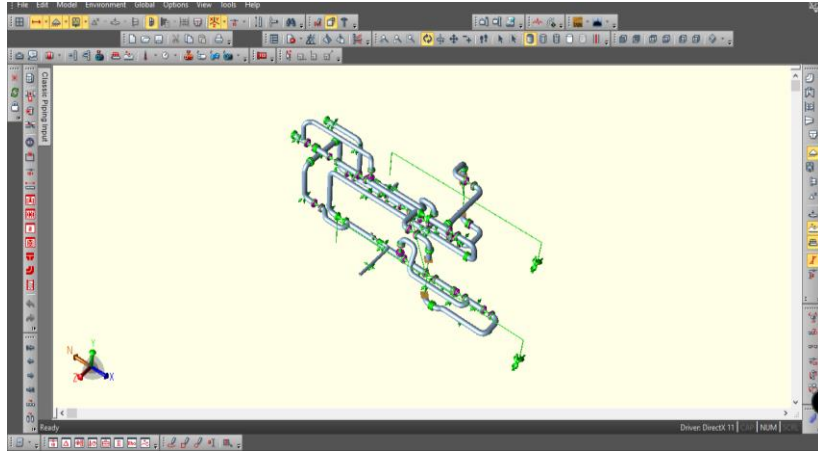


Basics of Pipe Stress Analysis

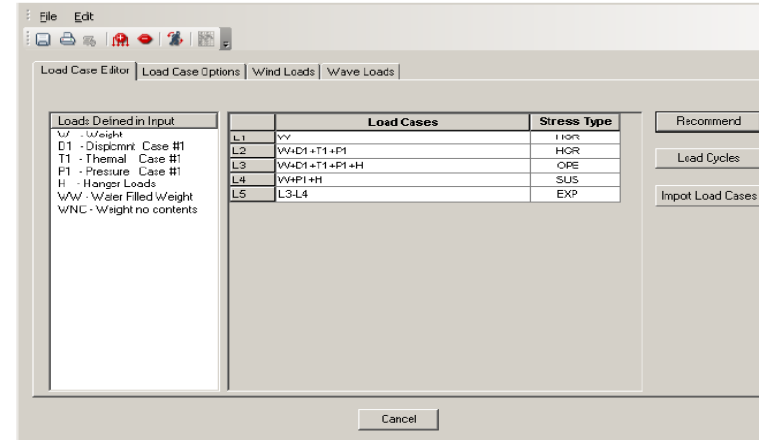
By
Ashish Shrivastava

Basics of Pipe Stress Analysis

1) Modelling



2) Load case



3) Analysis

Load Case Editor					
Load Case Options					
	Load Case Name	Output Status	Output Type	Comb Method	Snubbers Active?
L1	OPERATING CASE CONDITION 1	Keep	Disp/Force/Stress		☐
L2	W+D1+T1+P1+U1(OPE)	Keep	Disp/Force/Stress		☒
L3	SUSTAINED CASE CONDITION 1	Keep	Disp/Force/Stress		☐
L4	EXPANSION CASE CONDITION 1	Keep	Disp/Force/Stress	Algebraic	☐
L5	L2-L1(OCC)	Keep	Disp/Force/Stress	Algebraic	☐
L6	L3-L5(OCC)	Keep	Disp/Force/Stress	Scalar	☐

4) Output

Static Output Processing						
CAESAR II DISPLACEMENT REPORT FILE: FRANK						
CASE 1 (SUS) W DATE: JAN 6, 2005						
-----Translations (in.)-----Rotations (deg.)-----						
NODE	DX	DY	DZ	RX	RY	RZ
1	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
2	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
3	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
4	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
5	0.0000	0.0000	0.0000	-0.0002	0.0000	-0.0002
6	0.0000	0.0000	0.0000	-0.0002	0.0000	-0.0002
7	0.0000	0.0000	0.0000	0.0002	0.0000	0.0002
8	0.0000	0.0000	0.0000	0.0002	0.0000	-0.0002
9	0.0000	-0.0001	0.0000	-0.0003	0.0000	0.0000
10	0.0000	-0.0001	0.0000	0.0000	0.0000	0.0003
11	0.0000	-0.0001	0.0000	0.0003	0.0000	0.0000
12	0.0000	-0.0001	0.0000	0.0000	0.0000	-0.0003
13	0.0000	-0.0003	0.0000	0.0000	0.0000	0.0000

Basics of Pipe Stress Analysis

Basics of pipe stress analysis can be broadly divided into four categories

- 1. Academic Base**
- 2. Codes and Industrial requirements**
- 3. Applicable Code and Standard Compliance**

Academic Base

Academic Base

Typical Stresses in Pipe

Stress due to axial force

$$\sigma = F / A$$

F = Axial force acting on cross section (kg)

A = Cross-sectional of pipe (mm²)



Stress due to bending and torsion

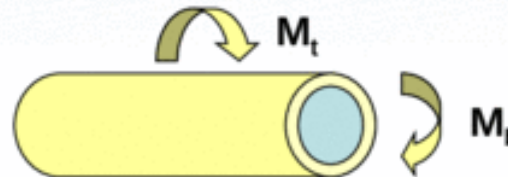
$$\sigma_b = M_b / Z$$

$$\sigma_t = M_t / 2Z$$

M_b = Bending moment (kg-mm)

M_t = Torsional moment (kg-mm)

Z = Section modulus of pipe (mm³)



Stress due to internal pressure

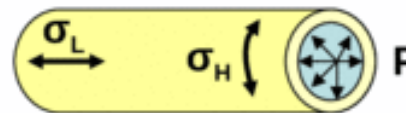
$$\sigma_H = PD_i / 2t \text{ (Hoop)}$$

$$\sigma_L = PD_i / 4t \text{ (Longitudinal)}$$

P = Design pressure (kg/mm²)

D_i = Inner diameter (mm)

t = Thickness of pipe (mm)



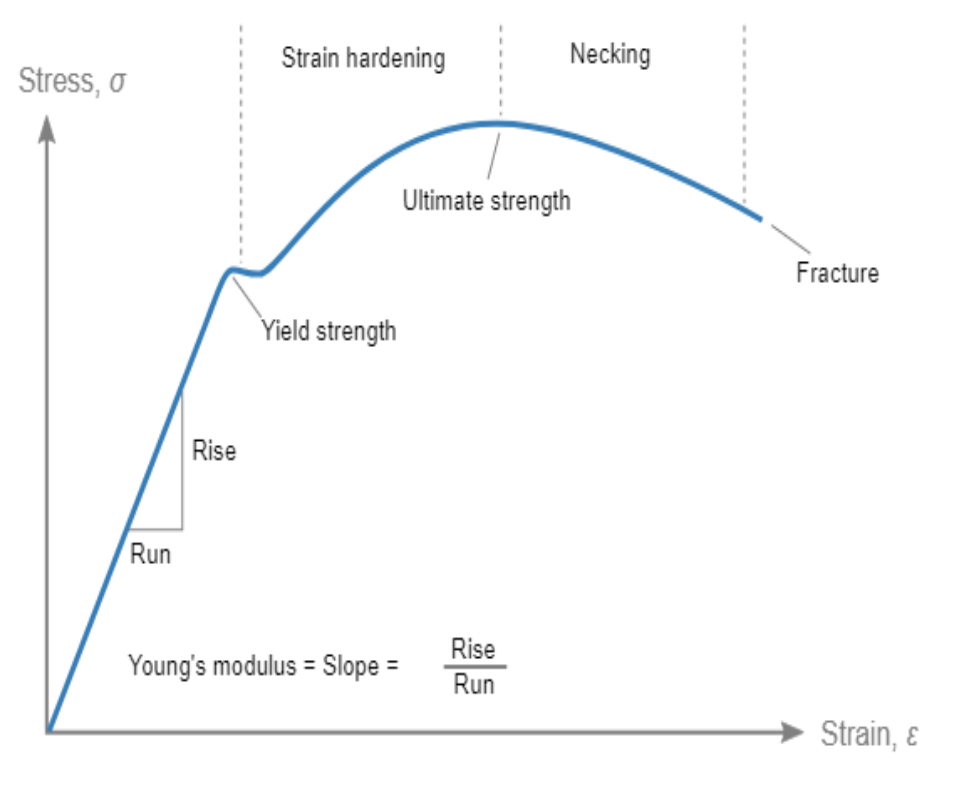
Academic Base

Stress-strain behavior of material:

Typical stress-strain behavior of a ductile material under tensile loading is shown in the famous stress-strain curve below, taking A 53 Gr B – a commonly used CS material.

