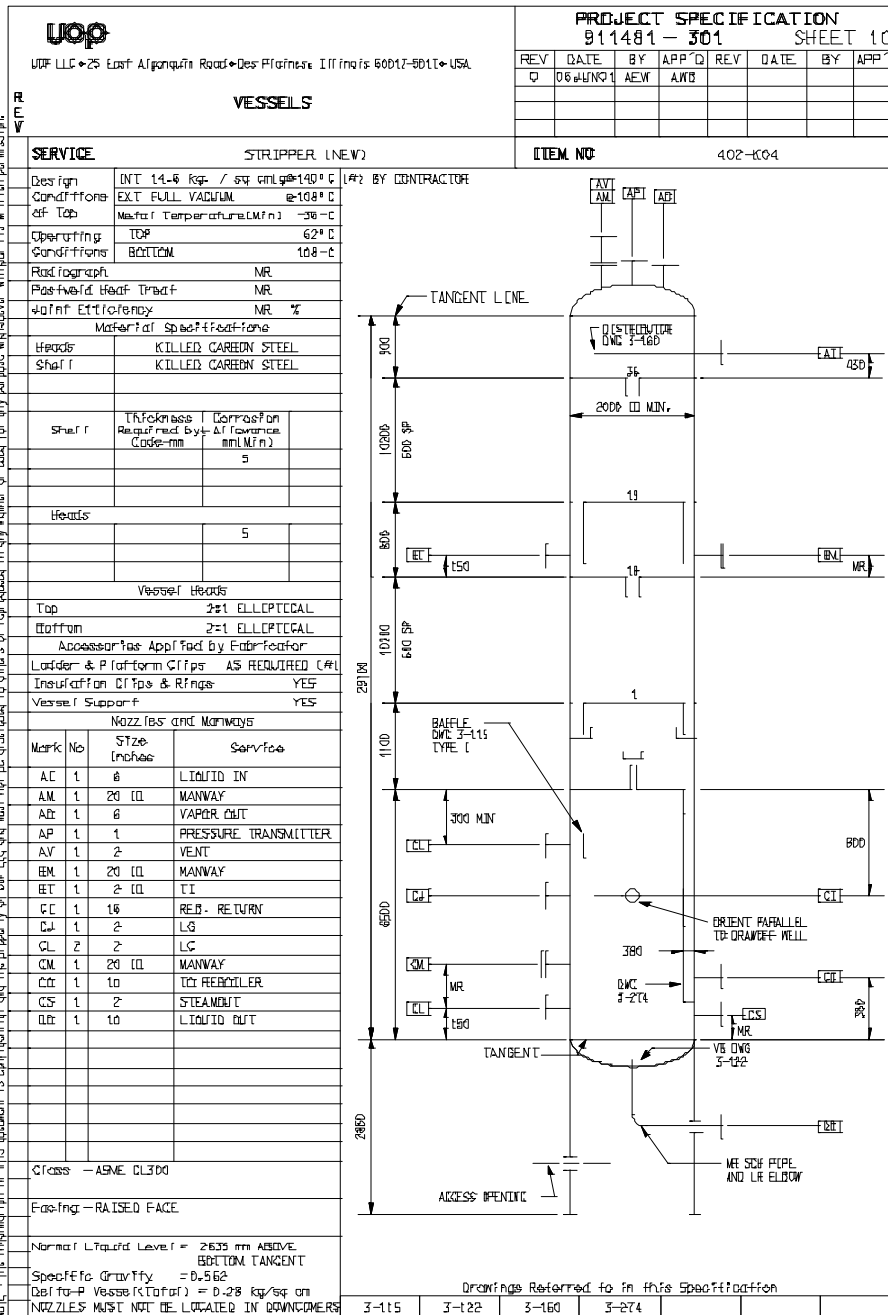
- **Thorough inspection is difficult – visible layers do not reflect or represent condition of other layers**
- **Very difficult to evaluate for future service (i.e. fitness for service or rerating) due to difficulty accurately ascertaining the current condition**
 - **Division 2 designs are especially difficult because of the detailed analysis required**
- **Very difficult to repair or modify**
- **May need to account for differential thermal expansion between layers**

Stripper:002

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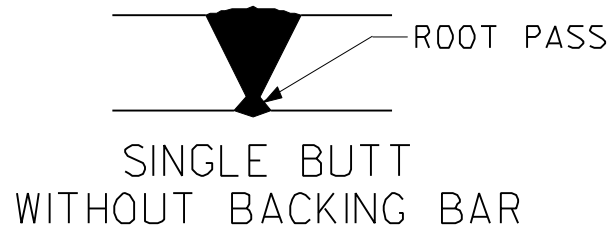
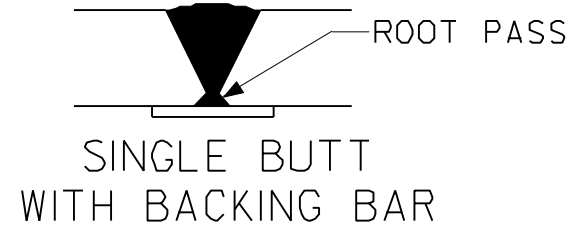
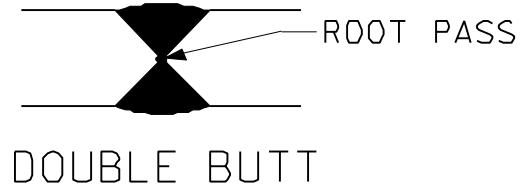
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Vessel Seam Welds



Welding Methods

- **All processes use an arc between the electrode and base metal to produce the heat for fusion**
 - **Some electrodes become a part of the weld (consumable) while others do not (non-consumable)**
- **All processes are dependent upon a competent welder, qualified per the governing code**
- **Procedures are written and welders tested for each type of weld used.**
- **Low hydrogen is desired to prevent flaws and cracking, hence electrodes must be kept dry**

Welding Methods

(continued)

- **Shielded Metal Arc (SMAW)**
 - **Shielding of arc provided by gases from electrode covering decomposition**
 - **Molten flux or slag provides more shielding**
 - **Electrode is consumed**
 - **Usually done manually**
 - **Can be done in any position**
 - **Good ductility and resistance to weld shrinkage cracks**

Welding Methods

(continued)

- **Gas Metal Arc (GMAW)**
 - **Shielding is from a gas stream**
 - **Electrode is consumable and becomes filler material**
 - **Usually done automatically (machine) with a continuously fed electrode**

Welding Methods

(continued)

- **Gas Metal Arc (GMAW) (continued)**
 - **Can be done in any position with proper shielding gas selection (e.g. argon is heavier than air and is not used for overhead welding)**
 - **Weld spatter is a concern**
 - **Sometimes known as MIG (Metal Inert Gas)**
 - **Use often limited due to concerns about difficult to detect cold lap or lack of fusion**

Welding Methods

(continued)

- **Submerged Arc (SAW)**
 - **Shielding from a granular, fusible flux (fused flux provides additional protection)**
 - **Arc cannot be seen, hence its “submerged”**
 - **Usually a continuous, automatic (machine) process**
 - **No weld spatter, but shielding flux may not stay in place if in other than a flat position**
 - **Flux is a material that prevents formation or aids removal of oxides and other undesirable substances**

Welding Methods

(continued)

- **Gas Tungsten Arc (GTAW)**
 - **Shielding from a gas stream (typically argon)**
 - **Uses a non-consumable tungsten electrode**
 - **Filler metal may be added**
 - **Used for thin materials (< 3-4mm) in all positions**
 - **Usually manual but may be automatic**
 - **Also known as TIG (Tungsten Inert Gas)**

Welding Methods

(continued)

- **Flux Cored Arc (FCAW)**
 - **Shielding gas from decomposition of the electrode and, occasionally, an external gas**
 - **Often produces a slag covering the weld**

Welding Methods

(continued)

- **Electric Resistance Welding**
 - **Heating of the base metal by resistance to an electric current**
 - **Does not melt the metal**
 - **Narrow, sometimes hard to detect weld or fusion line**
 - **Very limited applicability to pressure vessels**

Pressure Testing

- **Pressure testing is required by the ASME Code**
- **Testing to be performed after all fabrication, welding, and heat treatment is completed**
 - **Testing should occur prior to any painting or priming**
- **Testing to be observed by the authorized inspector**

Pressure Testing

(continued)

- **Test pressure may be based upon either**
 - **the design pressure**
 - **MAWP of the full, corroded or uncorroded thickness**

- **Two types of pressure are accepted:**
 - **Hydrostatic**
 - **Pneumatic**

Hydrostatic Pressure Testing

- Vessel is filled with water and pressured to the required value
- Section VIII Division 1 minimum required test pressure at all locations = $1.3 \cdot DP \cdot S_C/S_H$
- Use the lowest S_C/S_H ratio
- May be based upon design pressure or testing of full (uncorroded) thickness of vessel
- Recommended test temperature is 30°F over MDMT
 - Temperature is of the metal, not the test fluid

Hydrostatic Pressure Testing

(continued)

- **Check flanges and shell for overstress due to test pressure + hydrostatic head (especially significant for tall columns)**
 - No area may be stressed to more than 90 percent of the material's yield stress
- **Test is safer due to incompressibility of water (or other fluid)**
 - Little energy is stored in the test fluid under pressure
- **Easy to see and detect leaks; large water molecule may not reveal some small openings**
- **May add a dye or luminescent material to see leaks**

Hydrostatic Pressure Testing

(continued)

- **Must vent properly during filling to insure complete filling (including voids in internals)**
- **Avoid overstressing or lifting internals during filling**
- **Supports (e.g. support skirt and structure) must be adequate for liquid full vessel (may be difficult to provide in situ)**
- **Adequate supply of suitable water may be difficult to obtain**
 - **For example, where stainless steel is present, chlorides are limited to 50ppm**

Hydrostatic Pressure Testing

(continued)

- **Avoid damage (e.g. pulling a vacuum) during drainage; fully removing liquid and drying may be difficult**
 - **If not thoroughly dried, corrosion (rust) may occur**
- **Some environments and internals (e.g. refractory) may make hydrostatic testing undesirable**
- **Water must not freeze**

Pneumatic Pressure Testing

- **Test pressure is provided by compressing air or another gas**
- **Section VIII Division 1 minimum required test pressure at any point = $1.1 \cdot DP \cdot S_C/S_H$**
 - **As with hydrostatic testing, pressure may be based upon the design pressure or the full corroded or uncorroded thickness**
 - **Use the lowest S_C/S_H ratio**
- **Metal test temperature must be at least 30 °F over the MDMT**

Pneumatic Pressure Testing

(continued)

- **Very dangerous due to stored energy in the compressed gas**
- **Heat of compression, and subsequent cooling, may mean a loss of test pressure**
- **Existence of a leak may be detected by a loss of (i.e. difficulty maintaining) internal pressure**

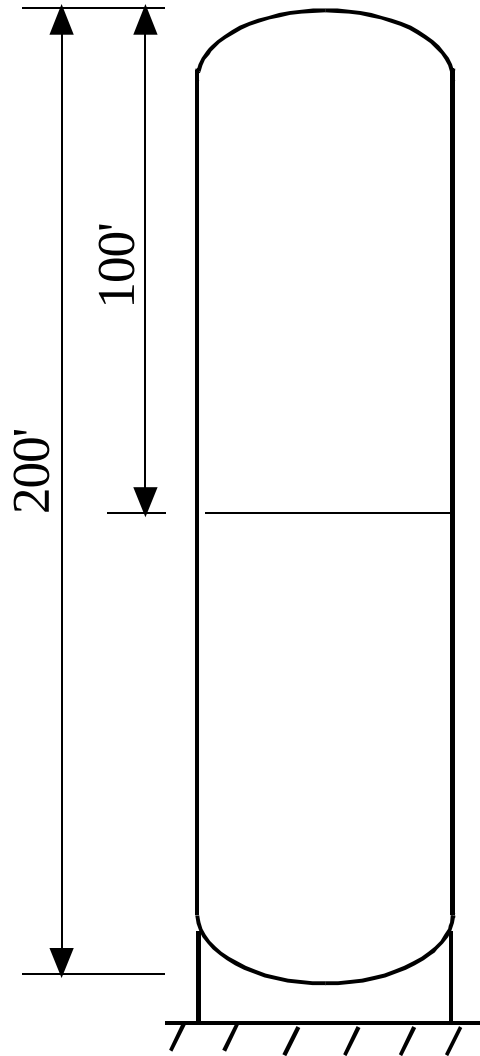
Pneumatic Pressure Testing

(continued)

- **May be difficult to see leak location—colored smoke sometimes added**
- **No extra weight or hydrostatic pressure to consider**
- **Venting and concern with the filling method are not a concern, nor is finding, draining, or disposing of the test medium**
- **Does not damage refractory or impact the process**



Hydrostatic Test Example



Design Conditions:

$$\begin{aligned}
 P &= 50 \text{ psig} \\
 T &= 650^\circ\text{F (Top 100')} \\
 &= 1050^\circ\text{F (Bottom 100')}
 \end{aligned}$$

Material:

SA387 GR11 CL2 (Bottom)

SA516 GR70 (Top)

Allowable Stress at Design

Temperature:

$$S_H \text{ (top)} = 18,800 \text{ psi}$$

$$S_H \text{ (bottom)} = 4,200 \text{ psi}$$

Allowable Stress at Test

Temperature (70°F)

$$S_T \text{ (top)} = 20,000 \text{ psi}$$

$$S_T \text{ (bottom)} = 21,400 \text{ psi}$$

$$P_{HYDRO} \text{ (at top head)} = 1.3(50)(1.064) = 69.2 \text{ psig}$$

Actual pressure at bottom,

$$\text{including hydrostatic head} = 69.2 + 0.433 \frac{\text{psi}}{\text{ft}} \times 200' = 155.8 \text{ psig}$$

Bottom head must be capable of taking this pressure.

All flanges must be checked for hydrotest condition.

NOTE: P_{HYDRO} for a single vessel made of SA387G11CL2 material, with Design Temperature = 1050F and $P = 50$ psig $P_{HYDRO} = 1.3(50)5.095 = 331.2$ psig. Including hydrostatic head $P_{Bottom} = 331.2 + 86.6 = 417.8$ psig.

Hydrotest Pressure, P_{HYDRO}

$$P_{HYDRO} = 1.3 P \frac{S_{TEST}}{S_{HOT}} \quad \text{Lowest Ratio}$$

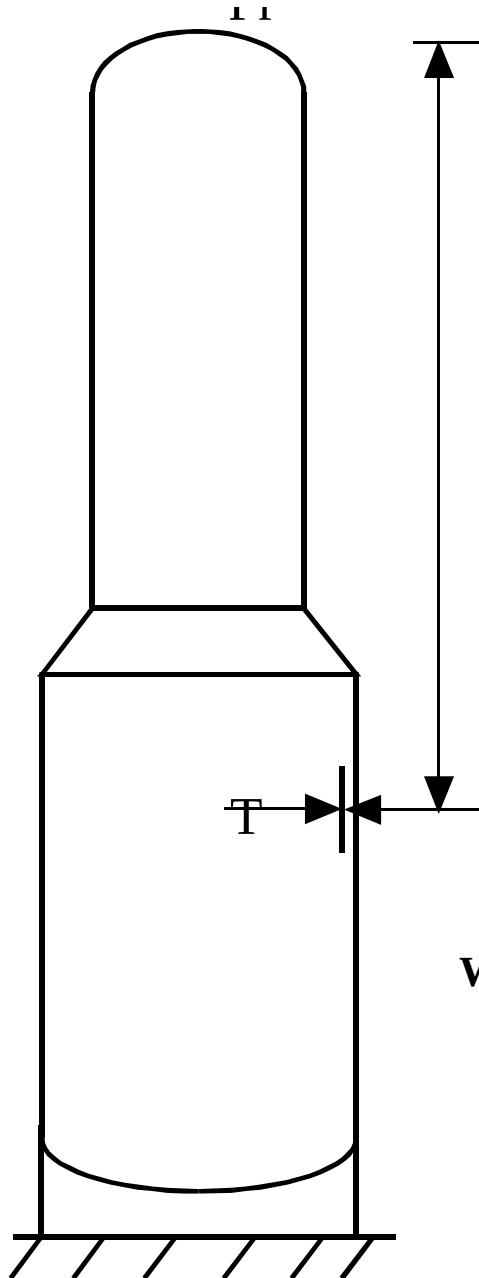
Top of the Vessel,

$$\frac{S_{TEST}}{S_{HOT}} = \frac{20,000}{18,800} = 1.064$$

Bottom of the Vessel,

$$\frac{S_{TEST}}{S_{HOT}} = \frac{21,400}{4,200} = 5.095$$

Full Thickness Hydrotest Pressure



H For each shell section, head cone, etc., determine the maximum allowable pressure at the test temperature (MAW P_C). For a shell section:

$$P_C = \frac{SE}{R + 0.6T}$$

Where: P_C = Maximum permitted pressure for material thickness

S = Material allowable stress at test temperature (ambient)

T = Material thickness

E = Joint efficiency

R = Radius

Full Thickness Hydrotest Pressure

(continued)

- **Calculated test pressure at top of vessel**

$$P_r = 1.3P_C - \text{liquid head}$$

For hydrotest of a cylindrical section:

$$P_r = \frac{1.3SET}{R + 0.6t} - 0.433(S.G.)H$$

Where: S.G. = Specific Gravity of the test medium

- **Hydrostatic test pressure at the top of the vessel = minimum of all calculated test pressures**

Vessel Supports

- **Straight skirts below the vessel are most common for vertical vessels**
- **Skirt is best centered on the shell thickness**
- **Skirt Details**
 - **For vessels subjected to high (creep range) temperatures, cyclic loading, or with thick shells a contoured joint is used to reduce stress concentrations**
 - **Insulation details locate thermal gradients away from the joint**

Vessel Supports ***(continued)***

- **Skirt Details (continued)**
 - **Other heavy wall or “severe” service equipment uses a less stringent detail, usually a flat exterior face with a weld height at least twice its width**
 - **Remaining equipment uses a “standard” fillet welded joint**
- **Flared skirts (conical skirt attached to the side of the vessel) are often used for equipment supported on a tabletop or a structure (e.g., reactors with unloading space beneath them)**

Vessel Supports

(continued)

- **A flared skirt allows the vessel to project below the support level, reducing the wind overturning moment on the vessel**
- **An alternative to a flared skirt is support from lugs**
 - **Tension and compression rings are required to avoid high local stresses**
- **Small vessels are occasionally supported by legs**
 - **This alternative should be considered only for short, small diameter, lightly loaded items**

Vessel Supports *(continued)*

- **Horizontal vessels are supported by saddles located near the ends**
- **Design of this system is fairly specialized in order to avoid shell distortions at the saddles and “sagging” of the vessel between saddles**

Vessel Support Skirts

- **Vents are required at the top of the enclosed space to allow escape of any gases and to promote air flow and cooling**
- **Flanges are not permitted beneath skirts because they are a leak source and are not easily accessible inside a skirt**
 - **The confined space promotes dangerous concentration of leaking vapors**

Vessel Support Skirts

(continued)

- **Skirt length must be sufficient to absorb any radial thermal growth of the vessel**
- **Upper portion of skirts is made of the same material as the vessel shell**
 - **Remainder of the uninsulated skirt may be carbon steel**
- **Provide a “hot box” at the skirt/shell junction for elevated temperature service**
 - **This moves the thermal stresses away from the mechanical stresses and HAZ at the junction**

Contoured Support Skirt Detail

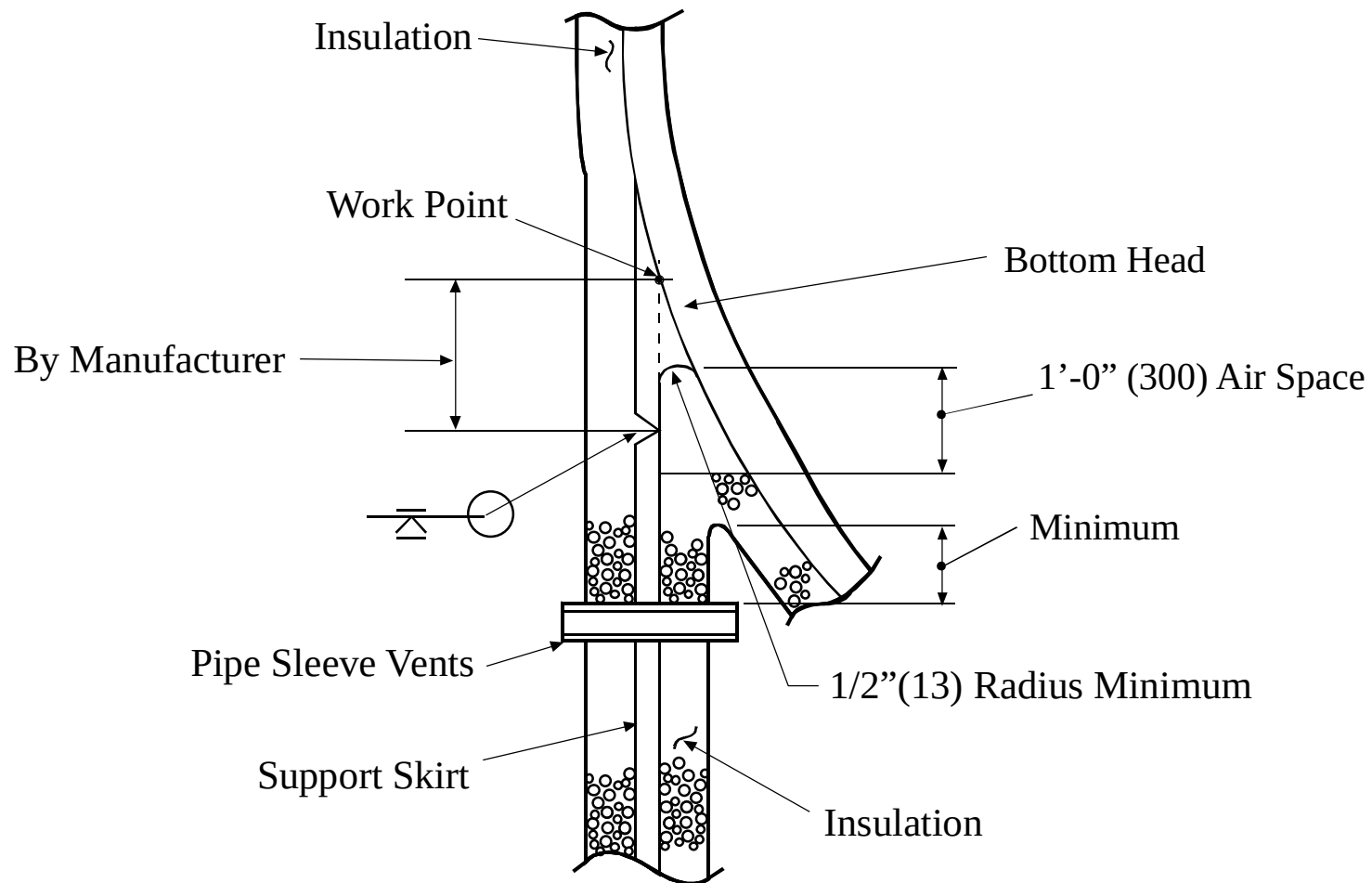
- **Joint between skirt and head shall have a smooth streamlined geometry**
- **Joint detail may be fabricated from:**
 - **A single forged component, butt welded as an integral portion of the vessel**
 - **Weld metal buildup**
 - **Built up plate construction**
- **Backing strips, if used, shall be removed after welding**

Contoured Support Skirt Detail *(continued)*

- **Welds shall be ground smooth and flush**
- **Weld surfaces shall be examined by magnetic particle or dye penetrant after final postweld heat treatment**
- **All pressure containing welds must be accessible for NDE in both the shop and the field**

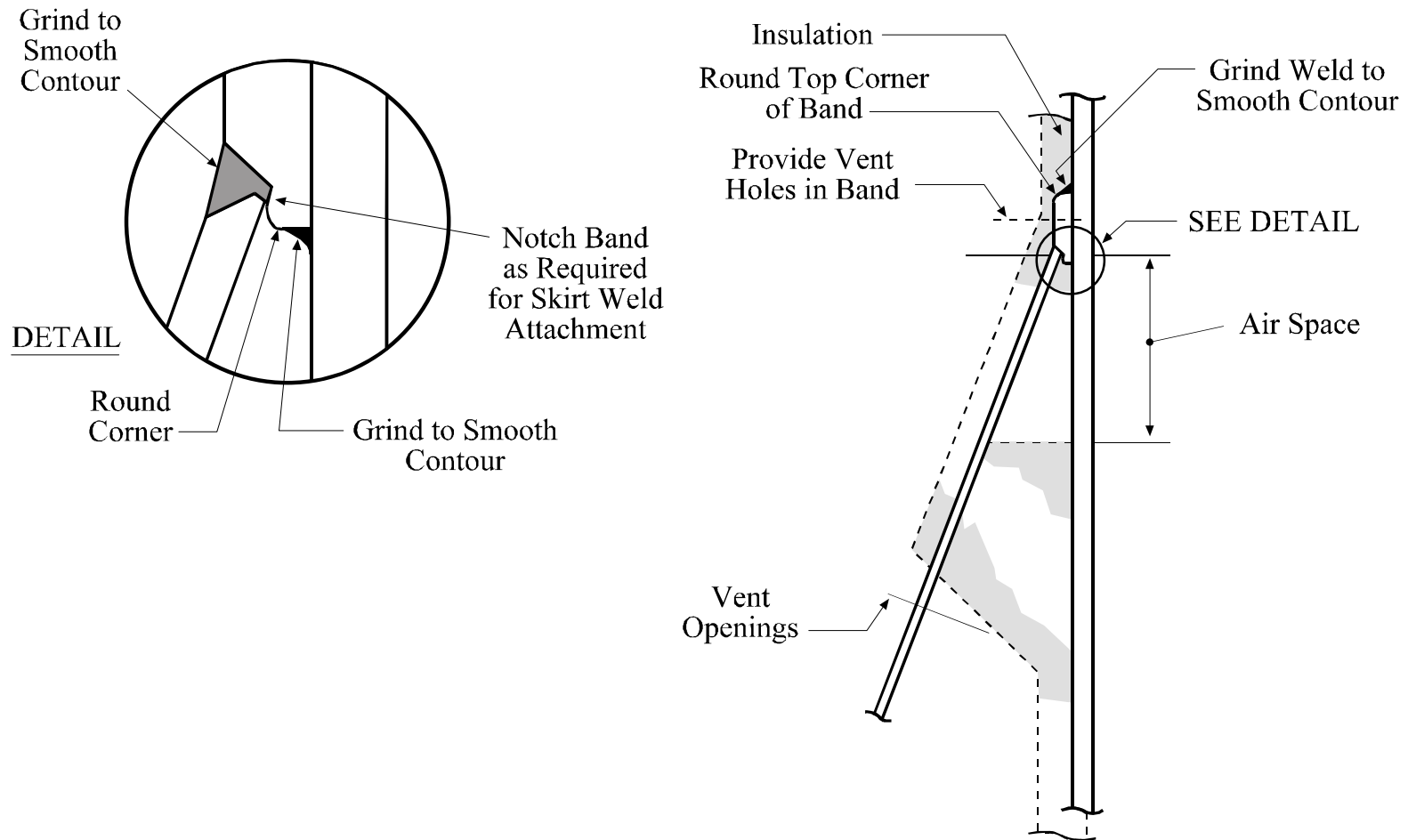
Vessel Supports

Contoured Skirt/Shell Junction



Vessel Supports

Flared Skirt



Revamps

- **Revamps include any re-evaluation and/or modification of an existing vessel**
 - **Rerates and evaluation for different operating conditions is included**
- **Perform a complete engineering evaluation of the vessel for any new design conditions or imposed loads**
- **All modifications must be designed and performed in accordance with the governing codes including the inspection codes (NB-23 or API-510)**

Revamps (continued)

- **Consider a formal fitness for service evaluation, especially if the vessel operated in the creep range, has been deformed, has significant corrosion damage, experienced operational upsets including overpressure or overheating, was subjected to cyclic loading or has been damaged (e.g. cracks) or deformed (e.g. bulges).**
- **API 579 provides a basis for evaluation of cracks and similar flaws, local thin areas (LTA's), bulges, creep and fatigue damage, etc.**

Revamps (continued)

- **Vessel must be thoroughly inspected, both visually and by nondestructive means, prior to commencement of the evaluation and any modifications.**
- **A complete metallurgical evaluation is also necessary to determine the present metallurgical condition after operation (e.g., creep, fatigue, embrittlement, etc).**

Revamps (continued)

- **Suitability for continued service under the same or new service conditions must be determined per the original code of construction.**
- **Very difficult to evaluate Division 2 vessels due to the detailed analysis originally required.**
- **Consider evaluation in accordance with the current design code to investigate the effect of code modifications (e.g. lower allowable stresses) since the original code.**

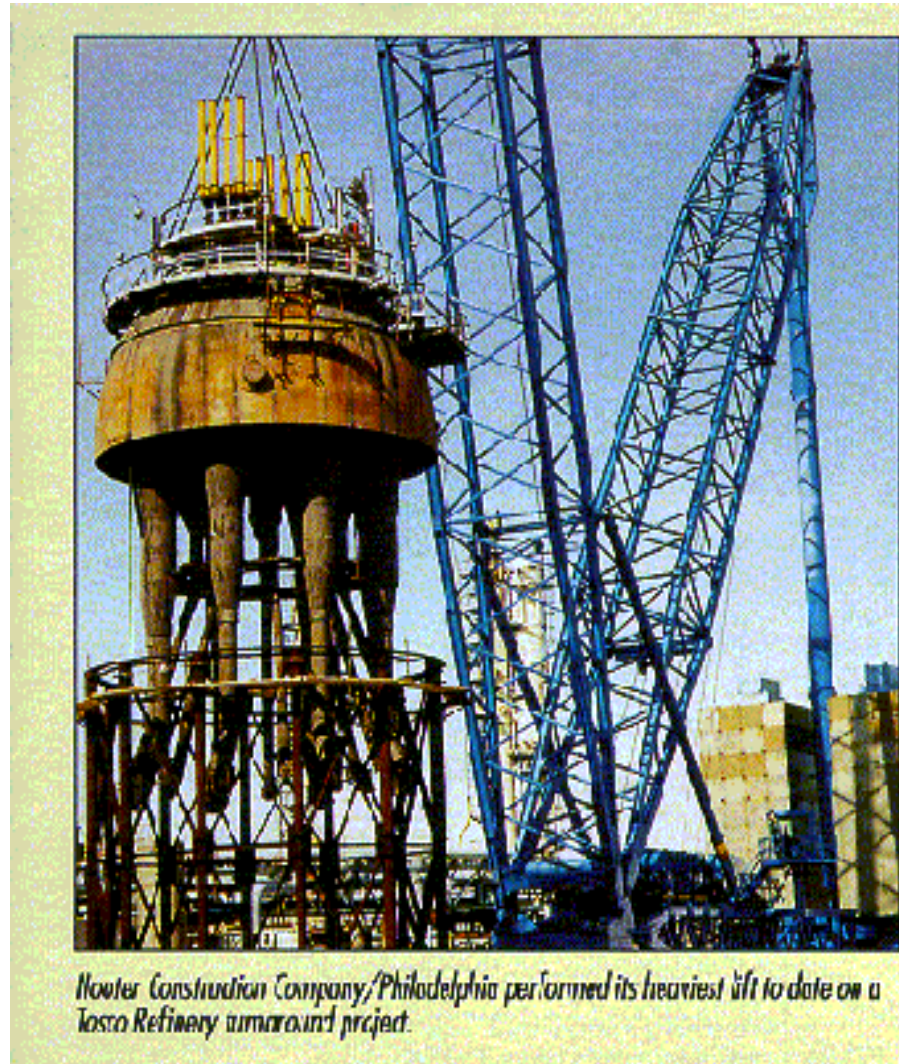
Revamps (continued)

- **Suitability of the materials for the intended atmosphere must be checked, even if it has not changed**
 - **For example, the Nelson curves for hydrogen atmospheres are occasionally revised so that a material may no longer be suitable for operation at the intended design conditions**
- **Review flange classes**
- **Review nozzle reinforcement**
- **If the vessel is relocated, review wind and earthquake**

Revamps (continued)

- **For service in the creep range, a remaining life evaluation is necessary as a minimum**
- **Proper fabrication methods must be used for the alteration, considering that the vessel has been in service**
 - **More care may be needed to prevent damage (e.g. maintenance of proper pre, during, and post-weld heat temperatures, sequence of welding, dehydrogenization, existence of coke)**
- **Thoroughly inspect and possibly test the modifications**

FCC Revamp



*Hooter Construction Company/Philadelphia performed its heaviest lift to date on a
Largo Refinery turnaround project.*

On-Stream Repair Concerns

- **Welding to operating equipment is dangerous**
 - **Welding may add to stresses already present**
 - **This may over-stress material or propagate an existing crack**
- **Welding will increase the local metal temperature, perhaps to the point the load carrying ability is compromised**
- **If there is a leak, welding arc may ignite vapors**
- **Hot taps are strongly discouraged**

On-Stream Repair Concerns ***(continued)***

- **Working in the presence of a process leak is very dangerous**
- **Must avoid creating thermal stresses during repair procedure or shutdown**
 - **Any patch must be the same material at the same temperature as the base material at the time of the repair**
- **Stress relief may be required**
- **May need to vent beneath a patch to allow escape of welding gases**

Stress Analysis

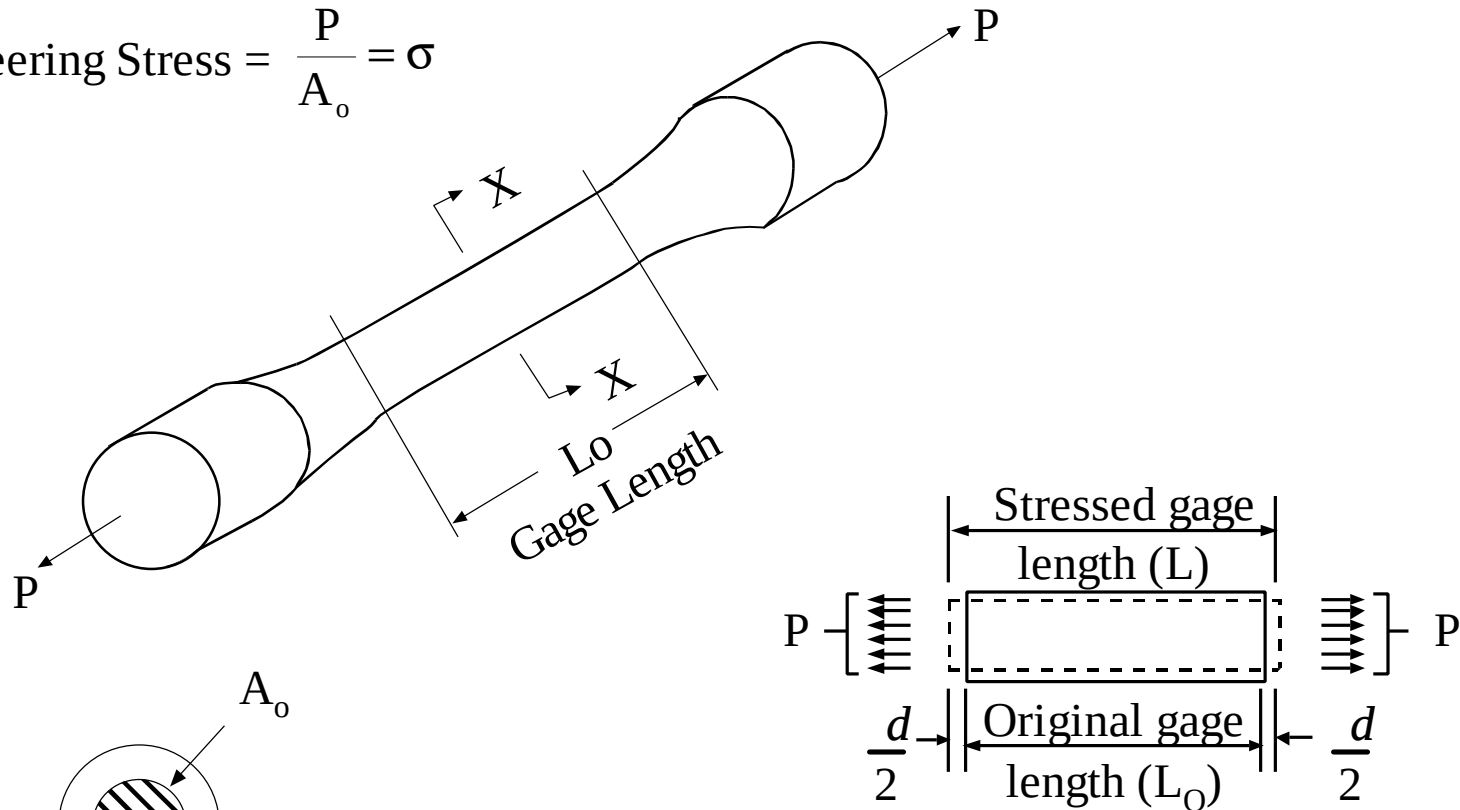
- **Terminology**
- **Primary Membrane Stresses in Shells**
- **Primary Membrane Stresses in Heads**
- **Code Design Equations for Shells and Heads**
- **Nozzle Reinforcement**
- **Discontinuity Stresses**
- **Code Allowable Stress Basis**
- **Wind and Earthquake**

Stress and Strain Definitions

- **Strain - Distortion per unit length. For a tensile test it's usually the elongation divided by the original stressed length. It may be applied directly or be the byproduct of an applied stress.**
- **Stress - Force divided by the area over which it is applied. For a tensile test the area is the original cross section. It may be applied directly or be the byproduct of an applied strain.**

Review of Strength of Materials

$$\text{Engineering Stress} = \frac{P}{A_0} = \sigma$$

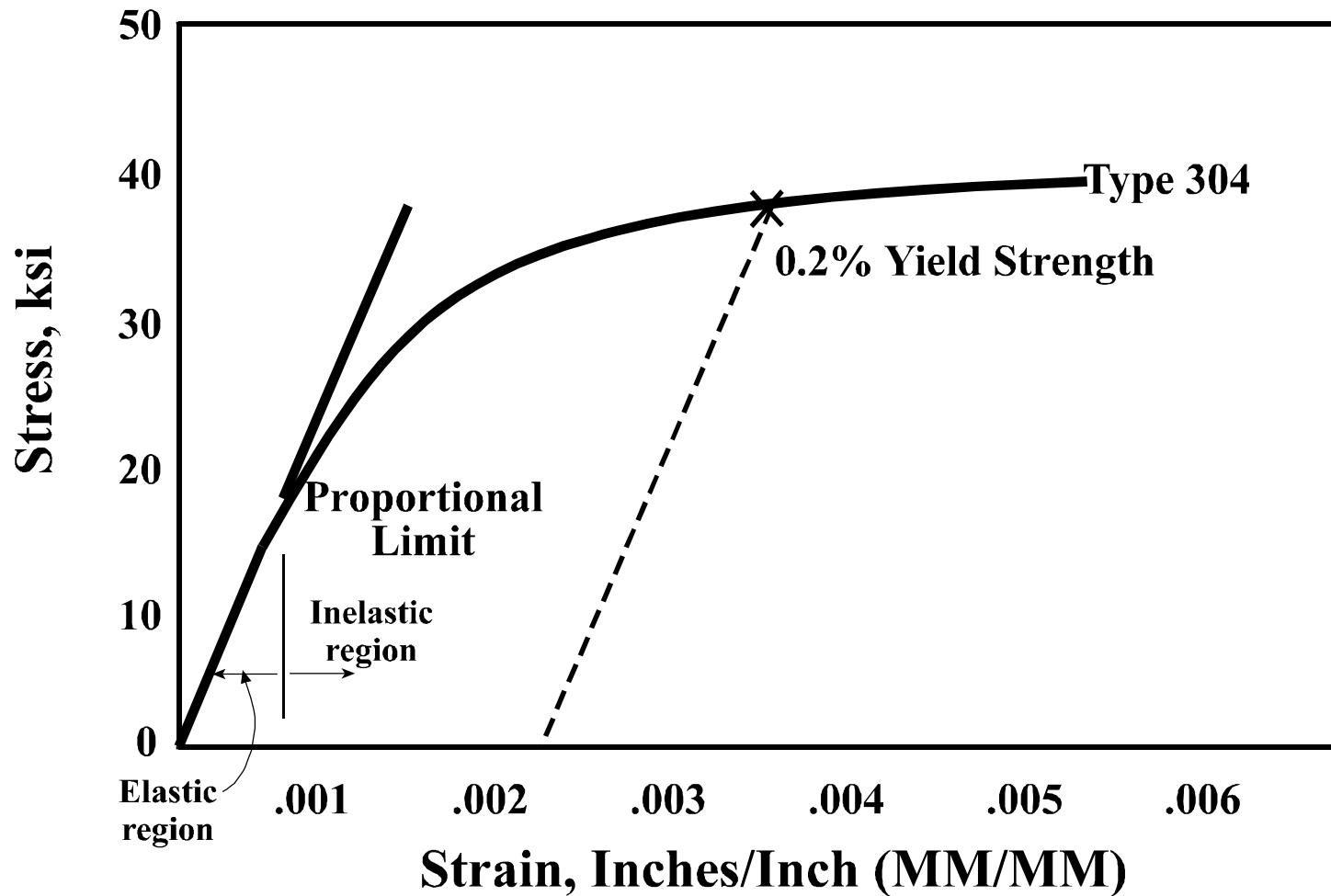


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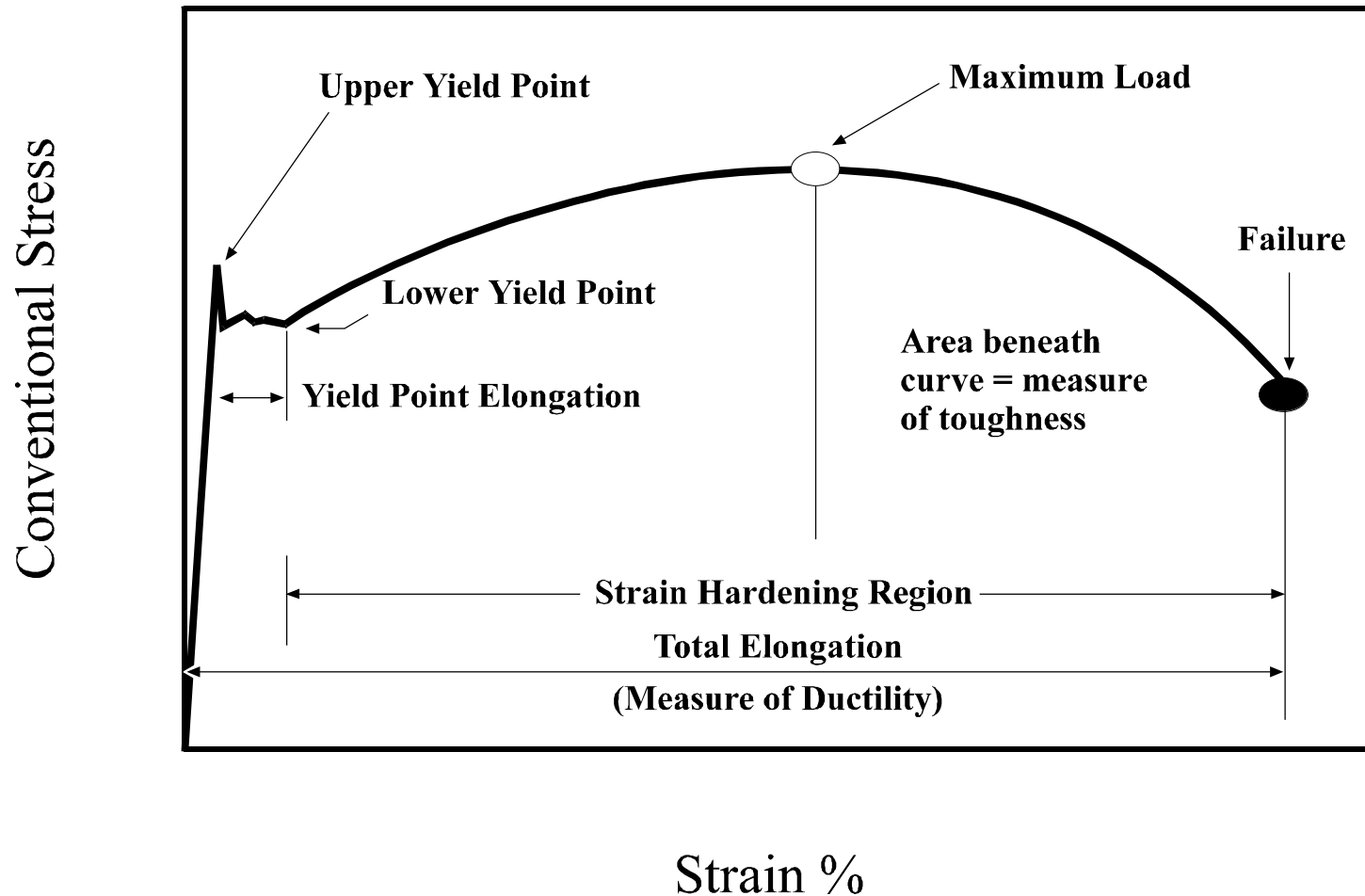
$$\text{Engineering Strain} = \frac{L - L_0}{L_0} = \frac{d}{L_0}$$

$$e = \frac{d}{L_0}$$

Typical Stress-Strain Curve for a Stainless Steel



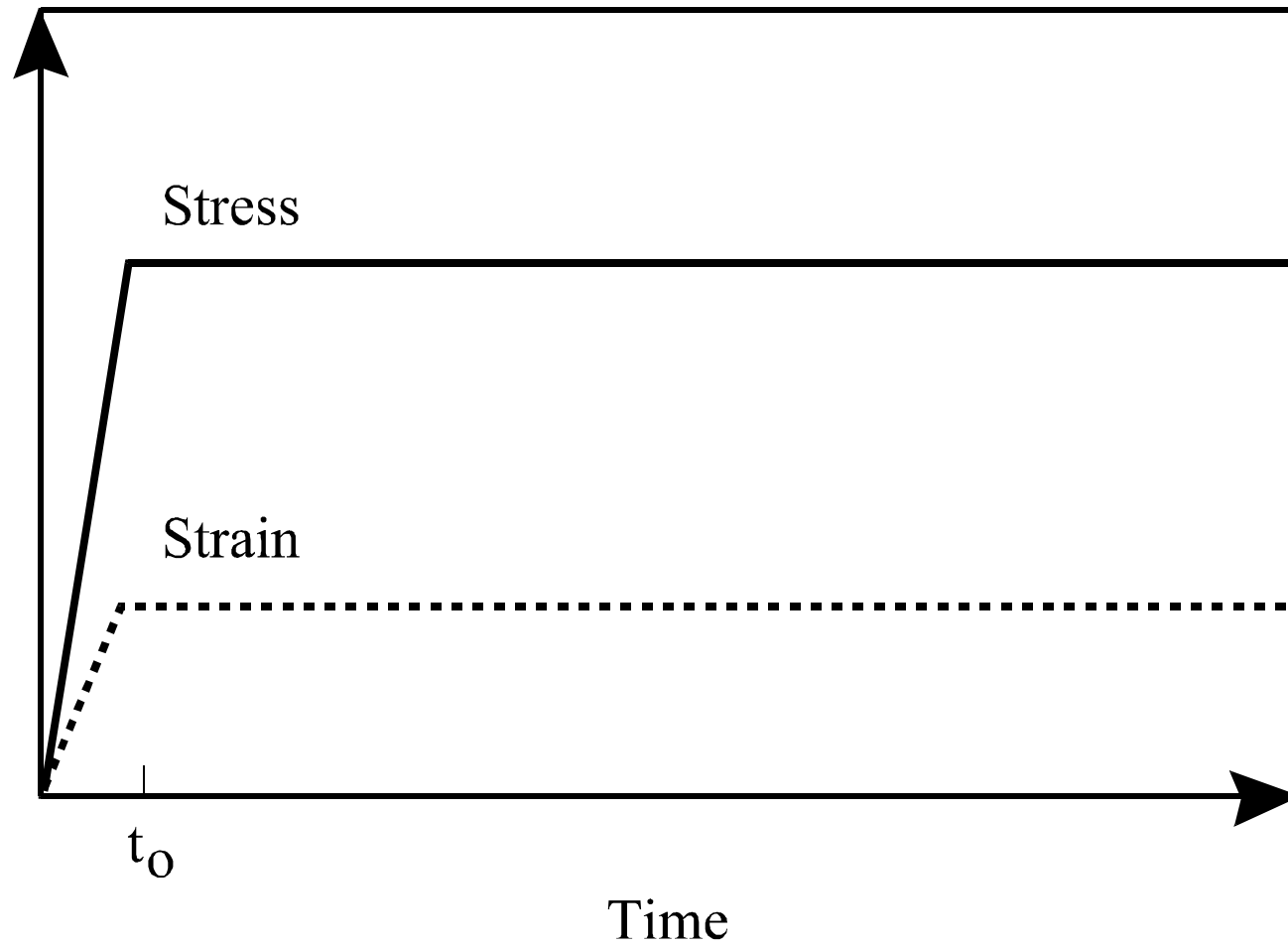
Typical Stress-Strain Curve for a Mild Steel



