Journal of Innovative Research in Engineering and Sciences 4(5), December, 2016.
ISSN: 2141-8225 (Print); ISSN: 2251-0524 (Online)
http://grpjournal.net/index.php/joires/index.
Original Research Article Global Research Publishing.

Review of Oil and Gas Composite Piping System

(pp 559-573)

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Abstract: Continued increase in the use of vessels for storage, industrial processing, and power generation under unusual conditions of pressure, temperature, and environment has given special emphasis to analytical and environmental methods for determining their operating stresses. Creeps, fracture, fatigue, explosion, vibration, expansion, buckling loads, etc constitute factors known to cause some form of stresses and deformations or failure in composite material vessels used in oil and gas industry. Good knowledge of the nature and magnitude of these stresses enables provision of the adequate methods for their prevention and control. The study aimed at modeling the wall thickness of an oil pipeline (pressurized cylindrical vessels) used in transferring oil from one location to another. In doing so, a review of some published works on such factors as creeps, fracture, fatigue failure, expansion, buckling loads, etc was also made. A liquid flow problem was formulated and solved using a number of the existing design failure theories used in the design of oil and gas piping systems. The solved problem shows vividly how the wall thickness of an oil pipeline under fluid pressure can be determined and selected from among the existing commercial standard pipe sizes. It is hoped that this work will answer some of the questions asked by engineering students during lectures on theories and design of pressure vessels.

Key words: Modeling, composite, dynamic, pressure, response, theory

Nomenclature

σ = applied stress
σ_E = longitudinal stress in pipe prior to ground movement
σ_Y = minimum yield stress of pipe material
σ_{all} = allowable stress of pipe material
υ = Poisson ratio
V = maximum allowable velocity
e = strain
E_L = longitudinal efficiency
E = the modulus of elasticity or Young's modulus
E' = modulus of soil reaction
EI = pipe wall stiffness
I = pipe cross section moment of inertia
ε = strain
C_E = empirical constant
C_o = instantaneous creep compliance
C = creep compliance coefficient
Cu = shear strength of soil
Cs = apparent propagation velocity for seismic waves
T = retardation time
$f(\tau)$ = distribution of retardation times
t = time
T = Temperature
γ = soil density
γ = the surface energy density of a given material
γ = unit weight of backfill
γ_w = unit weight of water
G_p = the plastic dissipation (and dissipation from other sources) per unit area of crack growth.
μ = kinematic viscosity
G = the strain energy release rate
ρ = density
R or r = pipe radius
P = internal design gage pressure, (statically applied or due to fluid flow), psi or psig
t_h = minimum required wall thickness, may

exclude manufacturing tolerance and allowances for corrosion
D = diameter of pipe
y = temperature coefficient or derating factor (Table 1)
H = burial depth
h = height of water above pipe
d = offset distance from surface load to buried pipe
α = coefficient of thermal expansion
α = factor 2 for shear waves, 1 for other seismic waves
$(t_h)_c$ = commercial pipe wall thickness, in
$t_{min} = t_h$ = minimum wall required by Code, in
C = corrosion or threading allowance, in
f = pipe wall thickness fabrication tolerance
A_{pipe} = Cross sectional area of pipe
a = distance along the trench from origin of deflection
$N_{critical, perfect}$ = compressive buckling force for a perfectly straight buried pipe
$N_{cricomp, perfect}$ = compressive buckling force for an initially deformed pipeline
λ = fraction that depends on initial curvature of the pipeline
F_d = load factor 0.6 for gravel or rock dump, down to 0.15 for very loose soil
L = minimum length required
P_T = Test Pressure, psi
S_T = Stress value at test temperature, psi
S = Stress value at design temperature, psi (see ASME B31.3 Table A-1)
L = length of pipe segment
X = mid-span deflection or vertical deflection, at a distance a
F_D = design factor; unless otherwise stated is a construction derating factor for steel which depends upon the location class unit.
SG = specific gravity of the liquid (relative to water)
S = specific gravity of gas (relative to air)
Δ = change in or maximum bow at mid-span
κ = curvature of bent pipe (1/R where R is the radius of curvature)
t_e = corrosion allowance
t_{th} = thread or groove depth
Q = fluid/energy flow rate
R = gas/liquid ratio
T_{ol} = manufacturers allowable tolerance, %
W_{LR} = longitudinal weld-joint reduction factor
Z = compressibility factor, dimensionless

1.0 INTRODUCTION

Pressure vessels (leakproof containers) which abound in use are made for the production, storage and/or transportation of liquids and gases at varying temperatures and pressures. The outer shells of these vessels are made of either metallic or non-metallic materials or a combination of both materials depending on the amount of pressures/forces to overcome. Nonsteel piping material systems are used for utility systems such as those for water, sanitary or storm water, air, draining or low-pressure oil or gas service applications. However, for the vast majority of engineering constructions of “pressure” piping systems, steel pipe are used. The basic shapes of the outer shells of these vessels could either be cylindrical, conical, spherical or toroidal in shape. Pressure vessels used in automobile, aeronautics and chemical engineering experience very high internal pressures during their operation. Presently, research is still on how to model cumulative damage in composites, in particular how to take into account the loading history effects related to the chronology of high and low loads (Guedes, 2009).