Stress: - The resistance developed in the material per unit area against the applied force is the stress in the material. It can be simply specified as force per unit area of the material.

Stress (s) = Force / Cross-sectional area

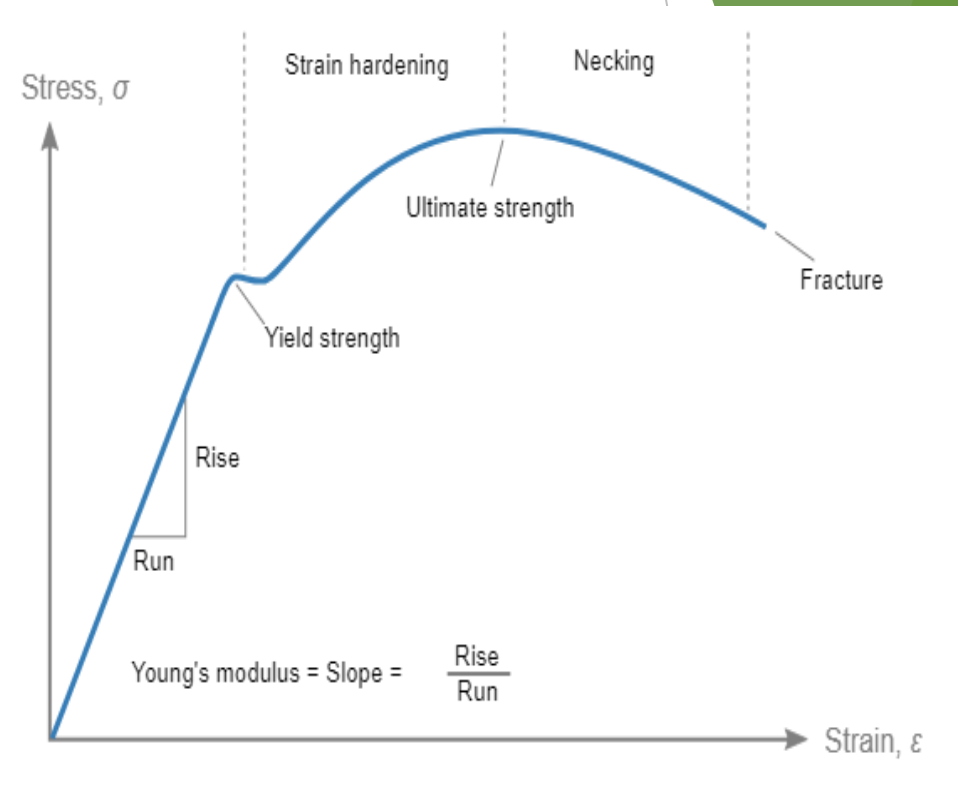


Academic Base

Strain: - A component subjected to load undergoes deformation. The deformation is quantified by strain defined as the change in length per unit length of the material.

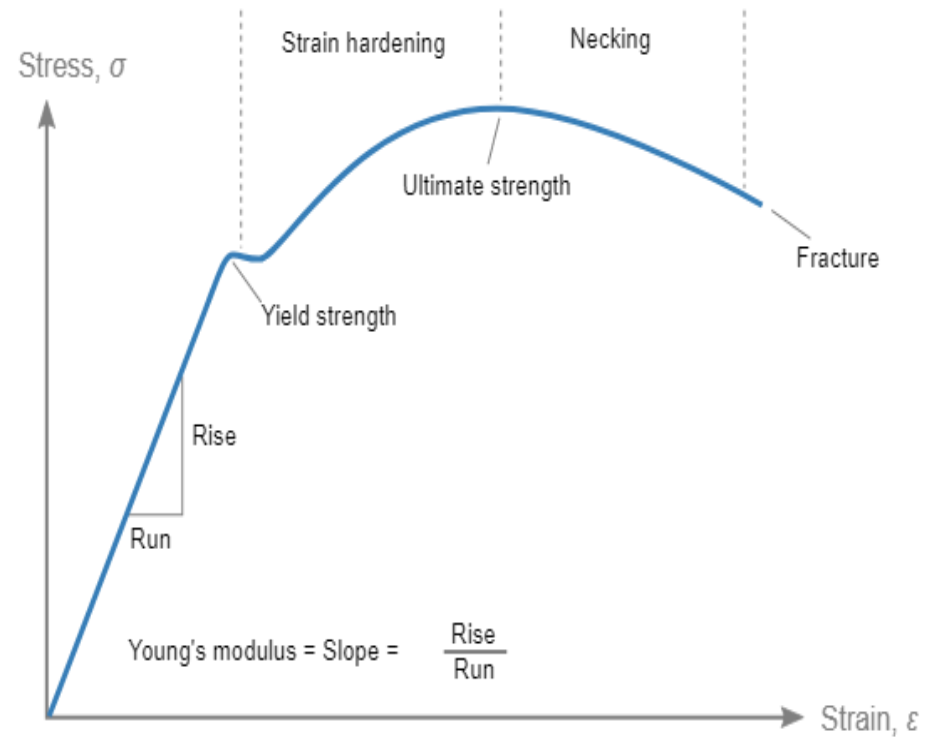
$$\text{Strain (e)} = \Delta l / L \text{ (Lateral strain)}$$

Modulus of elasticity (E): - Up to the certain limit of loading known as the proportional limit, the strain developed in the material is in direct proportion of the stress. This law is called Hooke's law and the constant of proportionality, E, is the modulus of elasticity or Young's modulus, which is a definite property of the material. Mathematically: $E = s / e$



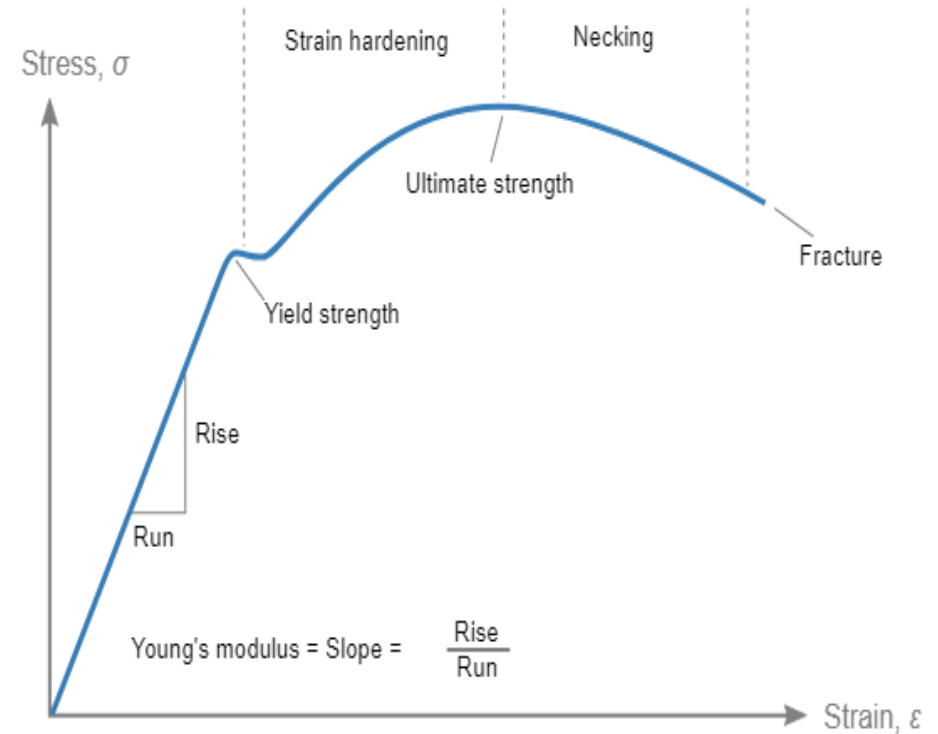
Academic Base

Yield strength: The point at which the specimen or material under tension or compression generates a large deformation without the addition of any load is called the yield point. The corresponding stress is called the yield stress, or yield strength, S_y . The yield point is easy to recognize for materials with a stress-strain curve similar to that shown in the above figure.



Academic Base

Ultimate Tensile strength: The maximum stress in the stress-strain curve of the material is the Ultimate Tensile Strength of the material. This is the point beyond which the material becomes unstable under load and breaks after uncontrolled yielding. This point signifies the beginning of the reduction in the cross-section area (Necking). This explains why the curve shows a drop in stress near the breakpoint toward the end of the curve.