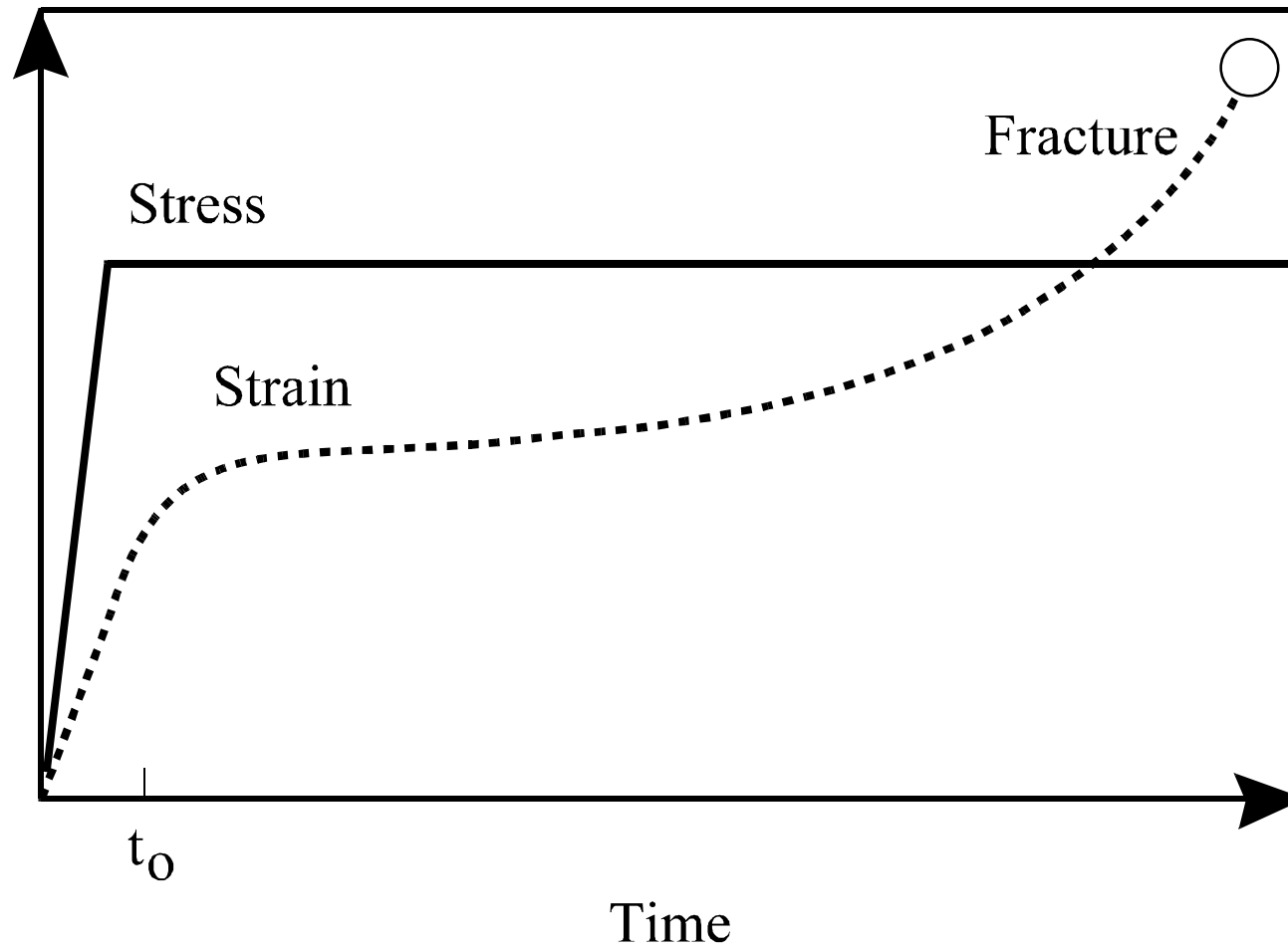
Stress - Strain Terms

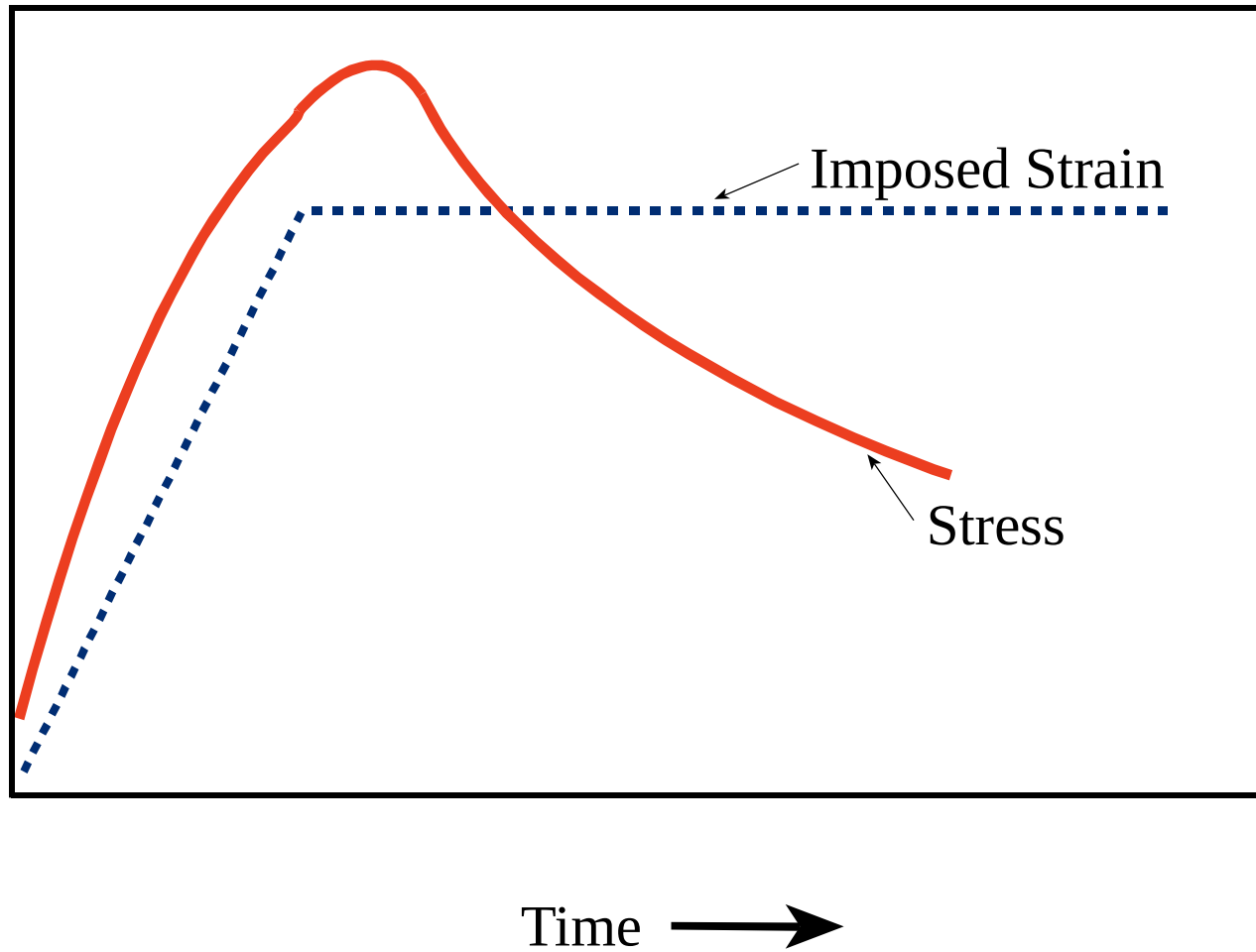
- **Creep - Continuous change in strain over time at elevated temperature under constant load or displacement conditions.**
- **Creep Strain - Increase in strain with time under constant loading conditions.**
- **Creep Relaxation - Reduction in hot stress with time under constant displacement conditions.**
- **Creep Rupture - Failure due to excessive accumulated creep strain.**

Histories from a Loading At Low Temperature



Histories from a Controlled Loading at Elevated Temperature





Stress - Strain Terms

- **Ductility - Ability to distort plastically before fracturing**
 - **Measured by elongation or area reduction in a tensile test.**
 - **Ductile material will distort dramatically before fracturing, giving warning of an overload.**
 - **Brittle material will distort very little before fracturing, giving little or no warning.**
 - **As temperature is lowered, ductile material can become brittle. This point is the transition temperature.**

Stress - Strain Terms

(continued)

- **Elasticity - Ability of a solid to deform in direct proportion to, and in phase with, increases or decreases in applied force, i.e., stress and strain are proportional**
- **Elastic Distortion - Strain is fully recovered when the stress is removed**
- **Plasticity - Ability of a material to deform inelastically without rupture**
- **Plastic (Inelastic) Distortion - Strain is not proportional to stress and is not recovered when the stress is removed, i.e. it is permanent**

Stress - Strain Terms

(continued)

- **Modulus of Elasticity (Young's Modulus) - Ratio of stress to strain before the proportional limit, e.g., the slope of the curve**
- **Proportional Limit - Stress at which stress and strain cease to be directly proportional (i.e., a straight line relationship)**

Stress - Strain Terms

(continued)

- **Strain Hardening - Increase in stress capacity due to internal strain redistribution in ductile materials**
- **Stress Rupture - Time dependent failure**
 - **Rupture is a function of time, temperature, and stress**

Stress - Strain Terms

(continued)

- **Toughness - Ability to absorb energy**
 - **Generally characterized by the area beneath the stress-strain curve**
 - **A common test method is the Charpy V-notch impact test**

- **Ultimate Strength - Maximum stress, based upon the original area, before failure**

Stress - Strain Terms

(continued)

- **Yield Strength - Stress at which a small additional stress increase results in a large additional strain**
 - Same as the proportional limit if there is a clear break between the elastic and inelastic portions of the stress-strain curve
 - If there is not a clear break between the elastic and inelastic portions of the curve it's defined as the stress at which a line beginning at 0.2% (0.002) strain and drawn parallel to the elastic portion of the curve intersects the stress-strain curve

Stress Analysis of Pressure Vessels

- **Basic Formulas for Stress**
- **ASME Code Pressure Design Equations**

Areas of Knowledge and Application

- **Analysis and design of pressure vessels can be complex**
- **Requires knowledge and application of:**
 - **Applied Mechanics**
 - **Strength of Materials**
 - **Fatigue**
 - **Fracture Mechanics**
 - **Plasticity**
 - **Creep**
 - **Provisions of all currently applicable codes**
 - **Stress Rupture**
 - **Metallurgy**
 - **Heat Transfer**
 - **Computational Methods (e.g., Finite Element Analysis)**
 - **Fabrication & Welding Techniques**
 - **Nondestructive Examination (NDE)**

Stress Analysis of Pressure Vessels

Types of Stress

■ **Primary Stress**

- **Caused by an applied force, strain is a response, i.e., a secondary event**
- **If excessive, can cause failure in a single application**
- **Necessary to satisfy equilibrium of forces and moments**
- **Not self-limiting**
- **Internal or external pressure**
- **Weight, wind, earthquake**

Stress Analysis of Pressure Vessels

Types of Stress (continued)

■ **Secondary Stress**

- **Caused by an applied strain, stress is a response, i.e., a secondary event**
- **Generally does not lead to failure in a single cycle**
- **Self-limiting (e.g. thermal stress)**
- **Local geometric effects, thermal stress, residual stresses from welding (often due to constraints)**

Loadings Causing Vessel Stresses

- **Internal or external design pressure**
- **Weight of the vessel and contents under operating or test conditions**
- **Superimposed static reactions from weight of attached equipment**
- **Internals**
- **Vessel attachments**

Loadings Causing Vessel Stresses

(continued)

- **Cyclic and dynamic reactions due to pressure or thermal variations**
- **Wind, snow, and seismic reactions**
- **Impact loads**
- **Temperature gradients and differential thermal expansion**
- **Residual stresses due to constraints**
- **Local stresses at geometric discontinuities**

Stress Analysis of Pressure Vessels

Terminology

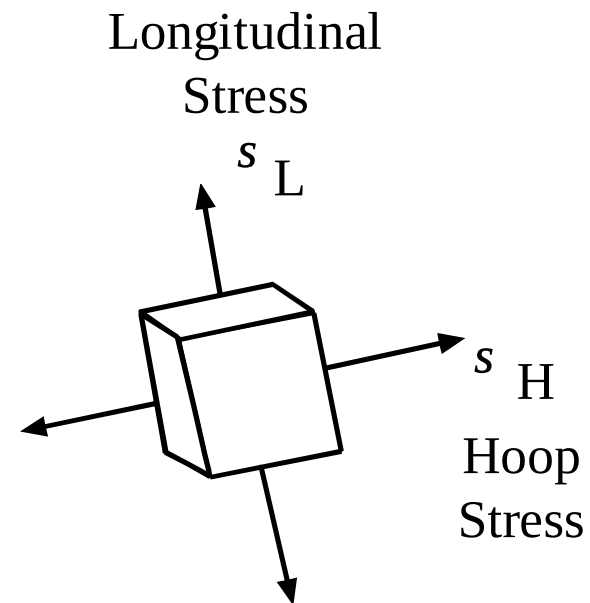
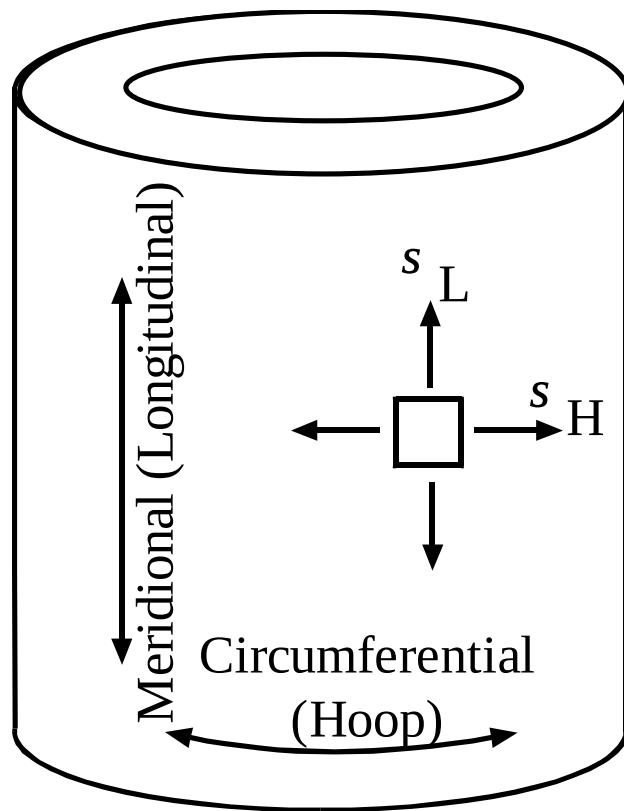
■ **Stress Types**

- **Membrane Stress**
 - **An essentially uniform stress averaged across the thickness of the cross-section**
- **Bending Stress**
 - **Stress level varies through the thickness of the cross-section**

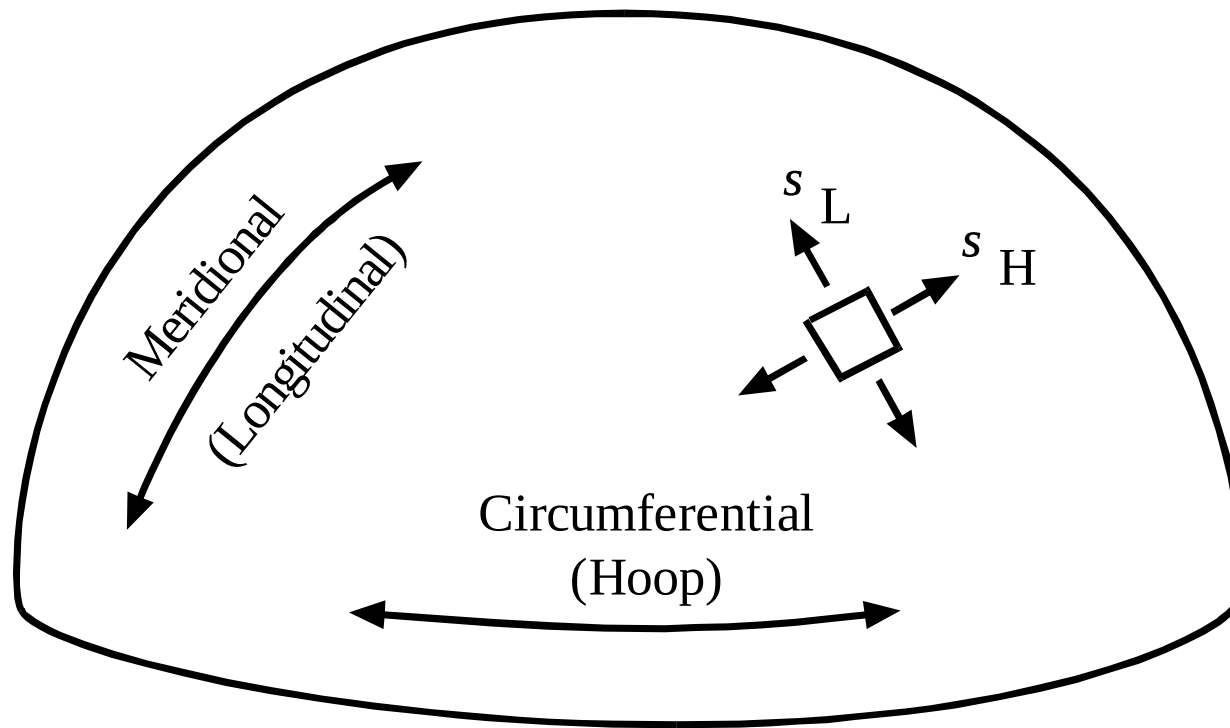
■ **Stress Direction**

- **Circumferential**
- **Longitudinal**

Cylindrical Vessel

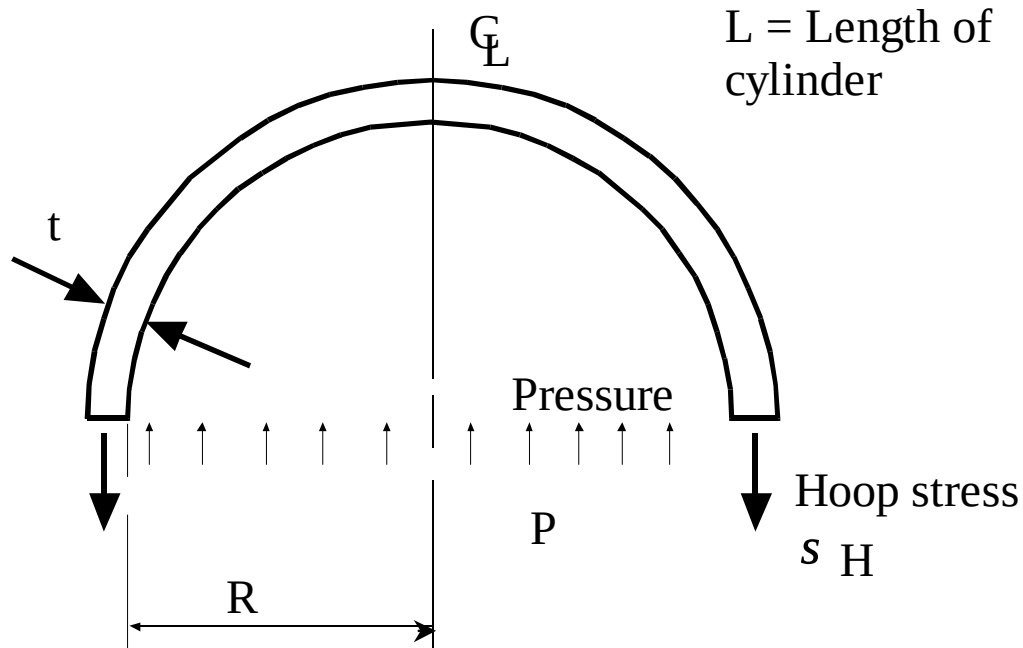


Spherical Vessel or Head



Hoop Stress

Sectional view of a pressure vessel
cylinder or sphere



Applied force = Pressure x fluid area
Reaction force = Stress x metal area

- a. Cylinder: Metal area = $2[tL]$
Pressure area = $2RL$

Equilibrium,

$$s_H[2(tL)] = P[2RL]$$

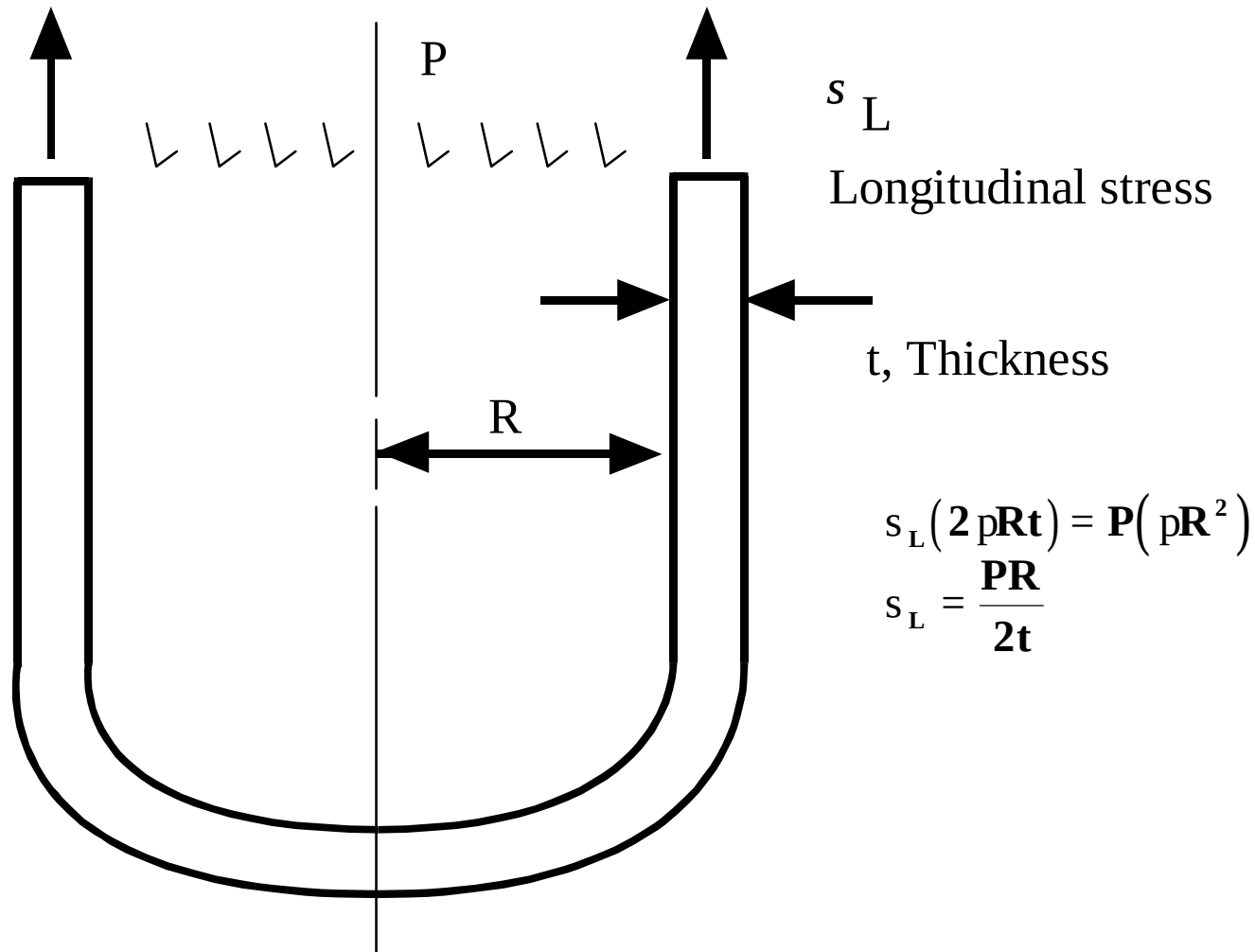
$$s_H = \frac{PR}{t}$$

- b. Sphere: Metal Area = $2\pi Rt$
Pressure area = πR^2

$$\sigma_H[2\pi Rt] = P[\pi R^2]$$

$$\sigma_H = \frac{pR}{2t}$$

Longitudinal Stress



Stresses in Pressure Vessels Due to Internal Pressure

Component	Hoop (Circumferential) Stress	Longitudinal (Meridional) Stress
Cylindrical Shell	$\frac{PR}{t}$	$\frac{PR}{2t}$
Spherical Shell or Hemispherical Head	$\frac{PR}{2t}$	$\frac{PR}{2t}$
2:1 Elliptical Head: At Center of Crown	$\frac{PR}{t}$	$\frac{PR}{t}$
At Knuckle	$-\frac{PR}{t}$	$\frac{PR}{2t}$

ASME Code Design Thickness Equations for Shells Section VIII, Division 1

■ Cylindrical Shells

- Circumferential stress (longitudinal joints)

$$t = \frac{PR}{SE - 0.6P}$$

$$\text{Limits } t \leq \frac{1}{2}R$$

$$P \leq 0.385SE$$

ASME Code Design Thickness Equations for Shells

- **Longitudinal Stress (circumferential joints)**

$$t = \frac{PR}{2SE + 0.4P}$$

$$\text{Limits } t \leq \frac{1}{2}R$$
$$P \leq 1.25SE$$

- **For circumferential stress (longitudinal joints), based on the outside radius**

$$t = \frac{PR_o}{SE + 0.4P}$$

ASME Code Design Thickness Equations for Shells

■ Spherical Shells

$$t = \frac{PR}{2SE - 0.2P}$$

Limits $t \leq 0.356R$
 $P \leq 0.665SE$

■ Spherical shells based upon the outside radius

$$t = \frac{PR_o}{2SE + 0.8P}$$

Pressure Vessel Heads

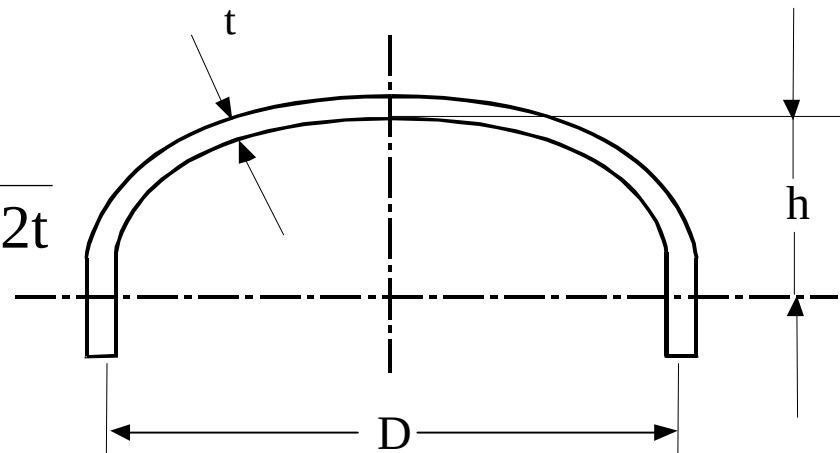
■ Pressure Vessel Heads Ellipsoidal

$$t = \frac{PDK}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{KD + 0.2t}$$

where

$$K = \frac{1}{6} \left[2 + \left(\frac{D}{2h} \right)^2 \right]$$

For a 2:1 ellipsoidal head $K=1$



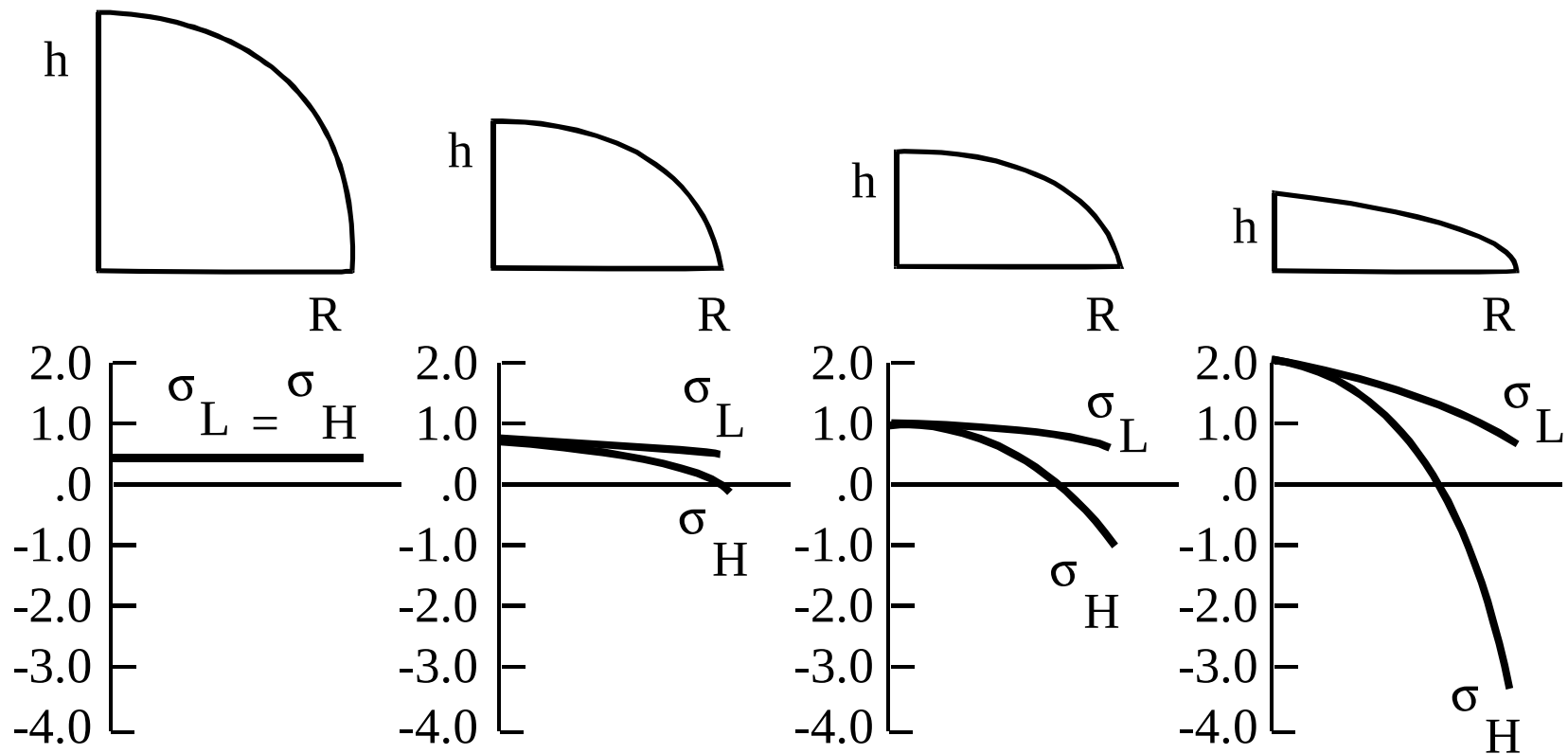
Elliptical Head/Cylinder Stress Ratios

R:h = 1:1

R:h = 1.42:1

R:h = 2:1

R:h = 3:1



ASME Code Design Thickness Equations for Heads

- **Pressure Vessel Heads**
 - **Conical (without transition knuckle)**

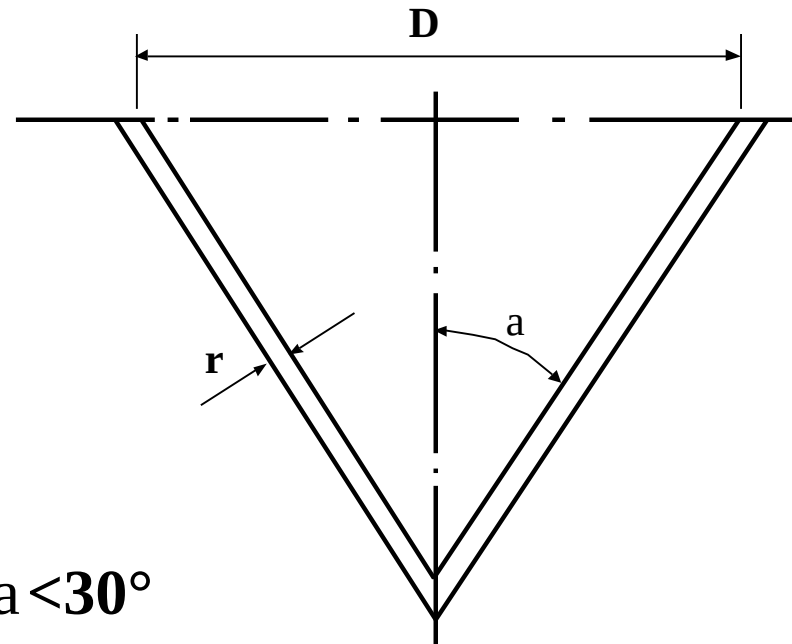
$$t = \frac{PD}{2 \cos a (SE - 0.6P)}$$

$$t = \frac{PD_o}{2 \cos a (SE + 0.4P)}$$

Limits:

Half Apex Angle, $a < 30^\circ$

$P \leq 1.25SE$



ASME Code Design Thickness Equations for Heads

■ Pressure Vessel Heads

– Toriconical heads (conical portion)

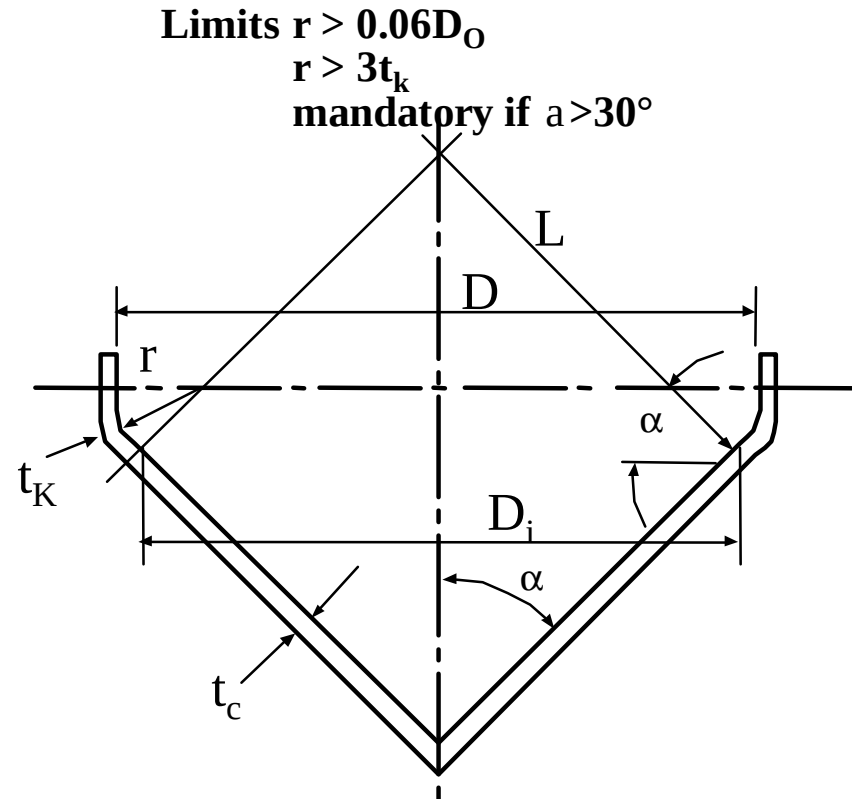
$$t_c = \frac{PD}{2 \cos a (SE - 0.6P)}$$

– Knuckle portion

$$t_k = \frac{PLM}{2SE - 0.2P}$$

$$L = \frac{D_i}{2 \cos a}$$

$$M = \frac{1}{4} \left(3 + \sqrt{\frac{L}{r}} \right)$$



Pressure Vessel Heads

■ Pressure Vessel Heads Torispherical

$$t = \frac{PLM}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{LM + 0.2t}$$

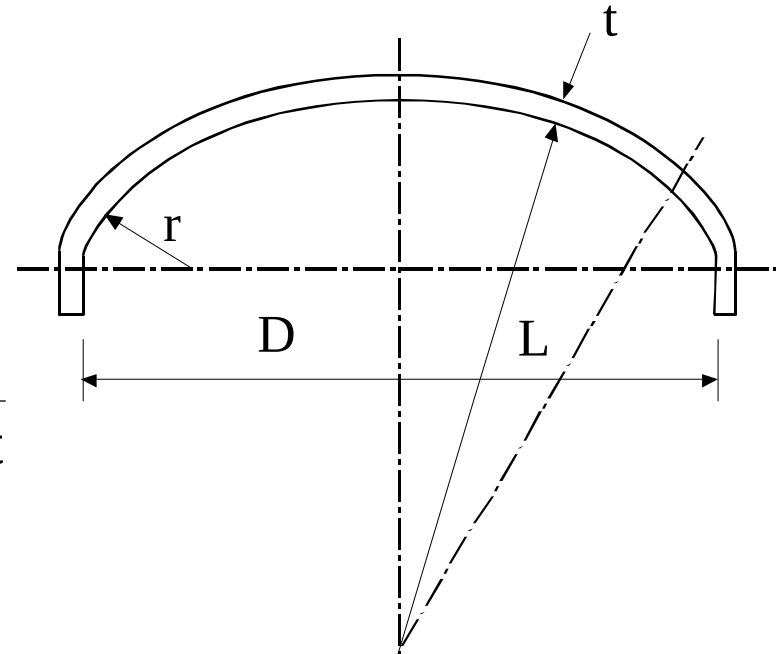
where

$$M = \frac{1}{4} \left(\frac{L}{r} \right)^3 + \sqrt{\frac{L}{r}}$$

for the typical case where

$r=0.06L$ and $L=\text{skirt OD}$,

$$t = \frac{0.885PL}{SE - 0.1P}$$



Symbols

- t = Minimum required thickness, exclusive of corrosion allowance**
- t_c = Minimum required thickness of cone, exclusive of corrosion allowance**
- t_R = Minimum required thickness of knuckle, exclusive of corrosion allowance**
- P = Internal design pressure**
- S = Tensile allowable stress value at design temperature**
- E = Joint efficiency**
- R = Inside radius**
- R_O = Outside radius**

Symbols *(continued)*

D = Inside diameter

D_O = Outside diameter

**D_L = Inside diameter of conical portion of
toriconical head = $D - 2r(1 - \cos a)$**

a = One half apex angle of cone

r = Inside knuckle radius

L = Inside crown radius

h = Minor axis of elliptical head

Vessel Weld Joint Categories

- **Assigned to permit application of specific rules and restrictions, including joint details and efficiencies**
 - **Category A — Longitudinal shell, heads, diameter transitions, hemispherical head to shell, highly stressed welds**
 - **Category B — Circumferential shell, head (other than hemispherical) to shell**
 - **Category C — Flanges**
 - **Category D — Nozzles to shell**

UW-3

PART UW — WELDED VESSELS

UW-5

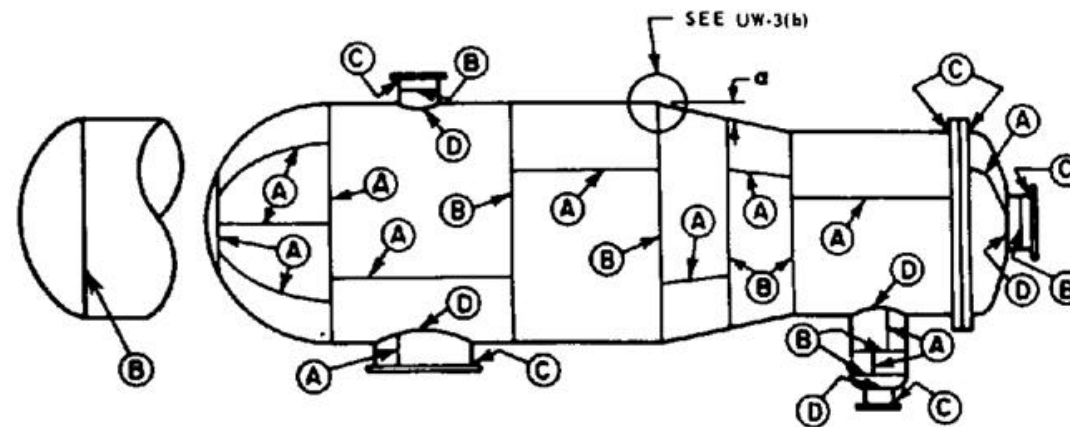


FIG. UW-3 ILLUSTRATION OF WELDED JOINT LOCATIONS TYPICAL OF CATEGORIES A, B, C, AND D

Efficiency of Welded Joints (E)

(Excerpt from ASME Code Table UW-12)

		Degree of Radiographic Examination		
No.	Type of Joint	Full	Spot	None
1	Double-welded butt joint or single-welded butt joint with backing strip which does not remain in place	1.00	0.85	0.70
2	Single-welded butt joint with backing strip which remains in place	0.90	0.80	0.65
3	Single-welded butt joint without use of backing strip	—	—	0.60

UOP permits only type 1 joints in hydrocarbon service.

