European Journal of Mechanical Engineering Research, 11 (2),1-27, 2024
Print ISSN: 2055-6551(Print)
Online ISSN: 2055-656X(Online)
Website: https://www.eajournals.org/
Publication of the European Centre for Research Training and Development-UK

Fundamental Principles and Mechanical Design Analyses of Oil and Gas Piping Systems: An Overview

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doi: https://doi.org/10.37745/ejmer.2014/vol11n2127 Published November 21, 2024

Citation: Isaac, S. O. and Sylvanus, C. A. (2024) Fundamental Principles and Mechanical Design Analyses of Oil and Gas Piping Systems: An Overview, European Journal of Mechanical Engineering Research, 11 (2),1-27

Abstract: *This study presents the fundamental principles and mechanical design analyses of oil and gas piping systems with the objective of investigating the basis for hydrostatic and pneumatic pressure integrity tests of the systems. Piping systems investigated are the pipes, pipe components (like pipe flanges and pipe bends) and pressure vessels. They constitute the static equipment for the transportation, transmission, production, processing and storage of hydrocarbon gases and liquids, and the associated fluid systems. They also provide means for process monitoring and control; as well as guide against environmental pollution, and ensure the safety of personnel and other equipment in the facility within which they are installed and operated. The study utilized the secondary data available in the accessed related literature and presented the formulations accordingly.*

Keywords: Pipe, piping systems, pipelines, pressure vessels, oil and gas, mechanical design, pressure tests.

INTRODUCTION

The introduction of the current piping and piping systems has made the transportation of large volumes of oil and gas fluids to a distance of up to thousands of kilometres in just a few hours very easy, reducing risk of material handling, environmental pollution and theft to the barest minimum (Isaac and Nwankwojike 2016 and Isaac et al 2017). Flow of the fluids is controlled in closed networks of conduits. Pipelines are now constructed underground, on the sea beds and overheads. Pressures and flow rates are easily measured. According to Nayyar (2000) and Geiger (2000), piping is a network of connected pipes joined with flanges, fittings, valves, and other mechanical equipment or components with adequate supports. Nayyar (2000) defined a pipe as a tube with circular cross-section that conforms to the dimensional specifications in ASME B36.10M and ASME B36.19M. According to the American Society of Mechanical Engineers (ASME B31.8, 2005) a "pipe is a tubular product made for sale as a production item". It is a closed conduit of circular cross section (Liu 2003). Generally, a pipe provides a channel through which fluids flow from one point to another; this channel can be circular or rectangular in cross section. But for pressure applications, pipes of circular cross sections are preferred for uniform radial pressure distribution. Rectangular cross sectional pipes are used only in applications where radial pressure distribution is not a primary concern. So the above definitions of a pipe are concerned with pressure applications. Then the interconnection of piping and other associated components for transmission, distribution, gathering, storage, control, sampling, production and processing of fluids and fluid transportation purposes are called piping systems (ASME B31. 8, 2005). They can be referred to as pipelines, which according to Liu (2003) comprise long connected pipe segments with pumps, valves, control devices and other equipment necessary for the operation of the facility. In this study the facility is oil and natural gas, particularly, hydrocarbon and associated fluids system facility.

Furthermore, pressure vessels are structures, conduits or housings and their direct attachments including the coupling points connecting them to other equipment, designed and built to hold internal fluid pressures (Roylance 2001 and Zeman et al 2006). They are cylindrical, spherical, ellipsoidal (or a combination of these shapes) leak proof fluid containers (such as beverage bottles and the sophisticated ones encountered in engineering constructions) "with pressure differential between inside and outside" (Pendbhaje et al 2014 and Harvey 1985). According to Thattil and Pany (2017), Harvey (1985) and ASME BPVC Section VIII Div. 1 (2013) pressure vessels are subjected to both internal and external pressures that are different from the atmospheric; and they are used in the oil, chemical and many other industries for fluid storage, industrial processing and power generation. Though pressure vessels are part of piping systems, ASME BPVC Section VIII Div. 1, 2 and 3 (2013) excluded pressure vessels covered within the scope of other section of the ASME Code; process tubular heaters; equipment whose main purpose is to transfer fluid from one point to another; vessels with internal or external pressures not more than $100kPa$; vessels that are smaller than $152mm$ in internal diameter, width, height and cross-sectional diagonals; water or air containing vessels whose design pressure and temperature are respectively not more than 300 psi and 99 ℃; hot water supply storage tank of not more than 120 gal (450 ltr), and a heat input and water temperature not exceeding $200000Btu/hr$ ($58.6kW$) and 210°F(99°C) respectively heated by steam or any other indirect means; vessels for human occupancy from its scope of pressure vessels. While unfired boilers; evaporators or heat exchangers; vessels that generate steam due to the presence of heat in a system or process are included within its scope of pressure vessels. It should be noted that pressure vessels are usually composed of complete pressure-containing shell together with flanges, rings and fastening devices for connecting and securing mating parts. Therefore, for the purpose and scope of this study, their design analysis will fundamentally concentrate on these members that make the whole in relation to the associated joints.