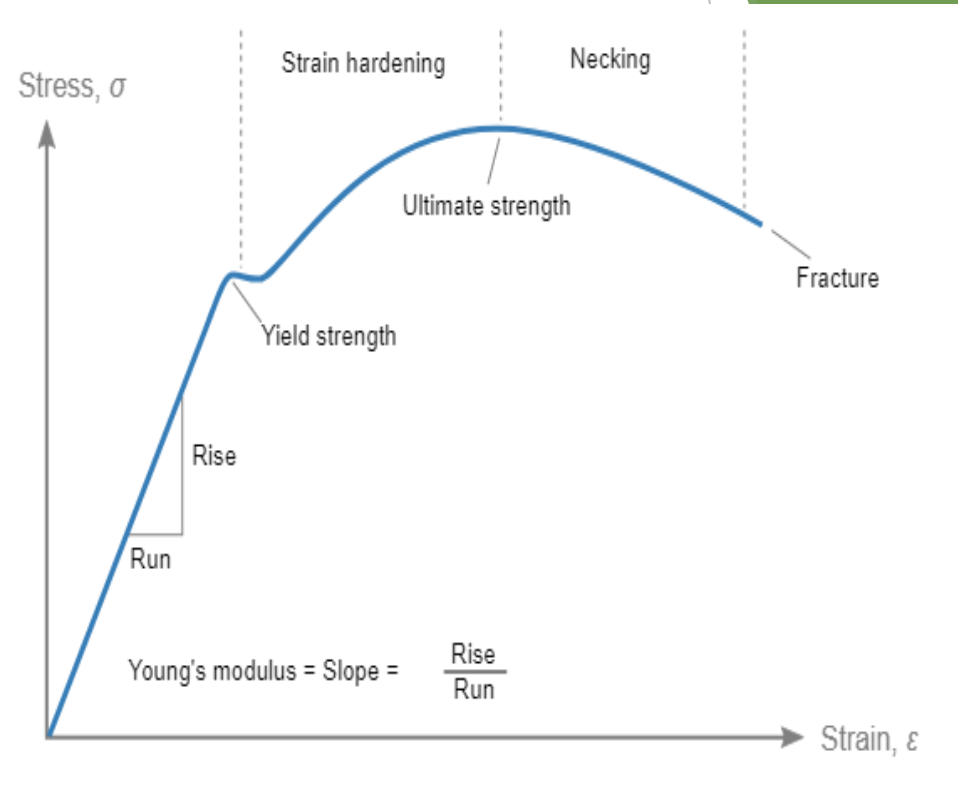


Academic Base

Allowable stress: The yield strength or Ultimate tensile strength of a material, as obtained from standard property charts is divided by a factor of safety to reach the allowable stress of the material.

Mathematically, Allowable stress, s
 $= \text{Yield Strength (or UTS)} / \text{Factor of safety}$

Factor of safety or safety factor is a factor widely followed by the method of design accounts for the uncertainties in the loading and material behavior.



Academic Base

Theories of Failure

There are three generally accepted failure theories which may be used to predict the onset of yielding in a material:

- 1 - OCTAHEDRAL SHEAR, or VON MISES THEORY
- 2 - MAXIMUM SHEAR, or TRESCA THEORY
- 3 - MAXIMUM STRESS or RANKINE THEORY

Academic Base

Theories of Failure

Octahedral Shear - Von Mises Theory:

“Failure occurs when the octahedral shear stress in a body is equal to the octahedral shear stress at yield in a uniaxial tension test.”

The octahedral shear stress is calculated as:

$$\tau_{\text{oct}} = \frac{1}{3} [(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2]^{1/2}$$

In a uniaxial tensile test specimen at the point of yield:

$$S_1 = S_{\text{Yield}}; S_2 = S_3 = 0$$

Therefore the octahedral shear stress in a uniaxial tensile test specimen at failure is calculated as:

$$\begin{aligned}\tau_{\text{oct}} &= \frac{1}{3} [(S_{\text{Yield}} - 0)^2 + (0 - 0)^2 + (0 - S_{\text{Yield}})^2]^{1/2} \\ &= \frac{2^{1/2}}{3} \times S_{\text{Yield}}\end{aligned}$$

Therefore, under the Von Mises theory:

Plastic deformation occurs in a 3-dimensional stress state whenever the octahedral shear exceeds $\frac{2^{1/2}}{3} \times S_{\text{Yield}}$.

Source: Coade
Seminar

Academic Base

Theories of Failure

Maximum Shear Stress - Tresca Theory:

“Failure occurs when the maximum shear stress in a body is equal to the maximum shear stress at yield in a uniaxial tension test.”

The maximum shear stress is calculated as:

$$\tau_{\max} = (S_1 - S_3) / 2$$

In a uniaxial tensile test specimen at the point of yield:

$$S_1 = S_{\text{Yield}}; S_2 = S_3 = 0$$

So:

$$\tau_{\max} = (S_{\text{Yield}} - 0) / 2 = S_{\text{Yield}} / 2$$

Therefore, under the Tresca theory:

Plastic deformation occurs in a 3-dimensional stress state whenever the maximum shear stress exceeds $S_{\text{Yield}} / 2$.

Source: Coade
Seminar

Academic Base

Theories of Failure

Maximum Stress - Rankine Theory

“Failure occurs when the maximum tensile stress in a body is equal to the maximum tensile stress at yield in a uniaxial tension test.”

The maximum tensile stress is the largest, positive principal stress, S_1 . (By definition, S_1 is always the largest of the principal stresses.)

In a uniaxial tensile test specimen at the point of yield:

$$S_1 = S_{Yield}; S_2 = S_3 = 0$$

Therefore, under the Rankine theory:

Plastic deformation occurs in a 3-dimensional stress state whenever the maximum principal stress exceeds S_{Yield} .

Academic Base

Theories of Failure

Most piping codes use slight modification of Maximum Shear

Stress theory for flexibility related failures.

Therefore the Maximum Stress Intensity criterion, as adopted by most piping codes, dictates the following requirement:

$$[(S_L - S_H)^2 + 4 \tau^2]^{1/2} < S_{Yield}$$

Academic Base

Friction Effects

- ▶ Resists motion
- ▶ When pipe expands and moves, friction resists it, causes forces on the pipe supports.
- ▶ Anti friction pads (Teflon, graphite material are used) to reduce friction
- ▶ Occasional cases friction advantage is not recommended

Academic Base

Galling Effect

- ▶ Galling is adhesive wear that is caused by microscopic transfer of material between metallic surfaces, during transverse motion (sliding). It occurs frequently whenever metal surfaces are in contact, sliding against each other, especially with poor lubrication.

Academic Base

Thermal Bowing

- Thermal bowing typically occurs when transporting fluid on a partially filled pipe but can also occur in exposed pipes in hot climates where one side of the pipe is exposed to the sun and the other side is in the shade. In this case, the top and bottom of the pipe section can experience a significant temperature differential. This temperature differential causes pipe thermal strains which produce pipe curvature.

Academic Base

Cold Spring

Cold spring is the intentional deformation of piping during assembly to produce a desired initial displacement and reaction. Cold spring is beneficial in that it serves to balance the magnitude of the reaction under initial and extreme displacement conditions. When cold spring is properly applied there is less likelihood of overstrain during initial operation; hence, it is recommended especially for piping materials of limited ductility.

Academic Base

Cold Spring

Where cold spring is used in the piping system, experience has shown that it cannot be fully assured. Therefore, the reactions shall be computed both with the assumption that only two-thirds of the design cold spring is present, and with four-thirds of the design cold spring present

Cold springing or misalignment can result in significant stresses in the ambient condition. The designer is responsible for ensuring that such stresses are accounted for before any credit is taken for reduction in minimum design temperature without impact testing.

Codes and Industrial Requirements

Codes and Industrial requirements

Piping Loads

Sustained Loads:

Loads that are present throughout the operations (weight of pipe, fluid, insulation, internal pressure etc.)