Weld Examination

- **Welds shall be examined by full or spot radiography**
 - **Full — Radiography of the entire length of the weld joint**
 - **Spot — Radiographic examination of one spot in each 50 feet or fraction thereof for each welder, weld method, or type of joint**
- **Ultrasonic examination may be substituted for radiography for the final closure seam if it is not possible to obtain interpretable radiographs**

Section VIII, Allowable Stress Basis

■ **Division 1**

- **The lower of the following at temperature:**
 - **2/3 yield**
 - **1/3.5 ultimate tensile**
 - **2/3 average rupture stress in 100,000 hours**
 - **80% minimum stress to rupture in 100,000 hours**
 - **Average stress for creep of 1% in 100,000 hours**

Section VIII, Allowable Stress Basis (continued)

■ **Division 1 (continued)**

- **Note: For many steels, yield and tensile strengths may first increase, then decrease as temperatures rise above ambient**
 - **Ambient allowable is used until a lower one is required (usually at 650 °F)**
- **In combination with wind or earthquake loads, allowable stress may be increased to 1.2 times the values listed in the code**

Allowable Stress Table

Table 1A

2001 SECTION II

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3;* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class)

Line No.	Nominal Composition	Product Form	Spec No.	Type/Grade	Alloy Desig./UNS No.	Class/Cond./Temper	Size/Thickness, in.	P-No.	Group No.
1	Carbon steel	Castings	SA-352	LCB	J03003	...	...	1	1
2	Carbon steel	Plate	SA-515	65	K02800	...	...	1	1
3	Carbon steel	Plate	SA-516	65	K02403	...	...	1	1
4	Carbon steel	Wld. pipe	SA-671	CB65	K02800	...	...	1	1
5	Carbon steel	Wld. pipe	SA-671	CC65	K02403	...	...	1	1
6	Carbon steel	Wld. pipe	SA-672	B65	K02800	...	...	1	1
7	Carbon steel	Wld. pipe	SA-672	C65	K02403	...	...	1	1
8	Carbon steel	Sheet	SA-414	E	K02704	...	...	1	1
9	Carbon steel	Plate	SA-662	B	K02203	...	...	1	1
10	Carbon steel	Plate	SA-537	...	K12437	1	$2\frac{1}{2} < t \leq 4$	1	2
11	Carbon steel	Wld. pipe	SA-691	CM5H-70	K12437	...	$2\frac{1}{2} < t \leq 4$	1	2
01	Carbon steel	Plate	SA/EN 10028-2	P295GH	...	...	$2\frac{1}{4} < t \leq 4$	1	1
01	Carbon steel	Plate	SA/EN 10028-2	P295GH	...	...	$\leq 2\frac{1}{4}$	1	1
01	...	...	...	...	...	...	...	...	...
14	Carbon steel	Plate	SA-455	...	K03300	...	$0.58 < t \leq \frac{3}{4}$	1	2
16	Carbon steel	Bar	SA-675	70	...	...	...	1	2
17	Carbon steel	Forgings	SA-105	...	K03504	...	...	1	2
18	Carbon steel	Forgings	SA-181	...	K03502	70	...	1	2
19	Carbon steel	Castings	SA-216	WCB	J03002	...	...	1	2
01	Carbon steel	Forgings	SA-266	2	K03506	...	...	1	2
21	Carbon steel	Forgings	SA-266	4	K03017	...	...	1	2
22	Carbon steel	Forgings	SA-350	LF2	K03011	...	...	1	2
23	Carbon steel	Forgings	SA-508	1	K13502	...	...	1	2
24	Carbon steel	Forgings	SA-508	1A	K13502	...	...	1	2
25	Carbon steel	Forgings	SA-541	1	K03506	...	...	1	2
26	Carbon steel	Forgings	SA-541	1A	K03506	...	...	1	2
27	Carbon steel	Cast pipe	SA-660	WCB	J03603	...	...	1	2
28	Carbon steel	Forgings	SA-765	II	K03047	...	...	1	2
29	Carbon steel	Plate	SA-515	70	K03101	...	...	1	2
30	Carbon steel	Plate	SA-516	70	K02700	...	...	1	2
31	Carbon steel	Wld. pipe	SA-671	CB70	K03101	...	...	1	2
32	Carbon steel	Wld. pipe	SA-671	CC70	K02700	...	...	1	2
33	Carbon steel	Wld. pipe	SA-672	B70	K03101	...	...	1	2
34	Carbon steel	Wld. pipe	SA-672	C70	K02700	...	...	1	2

Allowable Stress Table (Continued)

PART D—PROPERTIES

Table 1A

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3;* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class)

Line No.	Min. Tensile Strength, ksi	Min. Yield Strength, ksi	Applic. and Max. Temp. Limits (NP = Not Permitted) (SPT = Supports Only)			External Pressure Chart No.	Notes
			I	III	VIII-1		
1	65	35	NP	700	650	CS-2	G1, G17
2	65	35	1000	700	1000	CS-2	G10, S1, T2
3	65	35	850	700	1000	CS-2	G10, S1, T2
4	65	35	NP	700	NP	CS-2	S6, W10, W12
5	65	35	NP	700	NP	CS-2	S6, W10, W12
6	65	35	NP	700	NP	CS-2	S6, W10, W12
7	65	35	NP	700	NP	CS-2	S6, W10, W12
8	65	38	NP	NP	900	CS-2	G10, G35, T1
9	65	40	NP	NP	700	CS-3	T1
10	65	45	NP	700	650	CS-4	T1
11	65	45	NP	700	NP	CS-4	G26, T1, W10, W12
12	66.5	37.5	850	NP	1000	CS-2	G10, S1, T1
13	66.5	...	850	NP	1000	CS-2	G10, G43, S1, T1
14	...	...	...	...	...	...	...
15	70	35	NP	400 (Cl. 3 only)	650	CS-2	...
16	70	35	850	650 (Cl. 3 only)	1000	CS-2	G10, G15, G18, G22, G35, S1, T2
17	70	36	1000	700	1000	CS-2	G10, G18, G35, S1, T2
18	70	36	1000	700	1000	CS-2	G10, G18, G35, S1, T2
19	70	36	1000	700	1000	CS-2	G1, G10, G17, G18, S1, T2
20	70	36	1000	700	1000	CS-2	G10, G18, S1, T2
21	70	36	NP	NP	1000	CS-2	G10, T2
22	70	36	NP	700	1000	CS-2	G10, T2
23	70	36	NP	700	1000	CS-2	G10, T2
24	70	36	NP	700	1000	CS-2	G10, T2
25	70	36	NP	700	1000	CS-2	G10, T2
26	70	36	NP	700	1000	CS-2	G10, T2
27	70	36	1000	700	NP	CS-2	G1, G10, G17, G18, S1, T2
28	70	36	NP	NP	650	CS-2	...
29	70	38	1000	700	1000	CS-2	G10, S1, T2
30	70	38	850	700	1000	CS-2	G10, S1, T2
31	70	38	NP	700	NP	CS-2	S5, W10, W12
32	70	38	NP	700	NP	CS-2	S6, W10, W12
33	70	38	NP	700	NP	CS-2	S5, W10, W12
34	70	38	NP	700	NP	CS-2	S6, W10, W12

Allowable Stress Table (Continued)

Table 1A

2001 SECTION II

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3,* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class)

Line No.	Maximum Allowable Stress, ksi (Multiply by 1000 to Obtain psi), for Metal Temperature, °F, Not Exceeding												
	-20 to 100	150	200	250	300	400	500	600	650	700	750	800	850 900
1	18.6	18.6	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	...	...	...
2	18.6	18.6	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	13.9	11.4	8.7 5.9
3	18.6	18.6	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	13.9	11.4	8.7 5.9
4	18.6	...	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	...	...	...
5	18.6	...	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	...	...	...
6	18.6	...	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	...	...	...
7	18.6	...	18.6	...	18.6	18.6	18.6	17.9	17.3	16.7	...	...	...
8	18.6	18.6	18.6	...	18.6	18.6	18.6	18.6	18.6	16.9	13.9	11.4	8.7 5.9
9	18.6	18.6	18.6	...	18.6	18.6	18.6	18.6	18.6	16.9	...	...	...
10	18.6	...	18.6	...	18.6	18.6	18.6	18.6	18.6	16.9	...	...	...
11	18.6	...	18.6	...	18.6	18.6	18.6	18.6	18.6	16.9	...	...	...
01 12	19.0	19.0	19.0	19.0	19.0	19.0	19.0	19.0	18.5	16.9	13.9	11.4	8.7 5.9
01 13	19.0	19.0	19.0	19.0	19.0	19.0	19.0	19.0	19.0	16.9	13.9	11.4	8.7 5.9
01 14	...	...	...	...	...	...	...	...	...	...	...	...	...
15	20.0	20.0	20.0	...	20.0	19.9	19.0	17.9	17.3	...	...	...	...
16	20.0	20.0	20.0	...	20.0	19.9	19.0	17.9	17.3	16.7	14.8	12.0	9.3 6.7
17	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
18	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
19	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
01 20	20.0	...	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
21	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
22	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
23	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
24	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
25	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
26	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
27	20.0	...	20.0	...	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3 6.7
28	20.0	20.0	20.0	...	20.0	20.0	19.6	18.4	17.8	...	...	...	...
29	20.0	20.0	20.0	...	20.0	20.0	20.0	19.4	18.8	18.1	14.8	12.0	9.3 6.7
30	20.0	20.0	20.0	...	20.0	20.0	20.0	19.4	18.8	18.1	14.8	12.0	9.3 6.7
31	20.0	...	20.0	...	20.0	20.0	20.0	19.4	18.8	18.1	...	...	...
32	20.0	...	20.0	...	20.0	20.0	20.0	19.4	18.8	18.1	...	...	...
33	20.0	...	20.0	...	20.0	20.0	20.0	19.4	18.8	18.1	...	...	...
34	20.0	...	20.0	...	20.0	20.0	20.0	19.4	18.8	18.1	...	...	...

Section VIII, Allowable Stress Basis (continued)

■ Division 2

- Lower of the following at temperature (below the creep range):**
 - 2/3 yield**
 - 1/3 ultimate tensile**
- Above creep range, Division 1 allowables must be used**

■ Allowable stresses for materials permitted by the Code are listed in ASME Section II, Part D

External Pressure Design

■ Internal Pressure

- Allowable stress is a function of material properties

■ External Pressure

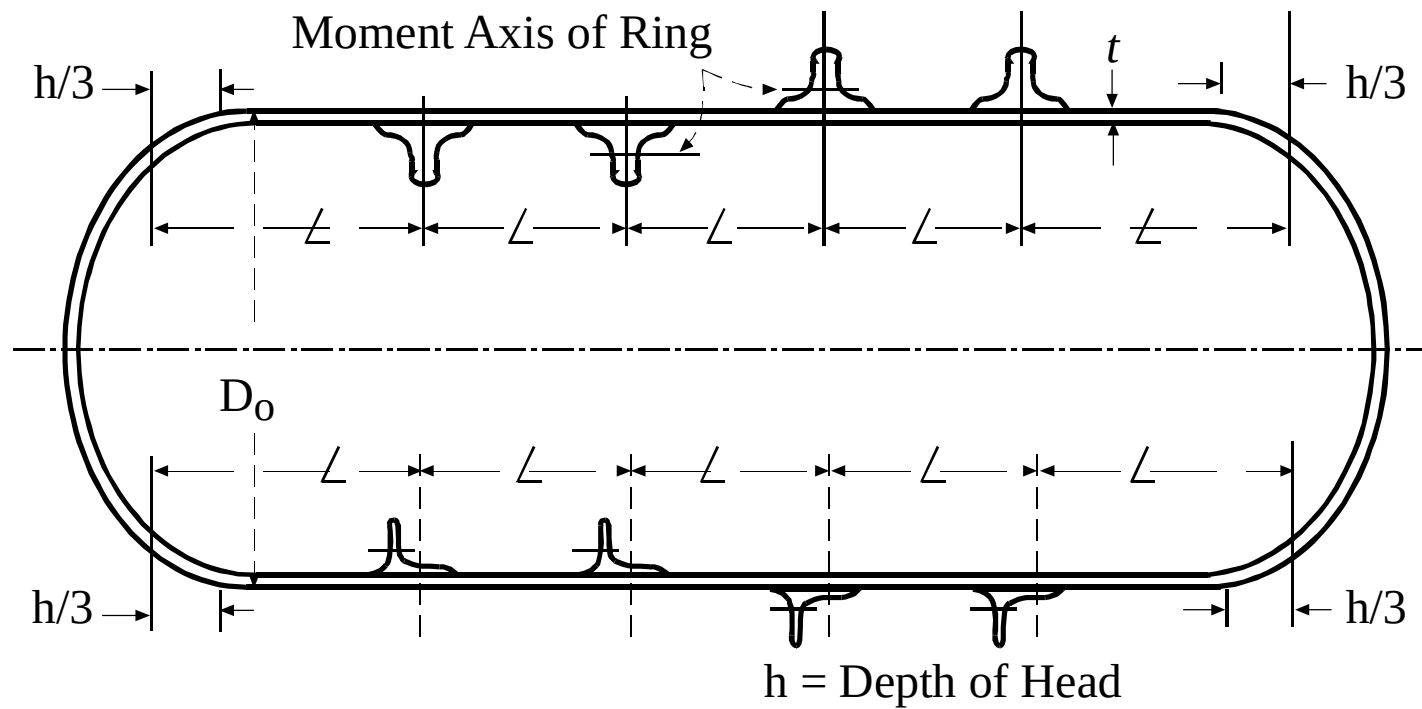
- Stability (buckling) becomes a concern
- Allowable stress is a function of material and geometrical properties

External Pressure Design

(continued)

- **Vessel diameter fixed**
 - **Variables are:**
 - **Length (between stiffeners)**
 - **Thickness**
- **Solutions**
 - **Increase t**
 - **Reduce length**
- **Length is decreased by adding stiffening rings**
- **Design procedure is trial-and-error**

External Pressure Design Length



External Pressure Design

Code Design Method

- **Step 1**
 - Assume a thickness t and determine the length between stiffeners, L
 - Calculate L/D_o , D_o/t
- **Step 2**
 - Find factor A from figure G of ASME Section II, Part D
- **Step 3**
 - Find B , using the proper chart for the material from ASME Section II, Part D

External Pressure Design Code Design Method (continued)

■ Step 4

- Calculate allowable external pressure,

$$P_{\text{ext}} = \frac{4B}{3(D_O/t)}$$

- or, for A values to the left of chart,

$$P_{\text{ext}} = \frac{2AE}{3(D_O/t)}$$

■ Step 5

- If $P_{\text{ext}} < \text{applied external pressure}$, repeat Step 1, using a larger t or a smaller L
- If $P_{\text{ext}} \geq \text{applied external pressure}$, design okay

External Pressure Design Code Design Method (continued)

- **Example: external pressure = 15psi**

$$T = 800\text{ }^{\circ}\text{F}$$

$$D_0 = 500\text{ mm}$$

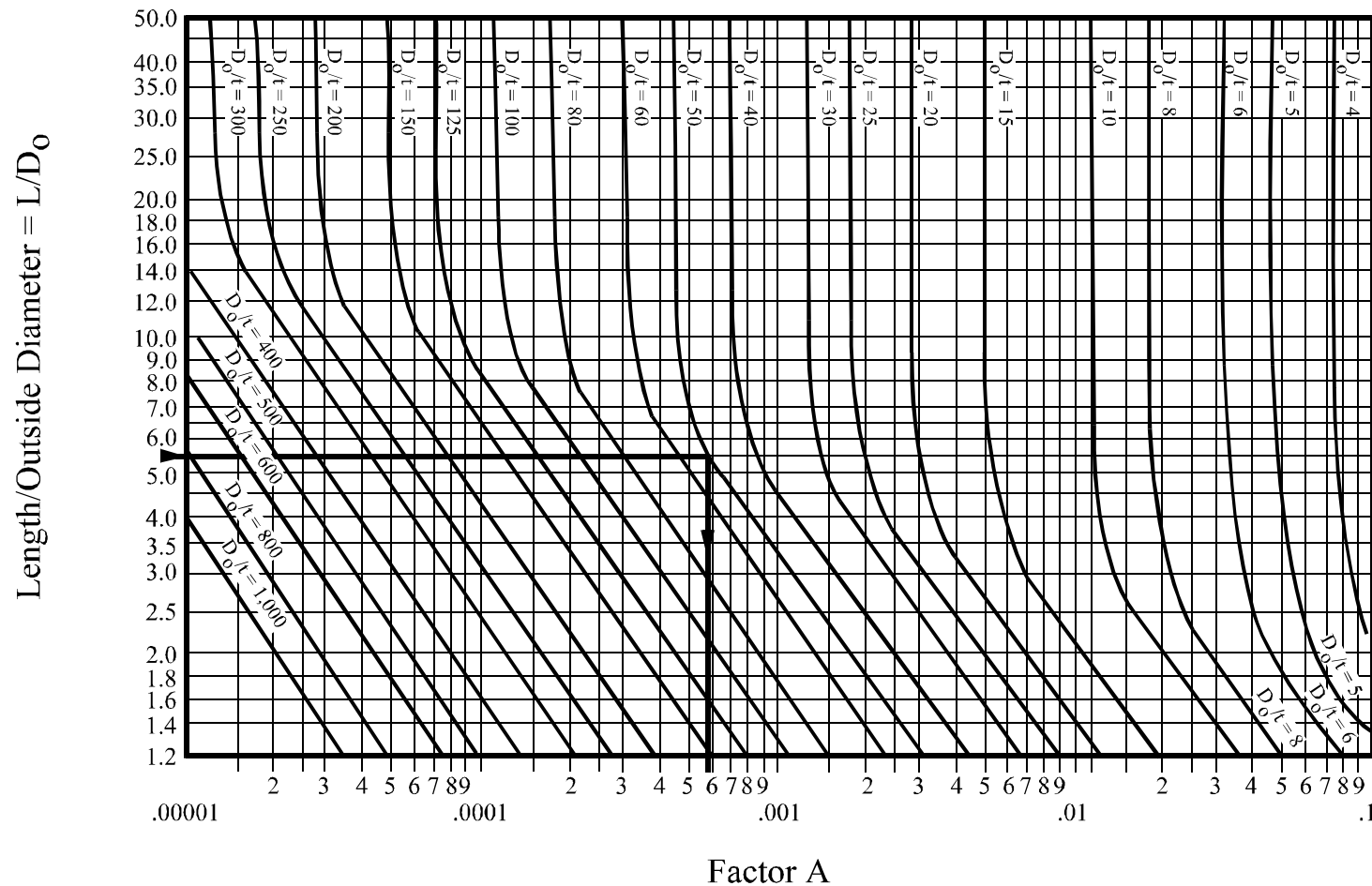
$$L = 2750\text{ mm}$$

$$t = 10\text{ mm}$$

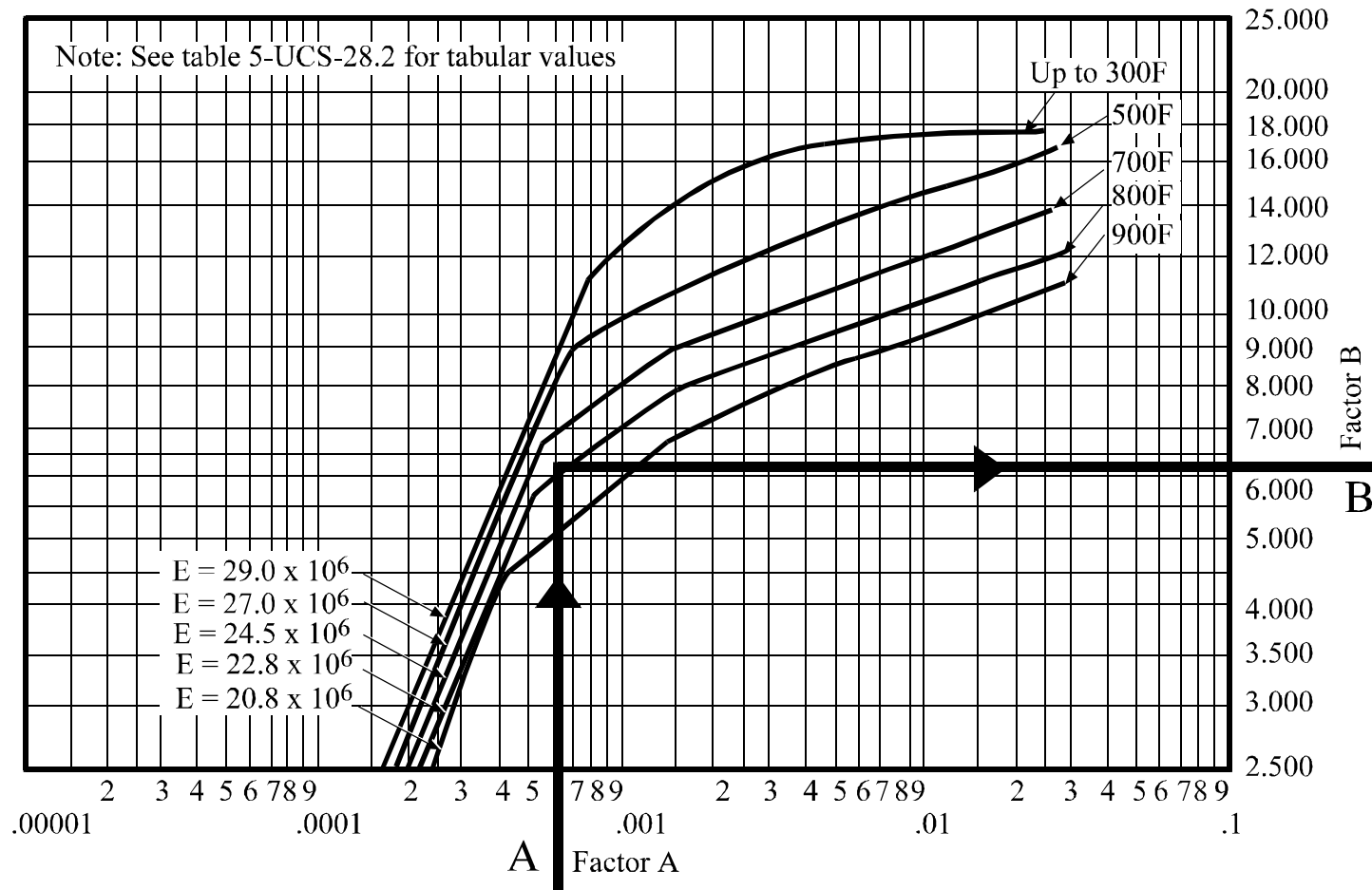
$$D_0/t = 50 \qquad L/D_0 = 2750/500 = 5.5$$

$$A = 0.0006 \text{ (per the following slide)}$$

External Pressure Design Code Design Method (continued)



External Pressure Design



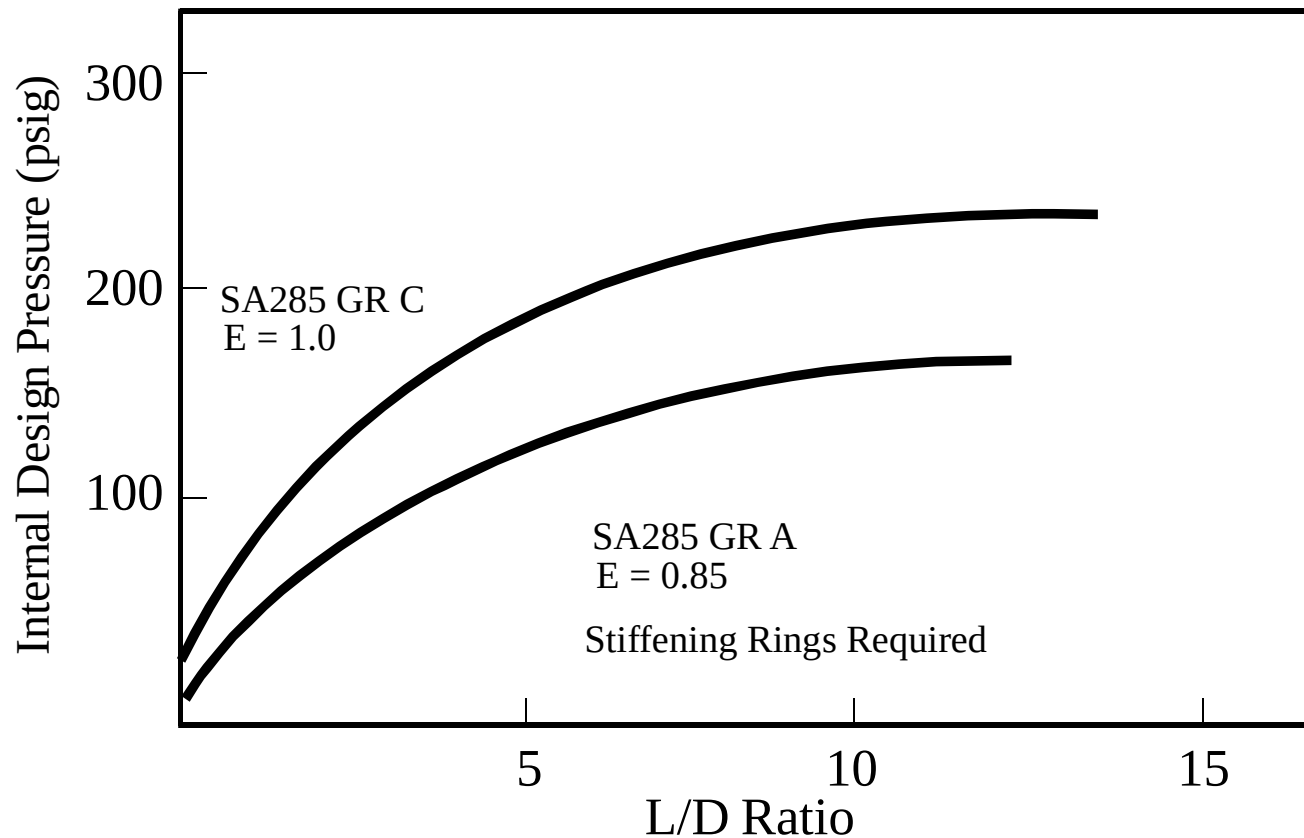
External Pressure Design Code Design Method (continued)

$$\mathbf{B = 6200psi}$$

$$\mathbf{P_{ext} = \frac{4(6200)}{3(50)} = \frac{24,800}{150} = 165psi > 15psi \text{ OK}}$$

Carbon Steel Vessels 500 °F

Full Vacuum Design



Illustrates the internal design pressure above which no stiffening rings would be required in accordance with the ASME code, for two material specifications.

Axial Compression

- **Maximum permitted axial compressive stress is the lower of the following:**
 - **Allowable tensile stress**
 - **Stress determined as follows:**
 - **Determine (outside radius/minimum required thickness)(R_o/t)**
 - **Determine $A=0.125/(R_o/t)$**
 - **Enter the appropriate external pressure chart and read “B,” the allowable compressive stress**
 - **Compare the allowable stress to the applied stress**
 - **If allowable stress is less than applied stress, increase t and repeat above steps**

Nozzle Reinforcement

- **Nozzle opening reduces the shell strength**
- **Replace cross-sectional area of metal removed**
- **Available reinforcement includes excess shell and nozzle thickness**
- **Limits of effective reinforcement defined by the code**

Nozzle Reinforcement (continued)

- **Factor “F” used for integrally reinforced nozzles since longitudinal stress is equal to half of the hoop stress**
- **Add additional reinforcement, if required**
- **Additional reinforcement may be provided by surface pads, insert plates, thickened full or partial shell courses, or thickened nozzle necks (integrally reinforced)**

Nozzle Reinforcement

(continued)

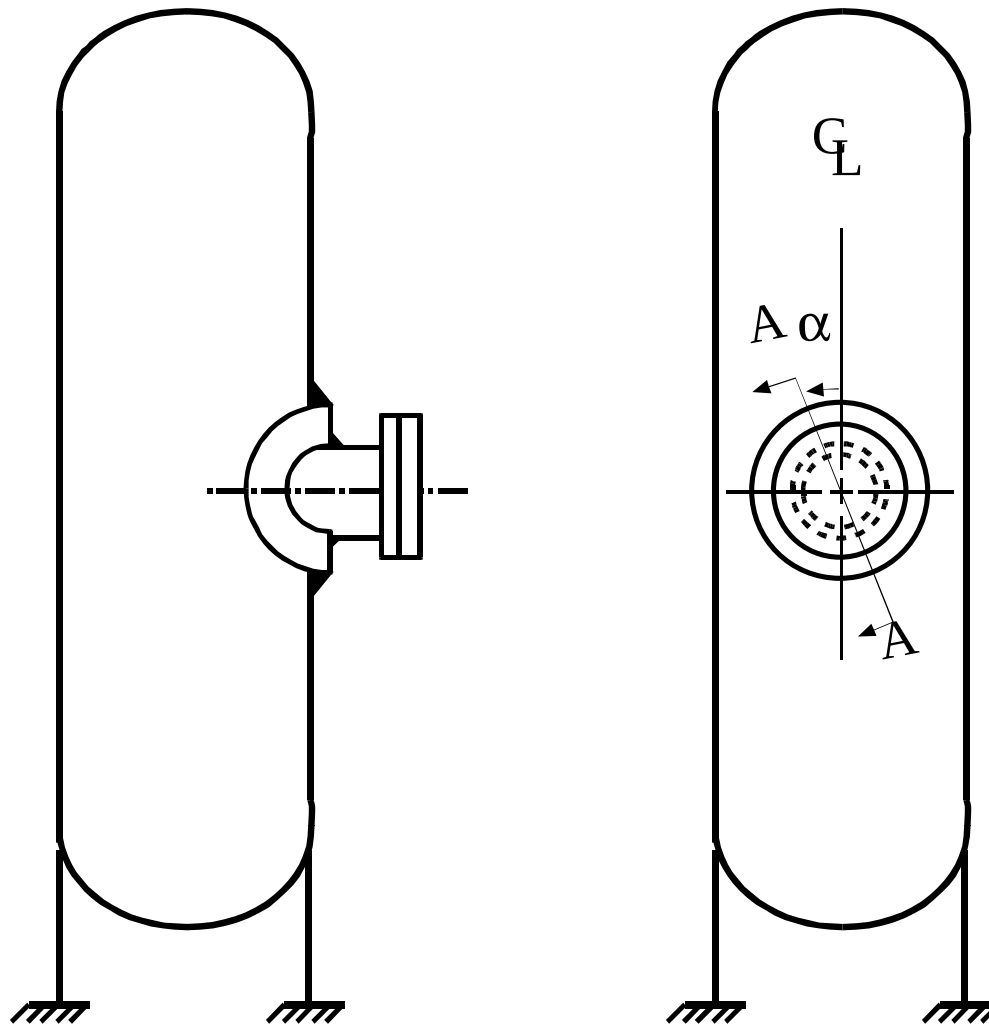
- **Per UW16, integral nozzle definition includes insert plate design**
- **Small openings do not require any additional reinforcement under the following conditions -**
 - **Finished openings equal to or less than 3.5 inches in diameter in vessel shells or heads with a required minimum thickness of 3/8 inch or less**
 - **Finished openings equal to or less than 2.375 inches in diameter in vessel shells or heads with a required minimum thickness over 3/8 inch**

Nozzle Reinforcement

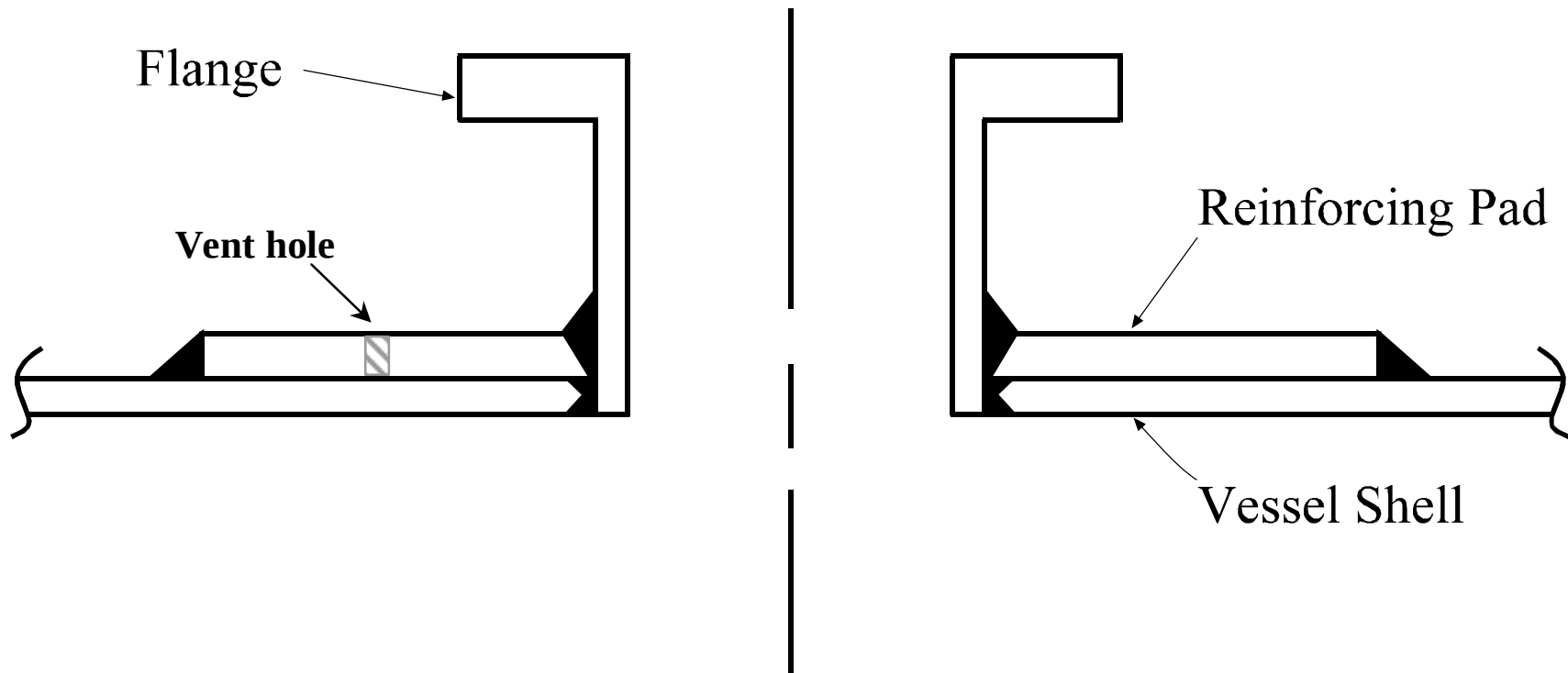
(continued)

- **Openings in flat heads where the opening diameter is less than one-half the head diameter shall be reinforced by replacing half of the area removed by the equation $A = 0.5dt$**
- **Reinforcement of large openings (UG-36) requires special consideration because area replacement is no longer a reasonable approximation**
 - **Large is defined as one-half the vessel diameter up to 60 inch diameter vessels and one-third the diameter for larger vessels**

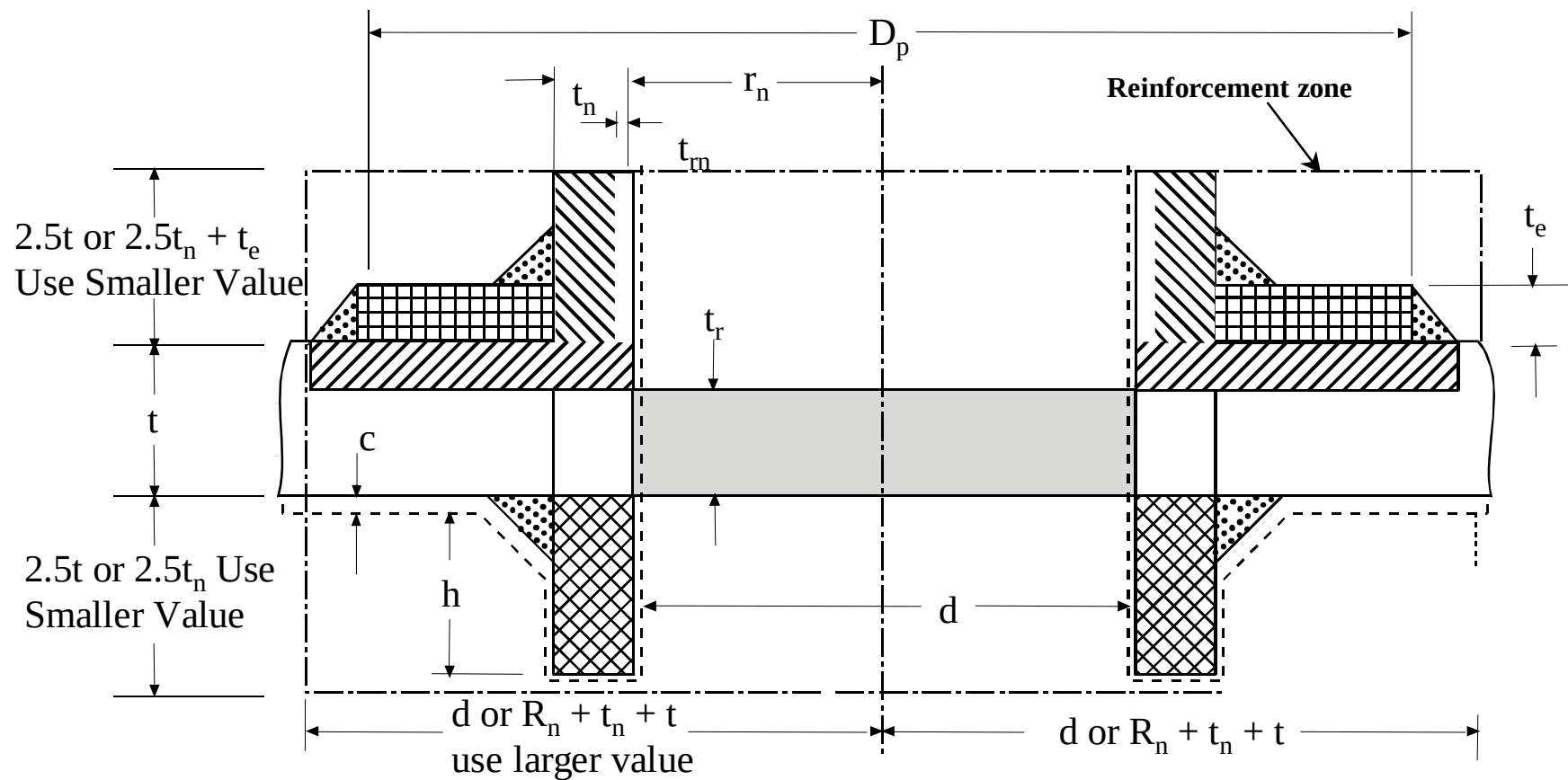
Nozzle Reinforcement



Section A-A



Nomenclature and Formulas for Reinforced Openings



Opening Reinforcement



A = Reinforcement area required



A₁ = Area available in shell



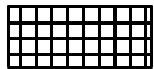
A₂ = Area available in outer nozzle



A₃ = Area available in inner nozzle



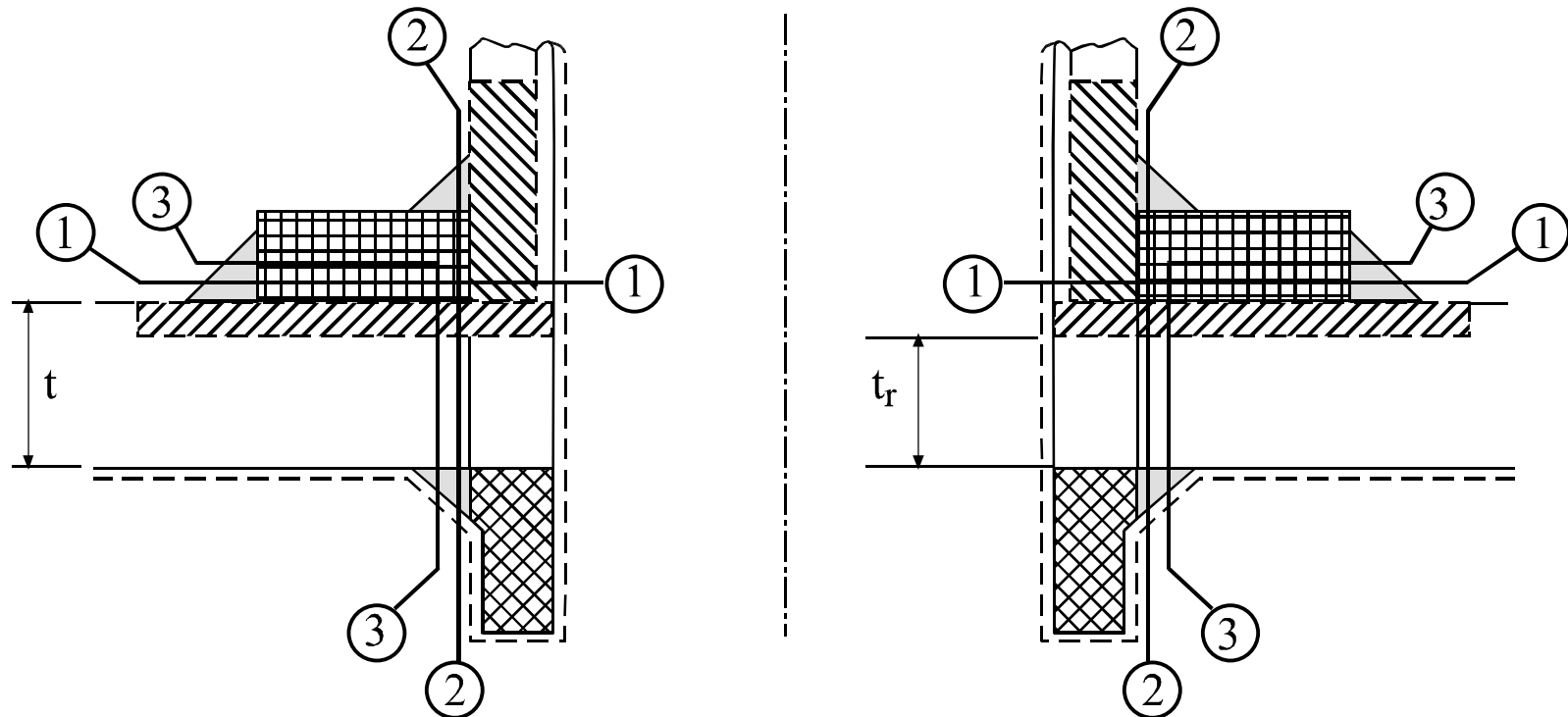
A₄ = Area available in welds



A₅ = Area available in pad (if required)

$$A_1 + A_2 + A_3 + A_4 + A_5 \text{ (if required)} \geq A$$

Nozzle Attachment Weld Loads and Weld Strength Paths to be Considered



- Strength calculations are required along each potential failure path when the nozzle to shell weld is not full penetration, or when a reinforcing pad is used.

Reinforcement of Multiple Openings

- **Total Reinforcement = Total of area(s) required by each opening**
- **Overlapping reinforcement area proportioned by the ratio of the opening diameters**
- **If reinforcement between openings is less than 50% of total, special rules apply**

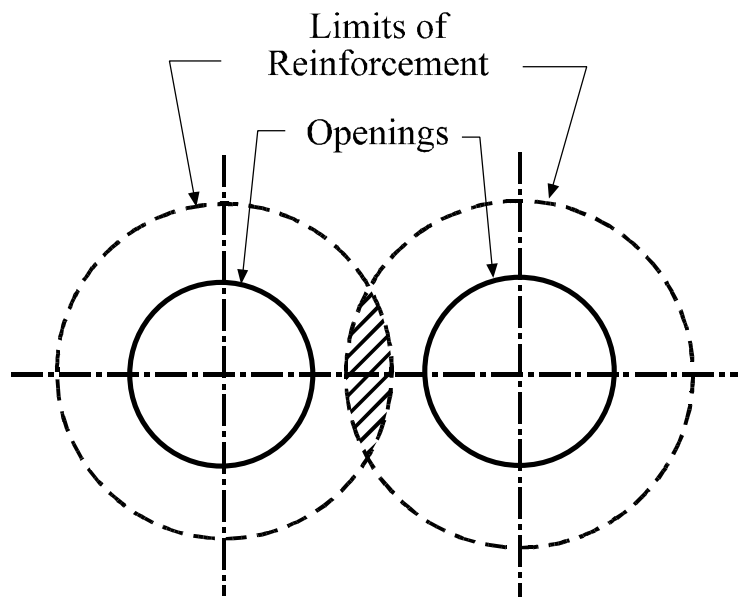
Reinforcement of Multiple Openings

(continued)

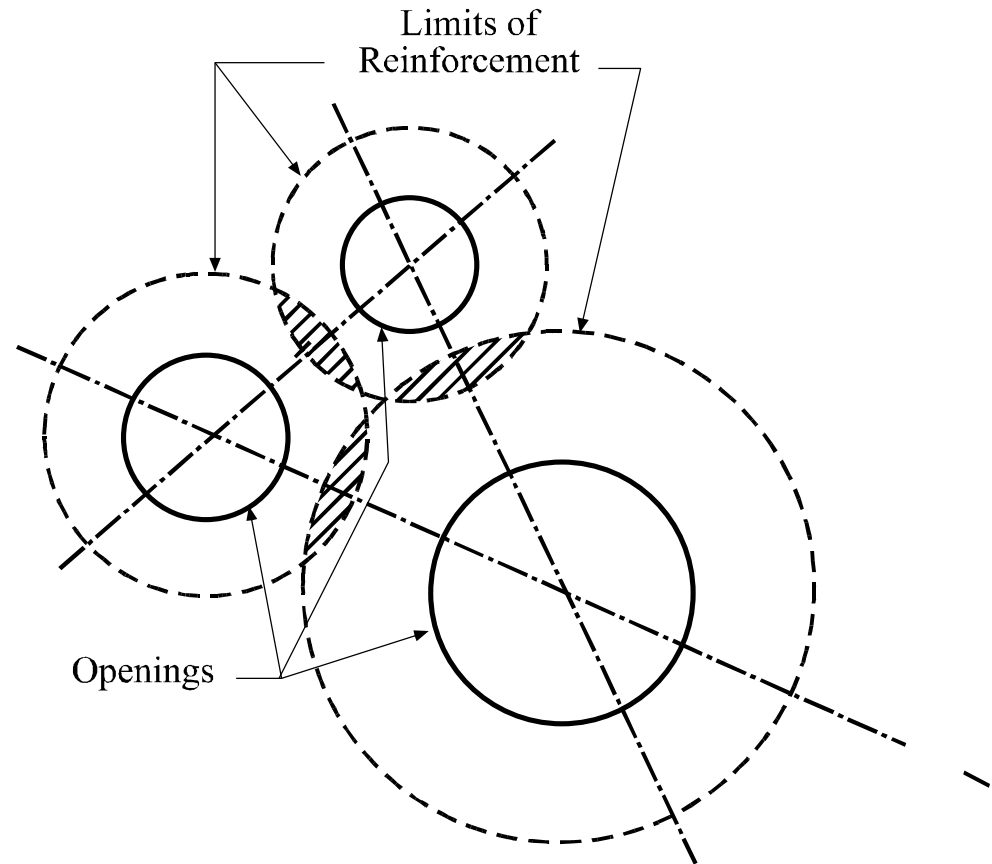
- **Each pair of three or more openings must be at least $1\frac{1}{3}$ times the average diameter apart**
 - **If not, then there is no credit for area between the openings**

- **In all cases, an opening that encompasses all of the actual openings may be assumed and reinforced**

Examples of Multiple Openings



(a) Two Openings Spaced at Less Than Two Times Their Average Diameter



(b) More Than Two Openings Spaced at Less Than Two Times Their Average Diameter

External Loads on Nozzles

- **Imposed loads on nozzles are generally not a problem for the vessel shell**
 - **Maintaining a flange seal usually governs**
- **Several analytical methods exist to evaluate local shell stresses from imposed loads**
 - **Welding Research Council Bulletin 107**
 - **Welding Research Council Bulletin 297**
- **WRC 297 is somewhat more accurate, but is limited to cylindrical shells**

Design of Tall Vertical Vessels

- **In addition to hoop (circumferential) stresses, tall vessels must consider longitudinal stress, which may govern the wall thickness**
- **Weight**
 - **Weight of the vessel will impose compressive stresses in the shell (tensile stresses when the shell is below the supports—i.e., it's hanging)**
 - **Weight of internals and contents supported by the shell above the point being considered also contribute to shell loadings**

Design of Tall Vertical Vessels

(continued)

■ **Pressure**

- **Internal pressure imposes tensile stresses on the shell**
- **External pressure imposes compressive stresses on the shell**

Design of Tall Vertical Vessels

(continued)

■ **Moment Loadings**

- **External loadings produce an overturning moment and resulting tensile and compressive longitudinal stresses at the bottom of tall vertical vessels.**
- **Common sources of large external moments are:**
 - **Wind**
 - **Earthquake**
 - **Eccentricity**
 - **Forces from piping weight, thermal expansion, and expansion joints**
- **Wind and earthquake are short term loadings; others are long term sustained loads.**

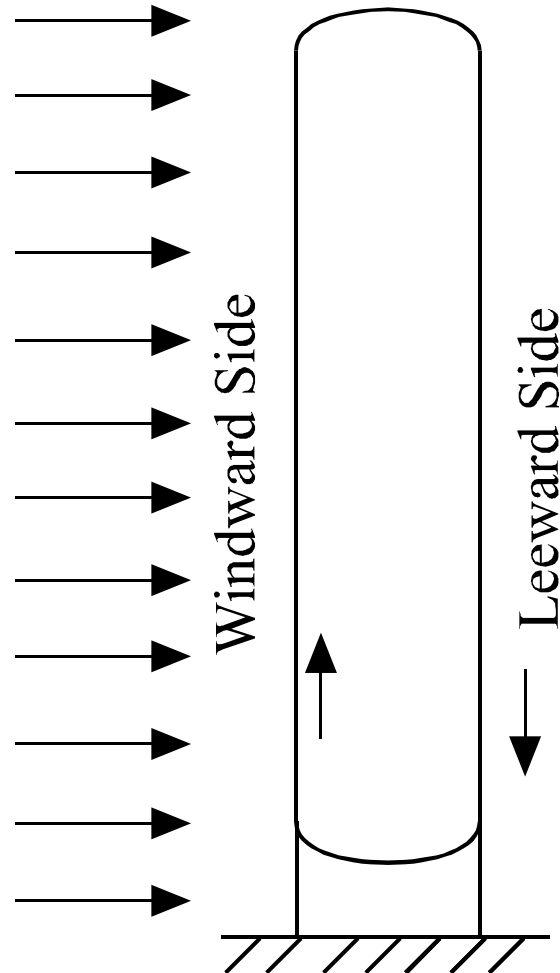
Design Cases

- **Erection – Greatest tensile uplift on the skirt and anchor bolts due to the least weight and the full moment**
- **Design – Greatest longitudinal tensile shell stress due to high internal pressure coupled with full moment**
- **Operating**
- **Shutdown – Greatest compressive loadings due to lack of internal pressure but full weight and moment**