ASME B31.8 (2005) classified piping systems as the transmission system, the distribution system, gathering system, gas storage lines and miscellaneous systems. Apart from the miscellaneous systems that are made up of instrument piping, control piping, sample piping, etc.; every other piping system constitutes pipeline network. ASME B31.8 (2005) defined pipeline as "all parts of physical facilities through which gas moves in transportation; and these include pipes, valves, fittings, flanges (including bolting and gaskets), regulators, pressure vessels, pulsation dampeners, relief valves, and other appurtenances attached to pipe, compressor units, metering stations, regulator stations and fabricated assemblies". This definition is interested in gas transmission, distribution, gathering and storage; but for the purpose of this study, it encompasses piping systems for all phases of hydrocarbon and associated fluid systems. Thus, this study is aimed at investigating the basic formulations governing the design and fluids pressure tests of piping systems for oil and gas fluid applications.

Historical Background

To understand the fundamentals of mechanical design analyses of piping systems, it is expedient to review the historical evolution pipes and pipelines, particularly, as they apply to the transportation of oil, gas and related fluid systems. Historically, the introduction of pipes in the transportation of hydrocarbon fluids dates back to 400 B. C. when the Chinese transported natural gas for lighting to Beijing with bamboo pipes wrapped with waxed cloth; meanwhile, Egypt had used clay pipes for drainage purposes as early as 4000 B.C. (Liu 2003). Antaki (2003) presented detailed evolutionary trend of pipeline technology from the Mesopotamian era through the Chinese, Egyptian, Indus valley, Cretans, Greek, Romans to the Middle Ages and beyond. The primitive approaches presented in the old days lacked the technical qualifications to reliably transport the fluids to very far distances. Liu (2003) noted that the 18th century recorded a tremendous improvement in the development of pipeline technology, marking the production of cast-iron pipes for water, sewer and gas transportation. So for the quest to improve the effectiveness, efficiency and reliability of the transportation systems of oil and natural gas fluids economically and technically, the development of pipelines was further advanced. This quest came to limelight with the advent of steel in the 19th century; then steel was introduced in the manufacture of pipes thereby improving the mechanical properties of the pipes since low carbon or low alloy steel pipelines are strong, resistant to defects, and easy to fabricate and repair (Kiefner and Trench 2001). This greatly enhanced the transportation of hydrocarbon fluids to long distances in 1879 using a six inches diameter pipe pipeline in the United States after oil was first discovered in Pennsylvania in 1858; and nine years later, an eight inches diameter long distance pipeline was constructed to transport natural gas from Pennsylvania to Buffalo (Liu 2003 and Kiefner and Trench 2001).

Kiefner and Trench (2001) and others presented the timeline in the evolution of pipeline technology in the 20th century with the first major boom in pipeline construction recorded from the late 1920s marking the formation of pipes with electric resistance-welding or flash-welding processes, hence the improvement in the reliability of the longitudinal seams; also material quality standards were developed and safe design standards were agreed upon. They noted that the late 1940s marked the beginning of the use of cathodic protection in controlling the corrosion of newly constructed pipelines; the 1950s marked the extension of cathodic protection to the older existing pipeline and the introduction of radiography in checking the integrity of welds, thereby improving the reliability of the pipelines. It is also noted that tremendous improvements were recorded in the 1960s when the use of low carbon or low alloy steels were used to manufacture tougher grades of pipes with highly reduced defects, high frequency electric resistance welding was used to form pipes thereby increasing the reliability of the longitudinal seams, then pipeline hydrostatic pressure test was introduced and corrosion control strategy was advanced by the introduction of improved methods of coating new pipes (Kiefner and Trench, 2001). The hydrostatic pressure test was being conducted on newly constructed pipelines before they were put into use. The formation of the American Society of Mechanical Engineers (ASME) in 1880 and subsequent national and international standard organizations and societies with their associated reviews in the 20th and early 21st centuries brought about streamlined standard codes for more reliable and efficient pipeline designs, maintenance and integrity management frameworks across various industries in use today (Antaki 2003).