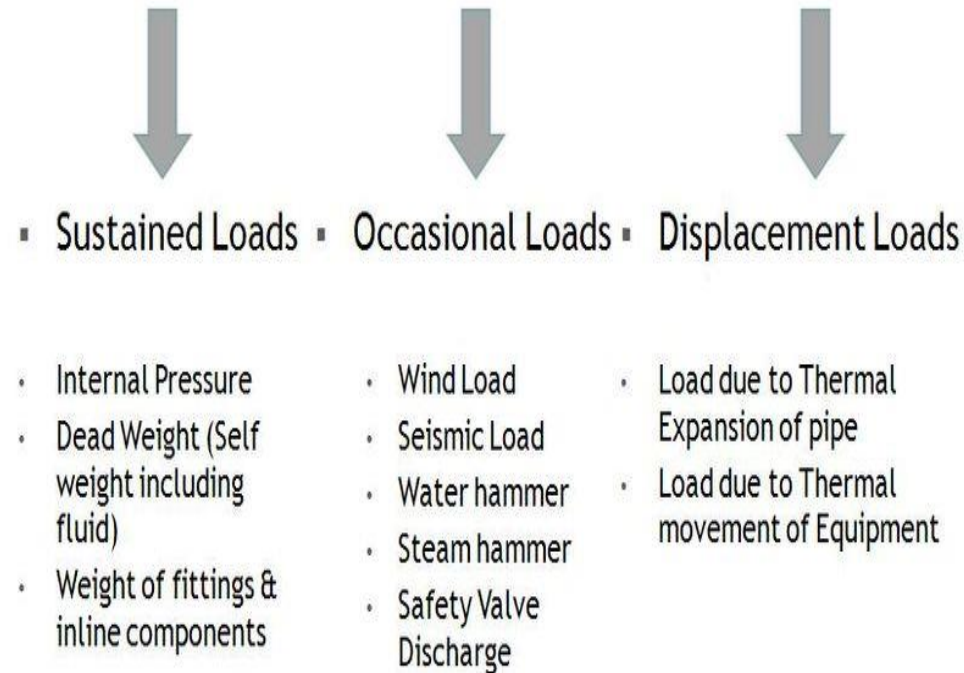
Occasional Loads:

Loads that are present occasionally during operations (wind, seismic, PSV, snow etc.)

Displacement Loads:

Loads that are produced due to restriction of thermal movement.

Loads on Piping System



Codes and Industrial requirements

Piping Loads

Static:

- ↓ Loads that act slowly the system gets time to react and loads and stresses.
- ↓ Piping remains in equilibrium.
- ↓ Sum of forces and moments always remains zero.

Dynamic:

- ↓ Dynamic loads are time driven.
- ↓ Piping does not have time to react to these loads.
- ↓ Results in unbalanced forces and moments
- ↓ Pipe displacement.
- ↓ The sum of the forces and moments may be greater/smaller than zero.

Codes and Industrial requirements

Piping Stresses

Primary Stresses:

- ↓ Primary stress is the reaction of piping system to the primary loads that are (weight, internal pressure).
- ↓ Stresses developed are not self-limiting.
- ↓ Allowable limits based upon yield strength

Secondary Stresses:

- ↓ Secondary stress are reaction of piping system produced due to secondary loads by restriction of thermal movement.
- ↓ Secondary stress are self limiting.
- ↓ Secondary stresses are cyclic in nature.
- ↓ Single application wont result in failure

Codes and Industrial requirements

Designing System

Sustained Loads

- ▶ Piping shall be adequately supported for self weight due to (pipe, insulation & fluid, internal pressure)
- ▶ Pipe shall have stresses and displacement within the allowable.
- ▶ Concentrated loads of valves and other fittings shall be properly supported.

Codes and Industrial requirements

Designing System

Expansion Loads

- ▶ Piping shall be adequately flexible to reduce thermal loads
- ▶ Piping shall have expansion loops and expansion joints to reduce the loads and redistribute the displacements
- ▶ Piping shall be designed to ensure thermal loads on equipment nozzle are within the limits

Codes and Industrial requirements

Providing Flexibility in Piping

Providing flexibility to a piping system means adding adjacent, perpendicular legs to absorb the thermal growth through bending.

- L shape Expansion Loop
- 2D Expansion Loop
- 3D Expansion Loop

2D & 3D Expansion Loops are required for Hot piping systems

Codes and Industrial requirements

Designing System

Occasional Loads

- ▶ Piping shall be designed for occasional loads like Wind, Seismic and snow as applicable.
- ▶ Piping shall be adequately rigid and strong to absorb these occasional loads

Codes and Industrial requirements

Support Span Calculations

$$L_{all} = (10 Z S_{all} / w)^{1/2}$$

Where:

L_{all} = allowable pipe span for weight loading, in

Z = section modulus of pipe, in³

S_{all} = approximate allowable stress of piping material for weight stresses (S_h , less pressure stresses, divided by intensification factor, for example), psi

Codes and Industrial requirements

Pipe Span Reduction Factor for Elbows, Concentrated Loads

- Pipe Span is reduced to $\frac{3}{4}$ of its length in presence of direction change (elbows)
- Pipe Span is reduced to $\frac{3}{4}$ of its length in presence of Concentrated Loads

Codes and Industrial requirements

Different Types and Function of Pipe Supports

Site Fabricated

- ▶ Shoe Support (Rest & Vertical loads)
- ▶ Guide Support (Lateral Loads)
- ▶ Axial Stop/Line Stop (Axial Loads)
- ▶ Trunnion (Vertical and Horizontal Loads)

Procured Supports

- ▶ Spring hangers (Rest & Vertical loads)
- ▶ Sway braces (Vibrations)
- ▶ Struts (Horizontal Loads)
- ▶ Expansion joints (Thermal Loads)

Codes and Industrial requirements

Piping Systems Hydrostatic & Pneumatic Testing

Hydrotest is a test performed to check and test pressure vessels piping & fittings for their strength and leaks. System is filled with water and pressurized up to 1.5 times of design pressure or as per applicable code.

Codes and Industrial requirements

Rotary Equipment and Natural Frequency

Piping connected to Rotary Equipment like Pumps, Turbine, Compressor etc. may be subject to vibrations. If frequency of equipment matches with piping then resonance may happen which will amplify the vibrations and can have highly decremental effects.

- So allowable span is reduced for rotary equipment connected piping system
- Natural frequency of piping system is maintained above 4Hz as rule of thumb for these system.

Codes and Industrial requirements

Solving Concentrated Loads and Reducing Loads on Equipment Nozzles

- As per Support Span (for stresses and displacement)
- Concentrated Loads like valves shall be supported both sides.
- Selected support shall be active in all scenarios
- First support location shall be judiciously selected to ensure proper alignment and load bearing capacity.
- Loads shall be within the allowable provided by the manufacturer.
- Even distribution of loads (axial stop for thermal null point)

Codes and Industrial requirements

Adding Flexible Connections (Vessels)

Whenever Pressure Vessel or Heat exchanger (Static Equipment) nozzle loads exceeds the allowable values provided by Vendors (Equipment manufacturer) or standard project specific tables (guidelines), the piping stress professional is permitted to use WRC 107/297 (or any other FEA) to check the stresses at the Nozzle-Shell junction point and check the stresses with allowable values provided by Codes. If the stresses are found to be within allowable limit then the load and moment values can be accepted without any hesitation.

However there are some boundary conditions which must be met before using WRC. This small write up will try to explain the required details for performing WRC 107 and WRC 297 using Caesar II and step by step method for performing WRC check.

Codes and Industrial requirements

Theory of Dynamic Analysis of Systems

A dynamic load changes quickly with time. The piping system does not have time to internally distribute the loads. Forces and moments are not always resolved, resulting in unbalanced loads and pipe movement. Because the sum of forces and moments are not in equilibrium, the internally-induced loads can be different—either higher or lower—than the applied loads

There are several methods for analyzing different types of system response under dynamic loads. Each method provides a trade-off of accuracy versus computing requirements.

The methods include modal natural frequency calculations, harmonic analysis, response spectrum analysis, and time history analysis.

Codes and Industrial requirements

Theory of Dynamic Analysis of Systems

Random

With this type of profile, the load unpredictably changes direction or magnitude with time. Even with the unpredictability, some load characteristics can predominate. Loads with random force/time profiles are best solved using a spectrum method or a static equivalent.

The major types of loads with random time profiles are wind and earthquake.

Codes and Industrial requirements

Theory of Dynamic Analysis of Systems

Harmonic

With this type of profile, the load changes direction and/or magnitude following a harmonic profile, ranging from its minimum to its maximum over a fixed time period. For example, the load can be described by a function of the form:

Where: $F(t) = A + B \cos(\omega t + \phi)$

$F(t)$ = force magnitude as a function of time

A = mean force

B = variation of maximum and minimum force from mean

ω = angular frequency (radian/sec)

ϕ = phase angle (radians)

t = time (sec)

Loads with harmonic force/time profiles are best solved using a harmonic method.

The major types of loads with harmonic time profiles are equipment vibration, acoustic vibration, and pulsation..