Design Cases *(continued)*

- **Long-Term Operation – Evaluating sustained loads (e.g., expansion joint operating forces)**
- **Short Term – Evaluated short term loads, e.g., expansion joint blow out forces**
- **Hydrotest – use reduced magnitudes of wind load**

Wind Loading



Design of Tall Vertical Vessels for Moment Loadings

- Combination of Longitudinal Stresses

- Internal pressure

Stress	Windward Side	Leeward Side
Due to Moment	+ (Tension)	- (Compression)
Due to Internal Pressure	+	+
Due to Weight	-	-

- External pressure

Stress	Windward Side	Leeward Side
Due to Moment	+	-
Due to External Pressure	-	-
Due to Weight	-	-

Resultant longitudinal stresses are the sum of each of the above

Design for Wind Loads

- **In the United States there are two commonly recognized standards for wind load design:**
 - **ASCE 7-98, “Minimum Design Loads for Buildings and Other Structures”**
 - **International Building Code (IBC)**

- **Apply applicable local codes must be followed**

Wind Load Design

■ Wind design per ASCE 7-98

The following information and factors must be determined for the site and application

- Basic wind speed (V)
- Importance factor (I)
- Exposure (A, B, C, or D)
- Velocity pressure coefficient (K_z)
- Gust factor (G)
- Directionality Factor (K_d)
- Force Coefficient (C_f)
- Projected area (A_f)
- Design wind pressure(s) (q_z)

Design for Wind Loads

(continued)

■ **Basic wind speed (V)**

- **50 year recurrence interval wind speeds in miles per hour at the standard height of 33 feet (10 m)**
- **Measures the speed of a 3 second gust**
- **Based upon Exposure C**
- **Given in ASCE 7-98 charts. USA varies from 85 mph to 150 mph**
- **Consult local codes**

■ **Importance Factor (I)**

- **Measure of the relative need for survivability or consequences of failure**
- **The greater the factor the higher the design load, increasing costs**
- **Petrochemical facilities use $I = 1.15$**

Design for Wind Loads

(continued)

■ **Exposure**

- **A measure of the surrounding conditions and wind obstructions**
- **Range from A (center of large cities) to D (unobstructed areas within 1500 feet of open water 1 mile or greater in width)**
- **The default for refinery design is Exposure C**
- **Within each exposure, non building structures are denoted as Case 2**

Design for Wind Loads

(continued)

■ **Velocity Pressure Coefficient (K_z)**

- **Accounts for the exposure and the height above grade**
- **Wind pressure increase with height for a given basic wind speed**
- **Given in tables in ASCE 7-98**

■ **Gust Factor (G)**

- **Accounts for the dynamic response to gusts**
- **For most refinery equipment $G = 0.85$**

Design for Wind Loads

(continued)

■ Velocity Pressure Coefficient (K_z)

Height Above Average Level of Adjoining Ground, in Feet	Exposure B	Exposure C
0-15	0.57	0.85
15-20	0.62	0.90
20-25	0.66	0.94
25-30	0.70	0.98
30-40	0.76	1.04
40-50	0.81	1.09
50-60	0.85	1.13
60-70	0.89	1.17

Design for Wind Loads

(continued)

- **Directionality Factor (K_d)**
 - Used with load combinations defined in ASCE 7-98
 - For round structures use $K_d = 0.95$
 - Use of $K_d = 1$ is only slightly conservative
- **Force Coefficient (C_f)**
 - Accounts for the streamlining effect of the shape
 - For round structures $C_f = 0.8$ in most cases

Design for Wind Loads

(continued)

■ Force Coefficient C_f

DESCRIPTION	C_f FACTOR
Round cross section (diameter/square root of wind pressure > 2.5), smooth surface	0.6
Round cross section (diameter/square root of wind pressure > 2.5), rough surface (projection/diameter = 0.02	0.8
Round cross section (diameter/square root of wind pressure < 2.5)	0.8
Square cross section, wind normal to face	1.4

Design for Wind Loads

(continued)

■ Projected area (A_f) or wind sail - simplified calculations

ITEM	WIND SAIL (FEET)
Vessel	Outside diameter of insulation
Piping	Outside diameter of primary line insulation
Ladders	1 foot
Platforms	1 foot

Design for Wind Loads

(continued)

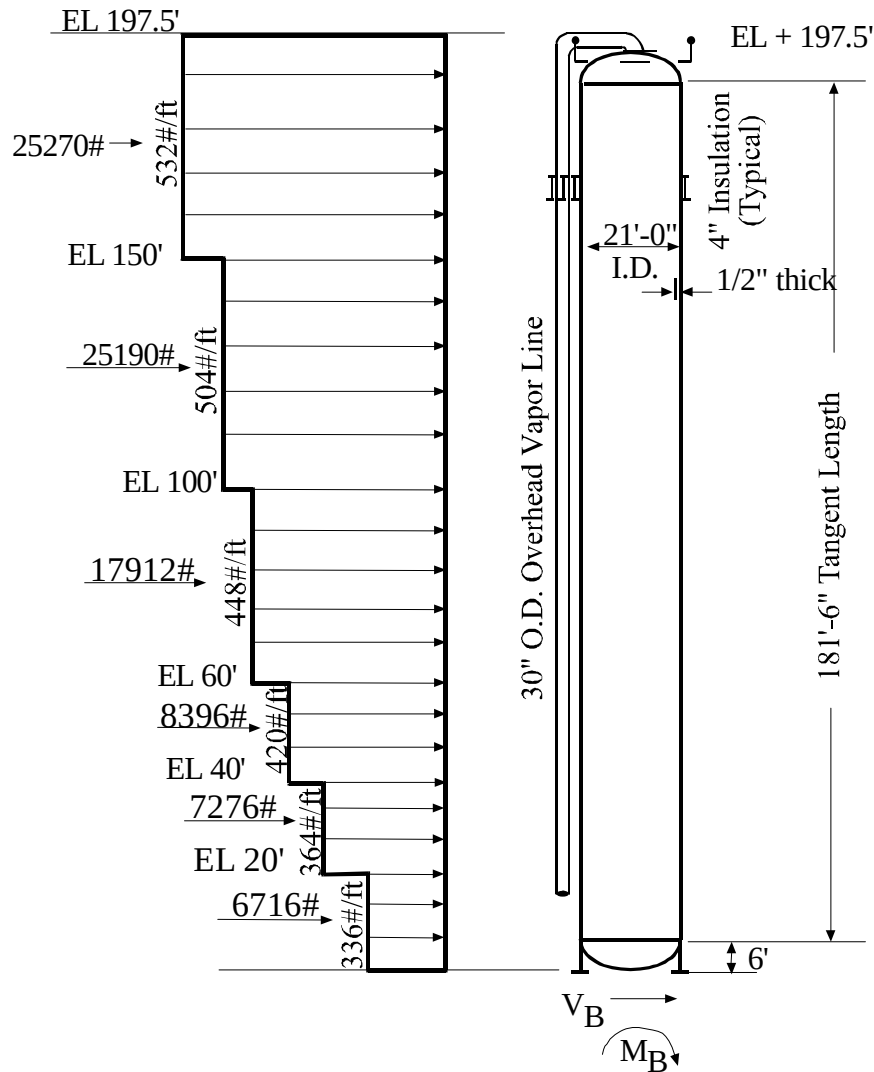
- **Design Wind Pressures**

$$q_z = 0.00256 K_z K_{zt} K_d V^2 I$$

- **Design Wind Force**

$$F = q_z C_f G A_f$$

Wind Loading Example



Total sail area =

$$21'(\text{ID}) = 2(1/2'')(\text{thickness}) + 2(4'')(\text{Insulation}) + 30''(\text{piping and insulation}) + 1'(\text{platforms + ladders}) = 25' 3''$$

$C_q = 0.8$

Effective width =

$$25' 3'' \times 0.8 = 20.2 \text{ ft}$$

$$V_B = 91,000^{\text{lb}}$$

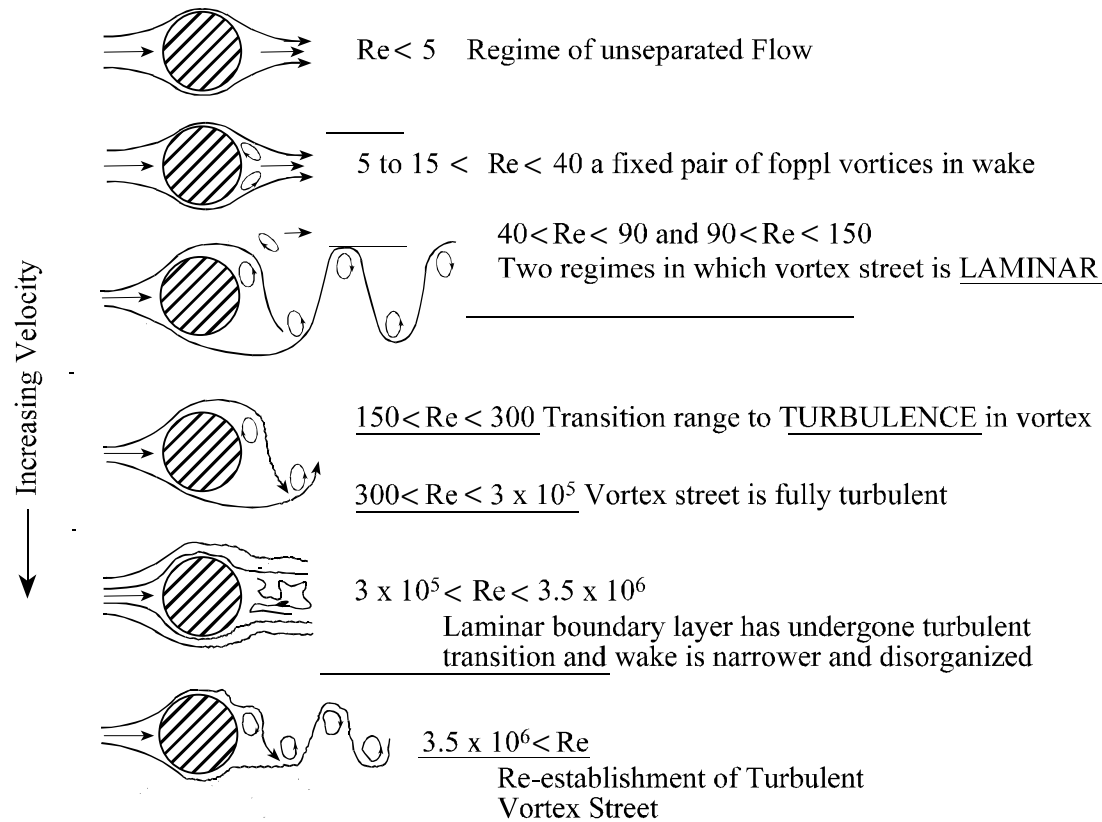
$$M_B = 9,800,000^{\text{lb-ft}}$$

If the vessel height was 150 feet, a 24 percent decrease:

$$V = 65,800^{\text{lb}} \quad \text{a 28 percent decrease}$$

$$M = 5,400,000^{\text{lb-ft}} \quad \text{a 45 percent decrease}$$

Regimes of Fluid Flow Across Circular Cylinders

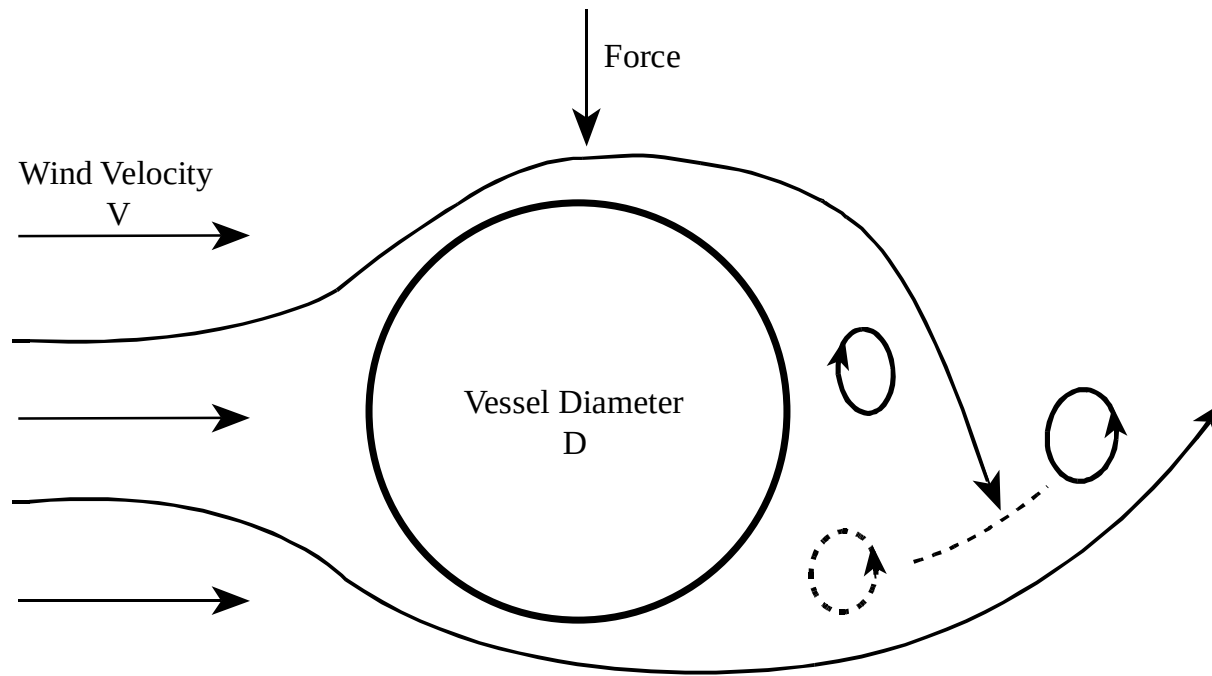


Re = Reynolds Number For Air $Re = 6380 \text{ VD}$

V = Wind Velocity (ft /sec)

D = Shell Diameter (ft)

Vortex Shedding



Frequency of Vortex Shedding

$$f = \frac{SV}{D}$$

Where,

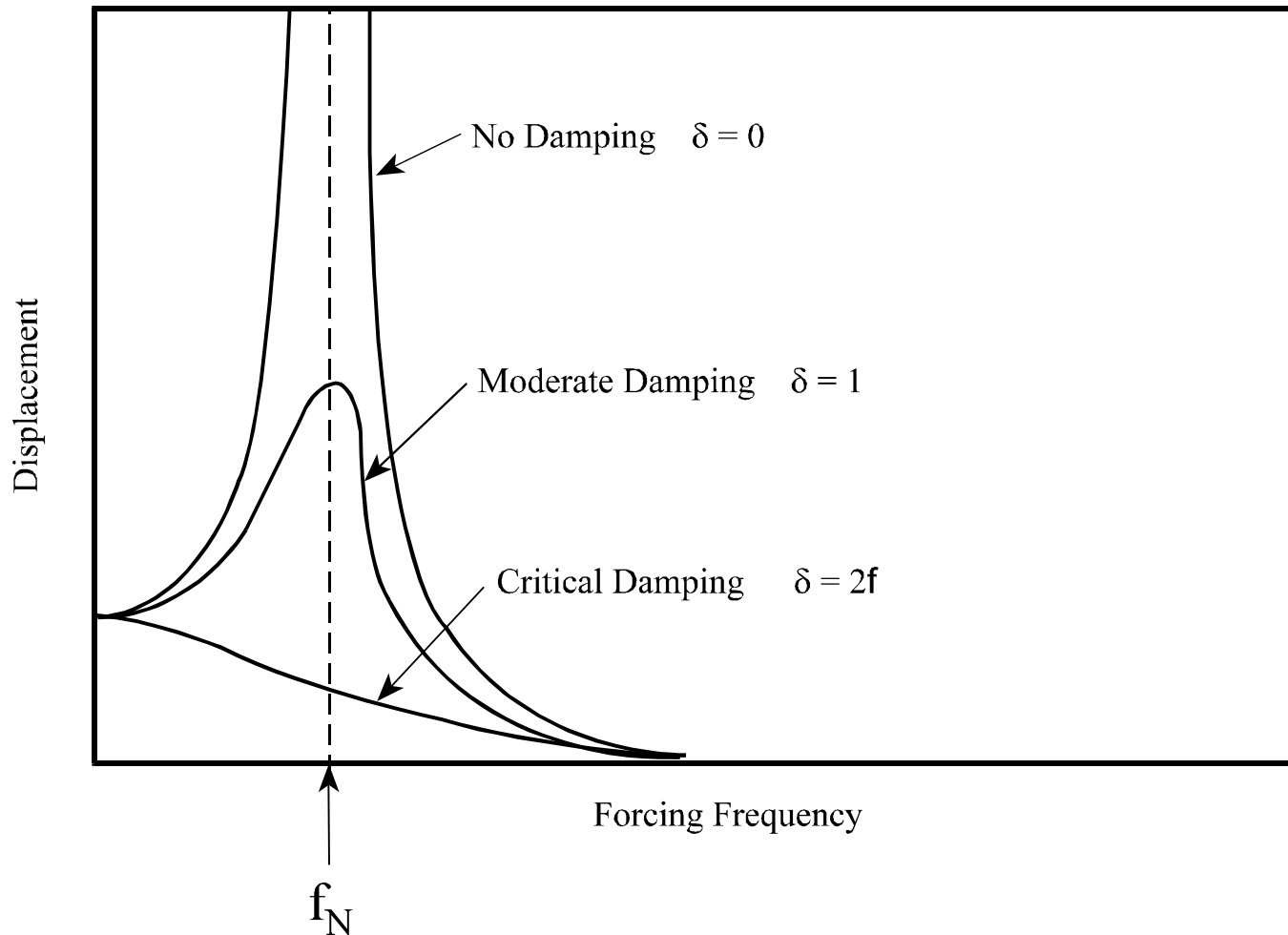
S = Strouhal Number = 0.2

V = Wind Velocity, feet/second

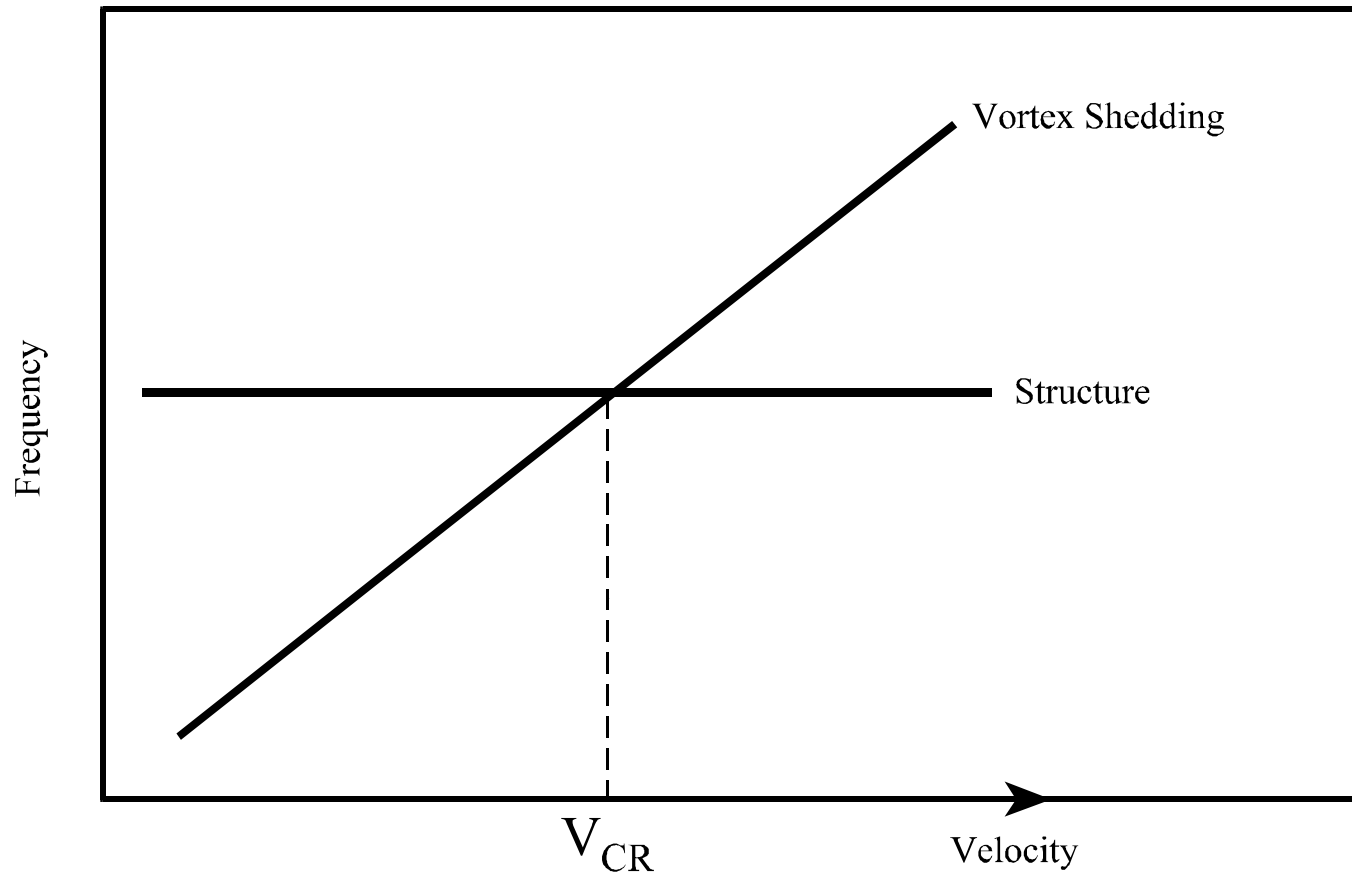
D = Shell Diameter, feet

Determining Structural Dampening

$$\text{Damping: } \delta = \text{Log}_e \frac{Y_N}{Y_{N+1}}$$



Critical Wind Velocity



Seismic (Earthquake) Design

- **Historically a simplified static force procedure has been used and is illustrated here**
- **The recently released International Building Code (IBC) requires a more detailed, dynamic analysis in many cases**
- **In the static force method an “equivalent” shear force is determined and distributed along the vessel**

Seismic Design (continued)

- **Static load method:**

$$\text{Shear Force, } V = \frac{ZIC}{R_W} W$$

where:

V = Design lateral force

Z = Seismic zone factor (varies from 0.075 to 0.40)

W = Dead load (weight)

R_W = Stiffness coefficient (4 for skirt supported vessels)

I = Importance factor (use 1.25)

C = Coefficient dependent upon soil conditions $C = \frac{1.25S}{T^{2/3}}$
and the vessel's period of vibration

S = Site coefficient, generally take as 1.5

T = Period for a vertical cylindrical vessel $= 0.258 \sqrt{\frac{L^4 W}{8EI}}$

Seismic Design

(continued)

- **Horizontal force is applied to the vessel as follows:**
 - **Force at the top, $F_t = 0.07TV$ (maximum $F_t = 0.25V$)**
 - **This approximates the effect of higher modes of vibration**
 - **Distribute the remainder in accordance with**

$$F_x = \frac{(V - F_t)W_x h_x}{\sum W_x h_x}$$

Seismic Design

(continued)

- **Horizontal force distribution is a function of the amount of mass at and the height of each location**
- **For a vessel with uniform mass distribution, the force distribution becomes triangular**
 - **The entire force ($V-F_t$) may be applied at $\frac{2}{3}$ the height**

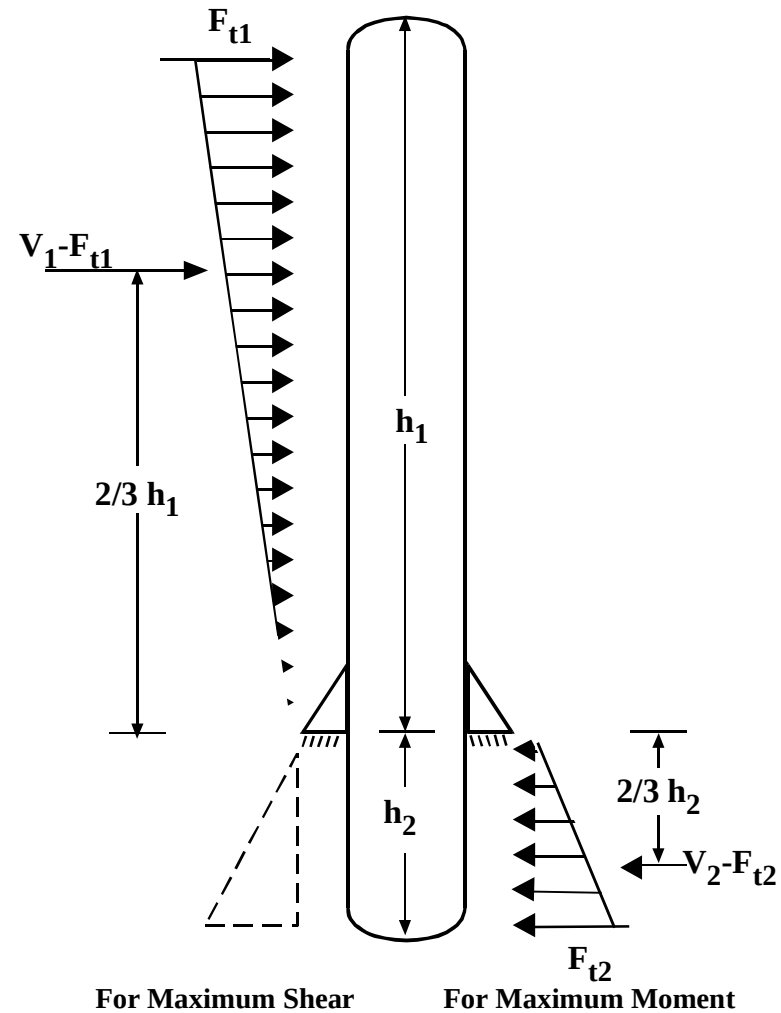
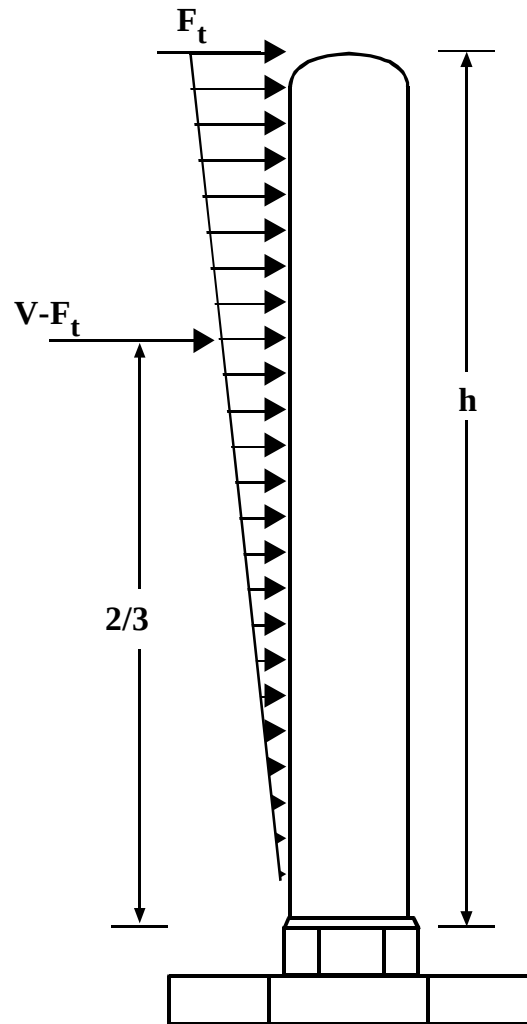
Seismic Design

(continued)

■ **Additional Points**

- **Movement or “sloshing” of liquid contents must be accounted for**
- **In general, use of the static force method has been adequate**
 - **However, design for force resistance leads to use of a stiffer structure, when more flexible structures are preferred for seismic conditions.**
- **If support is at intermediate point along vessel, treat portions above and below support separately**
 - **Unlike wind loadings, the upper and lower portions can move in opposite directions during seismic activity**

Seismic Design





Training Services

Piping

Purpose

To introduce piping designation, design, and support, including accommodation of thermal growth.

Piping Definition

Piping is an assembly of components, including pipe, valves, fittings, flanges and supports, used to convey, distribute or control fluid flows.



Definitions

■ Pipe

- **A hollow, generally cylindrical member used to convey a wide range of fluids (gas, liquid, or fine solids) over a wide range of temperatures and pressures**

Definitions

(continued)

■ **Tube**

- **Special, thin wall class of pipe**
- **Are generally small diameter**
- **OD equals its nominal size**
- **Made of soft, ductile materials, allowing them to be bent to the desired configuration**
- **Larger diameter tubes are used in heat transfer applications due to the smaller mass of metal, hence better heat transfer**

Definitions

(continued)

■ **Ducting**

- **A thin wall, often rectangular system used to convey vapor at near atmospheric pressure (i.e. a few inches of water).**

Piping Designation

- **Nominal pipe size (nps) is used to designated piping size**
- **For each pipe size, outside diameter is a constant**
 - **As wall thickness changes, the inside diameter and, therefore, the flowing area, varies**

Piping Designation

(continued)

- **For 14 inches and above, the nps is equal to the outside diameter**
- **For smaller sizes, the outside diameter varies with the size**
 - **For example, 10 inch pipe is 10.75 inches OD, 6 inch pipe is 6.625 inches OD, and 2 inch pipe is 2.375 inches OD**

Piping Thickness

- **Piping is available in many standard thicknesses**
 - **Common thicknesses are designated as schedules**

- **Schedule 40 is “standard” thickness for nps through 10 inch**
 - **For larger sizes, standard wall is 0.375 inch**
 - **In large diameter piping, sufficient thickness for structural stability and handling is necessary**

Piping Thickness

(continued)

- **Extra strong (X-strong) is another common size**
 - Is the same as Schedule 80 through 10 inch nps
 - For larger sizes, it is 0.500 inch wall

- **Double extra strong (XX-strong) is also sometimes called for in smaller sizes**
 - Is available 1/2 inch through 8 inch nps (except for 3.5 inch nps)
 - Has twice the wall thickness of X-strong (except for 8 inch nps)

Piping Thickness

(continued)

- **There is a ± 12.5 percent manufacturing tolerance on wall thickness of seamless pipe**
 - **Must be considered when determining required wall thickness, and when evaluating the cross sectional area of pipe wall and flowing area**
- **Small diameter pipe (1-1/2 inch and less) is usually a minimum of Schedule 80**
- **Pipe 2 - 10 inch uses Schedule 40 as minimum**
- **Very large pipe must be thick enough for stability and handling**

Design Properties of 8 Inch Pipe

	A	B	C	D	E	F	G	H	I	K	L	M	O
5S	.109	8.407	42.0	2.258	2.201	2.92	55.5	9.91	24.04	3.01	26.4	6.13	
10S	.148	8.329	40.1	2.258	2.180	3.94	54.5	13.40	23.59	3.00	35.4	8.21	
	.158	8.309	39.6	2.258	2.175	4.20	54.2	14.29	23.48	2.99	37.7	8.74	
	.165	8.295	39.3	2.258	2.172	4.39	54.0	14.91	23.40	2.99	39.3	9.10	
	.188	8.249	38.2	2.258	2.160	4.98	53.4	16.94	23.14	2.98	44.4	10.29	
	.203	8.219	37.5	2.258	2.152	5.37	53.1	18.26	22.97	2.98	47.7	11.05	
LW	.219	8.187	36.8	2.258	2.143	5.78	52.6	19.66	22.94	2.97	51.1	11.86	
	.238	8.149	35.9	2.258	2.133	6.27	52.2	21.32	22.58	2.97	55.2	12.80	
20	.250	8.125	35.4	2.258	2.127	6.58	51.8	22.37	22.45	2.96	57.7	13.39	
30	.277	8.071	34.2	2.258	2.113	7.26	51.2	24.70	22.15	2.95	63.4	14.69	
40 ST 40S	.322	7.981	32.4	2.258	2.089	8.40	50.0	28.56	21.68	2.94	72.5	16.81	
	.344	7.937	31.5	2.258	2.078	8.95	49.5	30.43	21.42	2.93	76.9	17.82	
	.352	7.921	31.2	2.258	2.074	9.15	49.3	31.1	21.3	2.93	78.4	18.19	
	.375	7.875	30.3	2.258	2.062	9.72	48.7	33.0	21.1	2.92	82.9	19.22	
60	.406	7.813	29.1	2.258	2.045	10.48	47.9	35.6	20.8	2.91	88.8	20.58	
	.469	7.687	26.8	2.258	2.012	12.02	46.4	40.9	20.1	2.89	100.3	23.25	
80 XS 80S	.500	7.625	25.8	2.258	1.996	12.76	45.7	43.4	19.8	2.88	105.7	24.52	
100	.593	7.439	22.8	2.258	1.948	14.96	43.5	50.9	18.8	2.85	121.4	28.14	
	.625	7.375	21.8	2.258	1.931	15.71	42.7	53.4	18.5	2.84	126.5	29.32	
120	.718	7.189	19.2	2.258	1.882	17.84	40.6	60.6	17.6	2.81	140.6	32.60	
140	.812	7.001	16.8	2.258	1.833	19.93	38.5	67.8	16.7	2.78	153.7	35.63	
XX	.875	6.875	15.4	2.258	1.800	21.30	37.1	72.4	16.1	2.76	162.0	37.57	
160	.906	6.813	14.7	2.258	1.784	21.97	36.5	74.7	15.8	2.75	165.9	38.48	

Piping Dimensions

Key

A	Schedule
B	Wall Thickness (inches)
C	Inside Diameter (inches)
D	(inside diameter)⁵(10³ inches⁵)
E	Outside Surface Area (square feet per foot)
F	Inside Surface Area (square feet per foot)
G	Metal Cross Section (inches²)
H	Flowing Cross Section (inches²)
I	Pipe Weight (lb/ft)
K	Water Weight (lb/ft)
L	Radius of Gyration (inches)
M	Moment of Inertia (inches⁴)
O	Section Modulus (inches³)

Piping Fittings

- **Fittings are piping system components that provide for junctions, size changes, and terminations**
 - **Examples are elbows, tees, caps, and reducers (eccentric and concentric)**
- **Fittings are made to standardized shapes and sizes**
 - **ASME B16.9, “Factory-Made Wrought Steel Buttwelding Fittings” is the governing standard**
 - **ASME B16.28, “Wrought Steel Buttwelding Short Radius Elbows and Returns” covers short radius elbows**

Piping Fittings

(continued)

- **Fittings are forged (wrought) components**
 - **May occasionally be fabricated or, in the case of bends, made from bent pipe**
- **Elbows are available as long radius (bend radius equals 1.5 times the nps) and short radius (bend radius equals the nps)**
 - **Long radius is standard because of smaller pressure drop and less potential for erosion**
 - **Short radius elbows have the same pressure rating as straight seamless pipe**

Piping Fittings

(continued)

- **For solids conveyance (e.g., catalyst) “sweep” elbows are often used**
 - **Are very long radius, gentle, changes of direction to avoid breaking up the solids**

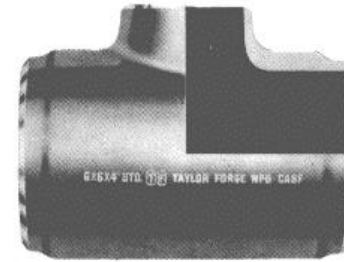
Components for Pipeline Systems



**Long Radius
Elbow**



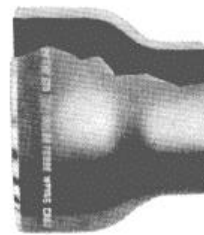
**Short Radius
Elbow**



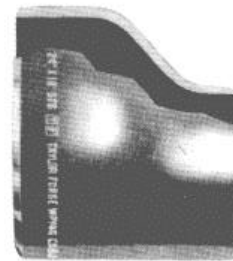
**Reducing Outlet
Tee**



Straight Tee



**Concentric
Reducer**



**Eccentric
Reducer**



**Welding
Cap**

Bending of Pipe

- **Instead of using fittings for changes of direction, straight piping is occasionally bent**
- **Method is very common for small (2 inch and less) piping**
- **With proper controls, (e.g. maintenance of cross section geometry, control of local strains and thinning) larger piping may also be bent**

Bending of Pipe

(continued)

- **Bending is normally done cold**

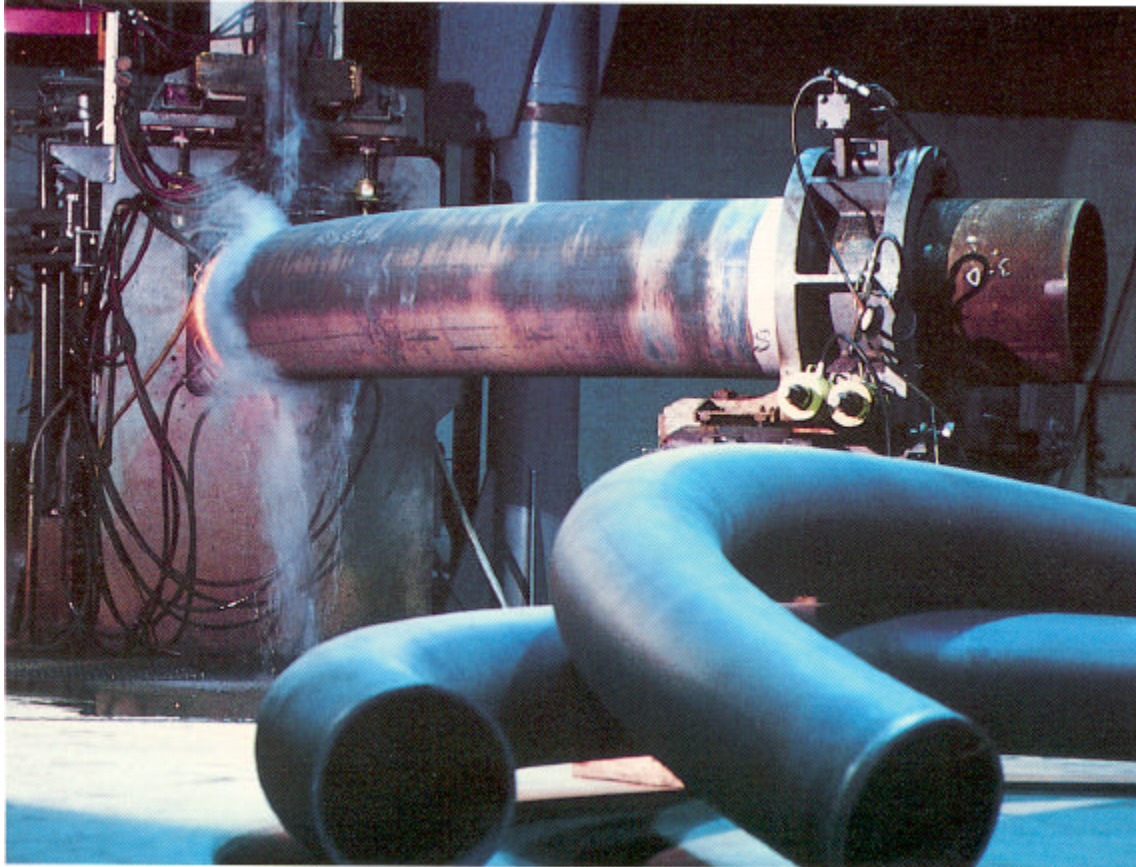
- **Hot bending may be considered if:**
 - **Procedure is tightly controlled and performed by trained, experienced people with proper equipment (e.g., not a welding torch)**
 - **Temperatures, holds, and heating and cooling rates, as well as the transition from cold to hot pipe, are tightly controlled**

Bending of Pipe

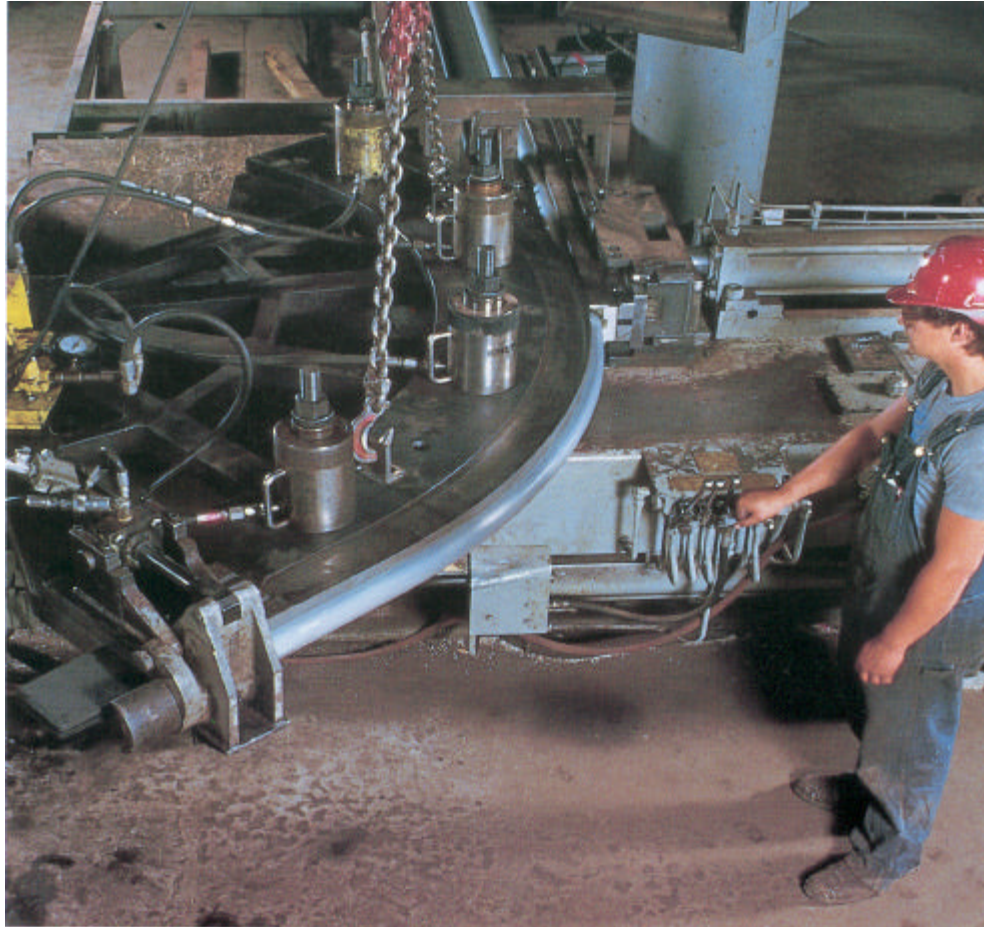
(continued)

- **Hot bending (continued)**
 - **Material must be suitable for the procedure (i.e. metallurgical structure and mechanical properties maintained)**
 - **Wall thinning or buckling, and cross section distortion (e.g., ovaling and flattening) must be controlled**

Bending of Pipe



Bending of Pipe



Miters

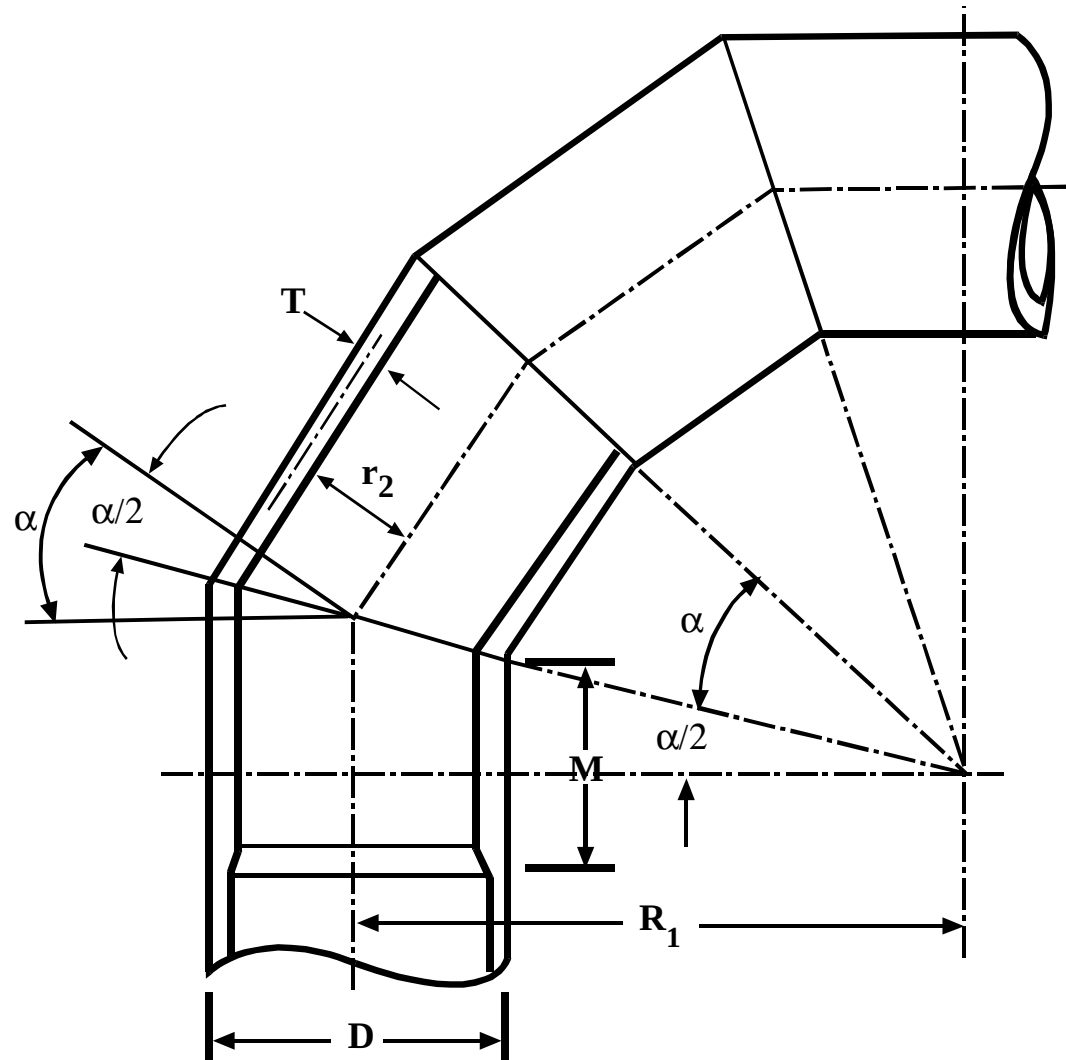
- **Are elbows fabricated from lengths of straight pipe**
- **Most commonly used for large diameter piping, where fittings are expensive or not readily available**

Miters

(continued)

- **Piping Code rules for determining permissible internal pressure in a miter bend are applicable only if the angle of each miter cut does not exceed 22.5° (total change of direction = 45°)**
- **Required thickness for a miter with a given internal pressure is greater than for a straight pipe**
 - **Actual thickness may be the same because of the use of standard pipe thickness, with a minimum depending upon size**

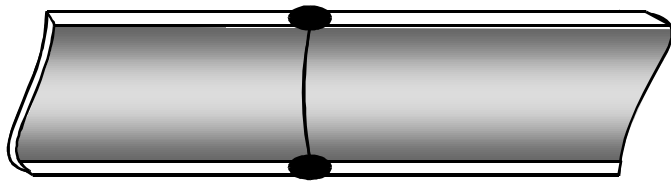
Multiple Miter Bend



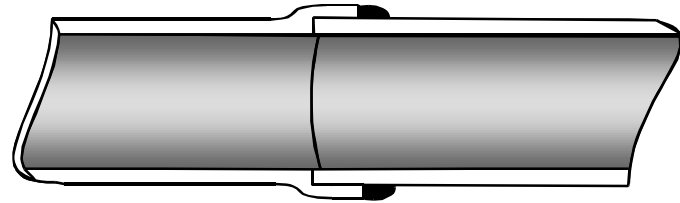
Joining of Piping

Piping is butt welded together using full penetration welds. Small piping ($1\frac{1}{2}$ inch and less) is generally socket welded.