Pressure vessels and piping systems are usually designed for minimum mass under strength constraints and require some preliminary mathematical considerations by which the loading conditions or the behaviour of structures under certain load conditions can be analyzed; by considering statically the effects of various loads on the structures or by considering the dynamic response of such structures under the influence of some loads. In static analysis, the effects of certain loading conditions on a structure are studied and determined while ignoring inertia and damping effects such as those caused by time varying loads. By assuming a steady loading and response conditions, the displacement stresses, strains and forces in structural components caused by loads that do not include significant inertia and damping effects can be analyzed and determined. The kinds of loading that can be applied in a static analysis (Rao et al, 2012) include: Entirely applied forces and pressures; Steady static inertial forces (e.g. gravity, rotational velocity); Imposed (non-zero) displacements; Temperatures (for thermal Strain); Fluences (for nuclear swelling)

Stresses are the forces acting in the walls of pressurized vessels. Engineer's Edge (2010) defines stress as the internal resistance or counter force offered by a material to the distorting effects of an external force or load, which depends on the direction of the applied load as well as on the plane it acts. At a given plane, both normal and shear stresses exist. Planes within a structural component under a mechanical or thermal load which contains no shear stress(es) are referred to as principal planes. Directions normal to such principal planes are principal directions and the stresses are principal stresses. In a general three-dimensional stress state there are always three principal planes along which the principal stresses act (Spence and Tooth, 1994).

Stresses are generally categorized as primary, secondary or peak stresses (Moss, 2004). Primary stresses are the most hazardous. They are stresses due to applied (internal/external) pressure, mechanical loads and wind which can cause rupture or total collapse of a pressure vessel. Secondary stresses are strain-induced stresses and can be developed at the junction of major components of a pressure vessel because of stress caused by relenting load or differential thermal expansion. Chattopadh-yay (2004) listed the different types of known stresses as pressure stresses, shear stresses, tensile stresses, radial stresses, normal stresses, bending stresses, principal stresses, tangential stresses, circumferential stresses, compressive stresses, local stresses, fatigue stresses, external stresses, tensile stresses and thermal stresses. Okafor, Abdulrahman and Ihueze (2016) design cylindrical pressure vessel for liquefied petroleum gas storage, their study also indicate that stress forces produce changes in the dimensions of the vessels in which they act. These changes in dimensions are known as strains. The value and extent of the operating stresses and strains enables the establishment of the behaviour of the material under consideration.

1.1 Mathematical Relationships Among Factors Considered in Design of Pressure Vessels

Relationships between stresses and strains due applied external loads have long been investi-gated and variously reported. The basic relation is well illustrated by the conventional tensile test specimen of Figure 1.

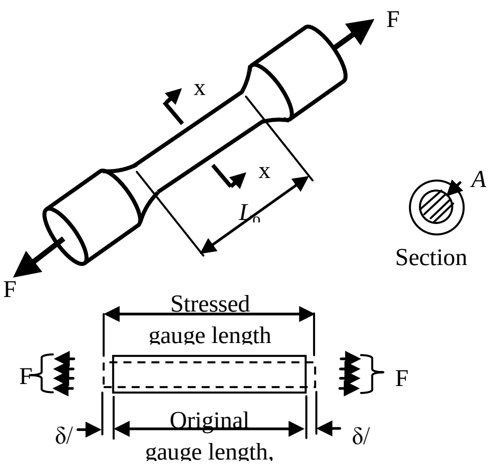

Figure 2: Tension specimen [Source: Harvey, 1987]

In terms of change in length, strain is mathematically defined as

$$e = \frac{\delta}{L_o} \qquad (1)$$

Stress is directly proportional to strain and is represented by the equation

$$E = \frac{\sigma}{e} \qquad (2)$$

Where E is the slop of the straight line portion of the stress-strain diagram of Figure 2.

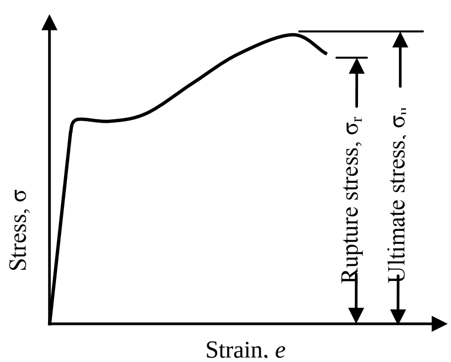

Figure 2: Determination of plastic strength

1.2 Creep, Buckling and Fatigue

Creep (sometimes called cold flow) which is the tendency of a solid material to move slowly or deform permanently under the influence of mechanical stresses can occur as a result of long-term exposure to high levels of stress that are still below the yield strength of the material. Creep is a very important aspect of material science, and it is a "time-dependent" deformation. Stages of creep are depicted in Figure 3.