Codes and Industrial requirements

Theory of Dynamic Analysis of Systems

Impulse

With this type of profile, the load magnitude ramps up from zero to some value, remains relatively constant for a time, and then ramps down to zero again. For rapid ramping times, this type of profile resembles a rectangle. Loads with impulse force/time profiles are best solved using time history or force spectrum methods. Major types of loads with impulse time profiles are relief valve, fluid hammer, and slug flow.

Codes and Industrial requirements

Theory of Dynamic Analysis of Systems

Modal natural frequency analysis

Modal natural frequency analysis measures the tendency of a piping system to respond to dynamic loads. The modal natural frequencies of a system typically should not be too close to equipment operating frequencies. As a general rule, higher natural frequencies usually cause less trouble than low natural frequencies.

Codes and Industrial requirements

Field Vibration Problems using Harmonic Analysis

Harmonic analysis addresses dynamic loads that are cyclic in nature, such as fluid pulsation in reciprocating pump lines or vibration due to rotating equipment. These loads are modeled as concentrated forces or displacements at one or more points in the system. To provide the proper phase relationship between multiple loads, a phase angle can also be used. Any number of forcing frequencies can be analyzed for equipment start-up and operating modes. Harmonic responses represent the maximum dynamic amplitude the piping system undergoes and have the same form as a static analysis: node deflections and rotations, local forces and moments, restraint loads, and stresses. For example, if the results show an X displacement of 5.8 cm at a node, then the dynamic motion due to the cyclic excitation is from +5.8 cm. to -5.8 cm. at that node. The stresses shown are one half of, or one amplitude of, the full cyclic stress range.

Codes and Industrial requirements

Response spectrum analysis

Response spectrum analysis allows an impulse-type transient event to be characterized by response versus frequency spectra. Each mode of vibration of the piping system is related to one response on the spectrum. These modal responses are summed together to produce the total system response. The stresses for these analyses, summed with the sustained stresses, are compared to the occasional stress allowables defined by the piping code. Spectral analysis can be used in a wide variety of applications. For example, in uniform inertial loading, ground motion associated with a seismic event is supplied as displacement, velocity, or acceleration response spectra.

Codes and Industrial requirements

Response spectrum analysis

The assumption is that all supports move with the defined ground motion and the piping system “catches up” to the supports. It is this inertial effect which loads the system.

The shock spectra, which define the ground motion, can vary between the three global directions and can even change for different groups of supports (such as independent or uniform support motion). Another example is based on single point loading. CAESAR II uses this technique to analyze a wide variety of impulse-type transient loads. Relief valve loads, water hammer loads, slug flow loads, and rapid valve closure type loads all cause single impulse dynamic loads at various points in the piping system. The response to these dynamic forces can be predicted using the force spectrum method..

Codes and Industrial requirements

Time History Analysis

Time history analysis is one of the most accurate methods, because it uses numeric integration of the dynamic equation of motion to simulate the system response throughout the load duration. This method can solve any type of dynamic loading, but due to its exact solution, requires more resources (such as computer memory, calculation speed and time) than other methods. Time history analysis is not appropriate when, for example, the spectrum method offers sufficient accuracy.

Codes and Industrial requirements

Hammer Loads

When the flow of fluid through a system is suddenly halted through valve closure or a pump trip, the fluid in the remainder of the system cannot be stopped instantaneously. As fluid continues to flow into the area of stoppage (upstream of the valve or pump), the fluid compresses causing a high pressure situation. On the other side of the restriction, the fluid moves away from the stoppage point, creating a low pressure (vacuum) situation. Fluid at the next elbow or closure along the pipeline is still at the original operating pressure, resulting in an unbalanced pressure force acting on the valve seat or the elbow. The fluid continues to flow, compressing (or decompressing) fluid further away from the point of flow stoppage, causing the leading edge of the pressure pulse to move through the line. As the pulse moves past the first elbow, the pressure is now equalized at each end of the pipe run, leading to a balanced (that is, zero) pressure load on the first pipe leg.

Codes and Industrial requirements

Hammer Loads

The unbalanced pressure, by passing the elbow, has now shifted to the second leg. The unbalanced pressure load continues to rise and fall in sequential legs as the pressure pulse travels back to the source, or forward to the sink.

The ramp up time of the profile roughly coincides with the elapsed time from full flow to low flow, such as the closing time of the valve or trip time of the pump. Because the leading edge of the pressure pulse is not expected to change as the pulse travels through the system, the ramp-down time is the same. The duration of the load from initiation through the beginning of the down ramp is equal to the time required for the pressure pulse to travel the length of the pipe leg.

Codes and Industrial requirements

Slug Flow Modelling

Slug Flow

Piping systems designed for single-phase

Multiple phases like slurry are slug susceptible

Fr

$$= dp / dt = \rho v^2 A [2(1 - \cos \theta)]^{1/2}$$

Where:

dp = change in momentum

dt = change in time

ρ = fluid density

v = fluid velocity

A = internal area of pipe

θ = inclusion angle at elbow

Codes and Industrial requirements

Slug Flow Modelling

Where:

dp = change in momentum

dt = change in time

ρ = fluid density

v = fluid velocity

A = internal area of pipe

θ = inclusion angle at elbow

Codes and Industrial requirements

Slug Flow Modelling

Constant fluid density

- ▶ Forces are constant on elbows
- ▶ Equal and opposite loads cancelling each other
- ▶ System remains in equilibrium

Variable fluid density

- ▶ Density changes with time
 - ▶ Forces are not constant on elbows
 - ▶ System does not remain in equilibrium
 - ▶ Dynamic force
-
- ▶ Slug (high density than gas) hitting the elbow
 - ▶ Momentum load increases multiple times
 - ▶ Last till the next elbow and drops to Zero again
 - ▶ Time duration is length of the slug / fluid velocity

Codes and Industrial requirements

Evaluating Relief Valve Discharge

- ▶ Relief valves are set to open when system pressure reaches a dangerous level.
- ▶ Relief valves vents the fluid to release the overpressure.
- ▶ Venting causes a jet force.
- ▶ Force ramps up from zero to max value while opening of valve.
- ▶ Jet force remains relatively constant till the outflow.
- ▶ Overpressure is reduced to normal
- ▶ Force ramps down from max to zero value while closing of valve.

The background features abstract, overlapping green geometric shapes, primarily triangles and polygons, in various shades of green, creating a modern, layered effect on the right side of the slide.

Applicable Code & Standards Compliance

Applicable Code & Standards Compliance

Typical Code and Standards

- Piping Design Code

- ASME B31.1 Power Piping
- ASME B31.3 Process Piping
- ASME B31.4 Pipeline (Hydrocarbon)
- ASME B31.8 Pipeline (Gas)
- ASME Section III Nuclear Component Design

ASME: The American Society of Mechanical Engineers

API: American Petroleum Institute

NEMA: National Electrical Manufacturers Association

WRC: Welding Research Council

- Related Code for Nozzle Evaluation

- API Std 610 Centrifugal Pump
- API Std 611 Steam Turbines
- API Std 617 Centrifugal Compressor
- API Std 618 Reciprocating Compressor
- API Std 650 Welded Steel Tanks
- API Std 560 Fired Heaters (Furnace)
- API STD 661 Air-Cooled Heat Exchangers (AFC)
- NEMA SM23 Steam Turbines
- ASME SEC VIII Pressure Vessel
- WRC 107, WRC 297 Local Stress on Nozzles