Weld Types



Buttweld



Socketweld

Piping Sizing

- **Process considerations are primary factor used to determine required pipe sizes**
- **Pressure drop through piping is the most common factor in line sizing**
 - **Often expressed in “equivalent length” of straight pipe**
 - **Pressure drops of fittings, valves, reductions, etc., may be expressed as an equivalent length of straight pipe**
 - **Convenient source of equivalent lengths is Crane Company’s Technical Paper 410**
 - **The total pressure drop is found by multiplying the equivalent length by the ΔP per unit length of straight pipe of the appropriate size**
 - **Pipe size is then selected for a low pressure drop, while remaining economical**

Representative Equivalent Lengths

Description of Product				Equivalent Length in Pipe Diameters (L/D)
Globe Valves	Stem Perpendicular to Run	With no obstruction in flat, bevel, or plug type seat	Fully open	340
		With wing or pin guided disc	Fully open	450
	Y-Pattern	(No obstruction in flat, bevel, or plug type seat)	Fully open	175
		- With stem 60 degrees from run of pipe line - With stem 45 degrees from run of pipe line	Fully open Fully open	145
Angle Valves		With no obstruction in flat, bevel, or plug type seat	Fully open	145
		With wing or pin guided disc	Fully open	200
Gate Valves	Wedge, Disc, Double Disc, or Plug Disc		Fully open	13
			Three-quarters open	35
			One-half open	160
			One-quarter open	900
	Pulp Stock		Fully open	17
			Three-quarters open	50
			One-half open	260
			One-quarter open	1200
Conduit Pipe Line Gate, Ball, and Plug Valves			Fully open	3**
Check Valves	Conventional Swing	0.5 ft ... Fully open	135	
	Clearway Swing	0.5 ft ... Fully open	50	
	Globe Lift or Stop; Stem Perpendicular to Run or Y-Pattern	2.0 ft ... Fully open	Same as Globe	
	Angle Lift or Stop	2.0 ft ... Fully open	Same as Angle	
	In-Line Ball	2.5 vertical and 0.25 horizontal ft ... Fully open	150	
Foot Valves with Strainer		With poppet lift-type disc	0.3 ft ... Fully open	420
		With leather-hinged disc	0.4 ft ... Fully open	75
Butterfly Valves (8-inch and larger)			Fully open	40
Cocks	Straight-Through	Rectangular plug port area equal to 100% of pipe area	Fully open	18
	Three-Way	Rectangular plug port area equal to 80% of pipe area (fully open)	Flow straight through	44
			Flow through branch	140
Fittings	90 Degree Standard Elbow			30
	45 Degree Standard Elbow			16
	90 Degree Long Radius Elbow			20
	90 Degree Street Elbow			50
	45 Degree Street Elbow			26
	Square Corner Elbow			57
	Standard Tee	With flow through run		20
		With flow through branch		60
Close Pattern Return Bend			50	

Piping Sizing *(continued)*

- **Other factors that may affect line sizing:**
 - **Erosion potential of the carried medium (e.g., entrained solids) or the need to prevent settling of components of the flowing mixture**
 - **Erosion reduction may call for low velocities and large diameter piping, while higher velocities to prevent settling argue for smaller diameter pipe**
 - **In some cases velocity can affect the the corrosion rate of the base material by “stripping” off the protective, passivating, film**

Piping Sizing (continued)

- **Piping is commercially available in sizes from less than 1 inch to 60 inches or more.**
- **Through most of that range, pipe is available in 2 inch increments (e.g., 6, 8, 10 inch). Beginning at 48 inches, the increment increases to 4 inches.**

Piping Sizing *(continued)*

- **When selecting a pipe size, consideration must be given to availability of flanges, valves, fittings, etc.**
- **For this reason, some sizes are generally avoided**
 - **2.5, 5, and 22 inch fall into this category**
- **Flanges are commonly available for 24 inch and smaller piping. They conform to the requirements of ASME B16.5. Flanges conforming to the requirements of ASME B16.47 are available for larger piping (26 inches and greater).**

Piping Sizing

(continued)

- **Minimum size normally used (other than for local instrument or utility lines) is 1 inch**
- **Small piping is provided as seamless (i.e., no longitudinal seam), while piping over 14 - 16 inches is normally welded (i.e., contains a longitudinal seam)**
 - **Larger seamless piping is available, but is increasingly more expensive as size increases**

Valves

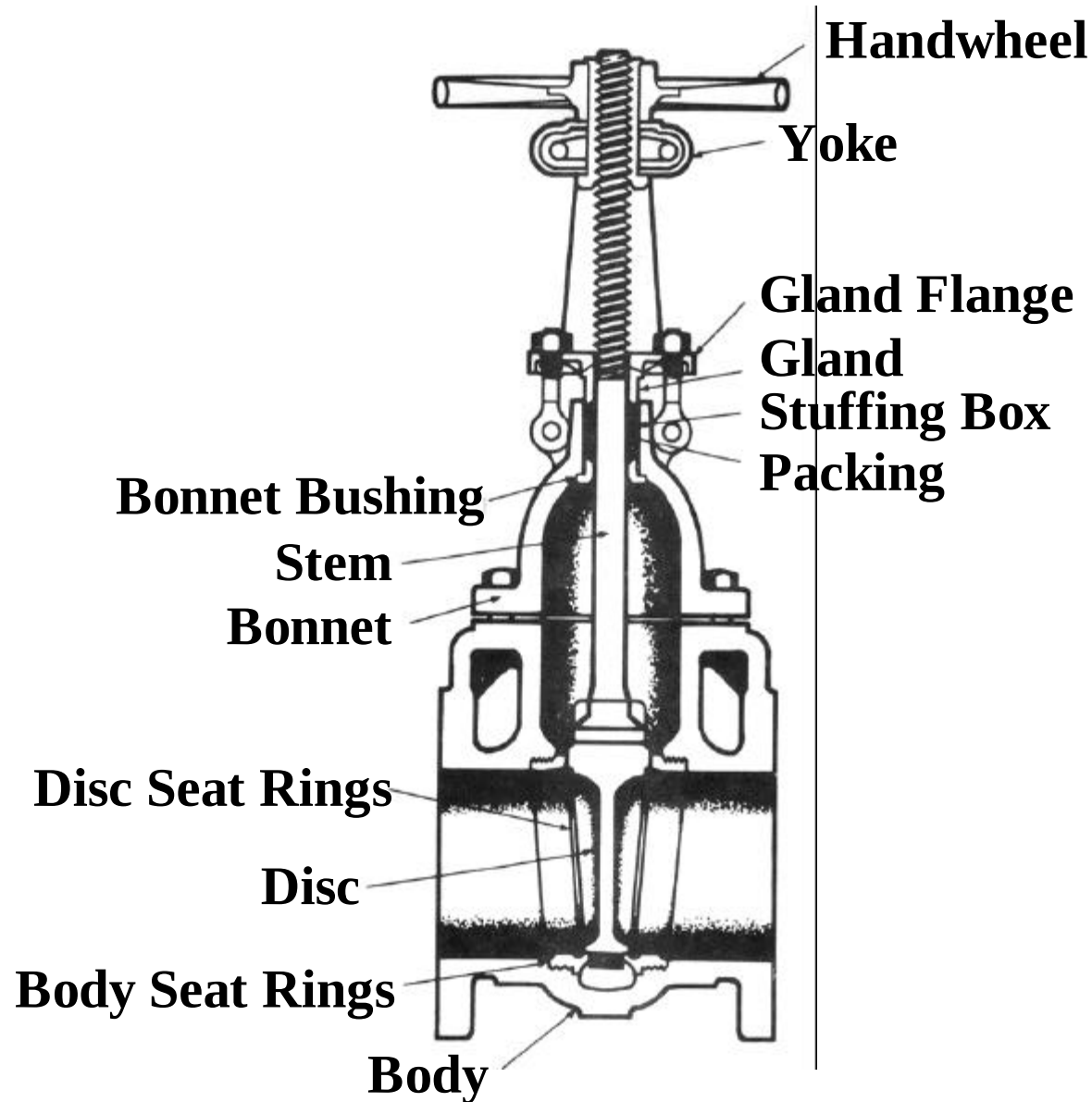
- **Valves are integral part of any piping system**
 - **Used to regulate flow, halt flow, or prevent backflow of fluids**
 - **Are also used for safety purposes to relieve pressure**
- **Valves must perform their intended function whenever they are called upon, whether that is multiple times a day or after a long idle period**

Valves

(continued)

- **Many types of valves, some very specialized**
- **Most common include gate, globe, check, ball, butterfly, relief valves**
- **Valve bodies are made to standard ratings, equivalent to the class rating system used for flanges**

Gate Valve



Gate Valves

(continued)

- **Most widely used valves in industrial plants**
- **Used to fully stop or fully permit flow**
 - **Best used where infrequent operation is required**
- **Not practical for throttling the flow because flowing area is not a straight line function of the amount of the gate's travel**
 - **Is difficult to know how open or closed the valve is**
 - **Partially open gates set up vibrations that can damage the valve**

Gate Valves

(continued)

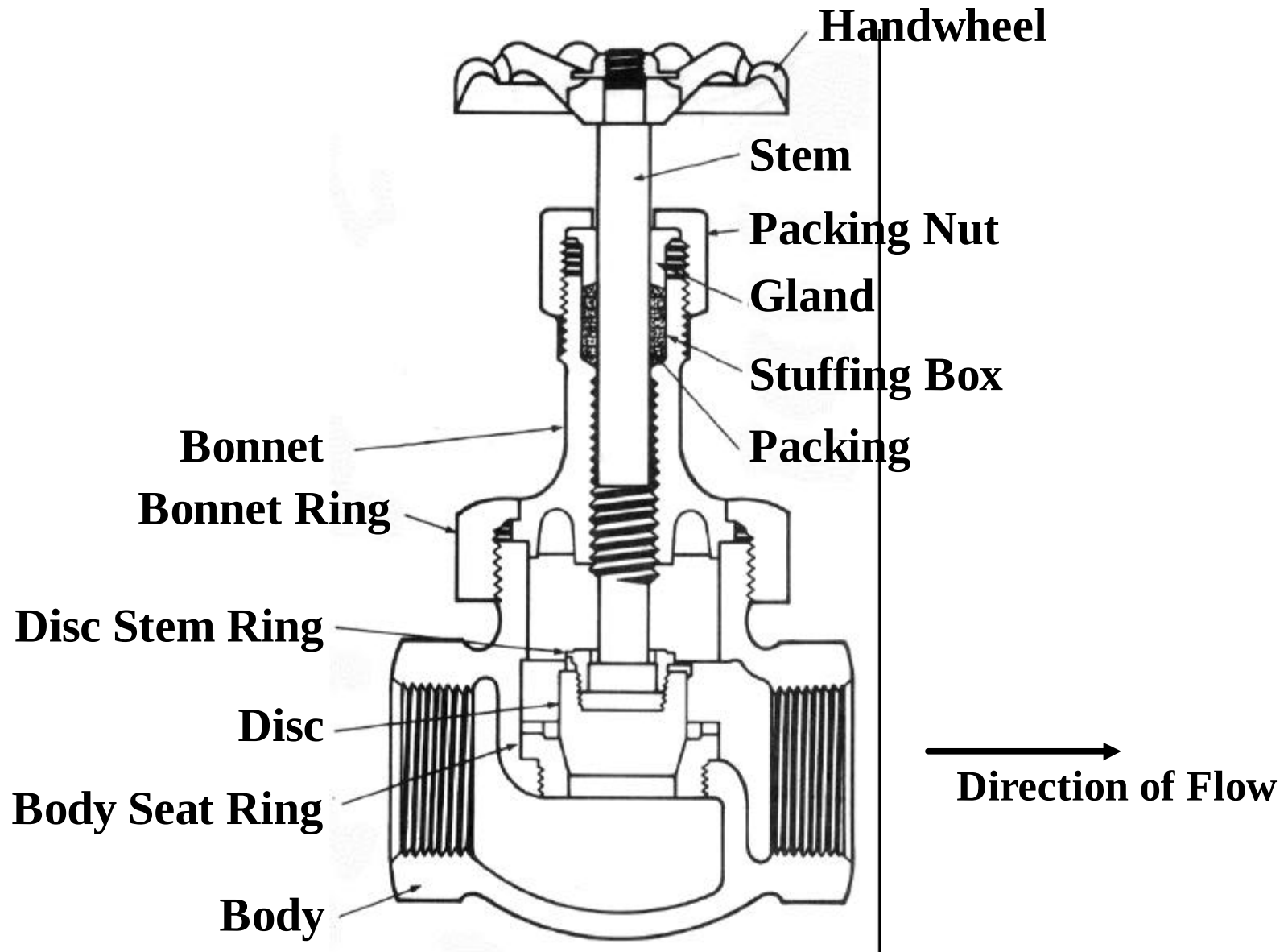
- **Fluid passes straight through the valve, minimizing the pressure drop**
- **A plate slides up and down perpendicular to the flow to open or close the flow path**
 - **Usually takes many turns of the handwheel to open or close valve**

Gate Valves

(continued)

- **Most discs and seats have a matched taper, making repair or resurfacing difficult**
- **Stems may be rising or non rising**
 - **Screw surfaces on rising stems are isolated from fluids in the line**
 - **A rising stem shows, at a glance, position of disc (open or closed)**
 - **Clearance must be provided for the rising stem, and exposed portions must be protected from damage or corrosion**
 - **In non-rising stem system, screw threads are exposed to the corrosive fluid**

Globe Valve



Globe Valves

(continued)

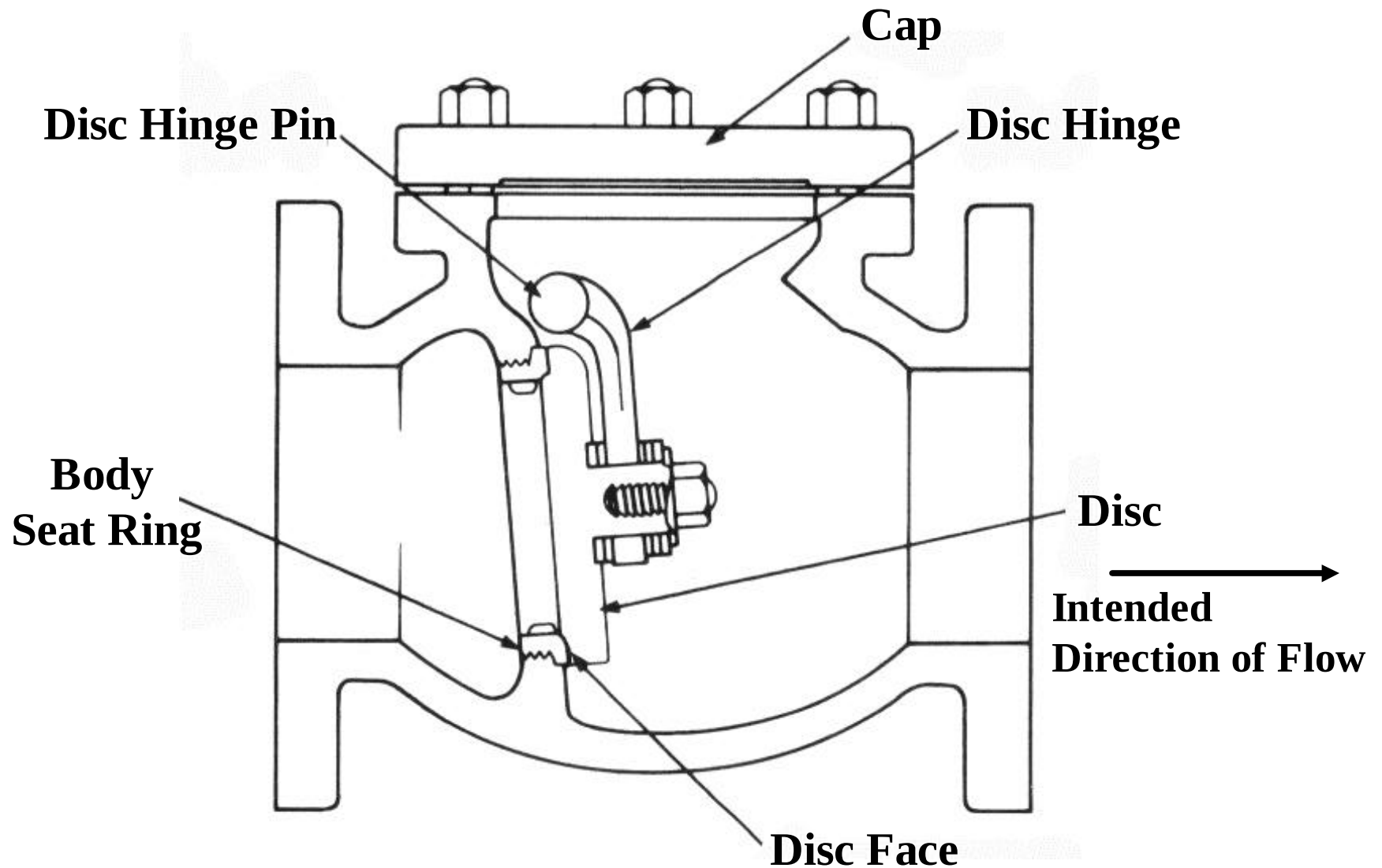
- **Seat of a globe valve is parallel to the flow path**
 - **All contact between seat and disc ends when flow begins**
- **Efficient for throttling of flow with minimum erosion**
- **Perform well where frequent operation is required**

Globe Valves

(continued)

- **Size of seat opening is proportional to the number of handwheel turns, making regulation of the flowing area easier**
- **Maintenance is easier than gate valves, often without removing the valve from the line**
- **Not recommended where flow resistance and pressure drop are a concern because flow path is not straight**

Swing Check Valve



Check Valves

- Check valves prevent the backflow of the carried fluid, i.e., they are “one way” valves
- The disc may be seated by gravity, by the fluid itself if it attempts to reverse, or sometimes by a piston or spring

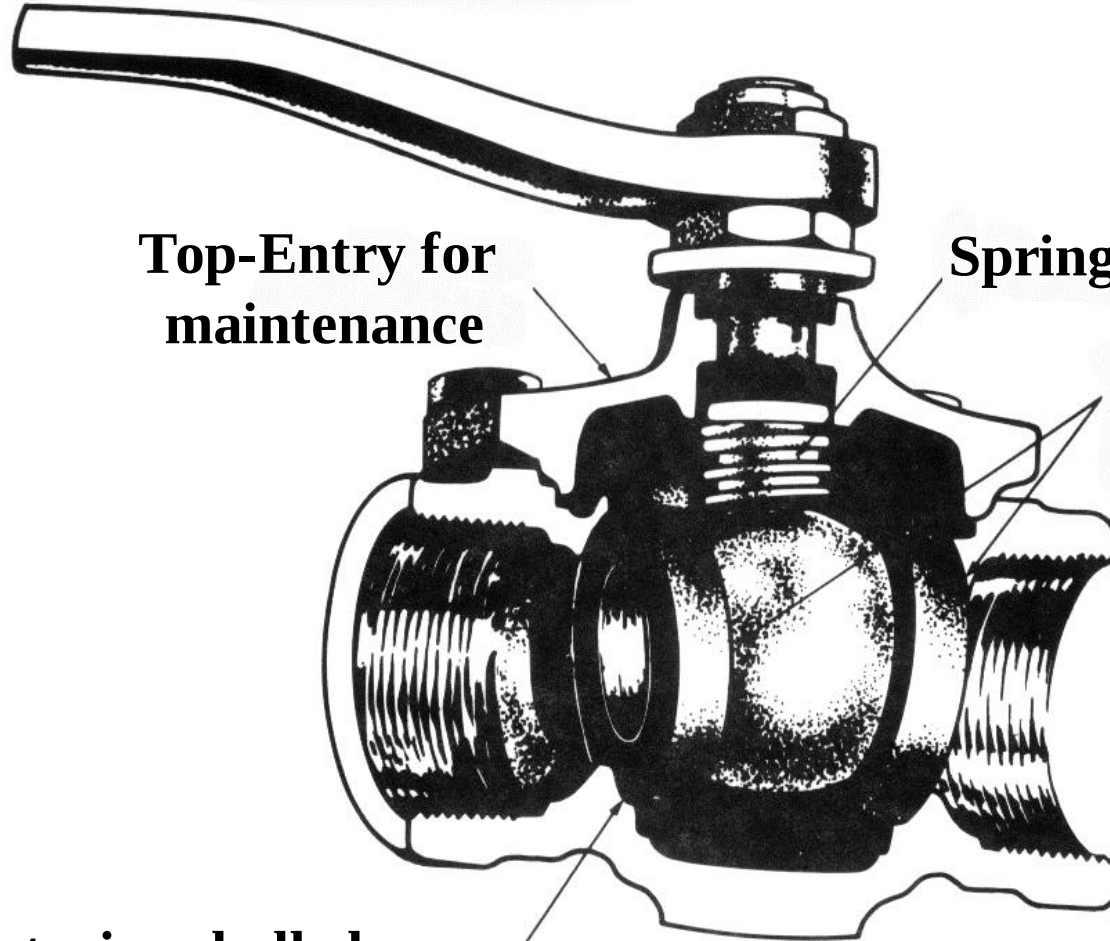
Check Valves

(continued)

- **Swing check valves are the most common in liquid service**
 - **Flow itself opens the valve, and keeps it open**
 - **Straight flow path means low pressure drop**
 - **Vapor usually has insufficient momentum to keep the valve open**
- **Lift checks are seated by gravity or the lack of flow**
 - **For a horizontal line, the change of direction results in increased pressure drop**

Ball Valve

Lever Handle for quick quarter-turn



**Top-Entry for
maintenance**

Spring

**"Wedge-Seat"
compensates
for seat wear**

**Seat wipes ball clean,
assuring tight shutoff**

Ball Valves

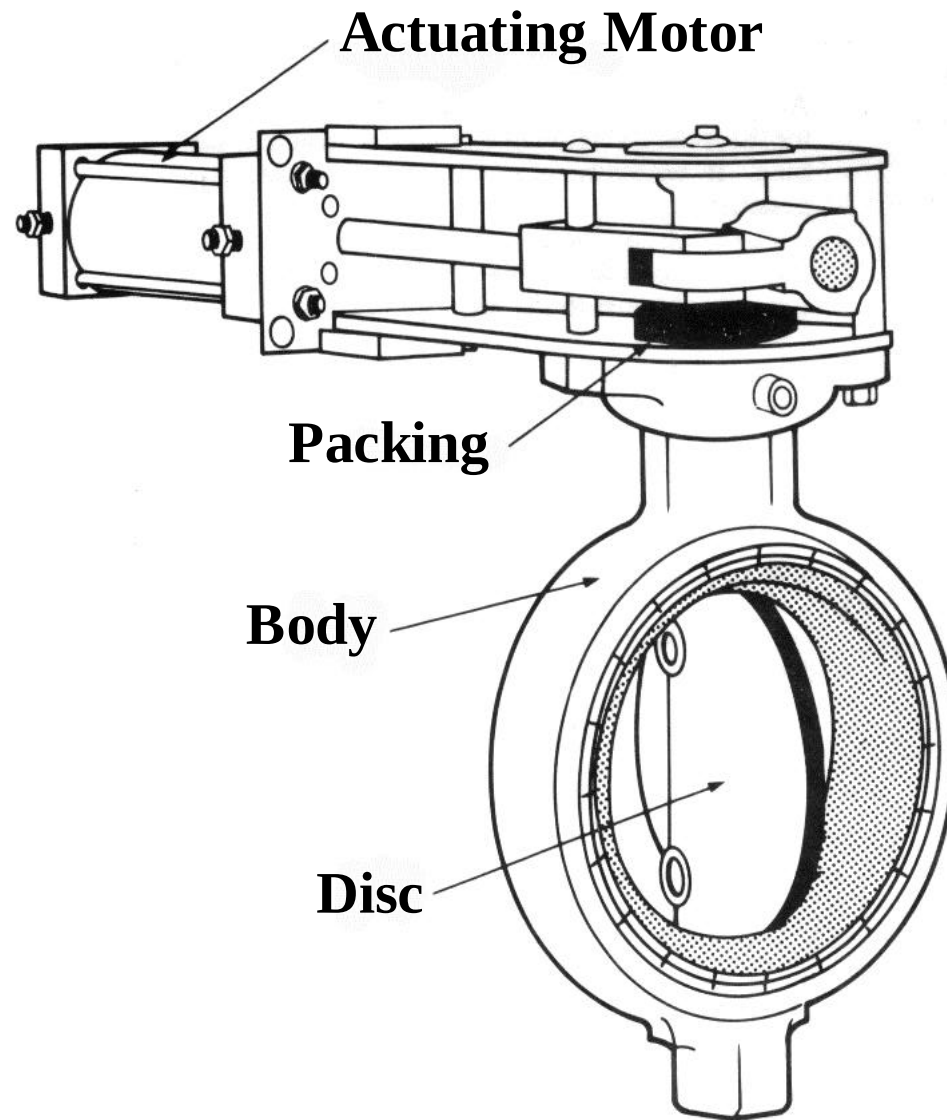
(continued)

- **Spring loaded ball, pierced with a line size hole through the center**
- **Seat is always out of the flowing stream, between the ball and the valve body**
 - **Can be good for flow carrying solids (prevents damage to the seat)**
- **Flow path is straight through the valve, minimizing the pressure drop**

Ball Valves ***(continued)***

- **Valves open and close with a quarter turn of the handle, making them quick on/off valves**
- **When installed in vertical piping, solids can settle on the ball (especially when closed) and be ground into the ball and seat upon use**
- **Are usually small so the force required to turn the handle is manageable for one person**

Butterfly Valve

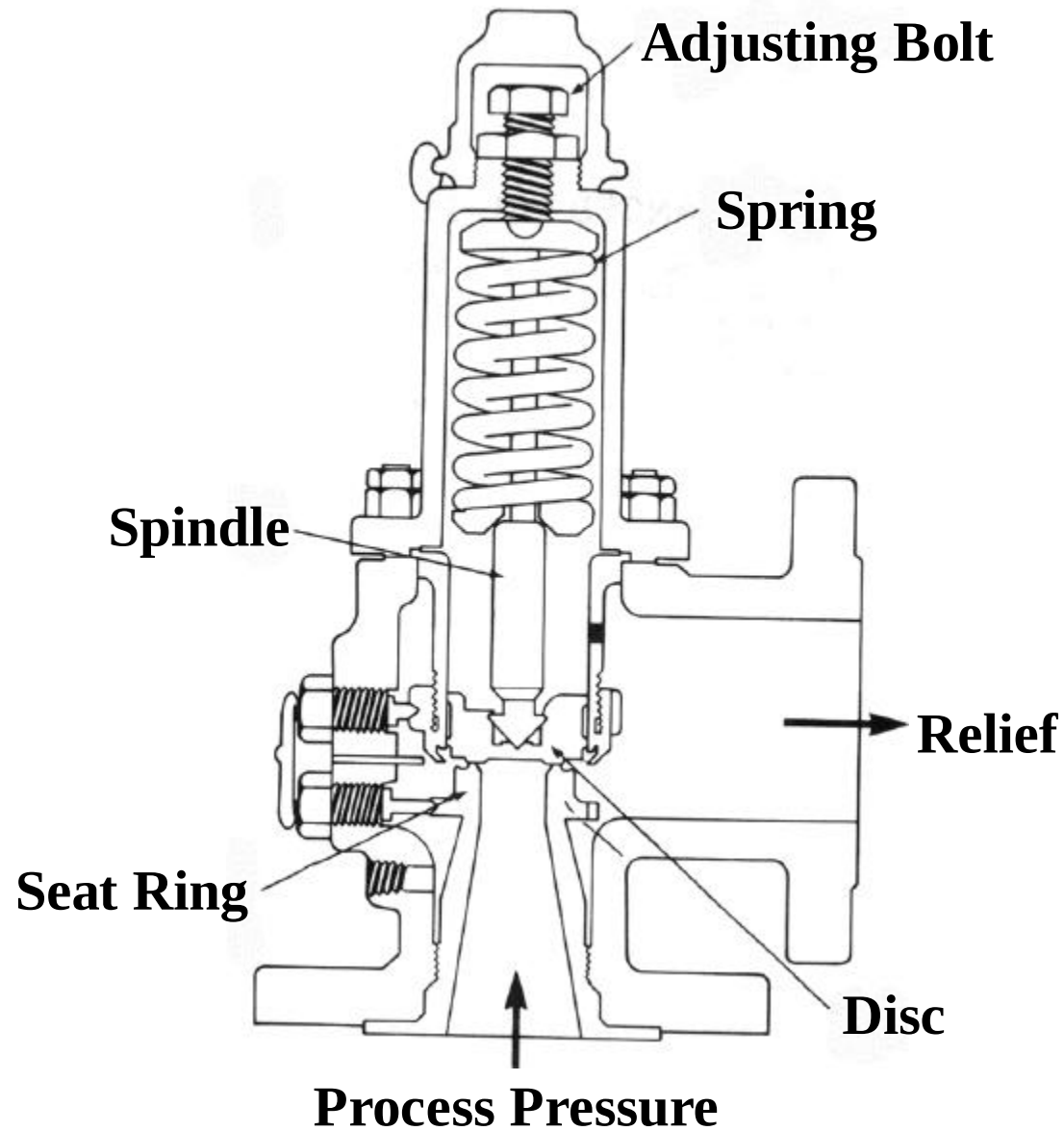


Butterfly Valves

(continued)

- **Consists of a disc, a shaft, and a body**
- **Usually used with an actuator, though they can be manual**
- **Used for flow control of gasses**
- **Not good for complete flow shut off**
 - **Another backup valve or, better, a blind flange is required**

Relief Valve



Relief and Safety Valves

- **Valves protect against overpressure of equipment**
 - **Overpressure could result in equipment damage or failure, with possible catastrophic consequences**
- **One valve can protect a full system (several pieces of equipment) if the system is “open”, i.e., no valves or other means of isolation from the relief/safety valve**
- **Valves are rarely, if ever, used, but must react quickly and properly when called upon, even after a long period of inactivity**

Relief and Safety Valves

(continued)

- **Generally use a spring to remain sealed**
 - **If pressure rises, force on the disc becomes greater than the spring force, causing the disc to lift and the contained fluid and pressure to vent**

- **Upon return to normal pressure, the spring causes the valve to reseal and seal**

Relief and Safety Valves

(continued)

- **Safety valves are for use with compressible fluids (gases) where a quick response is needed**
 - They open fully upon overpressure, relieving at full flow
- **Relief valves are for noncompressable fluids and open more slowly**
 - They do not fully open immediately
 - Less fluid is lost upon relief

Relief and Safety Valves

(continued)

- **Relief valves must be inspected and maintained to insure that they function properly when needed**
- **Relief and safety valves must be installed close to the equipment they protect so that pressure drop in the piping to the valve is not a factor**
- **Backpressure (e.g. trapped liquid, relief system pressure) is not permitted because it is additive to the spring force and will affect the pressure at which the valve opens**

Pipe Classes

- **Pipe classes are a means of conveying all information necessary to specify the components of a section of a piping system.**
- **Classes are usually organized based upon service, metallurgy, flange class, corrosion allowance, and temperature**
- **Classes contain designation of the piping, component, and bolting material, gasketing, flange facing, valve details, minimum piping thicknesses, etc.**

Pipe Classes

(continued)

- **Contractors have prepared standardized classes**
 - **Special classes may be developed as needed**

- **Line indexes are prepared for each line of a specific project**
 - **They list the class, thickness, design temperature and pressure, retirement thickness and other details**

Pipe Classes

(continued)

Flanges	ANSI Class 150 forged raised face. ASTM A 105. 2 inch and smaller socket welding. 2 1/2 inch to 24 inch welding neck bored to suit pipe. 26 inch and larger design per ASME Section VIII, Division 1.
Gaskets	Spiral wound per API 601 except with graphite filler.
Fittings	Class 3000 forged socket weld. ASTM A 105. Welding same wall thickness as pipe. ASTM A 234 WPB or WPB-W. Branch connection welding fittings. ASTM A 105.
Globe Valves	2 inch and smaller Class 800 forged socket weld. ASTM A 105. Vogt series SW-12141, Smith 0G80W, DSI 4211-A8 or Newco 28S-FS2. 2 1/2 inch to 6 inch Class 150 cast flanged. ASTM A 216 Grade WCB. Crane 143-XU, Stockham Triangle 1812V-U, Poyam 22 CIBP-IR12-U or Newco 21F-CB2.
Gate Valves (Note 2)	2 inch and smaller Class 800 forged socket weld. ASTM A 105. Vogt series SW-12111, Smith 0800W, DSI 4111-A8 or Newco 18S-FS2. 2 1/2 inch to 24 inch Class 150 cast flanged. ASTM A 216 Grade WCB Crane 47-XU, Stockham Triangle 1822V-U, Poyam 22 SBW-1R12-U or Newco 11F-CB2.
Check Valves	2 inch and smaller Class 800 forged socket weld. ASTM A 105. Vogt series SWB-701, Smith 0B80W, DSI 4511-A8 or Newco 48S-FS2. 2 1/2 inch to 16 inch Class 150 cast flanged. ASTM A 216 Grade WCB. Crane 147-XU, Stockham Triangle 1842V-U, Poyam 22RBC-1R12-V or Newco 31F-CB2.
Wafer Check Valves	1 1/2 inch and larger Class 150 Class 150 cast. ASTM A 216 Grade WCB. Stockham Duo-Chek II 15 SPF-4.
Plug Valves (Temp.limit to 350°F/177°C)	4 inch and smaller with manufacturer's standard wrenches; 6 inch and larger gear operated. Manufacturer's standard lubricant for specified flowing material, pressure and temperature. Class 150 cast flanged. ASTM A 216 Grade WCB. 1 inch to 4 inch Nordstrom 1925 or Serck Andco MRW/MSW 233 CC. 6 inch to 12 inch Nordstrom F1949 or Serck Andco MSW/MSG 233 CC. 14 inch to 30 inch Nordstrom F4149 or Serck Andco HVG 233 CC.
Bolting	Stud bolts ASTM A 193 B7 threaded full length with 2 ASTM A 194 2H nuts.

Piping Codes

- **Most refinery and petrochemical piping is designed and fabricated in accordance with ASME B31.3, “Process Piping”**
 - Is often referred to as the “Piping Code”
- **Steam system piping is designed and fabricated in accordance with ASME B31.1, “Power Piping”**

Piping Codes (continued)

- **Pipe conforms to ASME B36.10M, “Welded and Seamless Wrought Steel Pipe” and ASME B36.19M, “Stainless Steel Pipe”**
- **Material designations and requirements are in accordance with American Society for Testing of Materials (ASTM) standards**
- **Existing pipe may be evaluated in accordance with ASME B31G, “Manual for Determining the Remaining Strength of Corroded Pipelines”**
- **Piping inspection guidelines are given in API 570, “Piping Inspection Code: Inspection, Repair, Alteration, and Re-Rating of In-Service Piping Systems”**

Pressure Design

- **Code contains a listing of accepted standards**
- **A component that conforms to one of these standards, and that complies with that standard's temperature and pressure ratings, may be used without further evaluation**

Pressure Design

(continued)

- **Components not in accordance with listed standards, and proprietary items for which Code rules are not applicable, shall be designed by rules consistent with Code philosophy**
 - **Design shall be substantiated by service experience under similar conditions, detailed stress analysis, or proof testing**
- **Other components (straight pipe, bends, miters, branches) may be designed by Code equations**

Design Conditions

- **Each component shall be designed for the most severe expected coincident pressure and temperature condition**
 - **Most severe condition is one resulting in the greatest required thickness (piping) or highest required class (flanges, valves)**
- **Governing coincident conditions may not be the highest or lowest pressure or temperature**

Design Conditions

(continued)

- **Different components in same piping system may be governed by different conditions**
- **Advantage may be taken of lower temperatures of uninsulated components**
 - **Temperature may be determined by test, heat transfer calculation, or Code guidelines**
- **Short term variations above design stress or rating are allowed if Code requirements are met**

Useful Piping Code Features

- **Piping Code explicitly permits a design temperature reduction for uninsulated piping, valves, flanges and bolting**
- **Design temperature permitted, as a percentage of the fluid temperature:**

Bolting - 80%

Flanges - 90%

Piping and Valves - 95%

Useful Piping Code Features

(continued)

- **Piping code also explicitly allows for occasional short-term variations above pressure-temperature design conditions if:**
 - **Total number of variations is less than 1000**
 - **Pressure does not exceed test pressure or yield strength at temperature, and**
 - **Non-ductile components are not present**

Useful Piping Code Features

(continued)

- **Stresses may exceed allowable by:**
 - **33% - if no more than 10 hours at a time and 100 hours in a year**
 - **20% - if no more than 50 hours at a time and 500 hours in a year**

Required Piping Thickness

Piping thickness required by ASME B31.3 for internal pressure containment is: $T_m = PD / 2 (SE + PY)$

Where: T_m = Minimum required wall thickness

P = Design pressure

D = Pipe outside diameter

S = Allowable stress at design temperature, per ASME B31.3

E = Quality factor, depending upon radiographic examination or casting type (for valves)

Y = Factor dependent upon material and temperature; for ductile materials and for temperatures below 900 °F, it is 0.4

Equation is valid for $T < D/6$ and $P/SE < 0.385$.

Required Piping Thickness

(continued)

- **Total thickness is the minimum required thickness plus the corrosion allowance, corrected to account for the 12.5% tolerance on thickness**
 - **Standard schedule of pipe is then selected to provide this thickness**

- **Pipe is then evaluated for any longitudinal or thermal loads**

Required Piping Thickness

(continued)

- **Due to the normally small diameter and relatively thick wall of piping, external pressure is usually not a concern**
 - **When it is, use rules and procedures in Pressure Vessel Code**

- **Piping retirement thickness must be determined**
 - **One major factor in determining necessary thickness is the thickness required by internal pressure and temperature**

Basis for Allowable Stresses

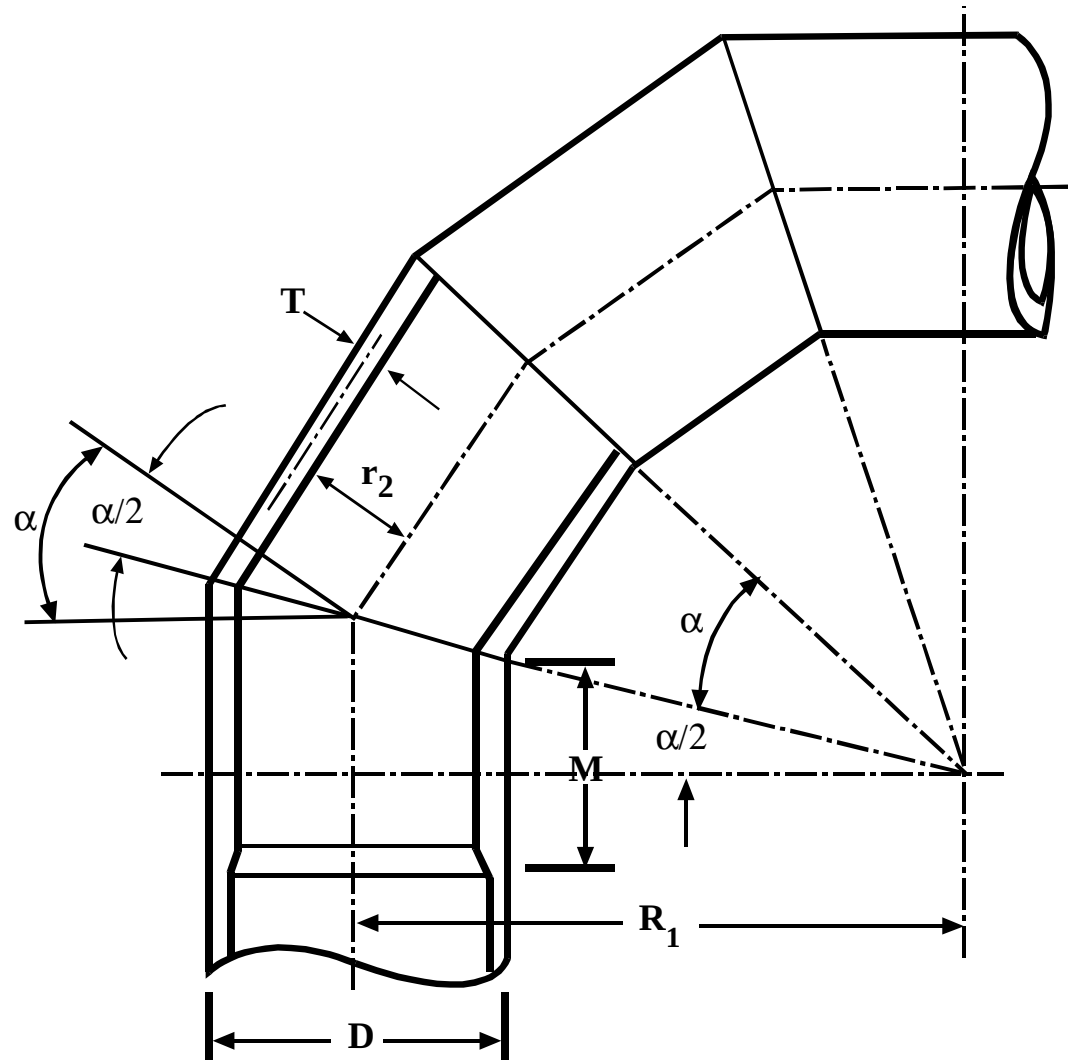
- **Piping Code allowable stresses are based upon the lowest of:**
 - **One third of room temperature and operating temperature tensile strength**
 - **Two thirds of room temperature and operating temperature yield strength**
 - **Average stress for creep rate of 0.01% in 1000 hours**
 - **67 percent of average stress for creep rupture in 100,000 hours**
 - **80 percent of minimum stress for creep rupture in 100,000 hours**

Basis for Allowable Stresses (continued)

- **Below the creep range, this criteria results in allowable stresses that are generally higher than those permitted by Division 1 of the Pressure Vessel Code**
 - **Are more closely related to Division 2 allowable range**

- **In the creep range, the allowable stress basis is the same as Division 1 of the Pressure Vessel Code**

Multiple Miter Bend



Miter Allowable Pressure

- Per the Piping Code (B31.3), the allowable internal pressure for a multiple bend miter is the lessor of:

$$P_m = \frac{SE(T - C)}{r_2} \frac{T - C}{\frac{C}{e}(T - C) + 0.643 \tan \phi \sqrt{r_2(T - C)}} \frac{\sigma}{\phi}$$

or

$$P_m = \frac{SE(T - C)}{r_2} \frac{R_1 - r_2}{\frac{C}{e}R_1 - 0.5r_2} \frac{\sigma}{\phi}$$

Miter Allowable Pressure

(continued)

- **Limitations include:**

$$Q \leq 22.5^\circ$$

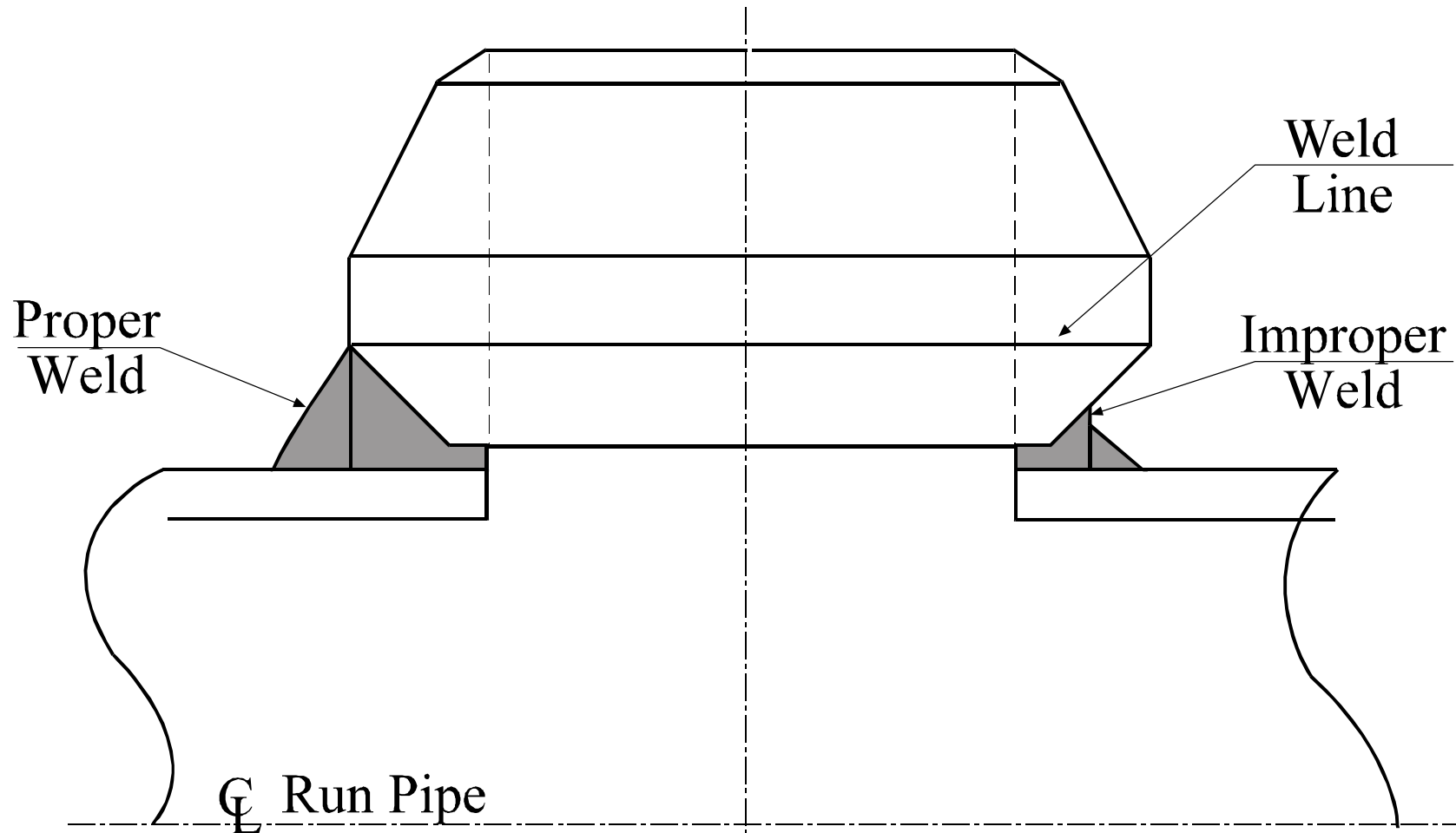
$$M = \text{larger of } 2.5(r_2 T)^{0.5} \text{ and } \tan Q (R_1 - r_2)$$