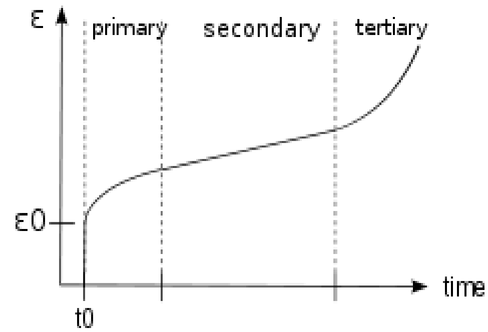

Figure 3: Strain as a function of time due to constant stress over an extended period for a viscoelastic material. [Source: Wikipedia]

The general creep equation is

$$\frac{d\varepsilon}{dt} = \frac{C\sigma^m}{d^b} e^{\frac{-Q}{kT}} \qquad (3)$$

C is a constant dependent on the material and the particular creep mechanism, m and b are exponents dependent on the creep mechanism, Q is the activation energy of the creep mechanism, d is the grain size of the material, k is Boltzmann's constant, and T is the absolute temperature.

Buckling may be defined as the failure of structure under axial compressive load; the load at which the shell structure becomes unstable under compressive loads and buckles is known as buckling load. Of all of the modes of failure, buckling is probably the most catastrophic. In design, it is a good idea to apply large safety factors to the theoretical buckling loads of compressed cylinders.

Fatigue is the weakening and eventual failure of a material as a result of repeated application of loads far smaller than those required to cause failure in a single stress/loading cycle. Fatigue results from plastic deformation (Orowan, 1939; Machlin, 1948), crack initiation and growth; and the principles of fracture mechanics may be used to predict fatigue behaviour (Harvey, 1987). Fatigue damages in poly-meric compo-sites for noncivil engineering applications have been extensively investi-gated (Reifsnider, 1979, 1991; Talreja, 1987; Jang, 1994; Martin, 1995; Tong et al, 2002).

2 REVIEW ON PRESSURE VESSEL DESIGN

2.1 Early Piping Design Equations

By early 1900s, several equations were already in use for the design of boilers, vessels and piping (Parsons, kikuchi-th.com). The United States Board of Supervising Inspectors of Steam Vessels rule applied at that time was

$$\mathrm{P} = \mathrm{k}S\frac{t_h}{3D} \qquad (4)$$

British Corporation's rule was

$$P = C(T-1)B/D \qquad (5)$$

Lioyd's rule was

$$P = C(T-2)B/D \qquad (6)$$

The Board of Trade's rule was

$$P = SB2t_h/(DC) \qquad (7)$$

In 1951, a Task Group of the American Society of Mechanical Engineers investigated a number of pressure stress equations, with the of making the design process more uniform. In seeking a balance between accuracy and simplicity, the group first assumed that as the pipe deforms under pressure the material maintains a constant volume at a material Poisson ratio ν of 0.5. This assumption enables the Saint Venant, Tresca and Von Mises equivalent stress can be written in a similar form as follows:

$$\sigma = (\sigma_\theta - \sigma_r) = \frac{\sigma_\theta - \sigma_r}{1.33} \qquad (8)$$

Maximum shear stress (Tresca):

$$\sigma = (\sigma\theta - \sigma r) = \qquad (9)$$

Maximum energy (Von Mises):

$$\sigma = (\sigma_\theta - \sigma_r)\frac{\sigma_\theta - \sigma_r}{1.15} \qquad (10)$$

The three equations can be written as

$$\sigma = \frac{\sigma_\theta - \sigma_r}{K} \qquad (11)$$

$$Where\ K = 1.0, 1.15\ or\ 1.33. \qquad (12)$$

Further still, the average principal stresses through the wall (kikuchi-th.com) are

$$\sigma_{\theta,avg} = [v - 1] \qquad (13)$$

$$\sigma_{r,avg} = - \qquad (14)$$

$$\sigma_{r,avg} = P[-0.75] \qquad (15)$$

Substituting eqns. (15) in (8) - (12) we obtain

$$\sigma = \frac{\sigma_{\theta,avg} - \sigma_{r,avg}}{K} = [-0.5] \qquad (16)$$

With adoption of the 1943 recommendation [Boardman], the 0.5 in eqn. (16) is replaced with 0.4 and *K* = 1. So, the maximum stress criterion becomes

$$\sigma = P[-0.4] \qquad (17)$$

In a more general form (17) is written as

$$\sigma = P[-y] \qquad (18)$$

The values of *y* are listed in Table 1

Table 1: Coefficient *y* for $t_h < D/6$ for temperatures *T* (°F)

Material	T ≤ 900	950	1000	1050	1100	T ≥ 1150
Ferritic Steel	0.4	0.5	0.7	0.7	0.7	0.7
Austenitic Steel	0.4	0.4	0.4	0.4	0.5	0.7
Ductile Metals	0.4	0.4	0.4	0.4	0.4	0.4
Cast Iron	0.0	-	-	-	-	-

2.2 Present Standard Piping Design Equations

To obtain the standard piping design equation for steam power piping systems for chemical plant and petroleum refinery, the ANSI/ASME Standard B31.1 and the ANSI/ASME Standard B31.3 respectively, we set the stress equal to its maximum allowable value σ_{all} multiplied by the weld quality factor E_{lw}, (where E_{lw},= E_L x W_{LR}) and rearranging the terms we obtain

$$t_h = \frac{PD_o}{2[\sigma_{all} E_{lw} + P_y]} \quad (19)$$

$$t_h = \frac{P(D + 2c)}{2[\sigma_{all} E_{lw} + P(1 - y)]} \quad (20)$$

Specifically, the ANSI/ASME Standard B31.3 code applies to major onshore and offshore facilities, providing wall-thickness formula that accommodates a high margin of safety (PetroWiki.org), as:

$$t_h = t_e + t_{th} \frac{PD_o}{2[\sigma_{all} E_{lw} + P_y]} \frac{100}{100 - T_{ol}} \quad (21)$$

The ANSI/ASME Standard B-31.3 code allows for increase in the allowable pressure for conditions that include any or both of the following:

- When the variation lasts no more than 10 hours at any one time and not more than 100 hours per year, it is permissible to exceed the pressure design at the temperature of the increased condition by no more than 33%.
- When the variation lasts no more than 50 hours at any one time and not more than 500 hours per year, it is permissible to exceed the pressure rating or the allowable stress for pressure design at temperature of the increased condition by not more than 20%.