Applicable Code & Standards Compliance

Allowable Thermal Expansion coefficient-Table C-1, ASME B31.3

Table C-1 Thermal Expansion Data																		
		$A = \text{Mean Coefficient of Thermal Expansion, } 10^{-6} \text{ in./in./}^{\circ}\text{F}$ $B = \text{Linear Thermal Expansion, in./100 ft}$ } in Going From 70°F to Indicated Temperature [Note (1)]																
		Temperature Range 70°F to																
Material	Coefficient	-325	-150	-50	70	200	300	400	500	600	700	800	900	1,000	1,100	1,200	1,300	1,400
Group 1 carbon and low alloy steels [Note (2)]	A	5.5	5.9	6.2	6.4	6.7	6.9	7.1	7.3	7.4	7.6	7.8	7.9	8.1	8.2	8.3	8.4	8.4
	B	-2.6	-1.6	-0.9	0	1.0	1.9	2.8	3.7	4.7	5.7	6.8	7.9	9.0	10.1	11.3	12.4	14.7
Group 2 low alloy steels [Note (3)]	A	6.0	6.5	6.7	7.0	7.3	7.4	7.6	7.7	7.8	7.9	8.0	8.1	8.2	8.3	8.4	8.4	8.5
	B	-2.9	-1.7	-1.0	0	1.1	2.0	3.0	4.0	5.0	6.0	7.0	8.1	9.2	10.3	11.4	12.5	13.5
5Cr-1Mo steels	A	5.6	6.0	6.2	6.4	6.7	6.9	7.0	7.1	7.2	7.2	7.3	7.4	7.5	7.6	7.6	7.7	7.8
	B	-2.7	-1.6	-0.9	0	1.0	1.9	2.8	3.7	4.6	5.5	6.4	7.4	8.4	9.3	10.3	11.4	12.4
9Cr-1Mo steels	A	5.0	5.4	5.6	5.8	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1	7.2	7.2
	B	-2.4	-1.4	-0.8	0	0.9	1.7	2.5	3.3	4.1	5.0	5.9	6.8	7.7	8.7	9.7	10.6	11.6
Straight chromium stainless steels																		
12Cr to 13Cr steels	A	5.1	5.5	5.7	5.9	6.2	6.3	6.4	6.5	6.5	6.6	6.7	6.7	6.8	6.8	6.9	6.9	7.0
	B	-2.4	-1.5	-0.8	0	1.0	1.7	2.5	3.3	4.2	5.0	5.8	6.7	7.6	8.5	9.4	10.2	11.1
15Cr to 17Cr steels	A	4.5	4.9	5.1	5.3	5.5	5.7	5.8	5.9	6.0	6.1	6.2	6.2	6.3	6.4	6.4	6.5	6.5
	B	-2.1	-1.3	-0.7	0	0.9	1.6	2.3	3.0	3.8	4.6	5.4	6.2	7.0	7.9	8.7	9.5	10.4
27Cr steels	A	4.3	4.7	4.9	5.0	5.2	5.2	5.3	5.4	5.4	5.5	5.6	5.7	5.7	5.8	5.9	5.9	6.0
	B	-2.0	-1.2	-0.7	0	0.8	1.4	2.1	2.8	3.5	4.2	4.9	5.6	6.4	7.2	8.0	8.7	9.6
Austenitic stainless steels (304, 305, 316, 317, 321, 347, 348, 19-9DL, XM-15, etc.)	A	7.5	8.0	8.2	8.5	8.9	9.2	9.5	9.7	9.9	10.0	10.1	10.2	10.3	10.4	10.6	10.7	10.8
	B	-3.6	-2.1	-1.2	0	1.4	2.5	3.8	5.0	6.3	7.5	8.8	10.2	11.5	12.9	14.3	15.8	17.2
Other austenitic stainless steels (309, 310, 315, XM-19, etc.)	A	7.1	7.6	7.8	8.2	8.5	8.7	8.9	9.1	9.2	9.3	9.4	9.5	9.6	9.7	9.8	9.9	10.1
	B	-3.4	-2.0	-1.1	0	1.3	2.4	3.5	4.7	5.8	7.0	8.2	9.5	10.7	12.0	13.3	14.7	16.1
Gray iron	A	...	...	...	...	5.8	5.9	6.1	6.3	6.5	6.7	6.8	7.0	7.2	...	...	...	...
	B	...	...	...	0	0.9	1.6	2.4	3.2	4.1	5.0	6.0	7.0	8.0	...	...	...	...
Ductile cast iron	A	...	4.9	5.3	5.7	6.0	6.3	6.6	6.8	7.0	7.1	7.3	7.4	7.5	...	...	...	...
	B	...	-1.3	-0.8	0	0.9	1.7	2.6	3.5	4.5	5.4	6.4	7.3	8.4	...	...	...	...

Applicable Code & Standards Compliance

ASME B31.3

ASME B31.3 Code for compliance shall meet requirements

- Design
- Material
- Fabrication
- Examination
- Inspection
- Testing.

Applicable Code & Standards Compliance

Allowable Stresses Values

As per the ASME B31.1 (Minimum of)

- ✓ $1/4^{\text{TH}}$ OF UTS OF MATERIAL AT OPERATING TEMPERATURE
- ✓ $1/4^{\text{TH}}$ OF UTS OF MATERIAL AT ROOM TEMPERATURE.
- ✓ $5/8^{\text{TH}}$ OF YIELD STRENGTH OF THE MATERIAL AT OPE TEMPERATURE
- ✓ $5/8^{\text{TH}}$ IF YIELD STRENGTH OF THE MATERIAL AT ROOM TEMPERATURE.

As per the ASME B31.3 (Minimum of)

- ✓ $1/3^{\text{RD}}$ OF UTS OF MATERIAL AT OPERATING TEMPERATURE
- ✓ $1/3^{\text{RD}}$ OF UTS OF MATERIAL AT ROOM TEMPERATURE.
- ✓ $2/3^{\text{TH}}$ OF YIELD STRENGTH OF THE MATERIAL AT OPE TEMPERATURE.
- ✓ $2/3^{\text{TH}}$ IF YIELD STRENGTH OF THE MATERIAL AT ROOM TEMPERATURE.

Applicable Code & Standards Compliance

Formulas from codes ASME B31.3

Sustained loads

The equation for the stress due to sustained loads, such as pressure and weight, SL,

$$S_L = \sqrt{(|S_a| + S_b)^2 + (2S_t)^2}$$

$$S_a = \frac{I_a F_a}{A_p}$$

$$S_b = \frac{\sqrt{(I_i M_i)^2 + (I_o M_o)^2}}{Z}$$

$$S_t = \frac{I_t M_t}{2Z}$$

Applicable Code & Standards Compliance

Concept of Stress Range, ASME B31.3

Bending stress that will not result in plastic deformation in hot condition $1.5S_h$

Bending stress that will not result in plastic deformation in cold condition $1.5S_c$

Total Sum hence the allowable maximum stress range $= 1.5(S_c + S_h)$

Conservatively as per ASME B31.3 $= 1.25(S_c + S_h)$

Now $1.0 S_h$ is dedicated to stress due to weight and internal pressure

$$SA = 1.25S_c + 0.25S_h$$

When $SL < 1.0S_h$

So allowable equation

$$SA = 1.25(S_c + S_h) - SL$$

Applicable Code & Standards Compliance

Formulas from codes ASME B31.3

Thermal Expansion loads

The axial, bending, and torsional displacement stress ranges shall be computed using the reference modulus of elasticity at 21°C (70°F), E_a , except as provided in para. 319.2.2(b)(4), and then combined in accordance with eq. (17) to determine the computed displacement stress range, S_E , which shall not exceed the allowable displacement stress range, S_A , in para. 302.3.5(d). See also eq. (1d) and Appendix S, Example 3 for the greatest computed displacement stress range.

$$S_E = \sqrt{(|S_a| + S_b)^2 + (2S_t)^2}$$

Applicable Code & Standards Compliance

Allowable Stress Range & Pipe Thickness Equation, ASME B31.3

(d) *Allowable Displacement Stress Range S_A .* The computed displacement stress range S_E in a piping system (see para. 319.4.4) shall not exceed the allowable displacement stress range S_A (see paras. 319.2.3 and 319.3.4) calculated by Eq. (1a):

$$S_A = f(1.25S_c + 0.25S_h) \quad (1a)$$

When S_h is greater than S_L , the difference between them may be added to the term $0.25S_h$ in Eq. (1a). In that case, the allowable stress range is calculated by Eq. (1b):

$$S_A = f[1.25(S_c + S_h) - S_L] \quad (1b)$$

In Eqs. (1a) and (1b):

S_c = basic allowable stress³ at minimum metal temperature expected during the displacement cycle under analysis

S_h = basic allowable stress³ at maximum metal temperature expected during the displacement cycle under analysis

f = stress range reduction factor,⁴ from Table 302.3.5 or calculated by Eq. (1c):⁵

$$f = 6.0(N)^{-0.2} \leq 1.0 \quad (1c)$$

where

N = equivalent number of full displacement cycles during the expected service life of the piping system⁶

K302.3.6 Limits of Calculated Stresses Due to Occasional Loads

(a) *Operation.* The sum of the longitudinal stresses due to pressure, weight, and other sustained loadings S_L , and of the stresses produced by occasional loads such as wind or earthquake, may be as much as 1.2 times the allowable stress given in Table K-1. Wind and earthquake forces need not be considered as acting concurrently.

(b) *Test.* Stresses due to test conditions are not subject to the limitations in para. K302.3. It is not necessary to consider other occasional loads, such as wind and earthquake, as occurring concurrently with test loads.