- **Separate equations apply to single miter bends**

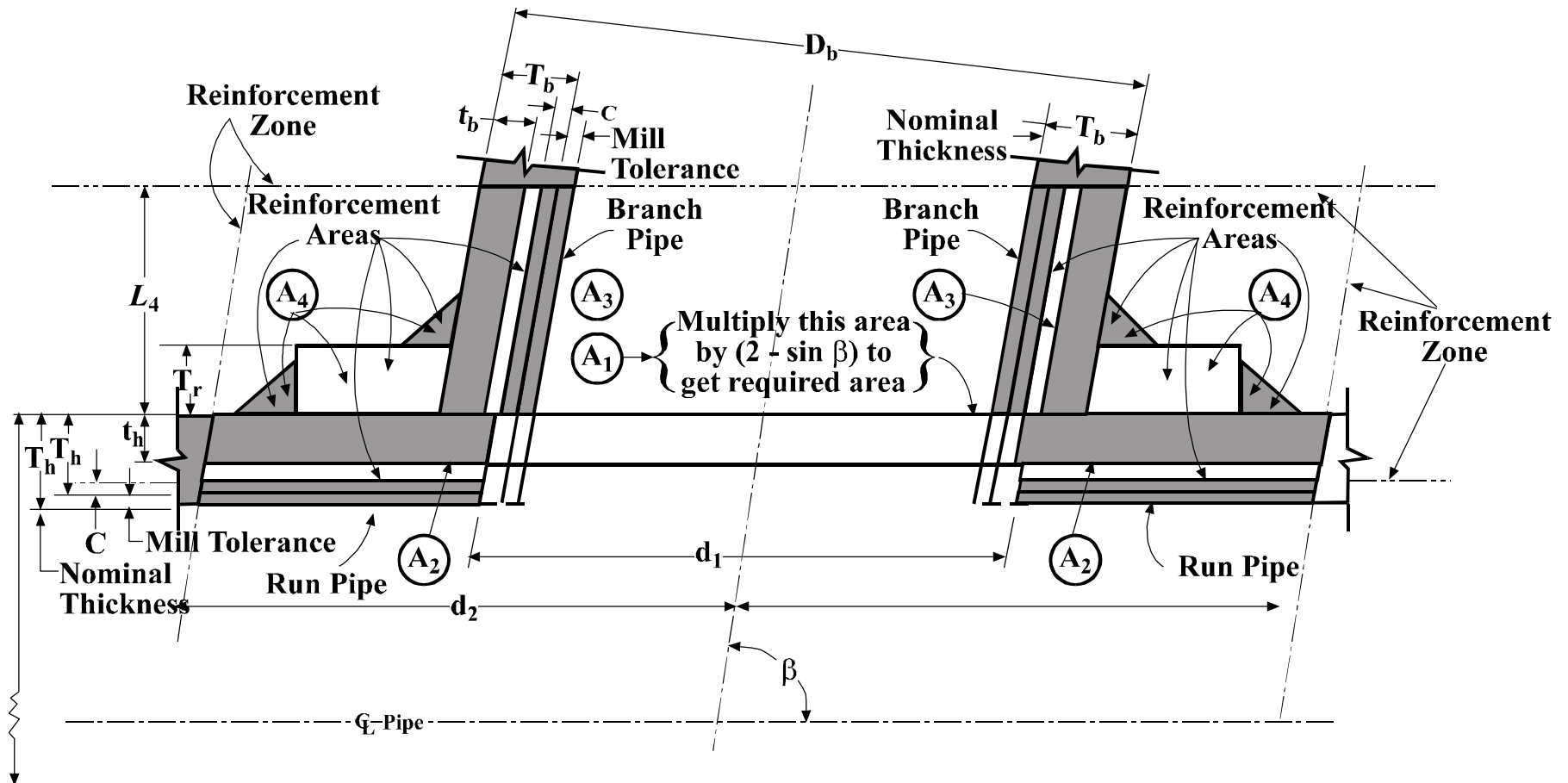
Branch Connections

- **Junctions, or branch connections made with a fitting (eg, a tee) complying with standard listed in the Piping Code may be used within its rated pressure without evaluation**
- **Use of proprietary fittings, such as weldolets, is acceptable, provided they have been qualified by burst test requirements of the Code**
 - **Be sure that they are fully and properly welded**
- **Branch junctions may be fabricated, but they require an evaluation for adequate reinforcement**
 - **Area replacement method is used**

Branch Connections (continued)



Branch Connection Nomenclature



Branch Connection Reinforcement

- Required reinforcement area, $A_1 = t_h d_1 (2 - \sin b)$

where:

t_h = pressure design thickness of header

d_1 = effective length removed from header

b = small angle between header and branch

Branch Connection Reinforcement ***(continued)***

■ **Available reinforcement:**

$$A_2 = \text{excess header thickness} = (2d_2 - d_1)(T_h - t_h - c)$$

where:

d_2 = reinforcement zone radius usually $= d_1$

**T_h = nominal header thickness including the
minus mill tolerance**

c = mechanical allowances (e.g. corrosion)

Branch Connection Reinforcement ***(continued)***

A_3 = excess branch thickness = $2L_4(T_b - t_b - c)/\sin b$

where:

L_4 = Height of reinforcement zone, usually $2.5(T_h - c)$

T_b = Total branch thickness (including minus mill tolerance)

t_b = pressure design thickness of branch

A_4 = area of weld metal and reinforcement within reinforcement zone

■ For a properly reinforced opening:

$$A_2 + A_3 + A_4 \geq A_1$$

Branch Connections

- **Extrusions may also be used for branch connection points**
- **Extrusions are formed by pulling a die through the wall of the pipe, creating a radius at the opening**
 - **Radius reduces local stresses, and the joining weld is a simple butt weld, located away from any mechanical stress raisers**
 - **Adequate reinforcement using area replacement method must be present**

Branch Connections

(continued)



Extruded Junction

Piping Testing

- **Piping Code requires that erected system be pressure tested**
- **Testing requirements are similar to those required for pressure vessels, i.e., hydrostatic or pneumatic testing based upon design pressure**

Piping Testing

(continued)

■ **Hydrostatic Testing:**

- **The Piping Code requires a test pressure of 1.5 times the design pressure times the ratio of cold to hot allowable stresses (S_C/S_H)**
- **The S_C/S_H ratio is “capped” at 6.5**
 - **Minimizes possibility of distortion or damage during hydrotest from a high test pressure caused by a low allowable stress due to creep considerations for elevated temperature operation**

Piping Testing

(continued)

■ **Pneumatic Testing**

- **The Piping Code requires a test pressure of 110 percent of design pressure, without correction for cold vs. hot allowable stress**
- **This is due to the inherent danger from the stored energy during a pneumatic test**
- **The initial test is normally performed at 25psig and gradually increased to the final test pressure (110 percent of design)**

Piping Testing *(continued)*

- **Nondestructive Examination**
 - **In certain circumstances (e.g., water exposure is undesirable and pneumatic testing unduly hazardous), Piping Code, unlike the Pressure Vessel Code, also permits nondestructive examination plus a leak test instead of full pressure testing**

Piping Testing

(continued)

- **During hydrotesting, care must be taken to insure that system is adequately supported for the additional weight of the water**
- **Flexible items, such as spring hangers, need to be fixed into place to prevent excessive distortion (remember to remove the fixity before returning to service)**
 - **Some components (e.g., expansion joint bellows, strainer internals) may need to be isolated from the test**

Piping Testing

(continued)

- **Additional pressure due to any head of water present must be considered when testing a piping system**
 - **May limit overall test pressure to avoid overstress of portions of the piping**

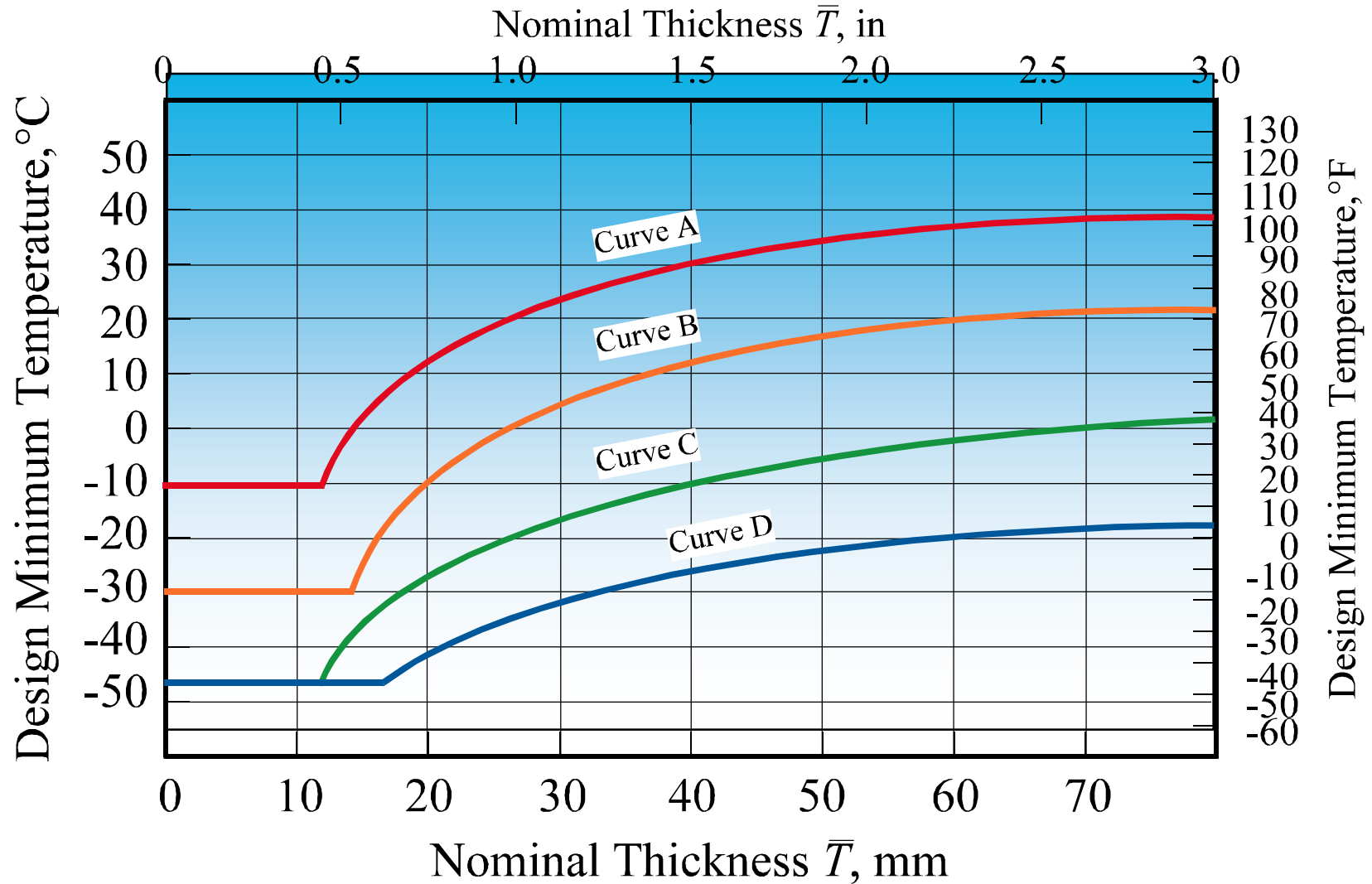
Cold Service Requirements

- **Piping must be adequate for design minimum temperature**
 - Temperature may be from cold service, auto-refrigeration, low ambient temperature, or other causes
- **Minimum temperature at which each material may be used without further evaluation is listed in stress tables of the piping code**
 - -20 °F is a common value
 - For some materials, reference is made to curves similar to Pressure Vessel Code's MDMT curves

Cold Service Requirements (continued)

- **For colder temperatures, material must be Charpy V-notch impact tested, and meet listed minimum impact capacity values, before it can be used.**
- **Alternatively, pretested materials, certified as suitable for a specified low temperature, may be used without testing. These materials have passed supplier performed Charpy V-notch impact tests at the specified temperature.**

Minimum Temperatures Without Impact Testing for Carbon Steel Materials



Examination and Heat Treatment

- **Process Piping Code contains requirements for piping system assembly welding preheat and postweld heat treatment**
- **Requirements are dependent upon material characteristics**
 - **Similar materials are grouped and assigned a “P” number, which is used to define the requirements**

Examination and Heat Treatment

(continued)

- **Required examination of welds consists of visual inspection and a random radiograph or ultrasonic examination of 5 percent of the butt welds of the completed system.**
 - **Further examination is as required by the piping design**
- **Heat treatment and examination of longitudinal welds are as required by ASTM specification and class of piping specified. ASME B31.3 may impose additional requirements.**
- **For some materials and services UOP imposes additional Radiography and heat treatment requirements.**

Piping Flexibility

- **Piping will be subject to thermal expansion (or contraction) caused by its own heating (or cooling) or that of the items to which it is connected**
- **Piping must be able to accommodate this movement without failure, overstress, flange leakage, or overloading the items to which it is connected**

Piping Flexibility (continued)

- **Accommodation of the movements is a job for a specialist**
- **Encompasses plant layout, routing of the piping and methods of supporting the piping**
- **Specialized computer programs are used to analyze the system**
- **Flexibility is strain, not stress, driven**

Piping Flexibility

(continued)

- **Object is to avoid:**
 - **Failure of piping or components due to overstress or fatigue**
 - **Leakage at joints**
 - **Detrimental stress or distortion in piping, valves, or connected equipment (e.g. pumps, turbines, expanders)**
 - **Creation of “pockets” (unvented or drained high or low spots), loss of required free draining, etc**

Piping Flexibility

(continued)

- **In addition to thermal expansion and support of the piping's weight, wind, earthquake, vibration, dynamic loading, and friction must be considered**

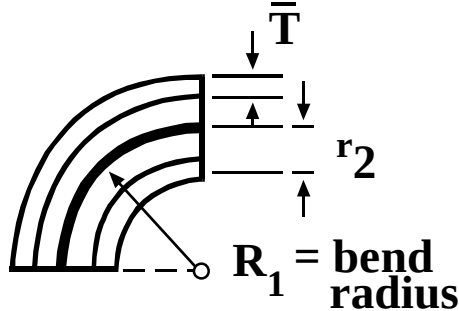
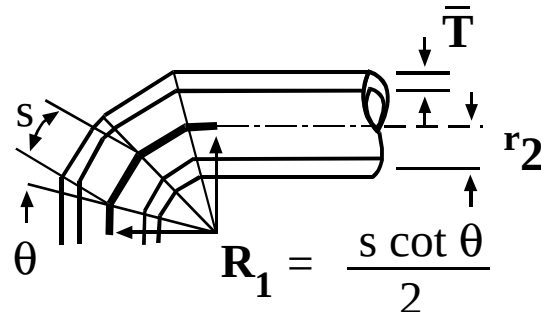
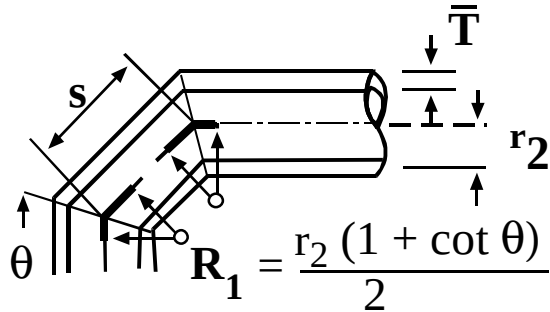
- **Analysis usually considers three cases:**
 - **Sustained (force driven) loads only**
 - **Displacement (thermal loads) and movements only**
 - **Operating case for reactions at equipment**

Piping Flexibility

(continued)

- **Stress concentrations at elbows and other points must be considered**
- **Piping Code contains stress intensification factors for determining effective stress at these points**

Flexibility and Stress Intensification Factors

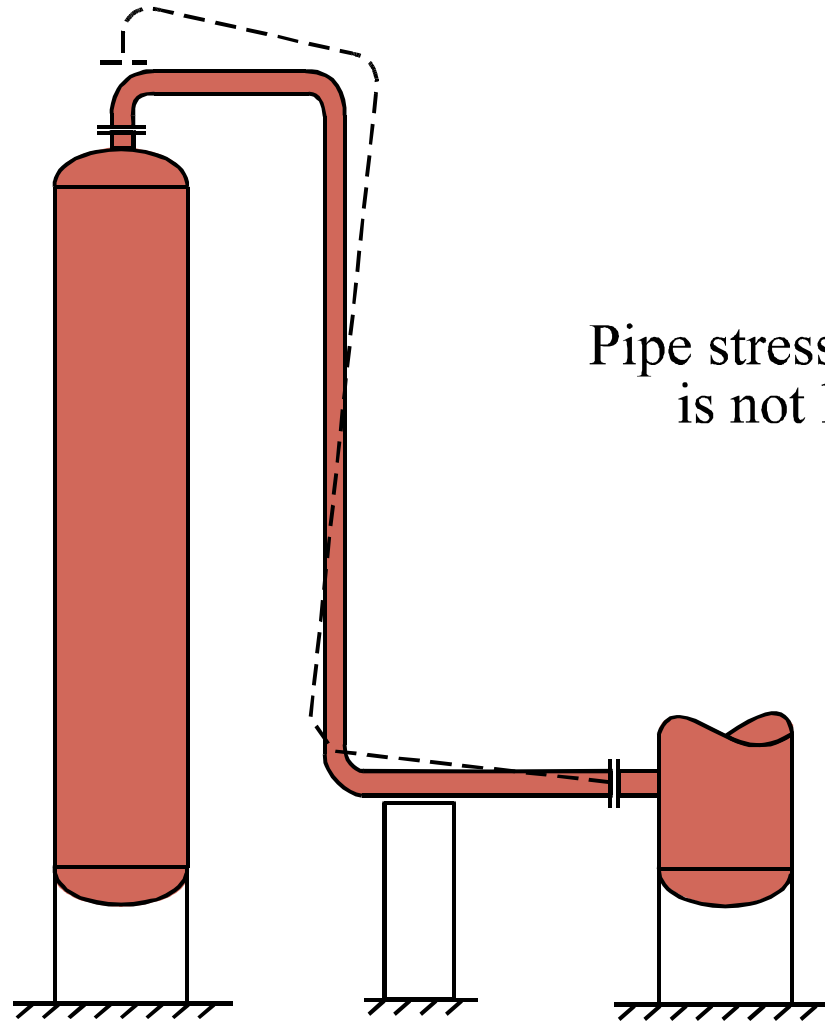
Description	Stress Intensification Factor			Flexibility Characteristic h	Sketch
	Flexibility Factor k	Outplane i_o	Inplane i_i		
Welding elbow or pipe bend	$\frac{1.65}{h}$	$\frac{0.75}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$\frac{\bar{T} R_1}{r_2^2}$	 $R_1 = \text{bend radius}$
Closely spaced miter bend $s < r_2 (1 + \tan \theta)$	$\frac{1.52}{h^{5/6}}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$\frac{\cot \theta}{2} \frac{\bar{T} s}{r_2^2}$	 $R_1 = \frac{s \cot \theta}{2}$
Single miter bend or widely spaced miter bend $s \geq r_2 (1 + \tan \theta)$	$\frac{1.52}{h^{5/6}}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$\frac{1 + \cot \theta}{2} \frac{\bar{T}}{r_2}$	 $R_1 = \frac{r_2 (1 + \cot \theta)}{2}$

Piping Flexibility

(continued)

- **Analysis is based upon operating temperature to properly account for all interactions and movements of lines and equipment**
- **Review the entire system—not just one line—because all portions are interrelated**
 - **Response and stresses depend on the other portions**

Sustained and Displacement Stresses when Pipe Lifts off of Support



Piping Flexibility (continued)

- Sustained load only case considers weight of the uncorroded piping, including contents, valves and attachments and longitudinal stresses due to pressure
- Also includes other force driven loads, e.g., wind, earthquake, friction, etc
- Use nominal thickness in the corroded condition for determining stresses
- Is no thermal movement; therefore, flexibility (i.e., moment of inertia) is not a factor

Piping Flexibility (continued)

- **Allowable stress for the weight case is the hot allowable stress listed in the Piping Code, without a joint efficiency factor**
- **A 33% increase may be used when considering wind or earthquake loadings**

Piping Flexibility

(continued)

- **For displacement, or strain driven, cases only the forces and loads resulting from the imposed displacements, such as thermal growth, are considered**
- **Consideration of multiple cases (e.g., startup, shutdown, operation, steamout, etc.) may be necessary in order to cover all possible combinations of temperatures for portions of the piping**
 - **Piping weight is not included**

Piping Flexibility

(continued)

- **Use nominal (uncorroded) thickness and the ambient temperature modulus of elasticity for maximum stiffness, hence highest displacement loads**
- **Thermal expansion is determined from the material's thermal expansion coefficient, the operating temperature, and lengths of pipe at each temperature**

Piping Flexibility

(continued)

- **The general expression for the thermal expansion of pipe is:**

$$DL = LaDT$$

where:

L = Length

DT = Change in temperature

a = Thermal expansion coefficient— change per unit length per degree

Thermal Expansion Coefficients (10^{-6} inches/inch-°F)

Temp., °F	Material							
	Carbon Steel Carbon-Moly- Low-Chrome (Through 3Cr-Mo)	5Cr-Mo Through 9Cr-Mo	Austenitic Stainless Steels 18Cr-8Ni	12Cr, 17Cr, 27Cr	25Cr-20Ni	UNS N04400 Monel 67Ni-30Cu	3½Ni	Copper and Copper Alloys
-450	...	...	...	...	...	...	...	6.30
-425	...	...	...	...	...	...	...	6.61
-400	...	...	...	...	...	...	...	6.93
-375	...	...	...	...	...	...	...	7.24
-350	...	...	...	...	...	...	...	7.51
-325	5.00	4.70	8.15	4.30	...	5.55	4.76	7.74
-300	5.07	4.77	8.21	4.36	...	5.72	4.90	7.94
-275	5.14	4.84	8.28	4.41	...	5.89	5.01	8.11
-250	5.21	4.91	8.34	4.47	...	6.06	5.15	8.26
-225	5.28	4.98	8.41	4.53	...	6.23	5.30	8.40
-200	5.35	5.05	8.47	4.59	...	6.40	5.45	8.51
-175	5.42	5.12	8.54	4.64	...	6.57	5.52	8.62
-150	5.50	5.20	8.60	4.70	...	6.75	5.59	8.72
-125	5.57	5.26	8.66	4.78	...	6.85	5.67	8.81
-100	5.65	5.32	8.75	4.85	...	6.95	5.78	8.89
-75	5.72	5.38	8.83	4.93	...	7.05	5.83	8.97
-50	5.80	5.45	8.90	5.00	...	7.15	5.88	9.04
-25	5.85	5.51	8.94	5.05	...	7.22	5.94	9.11
0	5.90	5.56	8.98	5.10	...	7.28	6.00	9.17
25	5.96	5.62	9.03	5.14	...	7.35	6.08	9.23
50	6.01	5.67	9.07	5.19	...	7.41	6.16	9.28
70	6.07	5.73	9.11	5.24	...	7.48	6.25	9.32
100	6.13	5.79	9.16	5.29	...	7.55	6.33	9.39
125	6.19	5.85	9.20	5.34	...	7.62	6.36	9.43
150	6.25	5.92	9.25	5.40	...	7.70	6.39	9.48
175	6.31	5.98	9.29	5.45	...	7.77	6.42	9.52
200	6.38	6.04	9.34	5.50	8.79	7.84	6.45	9.56
225	6.43	6.08	9.37	5.54	8.81	7.89	6.50	9.60
250	6.49	6.12	9.41	5.58	8.83	7.93	6.55	9.64

Thermal Expansion (inches/100 ft)

Temp., °F	Material							Copper and Copper Alloys
	Carbon Steel Carbon-Moly- Low-Chrome (Through 3Cr-Mo)	5Cr-Mo Through 9Cr-Mo	Austenitic Stainless Steels 18Cr-8Ni	12Cr, 17Cr, 27Cr	25Cr-20Ni	UNS N04400 Monel 67Ni-30Cu	3½Ni	
-450	...	...	...	...	...	...	...	- 3.93
-425	...	...	...	...	...	...	...	- 3.93
-400	...	...	...	...	...	...	...	- 3.91
-375	...	...	...	...	...	...	...	- 3.87
-350	...	...	...	...	...	...	...	- 3.79
-325	- 2.37	- 2.22	- 3.85	- 2.04	...	- 2.62	- 2.25	- 3.67
-300	- 2.24	- 2.10	- 3.63	- 1.92	...	- 2.50	- 2.17	- 3.53
-275	- 2.11	- 1.98	- 3.41	- 1.80	...	- 2.38	- 2.07	- 3.36
-250	- 1.98	- 1.86	- 3.19	- 1.68	...	- 2.26	- 1.96	- 3.17
-225	- 1.85	- 1.74	- 2.96	- 1.57	...	- 2.14	- 1.86	- 2.97
-200	- 1.71	- 1.62	- 2.73	- 1.46	...	- 2.02	- 1.76	- 2.76
-175	- 1.58	- 1.50	- 2.50	- 1.35	...	- 1.90	- 1.62	- 2.53
-150	- 1.45	- 1.37	- 2.27	- 1.24	...	- 1.79	- 1.48	- 2.30
-125	- 1.30	- 1.23	- 2.01	- 1.11	...	- 1.59	- 1.33	- 2.06
-100	- 1.15	- 1.08	- 1.75	- 0.98	...	- 1.38	- 1.17	- 1.81
-75	- 1.00	- 0.94	- 1.50	- 0.85	...	- 1.18	- 1.01	- 1.56
-50	- 0.84	- 0.79	- 1.24	- 0.72	...	- 0.98	- 0.84	- 1.30
-25	- 0.68	- 0.63	- 0.98	- 0.57	...	- 0.77	- 0.67	- 1.04
0	- 0.49	- 0.46	- 0.72	- 0.42	...	- 0.57	- 0.50	- 0.77
25	- 0.32	- 0.30	- 0.46	- 0.27	...	- 0.37	- 0.32	- 0.50
50	- 0.14	- 0.13	- 0.21	- 0.12	...	- 0.20	- 0.15	- 0.22
70	0	0	0	0	0	0	0	0
100	0.23	0.22	0.34	0.20	0.32	0.28	0.23	0.34
125	0.42	0.40	0.62	0.36	0.58	0.52	0.42	0.63
150	0.61	0.58	0.90	0.53	0.84	0.75	0.61	0.91
175	0.80	0.76	1.18	0.69	1.10	0.99	0.81	1.20
200	0.99	0.94	1.46	0.86	1.37	1.22	1.01	1.49
225	1.21	1.13	1.75	1.03	1.64	1.46	1.21	1.79
250	1.40	1.33	2.03	1.21	1.91	1.71	1.42	2.09

Displacement Stresses per B31.3

- Stresses shall be computed using the as installed modulus of elasticity; computed displacement stress range, S_E , is the greatest algebraic difference in stress:

$$S_E = \sqrt{S_b^2 + 4S_t^2}$$

S_b = Resultant bending stress range
= Resultant moment range $(S_{in}^2 + S_{out}^2)^{0.5}$ /section modulus

in = in plane out = out of plane

S_t = Torsional stress range
= Torsional moment range/2(section modulus)

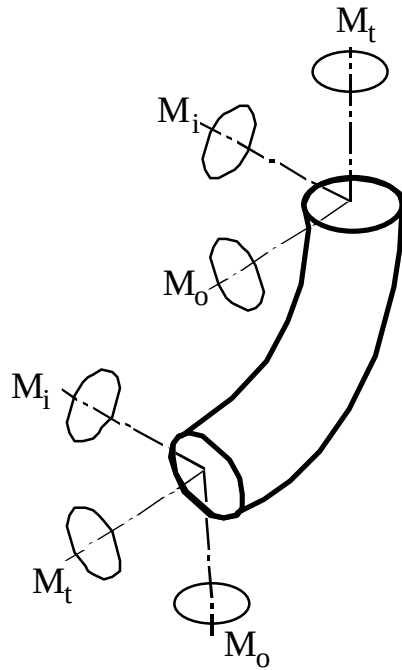
Displacement Stresses per B31.3 (continued)

- **Axial stress effects are generally insignificant and ignored**
- **For elbows, tees, and other components with applicable stress intensification factors (i) and applied moment M, the form of S_b is:**

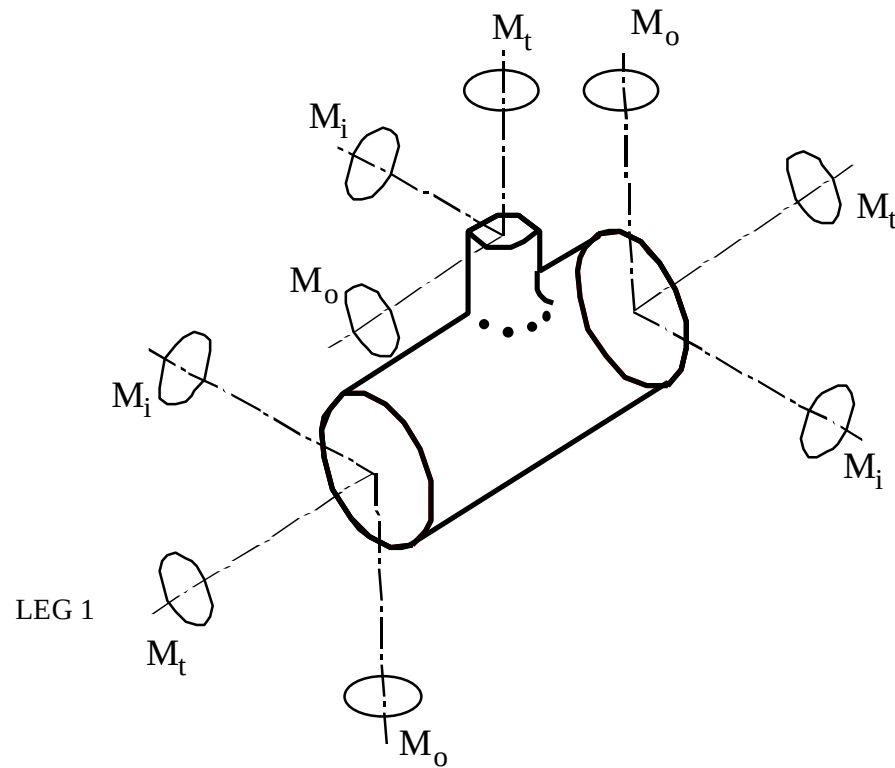
$$S_b = \frac{\sqrt{(\mathbf{i}_i \mathbf{M}_i)^2 + (\mathbf{i}_o \mathbf{M}_o)^2}}{\mathbf{Z}}$$

- **Subscripts i and o represent in and out of plane respectively; reference plane is that of the component, not of the piping system**
- **Computed stress range S_E is compared to the allowable stress range (S_A)**

Moments



Moments in Bends



Moments in Branch Connections

Piping Flexibility

- The allowable stress range, per Process Piping Code (B31.3) is:

$$S_a = \frac{1}{2} (1.25 S_c + 0.25 S_h)$$

If $S_h > S_l$ then the following formula may be used:

$$S_a = \frac{1}{2} [1.25 (S_c + S_h) - S_l] \text{ where:}$$

Piping Flexibility

(continued)

Where:

- S_a = Allowable displacement stress range
- f = Stress reduction factor accounting for the number of full displacement cycles (equals 1 if cycles are less than 7000 during expected service life)
- S_c = Code allowable stress at minimum temperature during the displacement cycle
- S_h = Code allowable stress at maximum temperature during the displacement cycle
- S_l = Longitudinal stresses due to sustained loadings (e.g., weight, pressure, etc.), based on nominal thickness minus corrosion allowance; thermal stress not included; under tolerance need not be subtracted to determine S_l

Piping Flexibility

(continued)

- **Higher allowable stress is permitted because stresses are self limiting**
- **Any local areas exceeding yield, or subject to creep, deform causing a stress reduction and redistribution**
- **Although stress range is unchanged, stress values will change**

Piping Flexibility

(continued)

- **Piping stresses are combined vectorially to determine resultant stresses**
 - **Compare to the allowable stress**

- **Changing wall thickness of straight pipe does not affect thermal bending stress levels**
 - **Both the section modulus and moment of inertia remain in same ratio**
 - **Reactions at anchors may change**

Piping Flexibility

(continued)

- **For very simple, uniform size, two anchor systems, with no intermediate restraints or branches, the Piping Code offers an empirical method of evaluation**

If $D_y / (L - U)^2 \leq K1$,

No formal flexibility analysis is necessary

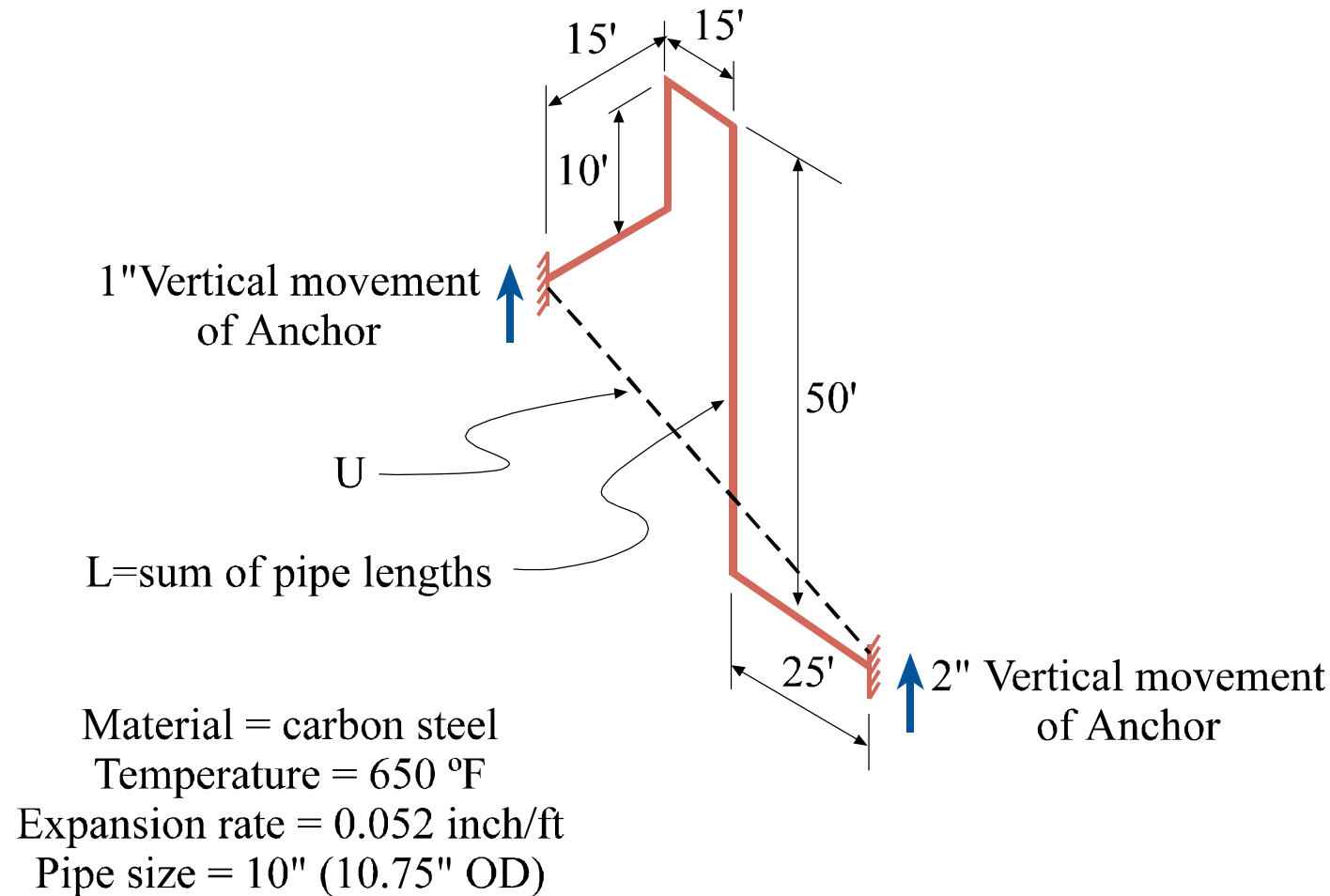
Piping Flexibility

(continued)

■ Where:

- **D = outside diameter of pipe (in or mm)**
 - **All piping must be the same size and thickness**
- **y = resultant of total displacement strains to be absorbed by the system (in or mm)**
- **L = developed piping length between anchors (ft or m)**
- **U = straight line distance between anchors (ft or m)**
- **K1 = 30 (S_a/E_a) for English units (208,000 (S_a/E_a) for SI units)**

Example of Simplified Stress Evaluation



Example of Simplified Stress Evaluation (continued)

$$\mathbf{L} = 15' + 10' + 15' + 50' + 25' = 115'$$

$$\begin{aligned}\mathbf{U} &= \sqrt{\mathbf{x}^2 + \mathbf{y}^2 + \mathbf{z}^2} = \sqrt{(15 + 25)^2 + (10 - 50)^2 + (15)^2} \\ &= \sqrt{(40)^2 + (-40)^2 + (15)^2} = \sqrt{1600 + 1600 + 225} \\ &\quad \sqrt{3425} = 58.5 \text{ feet}\end{aligned}$$

$$\Delta x = (15 + 25)(0.052) = 2.08''$$

$$\Delta y = (10 - 50)(0.052) - 2 + 1 = -3.08''$$

$$\Delta z = -15(0.052) = -0.78''$$

$$y = \sqrt{(2.08)^2 + (-3.08)^2 + (-0.78)^2} = 3.81''$$

Example of Simplified Stress Evaluation (continued)

D= 10.75" (10" nps pipe)

$$\frac{Dy}{(L-U)^2} = \frac{(10.75)(3.81)}{(115-58.5)^2}$$

$$= 0.0128 < 30(Sa/ Ea) \cong 30(0.001) = 0.03$$

- **Adequate flexibility has been provided.**
- **Method applies to piping stresses only**
 - **Does not address the reactions, which must be evaluated with a formal stress analysis**

Piping Flexibility

(continued)

- **In addition to an evaluation of the stresses, thermal movements must be reviewed to insure that there is adequate space to accommodate them, particularly on the pipe rack**

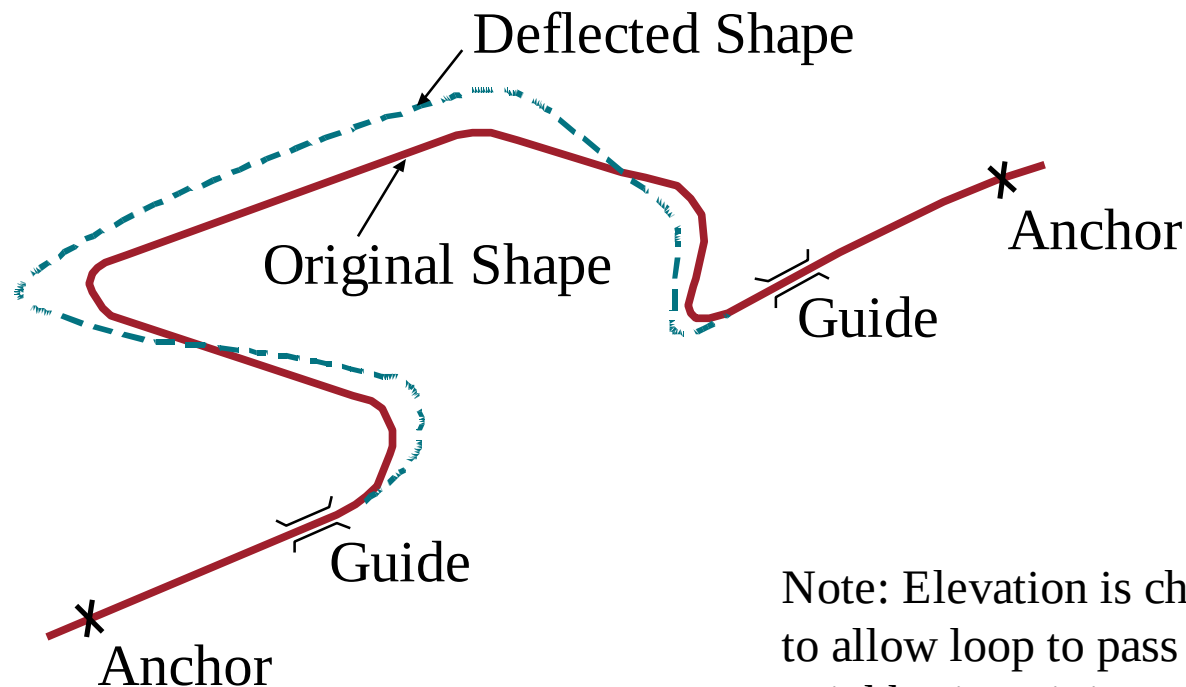
Providing Flexibility

- **Flexibility is generally provided by bending or distortion of the piping**
- **Installed components, such as valves and flanges, are very stiff and contribute little flexibility**
 - **They may be damaged by imposed bending or distortion**

Providing Flexibility (continued)

- **Most common method of increasing flexibility of a piping system is with changes in direction such as piping loops**
 - **Legs of the loop(s) allow absorption of movements**
- **Natural arrangement of the piping will often provide sufficient loops and flexibility**
 - **May be necessary to add artificial loops**
 - **Horizontal loops on pipe racks are an example**

Horizontal Expansion Loop

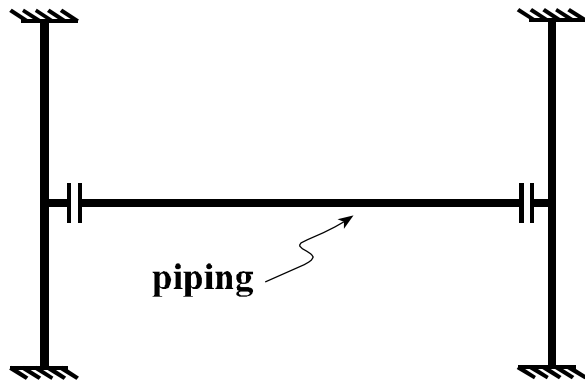


Note: Elevation is changed to allow loop to pass over neighboring piping.