Other important codes include:

Code for onshore oil pipeline facilities - the ANSI/ASME Standard B31.4 for liquid trans-portation systems for hydrocarbons, liquid pet-roleum gas, anhydrous ammonia, and alcohols. The wall-thickness formula as provided by this standard is stated thus

$$t_h = \frac{PD_o}{2FE_{lw} T\sigma_y} \quad (22)$$

where: t_h is inches, P in psi, D_O in inches, σ_y in psi, F = 0.72 all locations, E_{lw} = [1.0 for seam-less, ERW, double submerged arc weld and flash weld; 0.80 for electric fusion (arc) weld and electric fusion weld; 0.60 for furnace butt weld].

The ANSI/ASME Standard B31.8 code for designing natural-gas piping system facilities, such as compressor stations, gas-treatment facilities, measurement and regulation, and tank farms. The code provides wall-thickness formula as

$$t_h = \frac{PD_o}{2F_D E_{lw} T\sigma_y} \quad (23)$$

Where T is temperature derating factor.

Petrowiki.org also discussed extensively among others, the representations of design derating factor, F, and location classes for design and construction of the piping systems.

2.3 Pressurization of Fluid Storage Vessels

Vessels used in oil and gas industries can either be thin-walled or thick-walled. Such vessels consist of storage tanks, transmission pipelines, hydraulic and pneumatic cylinders, machine elements, such as rolling element bearings or gears pressed onto shafts, etc subjected to internal or external pressures, or both. Convention-ally, if

$$\frac{d_i}{t_h} > 20 \quad \text{(thin-walled cylinders)} \quad (24)$$

Accuracy from thin-wall analysis increases as di/t_h increases.

$$\frac{d_i}{t_h} < 20 \quad \text{(thick-walled cylinders)} \quad (25)$$

Next step after the inner diameter (ID) of the piping segment has been determined is to calculate the pipe wall thickness. Factors that affect pipe-wall-thickness include:

- The maximum and working pressures
- Maximum and working temperatures

- Chemical properties of the fluid
- The fluid velocity
- The pipe material and grade
- The safety factor or code design application

Petrowiki suggests that where there are no codes or standards that specifically apply to the oil and gas production facilities, the design engineer may select one of the industry codes or standards as the basis of design.

2.3.1 Wall thickness allowance

After establishing the minimum required wall thickness, the designer should add for a piping systems, a corrosion allowance and for a pipe-lines, a fabrication tolerance. The tolerance on wall thickness depends on the pipe material specifications. The commercial pipe wall size to procure is calculated thus:

$$(t_h)_c = t_h + C) + f \qquad (26)$$

2.3.2 Other pipeline design considerations

Critical pipeline performance objectives and critical engineering design parameters in pipe-line system design (Argonne, 2007) include the establishment of the following requirements:

- The throughput (volume per unit time for most petroleum products; pounds per unit time for petrochemical feedstocks);
- Origin and destination points;
- Product properties such as viscosity and specific gravity;
- Topography of pipeline route;
- Maximum allowable operating pressure (MAOP); and
- Hydraulic calculations to determine:
- Pipeline diameter, wall thickness, and required yield strengths;
- Number of, and distance between, pump stations; and
- Pump station horsepower required.

2.3.3 Cylindrical vessels under pressure

Figure 4 shows photograph of a cylindrical storage vessel used in oil and gas industry. When a cylindrical vessel, Figure 4 is subjected to internal pressure P_i, and is cut in section as depicted in Figure 5, P_i will generate three mutually perpendicular principal stresses namely circumferential (hoop) stress, radial stress and longitudinal stress in the vessel wall. If the vessel is thin-walled, it is assumed here that the stress is constant through the wall thickness and that the radial stress is small compared to the circumferential stress, considering a small element in plane stress with the principal stresses shown in Figure 5 (b).

Figure 4: Cylindrical pressure vessel [Source: Rao et al, 2012]

Figure 4 (c) is the front view of the cylinder with the forces acting on a small element due to P_i also indicated. The element has a length *dl* coming out of the paper. From literature, it has been shown that regardless of cylinder thickness, the hoop (circumferential) stress is twice the longitudinal (axial) stress.