304.1.2 Straight Pipe Under Internal Pressure

(a) For $t < D/6$, the internal pressure design thickness for straight pipe shall be not less than that calculated in accordance with either Eq. (3a) or Eq. (3b):

$$t = \frac{PD}{2(SE + PY)} \quad (3a)$$

$$t = \frac{P(d + 2c)}{2[SE - P(1 - Y)]} \quad (3b)$$

(b) For $t \geq D/6$ or for $P/SE > 0.385$, calculation of pressure design thickness for straight pipe requires special consideration of factors such as theory of failure, effects of fatigue, and thermal stress.

Applicable Code & Standards Compliance

Stress Range Reduction Factor, ASME B31.3

CYCLIC REDUCTION FACTOR TABLE			
Cycles N			Factor f
1	-	7,000	1.0
7,001	-	14,000	0.9
14,001	-	22,000	0.8
22,001	-	45,000	0.7
45,001	-	100,000	0.6
100,001	-	200,000	0.5
200,001	-	700,000	0.4
700,001	-	2,000,000	0.3

Applicable Code & Standards Compliance

Basic Allowable Stresses- Table A-1, ASME B31.3

ASME B31.3-2010

Table A-1 Basic Allowable Stresses in Tension for Metals (Cont'd)

Numbers in Parentheses Refer to Notes for Appendix A Tables; Specifications Are ASTM Unless Otherwise Indicated

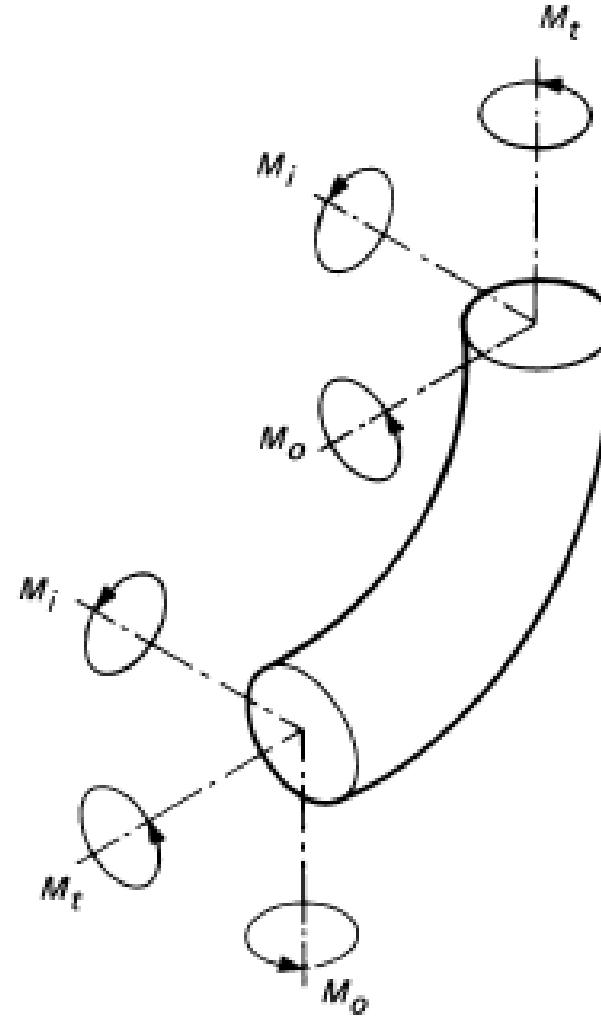
Material	Spec. No.	Type/Grade	UNS No.	Class/Condition/Temper	Size, in.	P-No. (5)	Notes	Min. Temp., °F (6)	Specified Min. Strength, ksi		Basic Allowable Stress, S, ksi, at Metal Temperature, °F [Note (1)]		
									Tensile	Yield	Min. Temp. to 100	200	300
Carbon Steel — Pipes and Tubes													
A285 Gr. A	A134	—	—	—	—	1	(Bb)(57)	B	45	24	15.0	14.7	14.2
A285 Gr. A	A672	A45	K01700	—	—	1	(57)(59)(67)	B	45	24	15.0	14.7	14.2
Butt weld	API 5L	A25	—	—	—	1	(Ba)(77)	-20	45	25	15.0	15.0	14.7
Smile & ERW	API 5L	A25	—	—	—	1	(57)(59)(77)	B	45	25	15.0	15.0	14.7
—	A179	—	K01200	—	—	1	(57)(59)	-20	47	26	15.7	15.7	15.3
Type F	A53	A	K02504	—	—	1	(Ba)	20	48	30	16.0	16.0	16.0
—	A139	A	—	—	—	1	(Bb)	A	48	30	16.0	16.0	16.0
—	A587	—	K11500	—	—	1	(57)(59)	-20	48	30	16.0	16.0	16.0
—	A53	A	K02504	—	—	1	(57)(59)	B	48	30	16.0	16.0	16.0
—	A106	A	K02501	—	—	1	(57)	B	48	30	16.0	16.0	16.0
—	A135	A	—	—	—	1	(57)(59)	B	48	30	16.0	16.0	16.0
—	A369	FPA	K02501	—	—	1	(57)	B	48	30	16.0	16.0	16.0
—	API 5L	A	—	—	—	1	(57)(59)	B	48	30	16.0	16.0	16.0
A285 Gr. B	A134	—	—	—	—	1	(Bb)(57)	B	50	27	16.7	16.5	15.9
A285 Gr. B	A672	A50	K02200	—	—	1	(57)(59)(67)	B	50	27	16.7	16.5	15.9
A285 Gr. C	A134	—	—	—	—	1	(Bb)(57)	A	55	30	18.3	18.3	17.7
—	A524	II	K02104	—	—	1	(57)	-20	55	30	18.3	18.3	17.7
—	A333	I	K03008	—	—	1	(57)(59)	-50	55	30	18.3	18.3	17.7
—	A334	I	K03008	—	—	1	(57)(59)	-50	55	30	18.3	18.3	17.7
A285 Gr. C	A671	CA55	K02801	—	—	1	(59)(67)	A	55	30	18.3	18.3	17.7
A285 Gr. C	A672	A55	K02801	—	—	1	(57)(59)(67)	A	55	30	18.3	18.3	17.7
A516 Gr. 55	A672	C55	K01800	—	—	1	(57)(67)	C	55	30	18.3	18.3	17.7
A516 Gr. 60	A671	CC60	K02100	—	—	1	(57)(67)	C	60	32	20.0	19.5	18.9
A515 Gr. 60	A671	CB60	K02401	—	—	1	(57)(67)	B	60	32	20.0	19.5	18.9
A515 Gr. 60	A672	B60	K02401	—	—	1	(57)(67)	B	60	32	20.0	19.5	18.9
A516 Gr. 60	A672	C60	K02100	—	—	1	(57)(67)	C	60	32	20.0	19.5	18.9
—	A139	B	K03003	—	—	1	(Bb)	A	60	35	20.0	20.0	20.0
—	A135	B	K03018	—	—	1	(57)(59)	B	60	35	20.0	20.0	20.0
—	A524	I	K02104	—	—	1	(57)	-20	60	35	20.0	20.0	20.0

Nozzle Evaluation and allowable loads

- Related Code for Nozzle Evaluation
- API Std 610 Centrifugal Pump
- API Std 611 Steam Turbines
- API Std 617 Centrifugal Compressor
- API Std 618 Reciprocating Compressor
- API Std 650 Welded Steel Tanks
- API Std 560 Fired Heaters (Furnace)
- API STD 661 Air-Cooled Heat Exchangers (AFC)
- NEMA SM23 Steam Turbines
- ASME SEC VIII Pressure Vessel
- WRC 107, WRC 297 Local Stress on Nozzles

In plane & Out plane Bending Moments

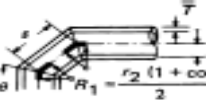
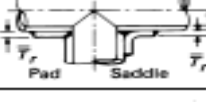
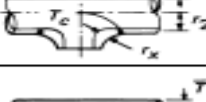
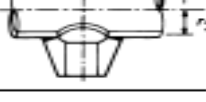
Simply, 'the bending moment which causes elbow to open or close in the plane formed by two limbs of elbow is called in-plane bending moment.' and 'the bending moment which causes one limb of elbow to displace out of the plane retaining other limb steady is called out-plane bending moment.'