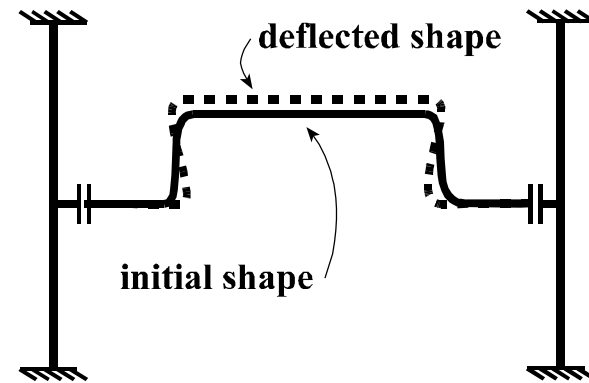
Providing Flexibility (continued)

- **Loops do, however, add to piping length and pressure drop**
- **Vertical direction changes may also create high or low points (local as well as global) that require vents or drains**

Loop Example

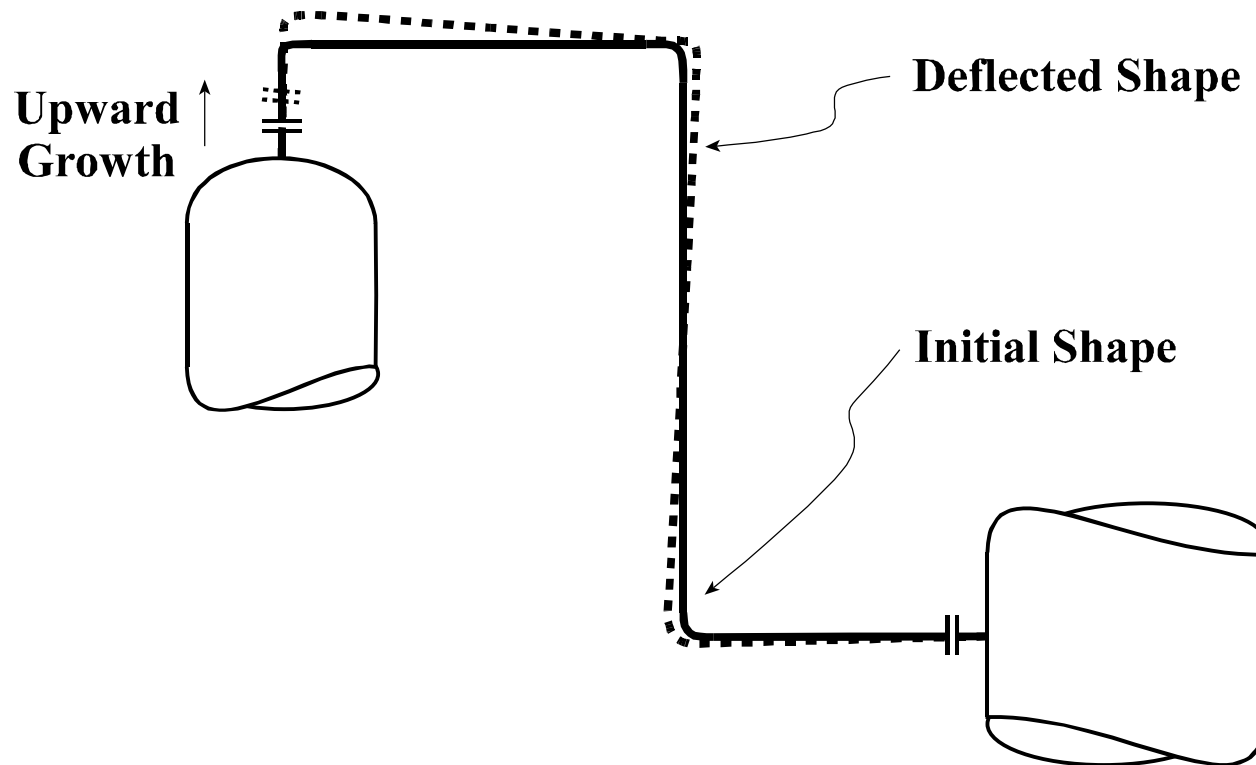


Inflexible System



Flexibility Improved with a Loop

Loop Examples



Natural “Loop” in the piping layout

Providing Flexibility (continued)

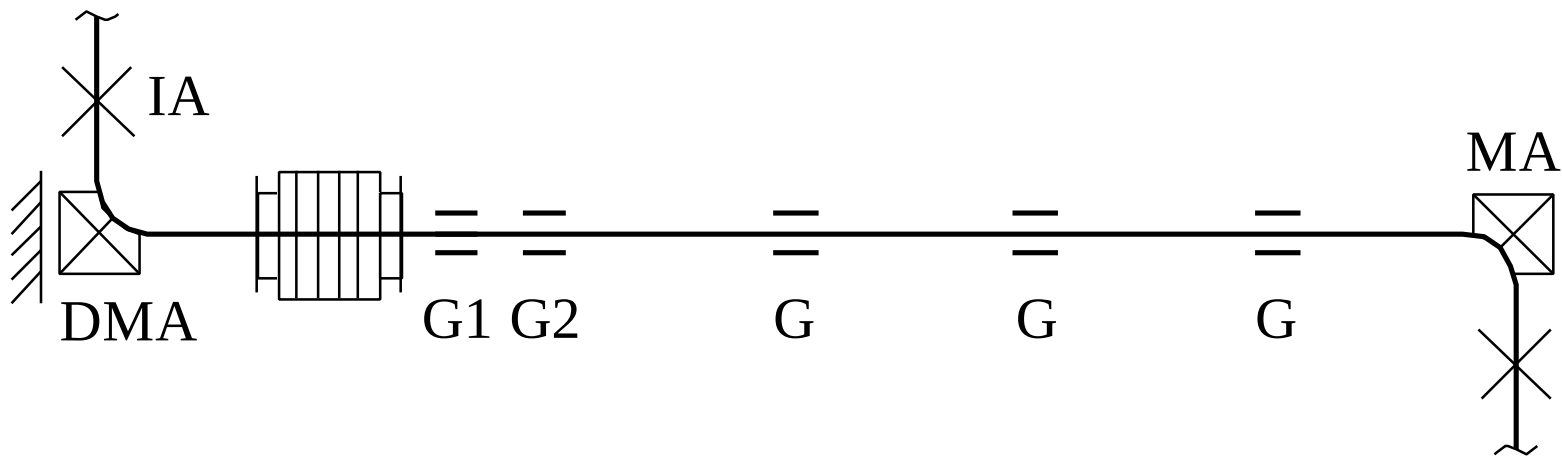
- **Elbows increase the flexibility, but are also points of stress concentration**

- **Piping Code contains**
 - **Flexibility factors**
 - **Accounts for reduced stiffness**
 - **Stress intensification factors**
 - **Accounts for higher stresses at these points**

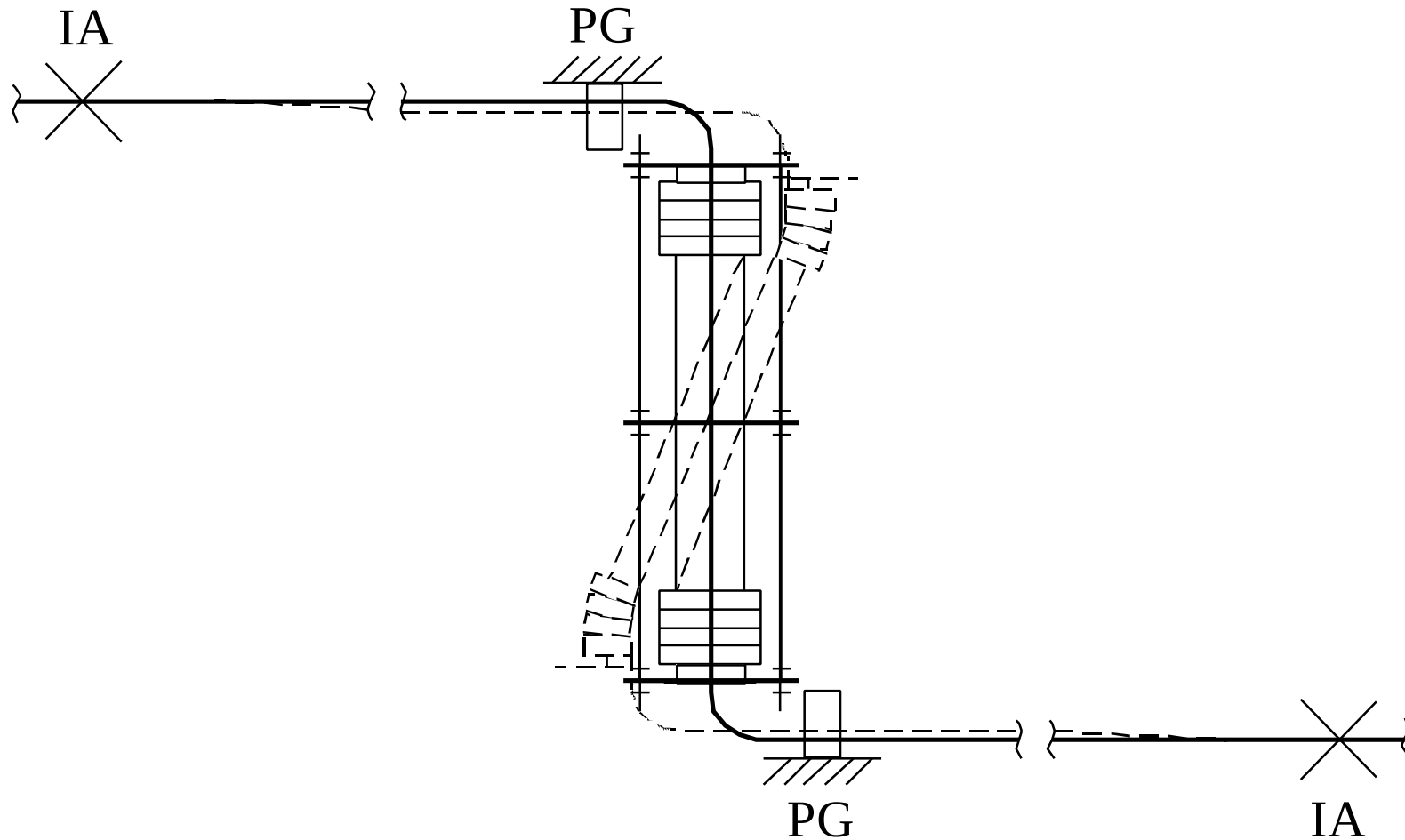
Providing Flexibility (continued)

- **In low pressure (usually below 50 - 60 psig), critical, cases, where loops are not an option, expansion joints may be considered**
- **Expansion joints consist of one or more flexible bellows that may compress, elongate, or rotate slightly to absorb piping movements**

Single Expansion Joint



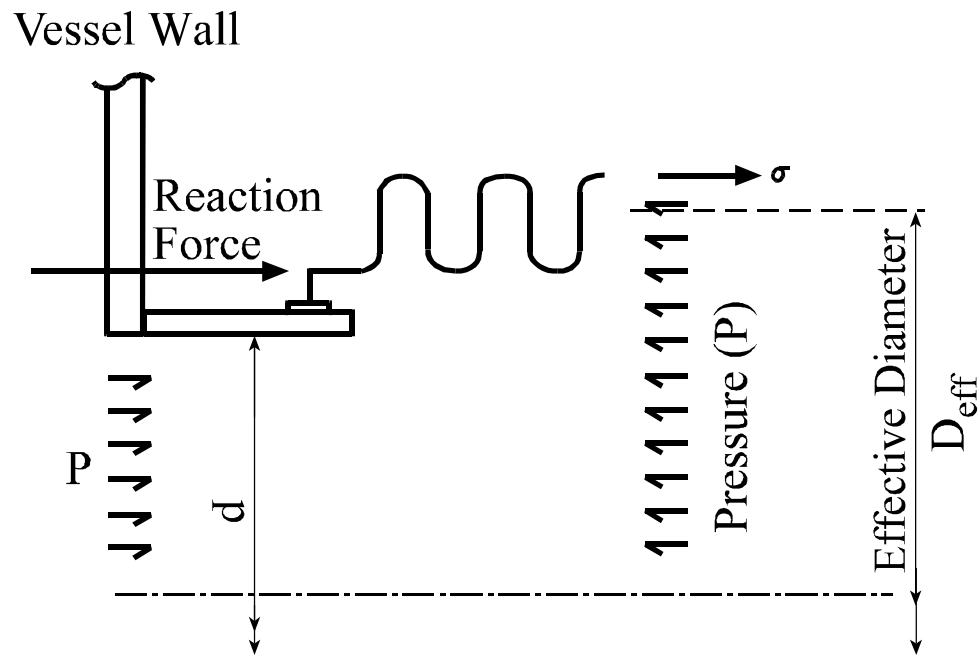
Universal Expansion Joint



Providing Flexibility (continued)

- **Expansion joints cannot transmit forces across the bellows**
 - **Piping forces, including “blowout” on the bellows, must be absorbed by piping system, via anchors**
 - **Some special joints (tied, hinged, or pressure balanced) do permit transmission of axial forces**

Expansion Joint Blowout Force



$$\begin{aligned} \text{Reaction Force on Nozzles} &= (P)A_{\text{Annulus}} & A_{\text{Annulus}} &= \frac{P}{4}(D_{\text{eff}}^2 - d^2) \\ \text{Force at Vessel Support} &= (P)A_{\text{Bellows}} \end{aligned}$$

$$\begin{aligned} \text{Blowout Force } F_B &= P \frac{P}{4} D_{\text{eff}}^2 & A_{\text{Bellows}} &= \frac{P}{4} D_{\text{eff}}^2 \end{aligned}$$

Providing Flexibility (continued)

- **Expansion joints are tested in place to confirm the adequacy of the main anchors.**
 - **Test pressure is no more than 150% of design pressure**

- **Joints are normally installed in vertical position to allow draining of corrugations**
 - **Otherwise, a means of drainage is necessary**

Providing Flexibility (continued)

- **Many types and configurations of expansion joints are available**
 - **Allows absorption of sometimes large movements**
- **Expansion joint designs are governed by the Expansion Joint Manufacturers Association (EJMA) Standards and Appendix X of B31.3**

Expansion Joint Bellows

- **Bellows must be flexible to allow deformation and movement absorption**
 - **Are very thin**
- **Bellows stiffness (spring constant -- axial, rotation, or torsional) must be overcome before the bellows will deform**

Expansion Joint Bellows

(continued)

- **Bellows must be strong to withstand internal pressure and local stress at the bends (which must retain a smooth radius)**
- **To provide needed strength and corrosion resistance, at generally elevated temperatures (causing significant movements to be absorbed) and still be thin and therefore flexible, bellows are constructed of high alloy material**

Expansion Joint Bellows

(continued)

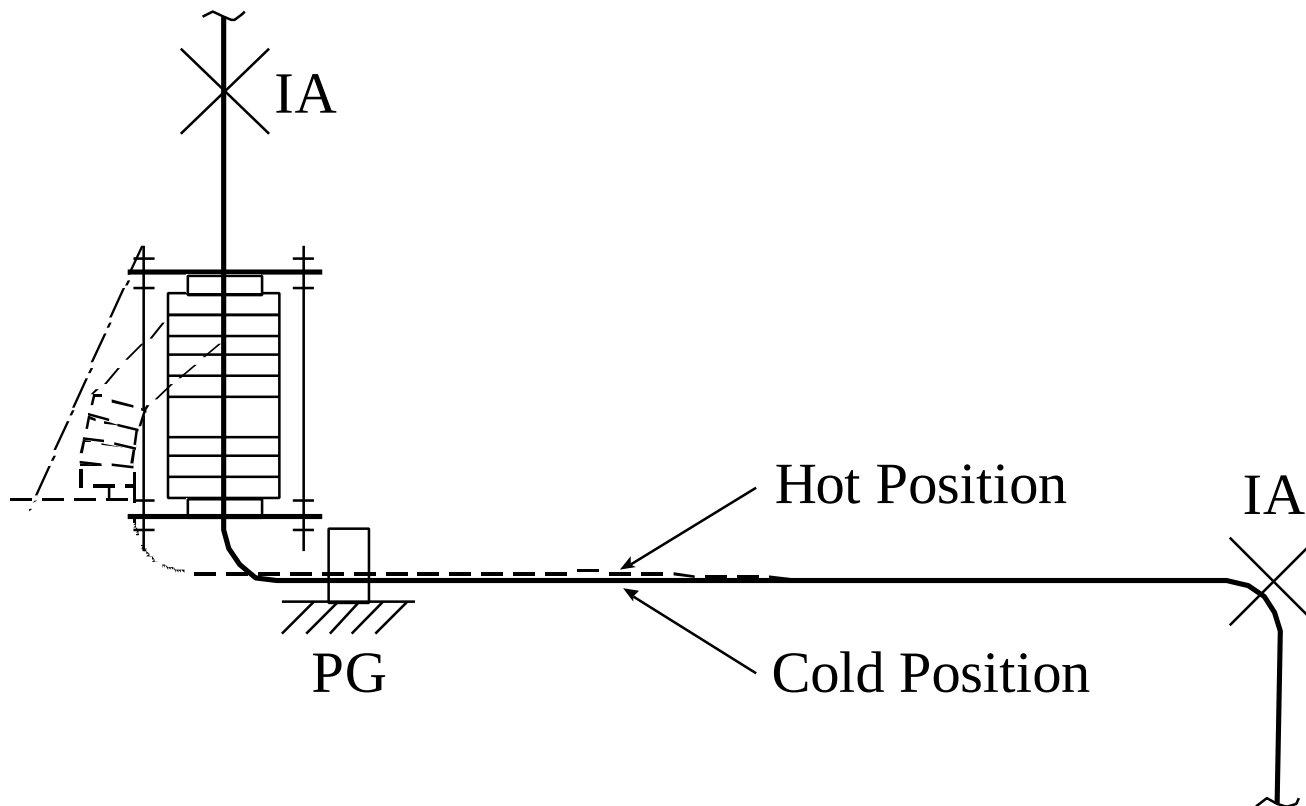
- **Multi-ply bellows (made of several independent layers) allow greater thickness, strength, and flexibility**
 - **Layers can move relative to each other**
 - **Inspection and maintenance is a problem**
 - **Leak detection between layers is required to detect inner layer failure**
 - **Weld joint details are more complex where the bellows joins the pipe**
 - **If one ply fails, the rest are not adequate to withstand the pressure**

Expansion Joint Bellows

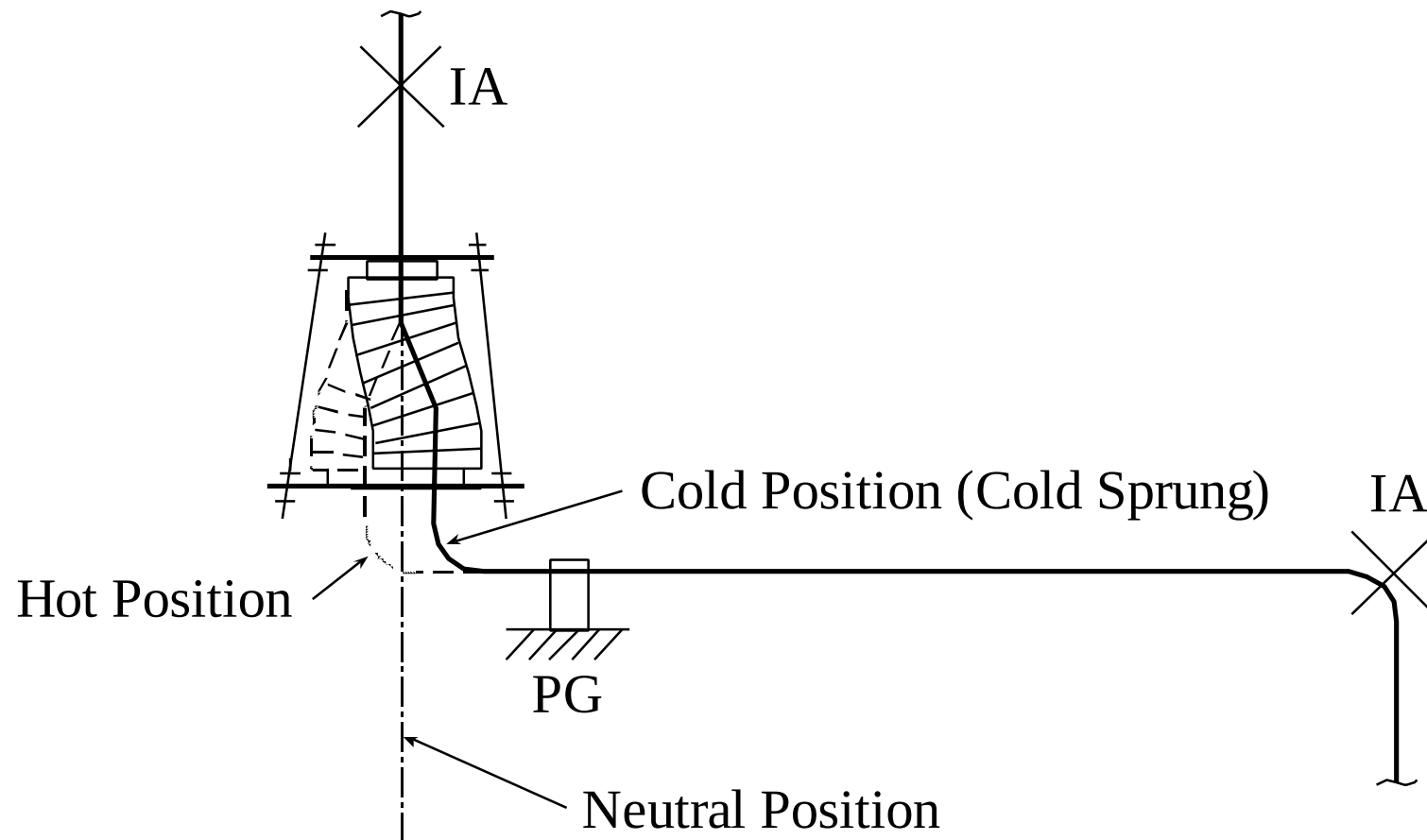
(continued)

- **Maintenance of thin bellows, and protection from damage, is critical**
 - Small amounts of corrosion or damage may be “fatal”
- **Bellows are generally purged to avoid contaminant accumulation**
 - Purge must not result in liquid water formation or accumulation which can result in very corrosive materials (e.g. polytheonic acid)
- **Reduce the net offset movement the bellows or universal joint must take by using an initial offset**

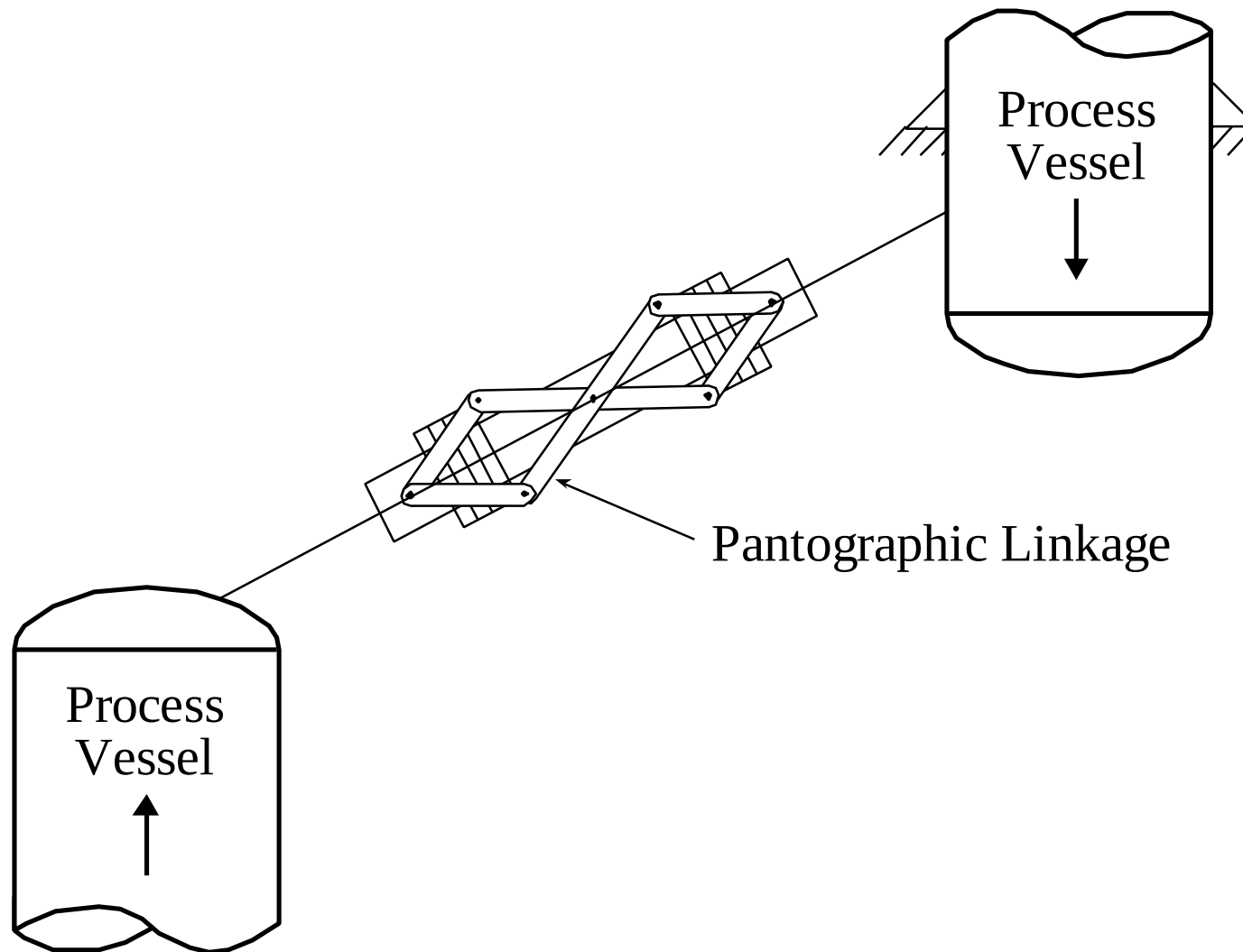
Universal Expansion Joint *(without Initial Offset)*



Universal Expansion Joint (with Initial Offset)

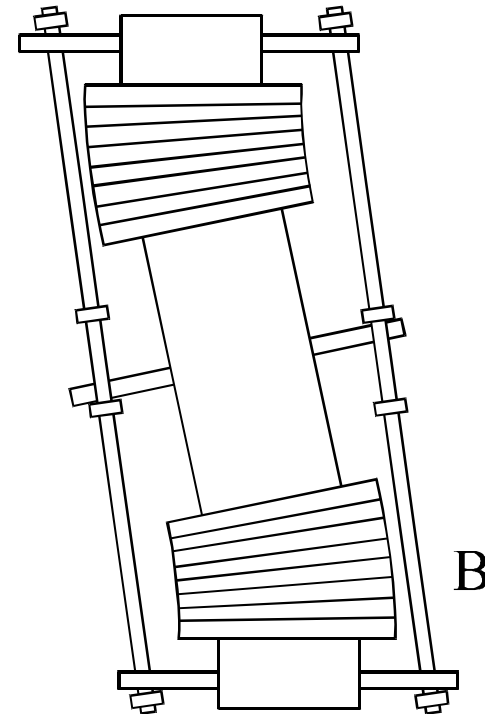
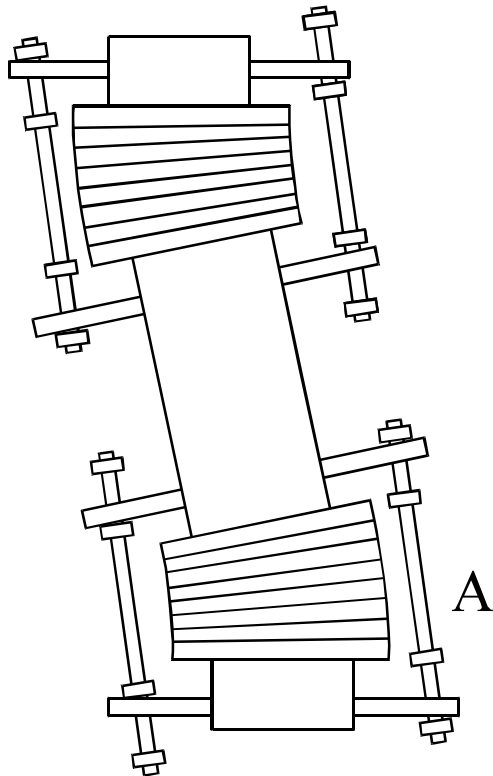


Pantographic Linkage

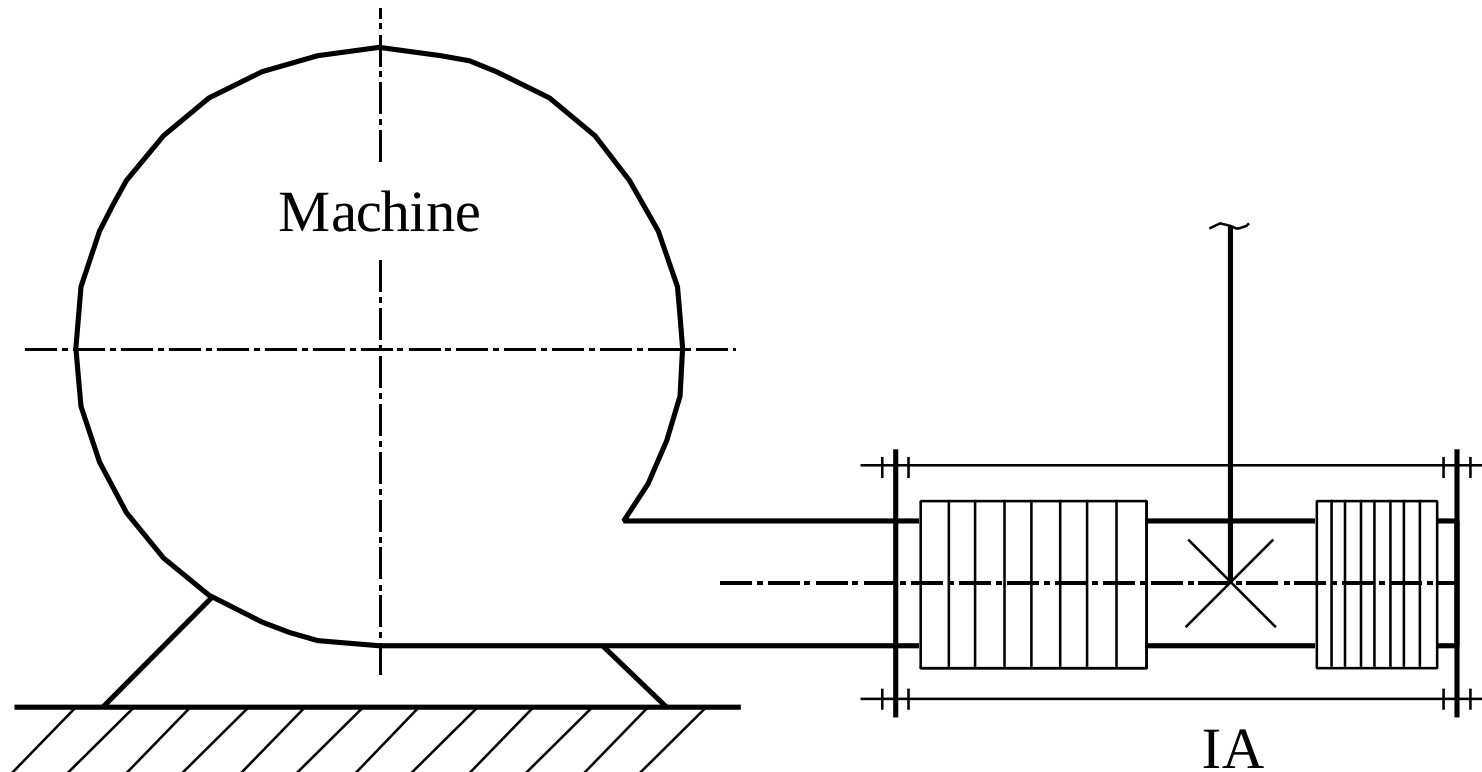




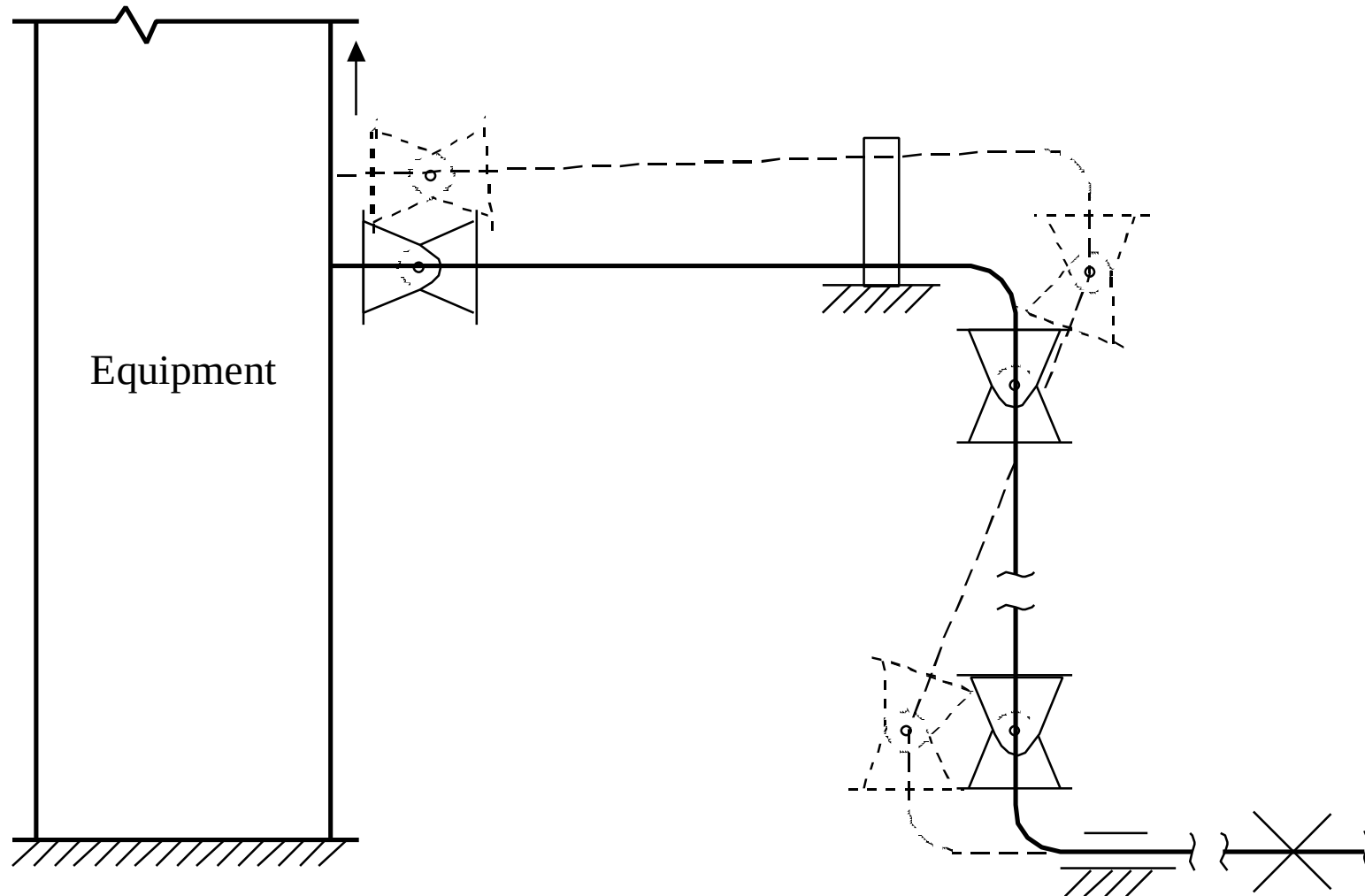
Tied Expansion Joint



Pressure Balanced Expansion Joint



Hinged Expansion Joint



Elastic Followup

- **Elastic followup is a phenomenon that may occur in “unbalanced” piping systems**
 - **Results in overstress or a creep or fatigue failure after a (possibly lengthy) period of satisfactory service**
 - **Normal stress/flexibility analysis will not indicate a problem**
 - **Makes detection or prediction more difficult**
 - **Stresses cannot be considered to be proportional to strains throughout the system**

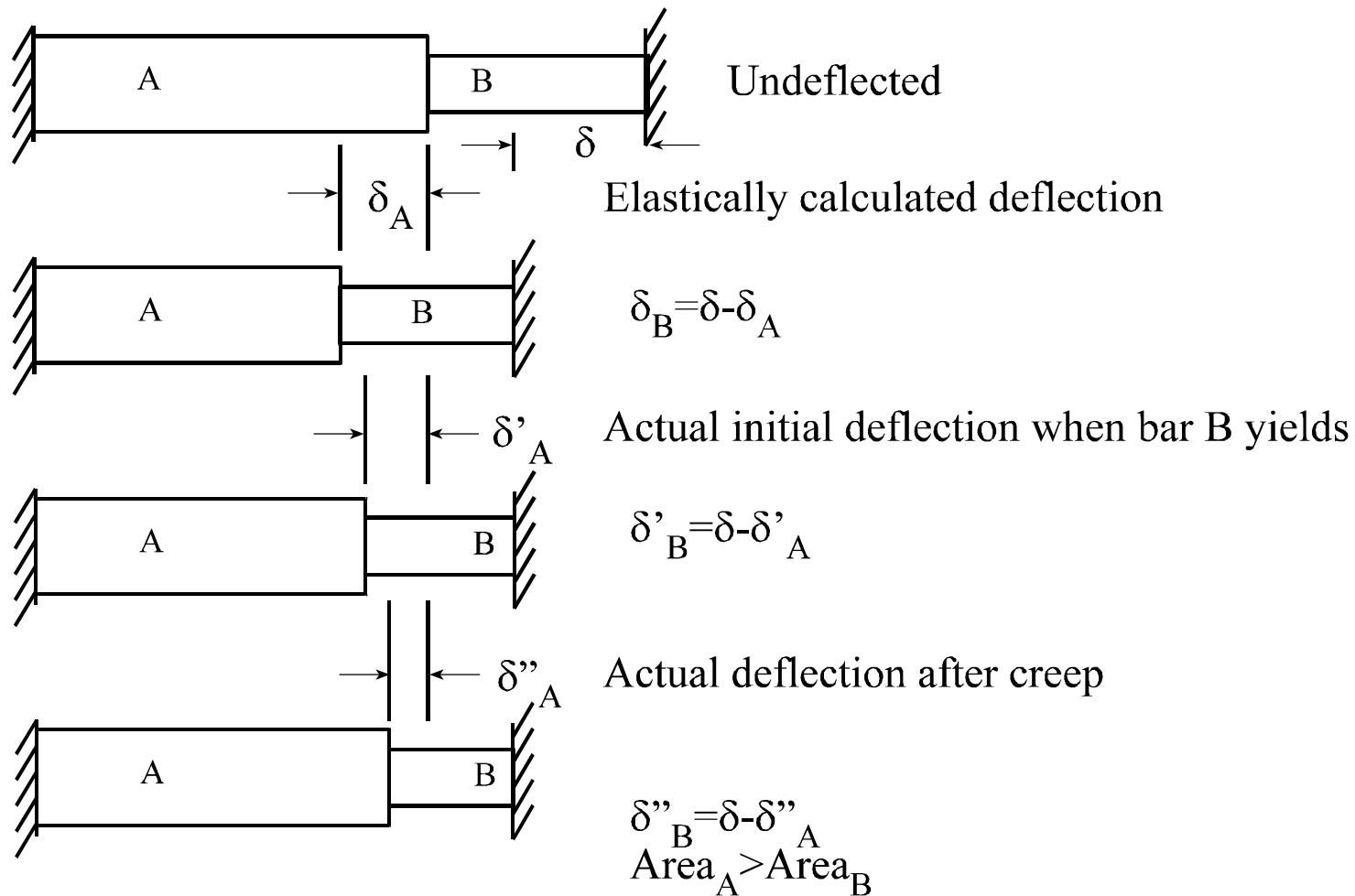
Elastic Follow-up (continued)

- **For elastic follow-up to occur, two conditions must be present**
 - **System, or a portion of it, must operate within the creep range of piping materials.**
 - **System must be unbalanced, i.e., some areas must be hotter, more highly stressed, or less stiff than others. These are the relatively weak areas.**

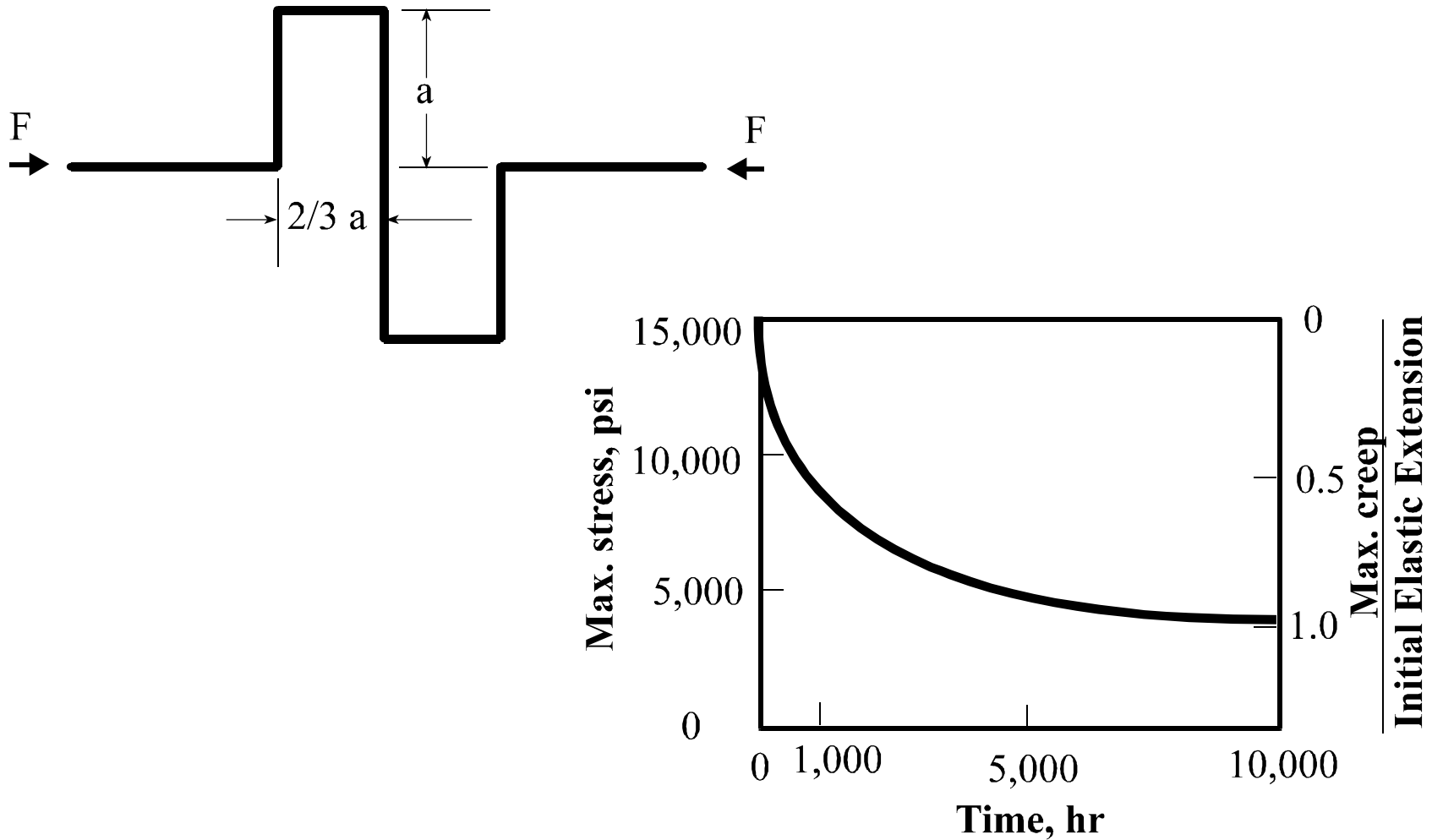
Elastic Followup (continued)

- **Over time, the more highly stressed, weaker, or hotter portions tend to creep more than remainder of system**
- **Strain in the system is concentrated in these locations (which may be local)**
 - **If weak areas perform elastically, increase in strain means increase in stress**
 - **Areas are more highly stressed (remainder of system less stressed) than predicted**
 - **If weak areas perform inelastically, they may develop creep (or fatigue) damage, even if increased strain results in little or no stress increase**

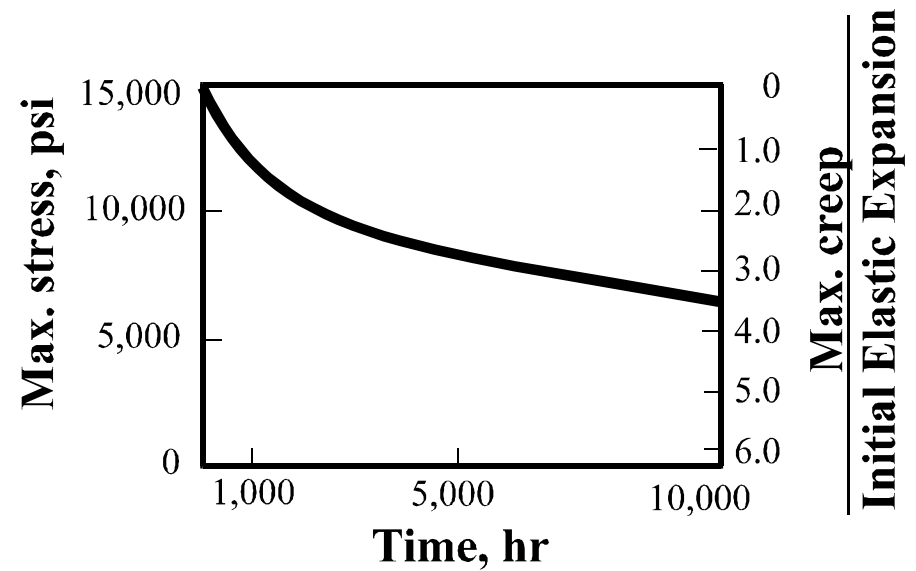
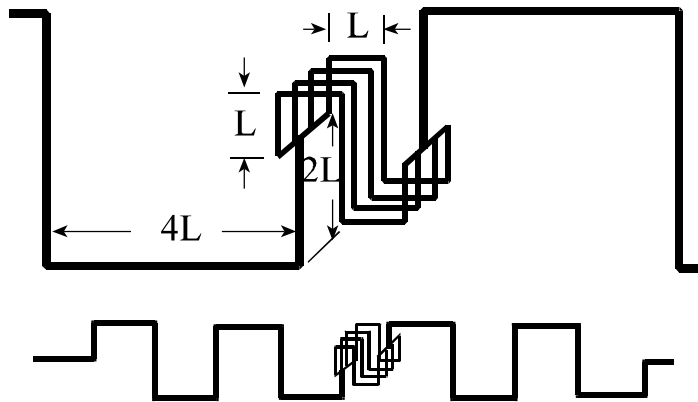
Elastic Followup Examples



Elastic Followup Examples



Elastic Followup Examples



Elastic Followup

(continued)

- **Normal flexibility/stress analysis is indicative of initial, pre-creep condition and is based upon stress, not strain, status**
 - **Strain redistributions due to creep are not accounted for**
 - **Damage, or failure, may occur**

Elastic Followup

(continued)

- **May be prevented by:**
 - **Avoiding use of unbalanced systems**
 - **Lowering long term operating temperatures below the creep range**
 - **Use of materials with higher creep range threshold temperatures**

Elastic Follow-up (continued)

- **May be prevented by (continued)**
 - **Design for overall low piping strains that remain low even with redistribution**
 - **Creative support/restraint systems to prevent strain redistribution, including the selective use of cold spring**
 - **Care must be taken not to impose additional stresses into the piping system**

Piping Support

- **Pipe supports must carry the weight of the piping, with contents, and account for thermal movements and loads**
- **As a general rule, minimizing restrictions to the piping's thermal movement will minimize pipe stresses and reactions**
 - **Means fewer guides, anchors and other restrictions is better**
 - **In some cases, problems can be resolved by removing restraints**

Piping Support

(continued)

- **Avoid placing portions of the piping system in compression**
 - **Buckling may occur**
 - **For example, support vertical runs of pipe from the top rather than the bottom**

- **Piping identified as “Free Draining” must gravity flow in the indicated direction during all operating, start-up, shut-down, out-of service, and other conditions.**

Piping Support *(continued)*

- **Often, load limitations on attached equipment (eg, pumps), locally high stress, or limits on the magnitude of the movement that can be permitted require that thermal movements be restricted via guides, anchors, etc**
- **Restricting thermal movements will result in imposition of a force on the resisting support and cause stresses within piping**

Piping Support

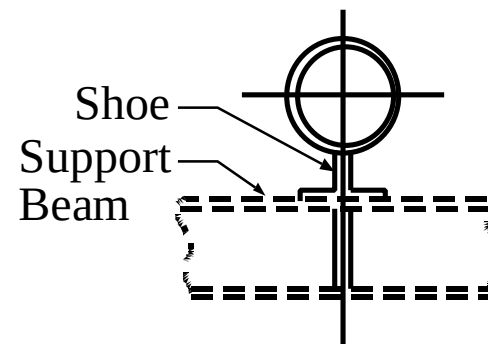
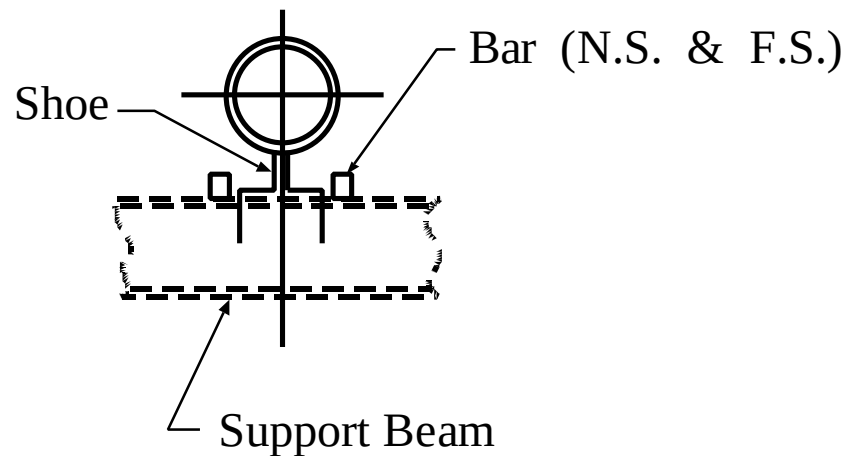
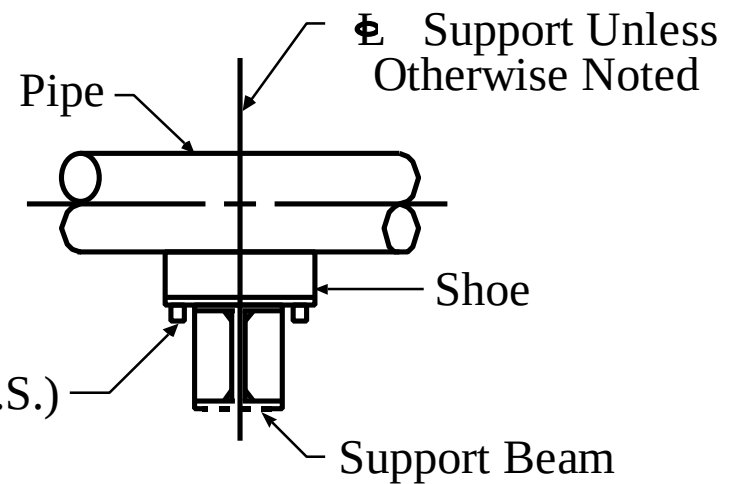
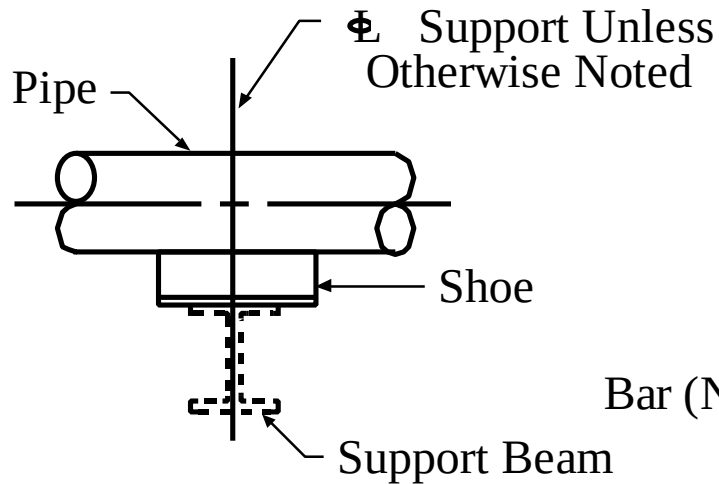
(continued)