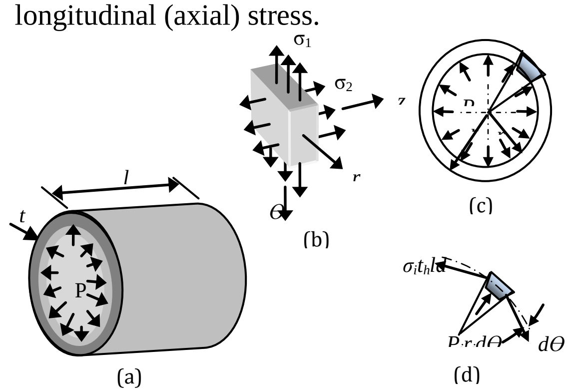

Figure 5: Schematic of the relationship between principal stresses in a pressure vessel [Source: Mahesh et al, 2014]

Thus the principal stresses for a thin-walled cylinder are

$$\sigma_1 = \sigma_\theta = \frac{P_i D_i}{2t_h} \quad (27)$$

$$\sigma_1 = \sigma_\theta = \frac{P_i D_i}{4t_h} \quad (28)$$

The radial stress varies through the wall, from P at the inner surface of the pipe to zero on the outer surface. The boundary conditions for the design of a thick-walled cylinder under internal and external pressurization are

1. $\sigma_r = -P_i$ at $r = r_i$
2. $\sigma_r = -P_o$ at $r = r_o$

The negative signs appearing in the boundary conditions imply that the pressurization is compressive. Applying the boundary conditions for a small $d\Theta$, $\sin(d\Theta/2) = d\Theta$ and tidying up gives

$$\sigma_r = \frac{P_i r_i^2 - P_o r_o^2 + (P_o - P_i)\left[\frac{r_o r_i}{r}\right]^2}{r_o^2 - r_i^2} \quad (29)$$

And

$$\sigma_r = \frac{P_i r_i^2 - P_o r_o^2 - \left[\frac{r_o r_i}{r}\right]^2 (P_o - P_i)}{r_o^2 - r_i^2} \quad (30)$$

Assuming $P_o = 0$, then eqns (29) and (30) will reduced to

$$\sigma_r = \frac{P_i r_i^2 \left[1 - \frac{r_i^2}{r^2}\right]}{r_o^2 - r_i^2} \quad (31)$$

$$\sigma_\Theta = \frac{P_i r_i^2 \left[1 + \frac{r_i^2}{r^2}\right]}{r_o^2 - r_i^2} \quad (32)$$

If $P_o > 0$ and $P_i = 0$, (29) and (30) will reduce to

$$\sigma_r = \frac{P_o r_o^2}{r_o^2 - r_i^2}\left[\frac{r_i^2}{r^2} - 1\right] \quad (33)$$

$$\sigma_\Theta = -\frac{P_o r_o^2}{r_o^2 - r_i^2}\left[\frac{r_i^2}{r^2} + 1\right] \quad (34)$$

The maximum circumferential (hoop) stress occurs at $r = r_i$ and the maximum radial stress occurs at $r = r_o$. These are expressed mathematically as

$$\sigma_{r,max} = -P_o \quad (35)$$

$$\sigma_{\Theta,max} = -\frac{2P_o r_o^2}{r_o^2 - r_i^2} \quad (36)$$

2.3.4 Spherical vessel under internal pressure

For a sphere:

$$r_1 = r_2 = r \quad (37)$$

$$\sigma_1 = \sigma_2 = \sigma \quad (38)$$

$$\text{Thus} \quad \sigma = \frac{P_i r}{2t_h} \quad (39)$$

2.3.5 Conical vessel under internal pressure

A conical vessel is represented in Figure 10, with the internal pressure and stresses acting on the walls shown. It could be seen that $r_1 = \infty$ and $r_2 = r/\cos ø$.

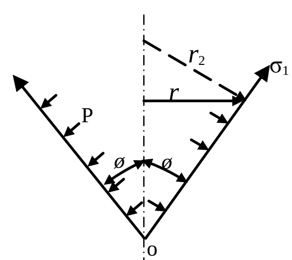

Figure 10: Stresses in a conical vessel

Thus

$$\sigma_2 = \frac{Pr}{t_h \cos ø} \quad (40)$$

From (40), it could be seen that:

(1) the hoop stress approaches that in a cylinder as ø approaches zero, and

(2) the stress becomes infinitely large as ø approaches 90^0 and the cone flattens out into a plate

$$2\pi\sigma_1 r t_h \cos ø = P\pi r_2 \quad (41)$$

$$\sigma_2 = \frac{Pr}{t_h \cos ø} \quad (42)$$

2.4 Causes of Pressure Vessels Failures

From literature, the main causes of failures of pressure vessels can be traced to any or a combination of the following factors: stress, faulty design, fatigue, creep, corrosion, erosion, cracking, poor maintenance/operator's error, change of service conditions, over temperature, operation above maximum allowable working pressures, safety valve, welding defects, impro-per installation, material(s) defect or improper, selection of material(s), low water condition, embrittlement, unknown conditions or under investigation, and unsafe modification, among others.