Stress Intensification Factors- SIF

The Stress Intensification Factor (SIF) is a multiplier factor on nominal stress for typically bends and intersection components so that the effect of geometry and welding can be considered

Table D300 Flexibility Factor, k , and Stress Intensification Factor, I

Description	Flexibility Factor, k	Stress Intensification Factor [Notes (1), (2)]		Flexibility Characteristic, h	Sketch
		Out-of-Plane, I_o	In-Plane, I_i		
Welding elbow or pipe bend [Notes (1), (3)–(6)]	$\frac{1.65}{h}$	$\frac{0.75}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$\frac{\sqrt{h_1}}{r_2}$	
Closely spaced miter bend $s < r_2 (1 + \tan \theta)$ [Notes (1), (3), (4), (6)]	$\frac{1.52}{h^{3/6}}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$\frac{\cot \theta}{2} \left(\frac{\sqrt{T}}{r_2} \right)$	
Single miter bend or widely spaced miter bend $s \geq r_2 (1 + \tan \theta)$ [Notes (1), (3), (6)]	$\frac{1.52}{h^{3/6}}$	$\frac{0.9}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$\frac{1 + \cot \theta}{2} \left(\frac{\sqrt{T}}{r_2} \right)$	
Welding tee in accordance with ASME B16.9 [Notes (1), (3), (5), (7), (8)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{3}{4} h_o + \frac{1}{4}$	$3.1 \frac{\sqrt{T}}{r_2}$	
Reinforced fabricated tee with pad or saddle [Notes (1), (3), (8), (9), (10)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{3}{4} h_o + \frac{1}{4}$	$\frac{(\sqrt{T} + \frac{1}{2} \sqrt{T_c})^{2.5}}{\sqrt{1.5} r_2}$	
Unreinforced fabricated tee [Notes (1), (3), (8), (10)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{3}{4} h_o + \frac{1}{4}$	$\frac{\sqrt{T}}{r_2}$	
Extruded welding tee with $r_o \geq 0.05 D_o$ $T_c < 1.5 T$ [Notes (1), (3), (8)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{3}{4} h_o + \frac{1}{4}$	$\left(1 + \frac{r_o}{r_2} \right) \frac{\sqrt{T}}{r_2}$	
Welded-in contour insert [Notes (1), (3), (7), (8)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{3}{4} h_o + \frac{1}{4}$	$3.1 \frac{\sqrt{T}}{r_2}$	
Branch welded-on fitting (integrally reinforced) [Notes (1), (3), (10), (11)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{0.9}{h^{2/3}}$	$3.3 \frac{\sqrt{T}}{r_2}$	

Adding Flexible Connections (Vessels)

Both WRC 107 and WRC 297 deal with “local” stress states in the vicinity of an attachment to a vessel or pipe. As indicated by their titles, WRC-107 can be used for attachments to both spherical and cylindrical shells while WRC-297 only addresses cylinder to cylinder connections. While both bulletins are used for nozzle connection. WRC-107 is based on un-penetrated shell, while WRC-297 assumes a circular opening in vessel. Furthermore, WRC-107 defines values for solid and hollow attachments of either round and rectangular shape for spherical shells but drops the solid/hollow distinction for attachments to cylindrical shells. WRC-297, on the other hand, is intended only for cylindrical nozzles attached to cylindrical shells.

Adding Flexible Connections (Vessels)

Boundary condition for using WRC 107:

To determine whether WRC 107 bulletin can be used for local stress checking the following geometry guidelines must be met:

- $d/D < 0.33$
- $D_m/T = (D-T)/T > 50$ (Here, T =Vessel Thickness, D_m =mean diameter of vessel)

Boundary condition for using WRC 297:

To determine whether WRC 107 bulletin can be used for local stress checking the following geometry guidelines must be met:

1. $d/D \leq 0.5$
2. $d/t \geq 20$ and $d/t \leq 100$ (Here t =nozzle thickness)
3. $D/T \geq 20$ and $D/T \leq 2500$
4. $d/T \geq 5$
5. Nozzle must be isolated (it may not be close to a discontinuity) – not within $2\sqrt{DT}$ on vessel and not within $2\sqrt{dt}$ on nozzle

Adding Flexible Connections (Vessels)

Difference between WRC 107 and 297:

The major differences other than the boundary conditions mentioned above are listed below:

1. WRC 107 calculates only the vessel stresses while WRC 297 calculates Vessel stresses along with nozzle stresses.
2. WRC 297 is applicable only for normally (perpendicular) intersecting two cylindrical shells whereas WRC 107 is applicable for cylindrical as well as spherical shells of any intersection.
3. The attachments for WRC 297 checking must be hollow but WRC 107 analyzes cylindrical or rectangular attachments which can be rigid or hollow.
4. WRC 297 is not applicable for nozzles protruding inside the vessel (Fig 1), Tangential Nozzle (Fig 2), Nozzle at angle (Fig 3).
5. Typically, WRC-107 is used for local stress calculations and WRC-297 is used for flexibility calculations.

Wind Loads

Wind loads are generated by multiplying the pipe exposed area, including insulation, and considering angle to the wind, by the equivalent wind pressure and the pipe shape factor. There are typically three different ways to get at the equivalent wind pressure:

- ASCE #7 (1995)
- Pressure vs. elevation table entry
- Velocity vs. elevation table entry

The total wind force on the element is calculated from

$$F = P_{eq} * S * A$$

Wind Loads

Where:

F = the total wind force on the element

P_{eq} = the equivalent wind pressure (dynamic pressure)

S = the pipe element wind shape factor

(between 0.5 and 0.7. A value of 0.65 is typical)

A = the pipe element exposed area

Wind Loads

ASCE #7 (formerly ANSI A58.1) modifies this concept slightly to consider facility importance, proximity of hurricanes, etc. Its formula for wind load is:

$$f = 0.00256 K_z (I V)^2 G_h C_d D$$

Where:

K_z = Exposure coefficient, based upon height above ground level and congestion of local terrain (varies from 0.12 for 0-15 feet height in city environment to 2.41 for 500 feet height in wide open terrain), dimensionless

Source: ASCE 7

Wind Loads

I = importance factor, based upon importance of structure and proximity to hurricane coast (varies from 0.95 for non-essential facility over 100 miles from a hurricane to 1.11 for essential facility on the hurricane coast), dimensionless

V = basic wind speed (excluding from the average abnormally high wind loading events such as hurricanes or tornadoes), from ANSI A58.1 map (ranging from 70 to 110), mph

G_h = gusting factor, based upon height above ground level and congestion of local terrain (varies from 1.0 for 500 feet height in wide open terrain to 2.36 for 0-15 feet height in city environment), dimensionless

Wind Loads

Velocity Vs. Elevation

If the user enters a velocity vs. elevation table then the velocity is converted to a dynamic pressure using the following equation:

$P = 1/2 \rho V^2$ where V is the wind velocity and ρ is the air density

Seismic Loads

ASCE #7: This standard calculates seismic g-factors in a manner similar to those of the building codes, based upon earthquake potential, structure importance, structure type, structure fundamental frequency, and soil parameters. The requirement is:

$$V = ZIKCSW$$

Seismic Loads

Where:

V = total lateral force or shear at the base, lb

Z = seismic zone coefficient:

K = structure type constant from Table 24 of ANSI A58.1, ranging from 0.67 to 2.5 (use $K=2.0$ for structures other than buildings)

Seismic Loads

$C = 1/[15 T^{(1/2)}]$, not greater than 0.12

T = fundamental period (inverse of frequency) of structure, sec

S = soil type coefficient from Table 25 of ANSI A58.1, ranging from 1.0 to 1.5 (note that the product of C and S need not exceed the value 0.14, so this value should be used as a conservative maximum).

W = total dead load

Seismic Loads

The "g" factor can be found by dividing both sides of this equation by W, so:

$$g = V/W = ZIKCS$$

For piping, the generic equation for the maximum g-factor is:

$$g = Z (1.0) (2.0) (0.14)$$

and, for the various values of Z

Seismic Loads

The "g" factor can be found by dividing both sides of this equation by W, so:

$$g = V/W = ZIKCS$$

For piping, the generic equation for the maximum g-factor is:

$$g = Z (1.0) (2.0) (0.14)$$

and, for the various values of Z

Seismic Loads

Seismic Zone	Product	g-load
4	(1)(1)(2)(0.14)	0.28
3	(3/4)(1)(2)(0.14)	0.21
2	(3/8)(1)(2)(0.14)	0.105
1	(3/16)(1)(2)(0.14)	0.0525
0	(1/8)(1)(2)(0.14)	0.035

Source: ASCE 7

The background features abstract, overlapping green geometric shapes, primarily triangles and polygons, in various shades of green, creating a modern and dynamic design. The shapes are layered, with some appearing more prominent than others, and they extend from the edges of the frame towards the center.

Thank You!