- **There are many ways to provide support**
- **Pipe “shoes” beneath the piping**
 - **Shoes allow axial and, if permitted, out of plane movement; may even lift off the support during operation**
 - **Guides, possibly with a clamp over the shoe, may be used to limit or restrict movements**
 - **Care must be taken to insure shoe cannot slide off support and “bind” at any time**

Piping Support (continued)

- **Provide low friction slide plates (Teflon or graphite) to minimize the effects of friction where movement is required**
- **Limit stops allow the pipe to move a certain amount, then halt further movement**

Pipe Shoe Details

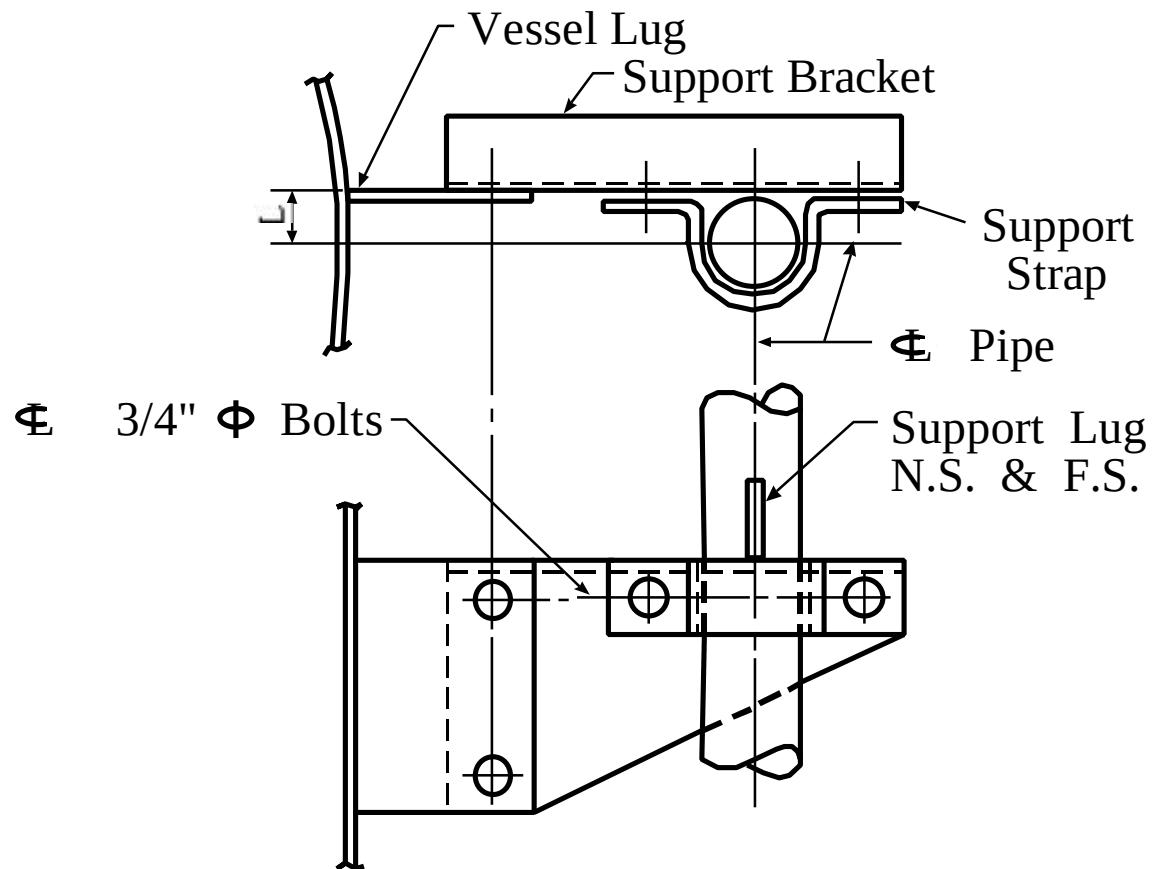


Piping Support

(continued)

- **Hangers provide a rigid vertical support, while allowing horizontal movement**
- **Anchors are any system that does not allow any piping movement at that point**
- **Piping supports must be provided to prevent excessive piping deformation from piping's weight (sag)**
 - **Supports must also allow retention of any free draining requirements, without creating pockets in piping system**

Pipe Support Assembly



Stress (psi) Due to Sag

Standard Wall Pipe—Filled With Water

Pipe Size Inches	SPAN BETWEEN SUPPORTS, FEET													Pounds of Water per Lineal Foot
	10	12	14	16	18	20	24	30	36	42	48	54	60	
$\frac{1}{2}$	2387	3438	4680	6113	7736	9551								.13164
1	1531	2205	3002	3921	4962	6126	8822	13785						.37345
1 $\frac{1}{2}$	1342	1932	2630	3436	4349	5369	7731	12081						.88260
2	903	1301	1771	2313	2927	3614	5205	8133	11711	15941				1.4541
3	620	892	1215	1587	2008	2480	3571	5580	8035	10936	14284			3.0032
4	503	725	986	1288	1631	2013	2900	4531	6525	8881	11600	14681		5.5172
5	424	610	831	1085	1373	1696	2442	3816	5495	7480	9769	12365	15265	8.6666
6	368	530	721	942	1192	1472	2120	3313	4771	6495	8483	10736	13255	12.530
8	319	459	625	816	1033	1276	1837	2871	4135	5628	7351	9304	11487	22.206
10	283	407	554	724	917	1132	1630	2547	3668	4993	6522	8254	10191	35.454
12	238	343	467	610	772	954	1374	2146	3091	4207	5496	6955	8587	49.760
14	213	308	419	547	701	855	1232	1925	2772	3774	4929	6239	7702	60.000
16	197	284	387	506	640	790	1138	1778	2561	3486	4554	5763	7115	79.187
18	181	261	355	464	588	726	1045	1633	2353	3201	4182	5292	6534	100.48
20	173	250	340	444	562	694	1000	1563	2251	3064	4003	5066	6254	125.30

Stress (psi) Due to Sag Standard Wall Pipe—Empty

Pipe Size Inches	SPAN BETWEEN SUPPORTS, FEET												
	10	12	14	16	18	20	24	30	36	42	48	54	60
$\frac{1}{2}$	2063	2971	4044	5282	6685	8254	11886						
1	1249	1799	2449	3199	4049	4998	7198	11247					
1 $\frac{1}{2}$	1010	1454	1979	2586	3273	4040	5818	9091	13092				
2	644	977	1262	1649	2087	2577	3711	5798	8350	11365	14844		
3	434	625	851	1111	1406	1736	2500	3907	5627	7659	10003	12661	15631
4	331	477	650	849	1075	1327	1911	2986	4390	5853	7644	9675	11945
5	265	381	519	678	858	1060	1526	2385	3435	4675	6106	7728	9541
6	220	317	432	656	667	823	1186	1853	2669	3893	5085	6436	7945
8	167	240	327	427	541	668	962	1503	2165	2998	3915	4956	6118
10	131	189	258	337	426	526	758	1185	1706	2323	3034	3840	4741
12	110	159	217	283	359	443	638	997	1436	1955	2554	3232	3990
14	101	145	198	259	332	405	583	911	1312	1786	2333	2953	3646
16	88	127	172	225	285	352	508	793	1143	1555	2032	2571	3175
18	78	112	153	200	253	313	450	704	1014	1380	1802	2281	2817
20	70	100	137	179	226	230	403	630	907	1235	1613	2041	2520

Maximum Pipe Spans (Feet)

PIPE SIZE PIPE SCHED.	1"	1½"	2"	3"	4"	6"	8"	10"	12"	14"	16"	18"	20"	24"
SCHED. 10	-	-	-	14	15	18	20	23	25	30	31	31	32	34
SCHED. 20	-	-	-	-	-	-	27	28	29	33	34	35	39	40
SCHED. 30	-	-	-	-	-	-	28	30	35	36	37	40	43	47
STANDARD	8*	11*	13*	19	21	26	29	33	35	36	37	38	39	40
SCHED. 40	8*	11*	13*	19	21	26	29	33	36	38	41	44	46	50
SCHED. 60	-	-	-	-	-	-	32	36	39	41	44	47	50	55
X-STG.	8*	11*	14*	20*	24	29	33	36	38	39	41	42	43	45
SCHED. 80	8*	11*	14*	20*	24	29	33	38	41	44	47	50	53	58
SCHED. 100	-	-	-	-	-	-	35	39*	43*	45*	48*	51*	54*	59*
SCHED. 120	-	-	-	-	25	31*	35*	40*	43*	45*	49*	52*	55*	60*
SCHED. 140	-	-	-	-	-	-	36*	40*	44*	46*	49*	52*	55*	61*
SCHED. 160	9*	12*	15*	20*	25*	31*	36*	40*	44*	46*	49*	52*	55*	61*

Basis: simple span; carbon steel pipe; water or lighter fluid service;
maximum temperature 650 °F

Criteria: maximum stress = 6000psi

***Maximum deflection = smaller of 1" or nominal pipe diameter**

Piping Vibration

- **Many potential causes of vibration**
 - **Connected equipment**
 - **Pumps**
 - **Compressors**
 - **Flow characteristics**
 - **Flashing**
 - **Intermittent flow**
 - **Two-phase flow**
 - **Water hammer**
 - **Etc.**

Vibration Control

- **Design piping system and its supports to have a critical frequency well away from the exciting frequency**
- **Isolate vibrating equipment from the piping system**
- **Maintain uniform process temperature and minimize piping length and elbows to control vibration caused by flow characteristics**

Vibration Control

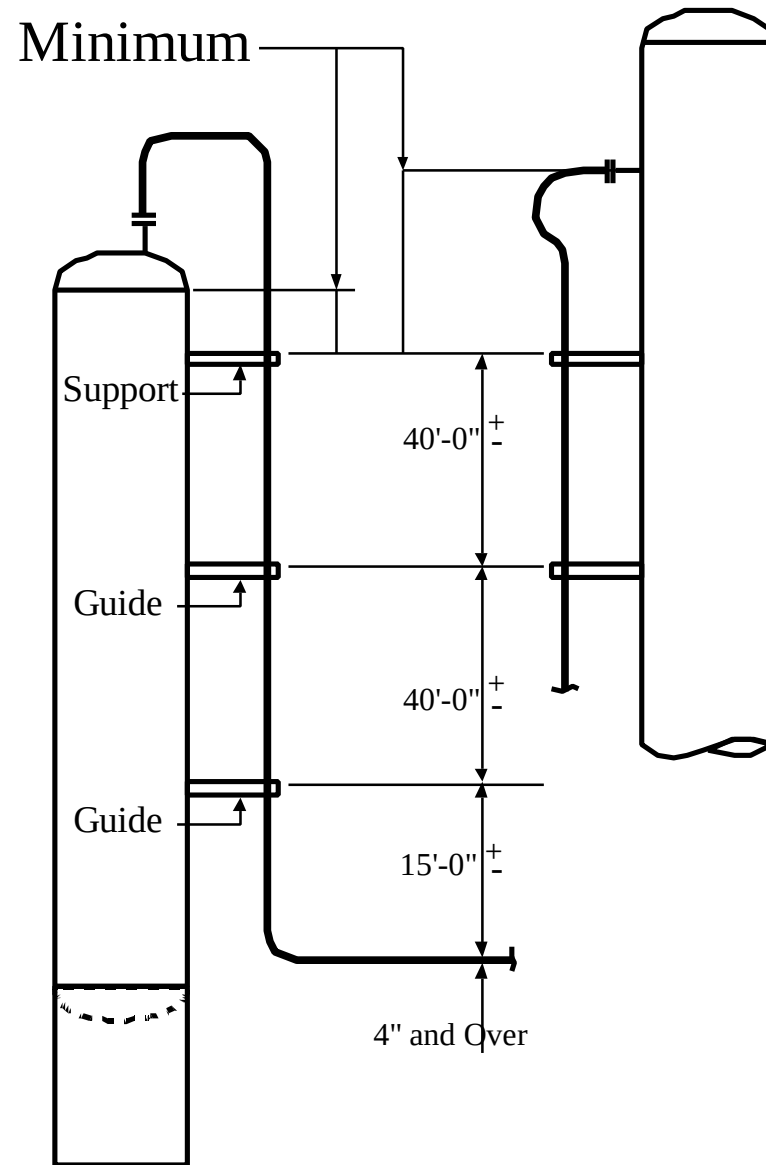
(continued)

- **Use snubbers or dampers to slow the systems response to excitation, while retaining flexibility**
- **Use guides to limit amplitude of vibration**
- **“Dead leg” tees rather than elbows may help reduce vibration**

“Floating” Systems

- **A “floating” system has either no intermediate supports or supports capable of absorbing only a fixed load (e.g., a constant spring support)**
- **Unless all weights and loading conditions are known, “floating” systems are avoided because any unanticipated or misestimated load can be absorbed only at the ends**
- **Load transmission through the system to the ends creates additional strains and stresses**

Location of Pipe Supports and Guides



Spring Hangers

- **Spring hangers are systems designed to provide support while allowing vertical movement**

- **Two kinds are available**
 - **Variable**
 - **Less expensive**
 - **Constant**

Spring Hangers

(continued)

- **Variable spring hangers provide a supporting force that varies as a function of deflection**
 - **Force is governed by the spring constant of a simple spring**
- **Allow for overtravel (greater than expected movement) when designing spring hangers**

Spring Hangers

(continued)

- **Support force will vary due to thermal movements**
- **Forces greater or less than those required to support the pipe's weight must be absorbed elsewhere in system**
 - **Results in stress changes in the piping and load changes at other supports**

Spring Hangers

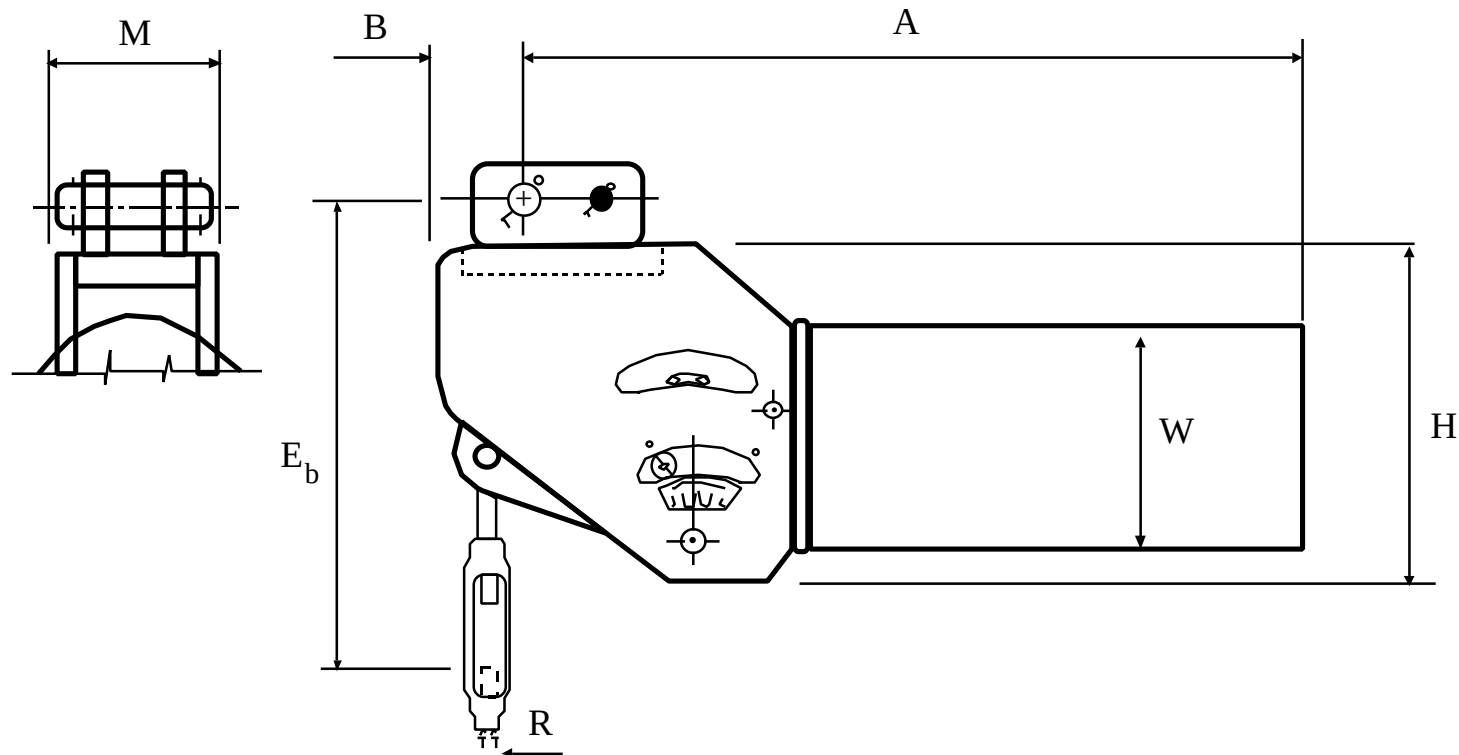
(continued)

- **Generally, variable hangers are commonly used if movements are less than 2 inches**
- **Constant spring hangers provide constant force throughout their range of motion**
 - **Constant supports are used for movements greater than 2 inches**
 - **Constant supports may also be advisable if additional loads cannot be imposed upon other supports (e.g. pump casings)**

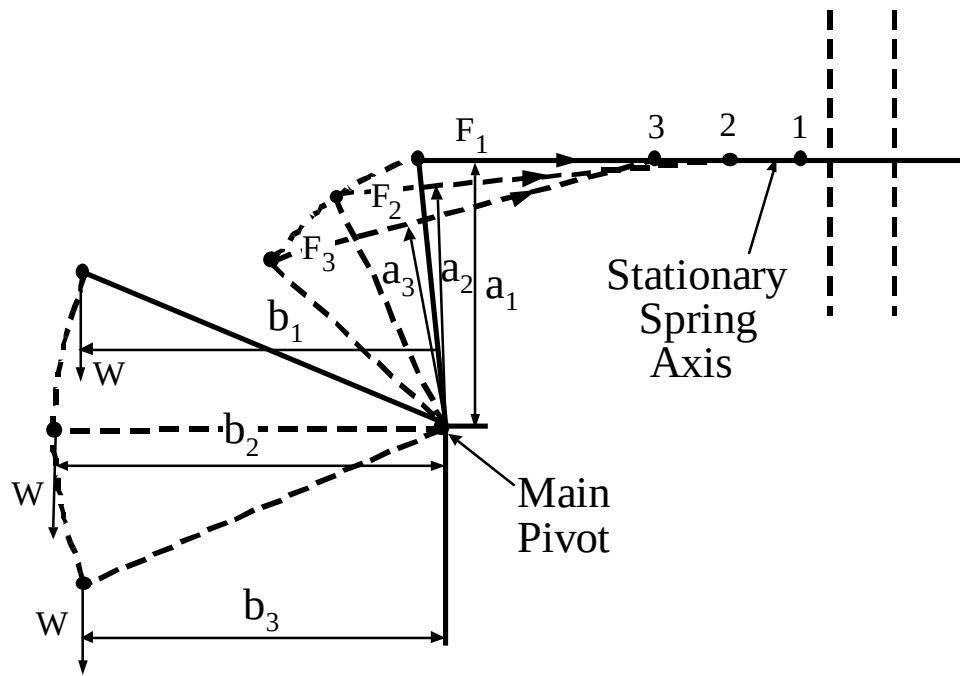
Variable Support Size Selection Table

Travel (Inches)					HANGER SIZE												
					1	2	3	4	5	6	7	8	9	10	11	12	
3600	3400	3200	3100	Economic Range	44	59	79	104	139	185	245	326	434	577	767	1020	
¾" Increments	½" Increments	¼" Increments	⅛" Increments		46	62	82	109	145	193	256	341	453	602	801	1065	
					48	64	85	113	151	200	266	354	471	626	832	1107	
					50	67	89	118	157	208	277	369	490	652	866	1152	
					52	69	92	123	163	217	288	383	509	677	900	1197	
					54	72	96	127	169	225	299	398	529	703	935	1243	
					56	75	99	132	175	233	310	412	548	729	969	1288	
					58	77	103	137	181	241	321	427	567	754	1003	1333	
					60	80	106	141	188	250	332	441	587	780	1037	1379	
2¼	1½	¾	⅝		62	82	110	146	194	258	343	456	606	805	1071	1424	
				64	85	113	150	200	266	354	470	625	831	1105	1469		
				66	88	117	155	206	274	365	485	644	857	1139	1515		
				68	90	120	160	212	282	375	499	664	882	1173	1560		
				70	93	124	164	219	291	386	514	683	908	1207	1605		
				72	96	127	169	225	299	397	528	702	934	1241	1651		
				74	98	131	174	231	307	408	543	722	959	1276	1696		
				76	101	134	178	237	315	419	557	741	985	1310	1741		
4½	3	1½	¾	78	103	138	183	243	323	430	572	760	1011	1344	1787		
				80	106	141	188	249	332	441	586	779	1036	1378	1832		
				82	109	145	192	256	340	452	601	799	1062	1412	1877		
				84	111	148	197	262	348	463	615	818	1088	1446	1923		
				86	114	152	201	268	356	474	630	837	1113	1480	1968		
				88	117	155	206	274	364	485	644	857	1139	1514	2013		
				90	119	159	211	280	373	495	659	876	1164	1548	2059		
				92	122	162	215	286	381	506	673	895	1190	1582	2104		
6¾	4½	2¼	1½	94	124	166	220	293	389	517	688	914	1216	1617	2149		
				96	127	169	225	299	397	528	702	934	1241	1651	2195		
				98	130	172	229	305	405	539	717	953	1267	1685	2240		
				100	132	176	234	311	414	550	731	972	1293	1719	2285		
				102	135	179	239	317	422	561	746	992	1318	1753	2331		
				103	138	183	243	323	430	572	760	1011	1344	1787	2376		
				Spring Rate — lbs. per inch of deflection													
				31.6	42	55.8	74.2	98.8	131	175	232	308	410	546	726		
15.8	21	27.9	37.1	49.4	65.5	87.5	116	154	205	273	363						
7.9	10.5	14	18.6	24.7	32.8	43.8	58	77	103	137	182						
5.3	7	9.3	12.4	16.5	21.8	29.2	38.7	51.3	68.3	91	121						

Constant Supports



Constant Supports (continued)



- Referring to the diagram to the left, take three positions high, mid and low, and equate the moments about the main pivot

$$F_1 a_1 = W b_1$$

$$F_2 a_2 = W b_2$$

$$F_3 a_3 = W b_3$$

$$\frac{F_1 a_1}{b_1} = \frac{F_2 a_2}{b_2} = \frac{F_3 a_3}{b_3} = W$$

Constant Support Size Selection Table

HGR. SIZE	TOTAL TRAVEL (In Inches) MAXIMUM RATED LOAD IN POUNDS FOR EACH SIZE															
	2	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8	8½	9	9½
4000																
4100																
4200																
1	90 130	75 100	60 85	55 70	45 65	40 55	35 50	35 45	30 40		25 35		20 30		20 30	
2	175	140	115	100	85	80	70	65	60		50		45		40	
3	235	190	160	135	120	105	95	85	80		70		60		55	
4	325	260	215	185	165	145	130	120	110		95		80		70	
5	445	360	295	255	225	200	180	160	150		130		110		100	
6	600	485	405	345	305	270	245	220	205		175		150		135	
7	755	605	505	435	380	335	305	275	255	235	215	205	190	180	170	160
8	905	725	605	520	455	405	365	330	305	280	260	245	230	215	205	190
9	1090	875	725	625	545	485	435	395	365	335	310	290	275	255	245	230
10	1310	1045	875	750	655	585	525	475	440	405	375	350	330	310	290	275
11	1465	1175	980	840	735	650	585	535	490	450	420	390	365	345	325	310
12	1620	1295	1080	925	810	720	650	590	540	500	465	435	405	385	360	345
13	1810	1450	1210	1035	905	805	725	660	605	560	520	485	455	430	405	385
14	2000	1605	1335	1145	1005	890	805	730	670	620	575	535	500	475	445	425
15	2240	1790	1495	1280	1120	995	900	815	750	690	640	600	560	530	500	475
16	2475	1980	1655	1415	1240	1100	990	900	825	765	710	660	620	585	550	525
17	2770	2215	1850	1585	1385	1230	1110	1010	925	855	790	740	695	655	615	585
18	3060	2445	2045	1750	1530	1365	1225	1115	1020	945	875	815	765	720	685	650
19	3400	2720	2265	1940	1670	1510	1360	1240	1135	1050	975	905	850	800	755	720
20	3735	2990	2490	2130	1870	1660	1495	1360	1245	1150	1070	995	935	880	830	790
21	4120	3295	2745	2350	2060	1830	1650	1500	1375	1270	1180	1100	1030	970	915	870
22	4500	3600	3000	2575	2250	2000	1800	1640	1500	1385	1285	1200	1125	1060	1000	950
23	4950	3960	3300	2830	2475	2200	1980	1800	1650	1525	1415	1320	1240	1165	1100	1045
24	5400	4320	3600	3085	2700	2400	2160	1965	1800	1665	1545	1440	1350	1270	1200	1140
25	5940	4755	3960	3395	2970	2640	2375	2160	1980	1830	1700	1585	1485	1400	1320	1255
26	6480	5185	4320	3710	3240	2880	2595	2355	2160	1995	1855	1730	1620	1525	1440	1370
27		5710	4755	4075	3565	3170	2855	2595	2380	2195	2040	1905	1785	1680	1585	1505
28		6230	5190	4445	3890	3460	3115	2830	2595	2395	2225	2075	1945	1830	1730	1645
29		6845	5705	4890	4280	3805	3425	3110	2855	2635	2445	2285	2140	2015	1905	1805
30		7645	6380	5465	4780	4250	3825	3475	3190	2945	2735	2555	2395	2250	2125	2020
31		8660	7215	6180	5405	4810	4330	3935	3600	3330	3090	2885	2705	2545	2405	2285
32		9480	7895	6760	5915	5265	4740	4305	3945	3640	3380	3150	2960	2780	2630	2500
33		10515	8770	7510	6575	5845	5260	4780	4380	4045	3750	3500	3285	3095	2920	2775

Spring Hangers

(continued)

- **Limited horizontal movement may be accommodated**
 - **Amount depends upon length of the support rod**
 - **Angle of support rod should be within 4 degrees of vertical (constant supports may accommodate more in the plane of the support)**

Piping Reactions

- **Sometimes the equipment itself is designed to absorb some (or all) of the unrestrained movements present at that point in the piping system. This can serve to reduce the stress within the piping system.**
- **Reactions at equipment consider weight only, sustained load, and weight plus thermal load cases**
 - **Pressure, friction forces, etc., are considered when conservative to do so**
- **Loads imposed upon vessels are normally not a concern**
 - **For large loads, specialized methods exist for determining and evaluating the imposed stresses and deflections**

Piping Reactions

(continued)

- **Of more concern is overstressing or rotating flanges, possibly causing leakage**
- **Attachments to rotating equipment are of particular concern**
 - **Casings have a very restricted load carrying capacity**
 - **“Excessive” loads may distort the case**
 - **Even very small distortions may affect operation of equipment or cause contact between rotor and case**

Piping Reactions

(continued)

- **Permissible casing loads are provided by individual vendors**
- **Loads are expressed as permissible component forces and moments at inlet and outlet or are resolved to resultant loads relative to the shaft**

Piping Reactions

(continued)

- **NEMA SM 23 provides default loads for turbines**
- **API 610, “Centrifugal Pumps for General Refinery Services” provides default loads for pumps**
- **API 617, “Centrifugal Compressors for General Refinery Services” permits 1.85 times the loads permitted by NEMA SM 23**

Nozzle Loadings

(U.S. Units)

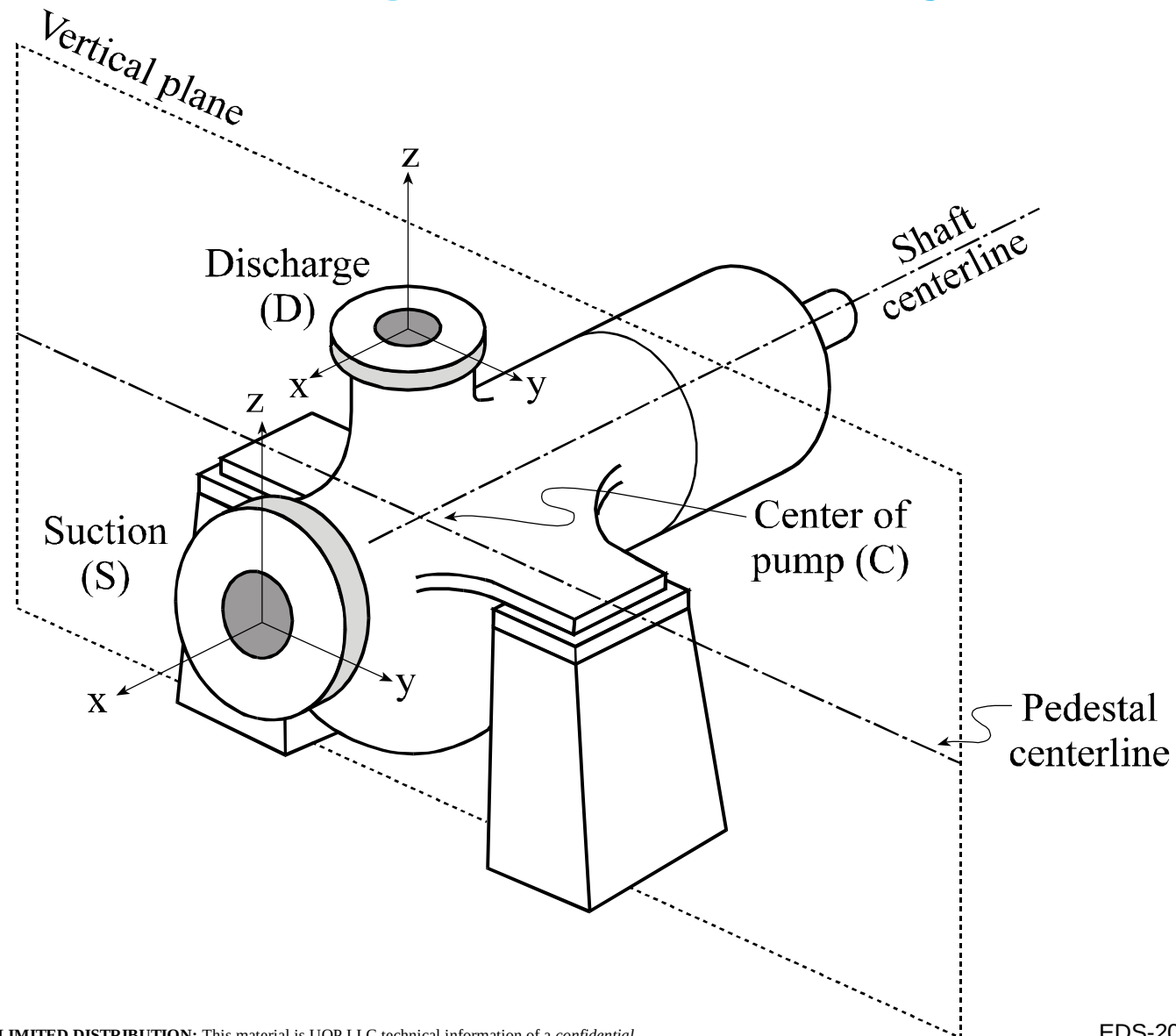
Note: Each value shown below indicates a range from minus that value to plus that value; for example 160 indicates a range from -160 to +160.

Force/Moment	Nominal Size of Flange (NPS)								
	2	3	4	6	8	10	12	14	16
Each Top Nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	130	200	260	460	700	1000	1200	1300	1500
FZ	200	300	400	700	1100	1500	1800	2000	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each Side Nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	200	300	400	700	1100	1500	1800	2000	2300
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each End Nozzle									
FX	200	300	400	700	1100	1500	1800	2000	2300
FY	160	240	320	560	850	1200	1500	1600	1900
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each Nozzle									
MX	340	700	980	1700	2600	3700	4500	4700	5400
MY	170	350	500	870	1300	1800	2200	2300	2700
MZ	260	530	740	1300	1900	2800	3400	3500	4000
MR	460	950	1330	2310	3500	5000	6100	6300	7200

Note 1: F = force in pounds; M = movement in foot-pounds; R = resultant. See Figures 2-2 – 2-6 for orientation of nozzle loads (X, Y, and Z).

Note 2: Coordinate system has been changed from API Standard 610, 7th Edition, convention to ISO 1503 convention.

Nozzle Loadings Coordinate System



Permissible Loads for Steam Turbines per NEMA SM 23

A) At any connection:

$$3F_R + M_R \leq 500 D_e$$

F_R = Resultant force (lbs)

M_R = Resultant moment (ft-lbs)

**D_e = Nominal connection size
(inches). If greater than 8
inches, use $(16+D)/3$.**

Permissible Loads for Steam Turbines per NEMA SM 23 (continued)

B) Combine resultant at centerline of exhaust

1) $2F_C + M_C \leq 250D_C$

F_C = Resultant force (lbs)

M_C = Resultant moment (ft-lbs)

**D_C = Diameter of the opening with
the same area as the sum of
inlet, extraction, and exhaust.
If greater than 9 inches, use
 $(18+D_C)/3$.**

Permissible Loads for Steam Turbines per NEMA SM 23 (continued)

2) Component Limits

$$F_X \leq 50D_C$$

$$M_X \leq 250D_C$$

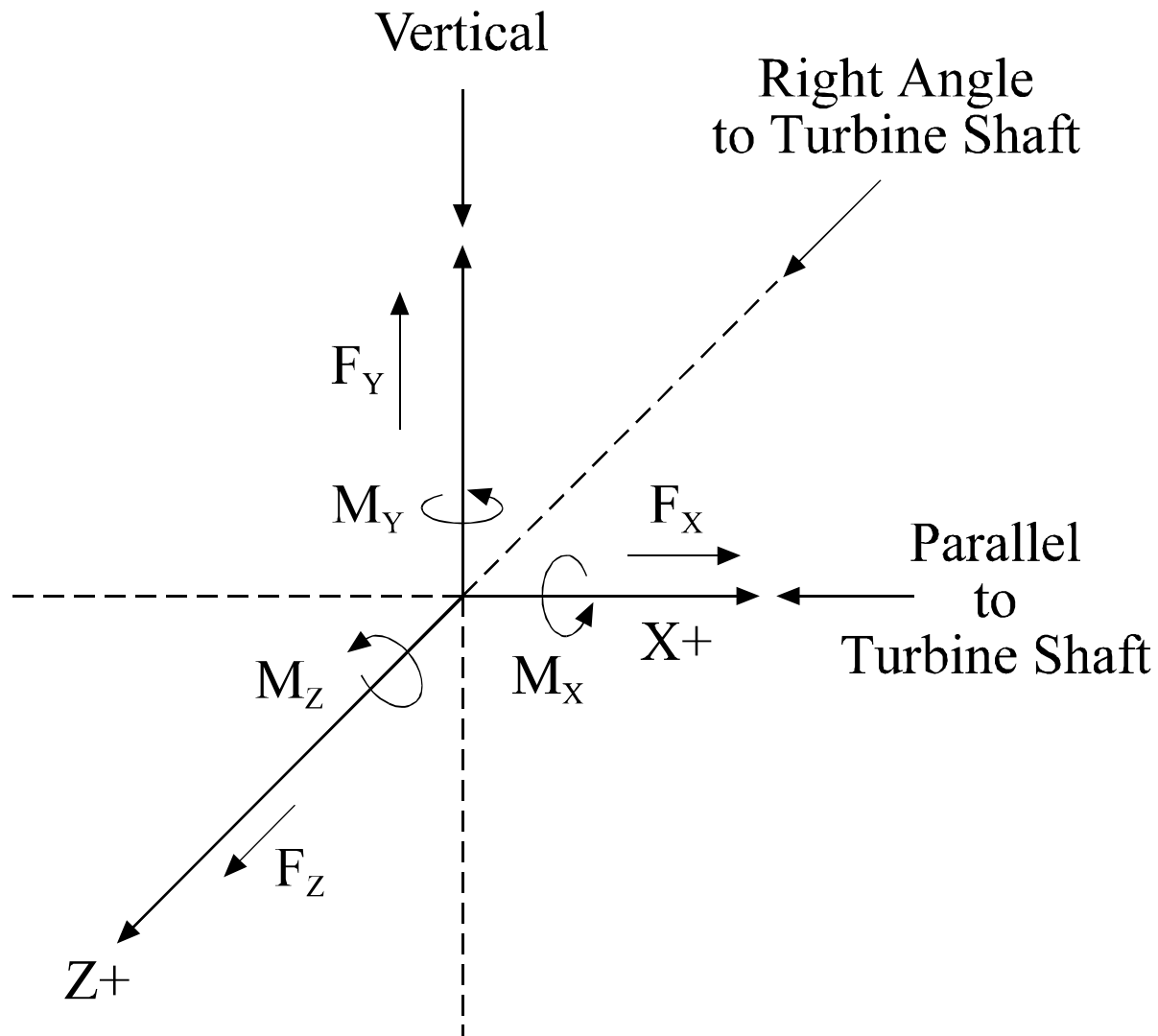
$$F_Y \leq 125D_C$$

$$M_Y \leq 125D_C$$

$$F_Z \leq 100D_C$$

$$M_Z \leq 125D_C$$

Force and Moment Terminology



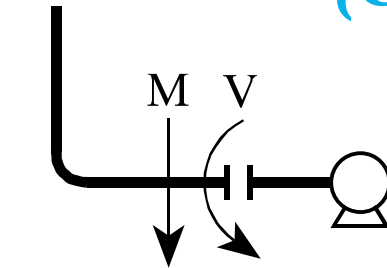
Cold Spring

- **Cold spring is the intentional, cold, deflection of the piping in a direction opposite to the direction of thermal movement, imposing opposite stresses**
- **First part of thermal movement will be through this deflected range**
 - **Results in a smaller net deflection and lower stress magnitudes**
 - **Less strain during initial deflection**
 - **Similar to camber provided in some beams**

Cold Spring (continued)

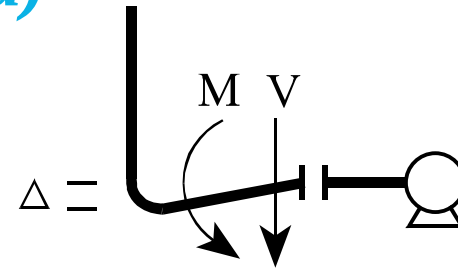
- **Effects of cold spring may only be considered for reaction loads**
 - **Piping stresses are compared to a stress range (maximum - minimum), which is not affected by the cold spring**
 - **Reactions, on the other hand, consider the absolute value of the load**

Cold Spring (Continued)



Initial

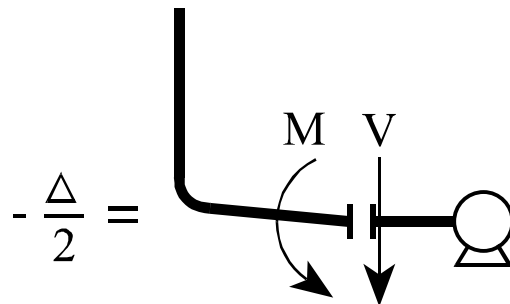
$$M = 0 \quad V = 0$$



Deflected

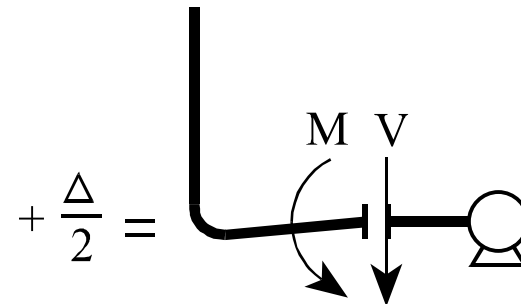
$$M = M \quad V = V$$

Without Cold Spring



Initial

$$M = -\frac{M}{2} \quad V = -\frac{V}{2}$$

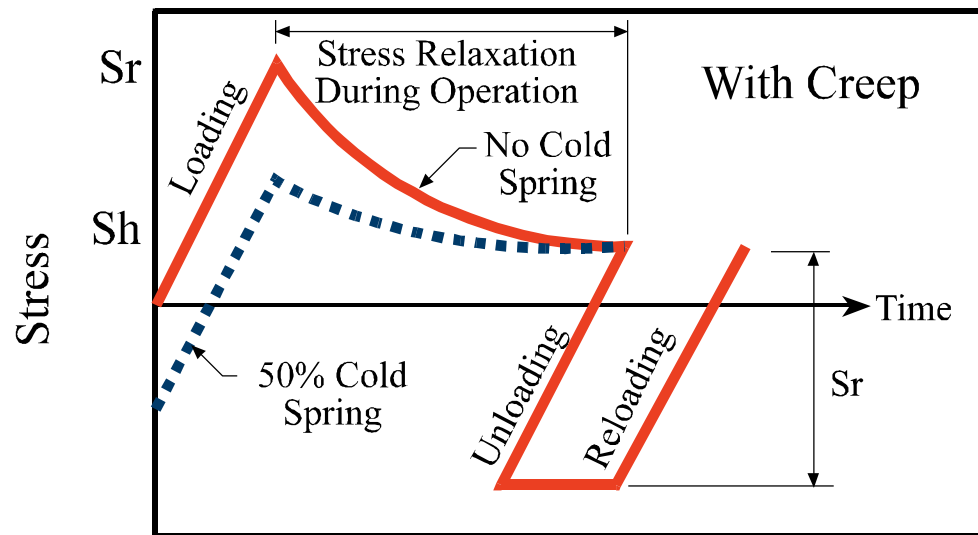
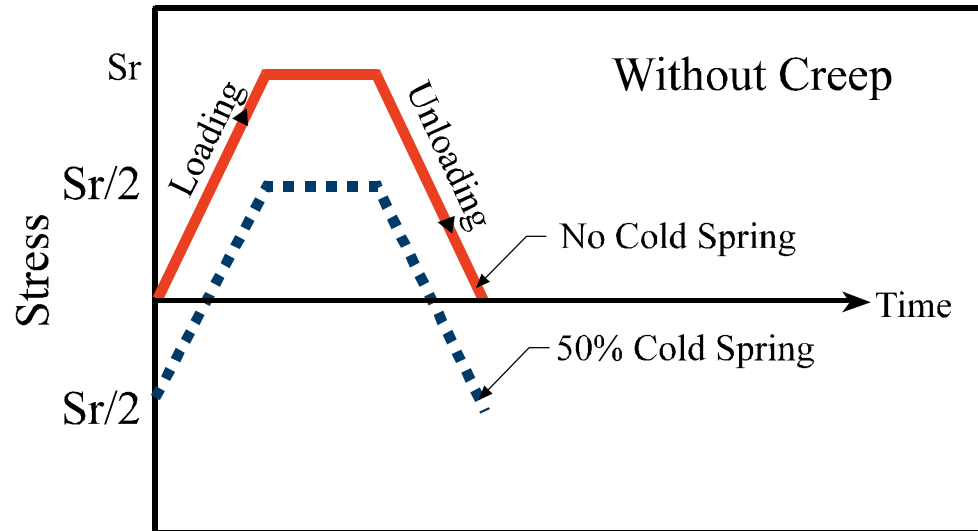


Deflected

$$M = \frac{M}{2} \quad V = \frac{V}{2}$$

With 50% Cold Spring

Cold Spring (continued)



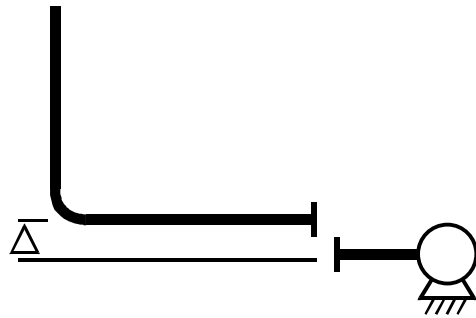
Cold Spring (continued)

- **Because of difficulties in insuring the proper cold spring is actually present, the Code allows consideration of only 2/3 of the intended cold spring**
- **When installing a cold spring, care must be taken to insure that piping still slopes in the intended direction**
- **Ensure that the flanges may still be bolted up without excessive difficulty (cold spring may introduce a cold misalignment)**

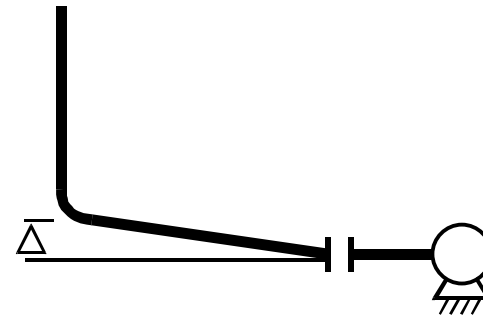
Cold Spring (continued)

- **Deformation must be a stress-strain imposing distortion (e.g. fabricating one leg short and pulling the other leg to mate), not a no stress or strain “off line” fabrication**

Cold Spring Fabrication



As fabricated - will
create cold spring
when assembled (i.e.
bolted at pump)



As fabricated - will
not create cold spring
when assembled (i.e.
bolted at pump)

Flue Gas Line Isometric