2.4.1 Failure prevention in Pipeline Vessels

The design and operation of gathering, trans-mission, and distribution of oil and gas in pipeline systems are usually governed by codes, standards, and regulations. (Petrowiki.org). To obtain the thickness of the pipe wall, the hoop stress, which is the largest stress in the pipe, must be limited to certain allowable stress *S*. According to the maximum shear stress theory (MSST), failure will not occur in a cylinder if

$$\sigma_1 - \sigma_3 < \frac{\sigma_y}{F_D} \rightarrow \frac{P_i D_i}{2t_h} < \frac{\sigma_y}{F_D} \qquad (43)$$

Or

$$\frac{P_i D_i}{2\sigma_y} < t_h \qquad (44)$$

Therefore, to avoid failure

$$t_h > \frac{P_i D_i}{2\sigma_y} \qquad (45)$$

Going by the Distortion-energy theory (DET), the von Mises stress for a biaxial stress state

$$\sigma_c = (\sigma_{12} + \sigma_{12} - \sigma_1\sigma_2)^{\frac{1}{2}} = \left[\frac{3{\sigma_{12}}^2}{4}\right]^{1/2} = \frac{\sigma_1}{1.15} \qquad (46)$$

Failure will occur if

$$\sigma_c < \frac{\sigma_y}{F_D} < t_h \qquad (47)$$

So, to avoid failure

$$t_h > \frac{\sigma_c F_D}{\sigma_y} \qquad (48)$$

2.5 Viscous Flow in Pipes

In a steady-state fluid flow equation it is assumed that the total mechanical energy for an incompressible, inviscid, isothermal flow with no heat transfer or work done is conserved. This is represented by the Bernoulli equation:

$$\left[\frac{V^2}{2g} + \frac{P}{\rho g} + z\right]_1 = \left[\frac{V^2}{2g} + \frac{P}{\rho g} + z\right]_1 + \Delta H_{ls} \quad (49)$$

Where the hydraulic loss between two different cross sections is the scalar quantity

$$\Delta H_{ls} = H_1 - H_2 \qquad (50)$$

Where $H_1 > H_2$ (51)

If $z_1 = z_2$, $D_1 = D_2$ and $v_1 = v_2$, the hydraulic loss

$$\Delta H_L = \frac{P_1 - P_2}{\rho g} \qquad (60)$$

The pressure change is partly due to elevation change and partly due to head loss associated with frictional effects, which are given in terms of the *friction factor f* that depends on Reynolds number and relative roughness. Thus

$$f = \varphi(R_e, \varepsilon/D) \qquad (61)$$

Meanwhile, Pipe relative roughness,

$$\varepsilon_R = \frac{\varepsilon}{D_i} \qquad (62)$$

And Reynolds number, $R_e = \frac{\rho V D_i}{\mu}$ (63)

where ρ is in lb/ft^3, D_i in ft, V in ft/s, μ in lb/ft-s

For liquids, $R_e = \frac{92.1(SG)_L Q_L}{\mu D_i}$ (64)

where: D_i is in inches, V in ft/s, μ in cp, SG for water = 1, Q_L in B/D

For gases, $$R_e = \frac{20100(SG)_g Q_g}{\mu D_i} \quad (65)$$

where: D_i is in inches, V in ft/s, μ in cp, SG relative to air (molecular weight divided by 29), Q_g in MMscf/D

Though $f = \varphi(R_e, \varepsilon/D)$, for laminar flow with $Re < 2300$, f is independent of the pipe relative roughness.

$$=> \quad f = \frac{64}{R_e} \quad (66)$$

For very large Reynolds numbers, i.e. completely turbulent flow, f is independent of the Reynolds number.

Going by Darcy -Weisbach equation:

$$\Delta H_{ls} = f\,\frac{L}{D}\frac{V^2}{2g} \quad (67)$$

Hence generally, if $z_1 \neq z_2$ and $V_1 = V_2$,

$$\frac{P_1 - P_2}{\rho g} = (z_2 - z_1) + f\,\frac{L}{D}\frac{V^2}{2g} \quad (68)$$

In terms of the internal diameter of pipe, (68) becomes

$$D_i^{\,5} = (11.5 \text{ x } 10^{-6}) f \frac{Q_L^{\,2}(SG)}{\Delta P} \quad (69)$$

For gases, with the assumptions of a steady-state flow, no work done and f = constant as a function of the length:

$$P_1^{\,2} + P_2^{\,2} = 25.5 \text{ x } \frac{ZTfLQ_L^{\,2}(SG)g}{D_i^{\,5}} \quad (70)$$

2.6 Velocity Consideration

Liquid velocity and gas velocity through a piping system can be expressed as in (71) and (72) respectively (PetroWiki. org). Thus

$$V_L = 0.012\frac{Q_L}{D_i} \quad (71)$$

$$V_G = 60\frac{Q_G TZ}{D_i^{\,2}P} \quad (72)$$

For a multiphase flow system, the erosional velocity of the mixture is defined as

$$V = \frac{C_E}{\sqrt{\sigma_m}} \quad (73)$$

$$\sigma_m = \frac{(12409)(SG)P + (2.7)RSP}{(198.7)P + ZRT} \quad (74)$$

Once the design velocity is chosen, the pipe diameter is determined from the relation

$$D_i = \left\{\frac{[11.9 + Q_L]}{1000V}\right\}^{0.5} \quad (75)$$

2.7 Buried Pipes

Oil and gas pipes can be buried under, placed above or on the surface of the ground. The decision on which to do depends on several factors: (a) buried pipes reduce plant congestion, allow for the shortest route (fewer bends) from point to point, avoid existing above ground obstructions, are protected from ambient temperature changes, are protected from wind loads, and if buried deeply, are protected from surface traffic and activities. In certain cases, burying the pipe may be the only viable option. (b) buried pipes have unique corrosion challenges that may dictate the use of coating and cathodic protection; require more elaborate repairs, with the need to locate the pipe, locate the leak, open the trench or resort to specialized trenchless repair techniques; can be accidentally damaged by digging; may leak for some time before the leak is detected; require careful trenching and backfill to avoid excessive soil settlement; and have to be designed for soil and surface loads, which requires a good understanding of the soil condition and properties.

2.7.1 Soil loading

The prism formula guides design of pipes installed in a trench with backfill. The formula states that the earth load on the pipe is equal to the weight of the soil prism right above the pipe (kikuchi-th.com). Figure 11 refers.

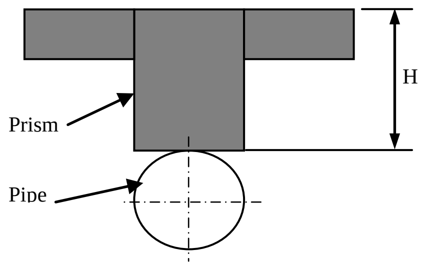

Figure 11: Soil prism above pipe
[Source: kikuchi-th.com]

Mathematically stated as

$$P_v = \gamma H \quad (76)$$

Where the pipe is below the water table, the soil pressure will be reduced by buoyancy and increased by the weight of water. Thus

$$P_v = \gamma H - 0.33\,\gamma H + \gamma_w h \quad (77)$$

If a pipe is tunneled through undisturbed soil, instead of being placed in ditch with backfill, the earth pressure is lower by a factor 2*c* (H/D) where *c* is the soil cohesion. However, kikuchi- th.com in referencing Moser, opines that whet-her the pipe is placed in a ditch or tunneled into place, the soil load on steel pressure on the pipe is 0.07 x 120 = 8.4 psi. The pressure (loads) transmitted to buried pipes crossing high-ways, runways, railroad tracks, construction sites, etc due to passage of heavy surface traffic (kikuchi-th.com) is

$$P_p = 0.48 \frac{P_s}{H^2\left[1 + \left(\frac{d}{H}\right)^2\right]^{2.5}} \quad (78)$$

Under the effect of soil and external loads, a buried pipe will tend to ovalize, causing through wall bending stresses, with

$$\sigma_b = 4E \frac{\Delta}{D} \frac{t_h}{D} \quad (79)$$

$$\frac{\Delta}{D} = \frac{0.15P}{\frac{EI}{R^3} + 0.061E`} \quad (80)$$

2.7.2 Thermal expansion and contraction

When the fluid temperature conveyed in a buried pipe differs from the soil temperature, the pipe will tend to contract or expand. In a straight pipe, fully restrained by the surrounding soil, unable to expand or contract, the temperature change will cause an axial stress

$$\sigma_A = E\alpha(T_2 - T_1) - v \frac{PD}{2t_h} \quad (81)$$

A significant rise in temperature can cause compressive stress in a buried pipe, which can cause the pipe to buckle up, what is referred to as upheaval buckling. The compressive force in the buried pipeline is

$$N = \sigma_A A_{pipe}$$
$$= \left[E\alpha(T_2 - T_1) - v \frac{PD}{2t_h}\right] \pi D t h \quad (82)$$

The critical compressive buckling force in a perfectly straight pipeline is (Friedman, WRC 425, kikuchi-th.com)

$$N_{critical\ ,perfect} = 2(EIK_e)^{0.5} \quad (83)$$

The critical compressive buckling force in a real pipeline with an initial curvature is a fraction of

$$N_{critical\ ,perfect} = \lambda N_{critical\ ,perfect} \quad (84)$$

The uplift resistance of the soil can also be expressed in terms of force per linear foot of pipeline. In a cohesionless soil (Schaminee, kikuchi-th.com)

$$P = \gamma HD\,(1 + f_d) \quad (85)$$

For cohesive soils

$$P = DC_u(1 + f_c) \le 5.14DC_u \quad (86)$$

2.7.3 Ground movement

Ground movement (either a gradual settlement or spread, or a sudden failure due for example to a landslide, an earthquake or mining operations) could cause a buried pipe to fail by plastic tension or by compressive buckling. The assess-ment of ground movement consists of two parts: first, the prediction of the deformed pipe profile; second, the resulting stresses or strains in the deformed pipe. The first part, predicting the pipe profile, is not a simple proposition. (kikuchi-th.com).

A pipe settlement X (Figure 12 refers) would be judged acceptable if it occurs over a distance at least equal to L, where (API 1117, kikuchi-th.com)

$$L = \sqrt{\frac{3.87\mathrm{x}10^7 DX + 7.74\mathrm{x}10^7 \mathrm{x}\, X^2}{F_D\sigma_y - \sigma_E}} \qquad (87)$$

The profile, along the pipe should be at least as gradual as that given by [API 1117]

$$X_a = \frac{16a^2X(L-a)^2}{L_4} \qquad (88)$$

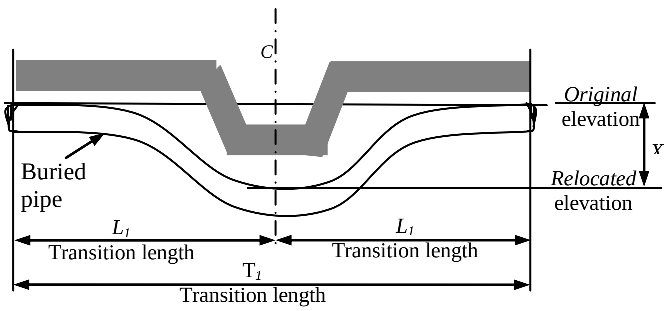

Figure 12: Mid-Span Deflection [kikuchi-th.com]

In addition to limiting the stress to F_DS_Y , the strain on the compressive side of the bent pipe should be less than the buckling strain (WRC 425, kikuchi-th.com)

$$\varepsilon_b = \frac{4\Delta}{L^2} = \frac{Dk}{2} < 2.42\left(\frac{t_h}{D}\right)^{1.6} \qquad (89)$$

2.8 Relationship among Design Pressure, Test Pressure and PSV Set Point

Design pressure, leak test pressure, and pressure relief device requirements discussed hereunder are as defined in the ASME B31.3 paragraph 302.2.4. The terms maximum allowable work-ing pressure (MAWP) and vessel within the ASME BPV Code Section VIII, Division I, are interpreted to mean design pressure and piping system respectively when applied to piping systems and in relation to pressure relieving devices.

There are various ways of leak testing piping systems. Two of such methods include the Hyrdrostatic and the Pneumatic methods. Hydrostatic Leak Test Pressure equations are defined thus:

$$P_T = 1.5P \qquad (90)$$

$$P_T = \frac{1.5PS_T}{S} \qquad (91)$$

Whereas the Pneumatic Leak Test Pressure is calculated based on 110% of the design pressure.

2.9 Seismic

2.9.1 Earthquake loading

The loading on pressure vessels and tanks due to earthquake occurs as a result of the vibratory response of the supporting deck structure. kikuchi-th.com states that earthquakes can fail buried pipes in one of two ways: either by (1) compressive buckling or a large ground movement that fails the pipe by tension at such points as corroded sections, poor weld joints and mechanical joints, or (2) a large cyclic movement caused by the passage of the seismic wave. To analyze wave passage, we assume that the upper bound of the strain in the pipe is equal to the soil strain caused by wave passage [ALA, ASCE]. Thus

$$\varepsilon_a = \frac{V_g}{\propto C_s} \qquad (92)$$

3 FLOW SYSTEM MODEL AND ANALYSIS

Oil of density, ρ = 59.31 lb/ft^3 and kinematic coefficient of viscosity, $\mu = 10^{-5}$ ft^2/s, is to be transferred from location A to location C at a flow velocity of 4.85 ft/s through a 10 inch dia-meter steel pipe which shall dip into a valley in the flow direction.

The distance of C from A is 14.3 miles and the lowest elevation point B down the valley from A is 2.2 miles. If the elevations of A, B and C are 460 ft, 86 ft and 528 ft respectively, it is required that the flow pipe be designed against maximum fluid pressure that may occur at any point along the pipeline. The operating temperature of the pipe, which should be selected from among the commercial steel pipes, should be 65^0 C.

3.1 Results and Discussions

Following are the results of the design analysis made:

R_e	=	385365.5	Eqn (63)
ε_R	=	0.000180	Eqn (62)
f	=	0.022032 (Moody Chart)	Eqn (67)
$(\Delta H_L)_{AC}$	=	704.0 ft	Eqn (67)
P_A	=	46375.3 psfg	Eqn (49)
$(\Delta H_L)_{AB}$	=	107.1 ft	Eqn (67)
P_B	=	74316.1 psfg (Fig.5)	Eqn (49)
P_i	=	516.1 psig (Fig.5)	Eqn (49)
t_h	=	0.129 in	Eqn (27)

The calculated pipe wall thickness does not include corrosion allowance and any other mechanical allowances. As can also be seen, the value is close to that of a standard pipe size of nominal bore 10 inch (DN 250 mm) with outside diameter of 273.0 mm. It is, therefore, recommended that this standard pipe size be used for the oil pipeline.

4 CONCLUSION

Loads, when applied onto a vessel, usually generate some form of stresses and strains inside the walls of the concerned vessels. A structure can respond dynamically in a given system to an action(s) produced by an applied time varying load(s). It is known that before a steady state condition is reached in a pressurized system for static considerations, a transient gradient first takes place. To ensure that all safety precautions are taken into consideration at the design stage of a pressure vessel manufacture, the need to consider those effects caused by time varying loads on a structure cannot be overemphasized. For a fiber reinforced composite pressure vessel, such reactions may come in the form of creeping, fracturing, fatigue failure, explosion, expansion, buckling, debonding etc of the constituting elements/materials. On the other hand, the loads could be a pressure force, vibration, stress corrosion mechanism, an impact etc from an external body(ies).

The study aimed at modeling the wall thickness of a lined pressurized cylindrical vessels used in transferring oil from one location to another. In doing so, a review of some published works on such factors as creeps, fracture, fatigue failure, explosion, expansion, buckling loads, etc was also made. A liquid flow problem was formulated and solved by applying a number of the existing design failure theories used in the design of oil and gas piping systems.

It is hoped that this work will assist engineering students and engineers in facilitating their understanding of the economic importance and application of stress and strain factors in the design of pressure vessels used in oil and gas industry.

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