For the specifics of this study, it is important to review the different types of pipelines that exist. Liu (2003) classified pipelines based on the commodity transported; there are water pipelines, sewer pipelines, natural gas pipelines, oil or crude oil pipelines, product pipelines, solid pipelines, etc.; based on the type of flow encountered; there are single-phase incompressible flow pipelines, single-phase compressible flow pipelines, two-phase solid-liquid mixture flow (hydro-transport) pipelines, two-phase solid-gas mixture flow (pneuma-transport) pipelines, two-phase liquid-gas flow pipelines, non-Newtonian fluids pipelines, and capsules flow pipelines; based on the environment of use; there offshore pipelines, inland pipelines, in-plant pipelines, cross-mountain pipelines, etc.; based on burial or support type; there are underground pipelines, above ground pipelines, elevated pipelines, and underwater pipelines; based on the material of manufacture; there are steel pipelines, cast iron pipelines, plastic pipelines, concrete pipelines, and other pipelines.

This study is mainly concerned with oil, natural gas and associated product steel pipelines with either single-phase or two-phase flow characteristics. PST (2015) classified them as hazardous liquid pipelines and natural gas pipelines. These are pipelines for transporting hydrocarbon fluids and other volatile flammable liquids like crude oil – sweet, sour, heavy and light oil; refined products – gasoline, diesel, heating oil, kerosene, jet fuel, etc.; natural gas; natural gas liquids and petrochemicals. They are further classified as transmission, distribution, gathering and production pipelines (PST 2015, De Wolf 2006, Wang and Economides 2009, CEPA 2015, and ASME B31.8 2005). The transmission lines are long distance pipelines of over 100,000 km in length and diameters of 100 mm to 1200 mm or more carrying oil and natural gas from production or storage points around the country or beyond at high pressures. Distribution lines are pipelines of about 12.7 mm to 152.4 mm in diameter and 450000 km in length that carry oil and natural gas to the domestic or industrial users at relatively low pressures. Gathering lines are pipelines of about 100 mm to 304 mm in diameter and 250000 km in length that transport oil and natural gas from different points of production or wells to other facilities for further processing or to transmission pipelines. The production lines are usually located near the wellhead and they are used to produce and prepare oil and natural gas for transport. They can be called the feeder lines and can range from about 152 mm to 304 mm in diameter and about 25000 km in length. It is important to note that these dimensional specifications depend largely on the size of the industry and applied location (De Wolf 2009). Thus, this study covers all the types of surface and overhead oil and gas pipelines containing flanged joints.

Design Frameworks

Since pipeline is basically a connection of several pipe sections, pressure vessels, valves, etc.; it is technically logical to analyse its design fundamentals based on the individual pipe sections and components that make up the pipeline. This review is concentrated on pipes, pressure vessels and their bolted joints; while valves in their kinds are treated either as pipes (while in open position) or pressure boundaries (while in closed position) in this study. The fundamental design analyses of pipelines (which must consider "the physical attributes, loading and service conditions, environmental factors and material-related factors to ensure the pressure integrity" of the pipelines) require the application of theories "from fluid mechanics, statics, dynamics, strength of materials, physical metallurgy, and knowledge of a number of codes and standards" (Casiglia 2000).The design considerations are concerned with the sizing, layout and dimensional specifications that conform to the predefined manufacturing codes and standards of the pipelines; the pressure and thermal changes (which may be internal or external) that affect the stress condition of the piping system, and they are often specified by the applicable design codes and standards or formulated by the designer based on the available design codes and standards. They are also concerned with understudying the internal and external operating conditions to which the piping system is exposed that eventually lead to its deterioration and failure over time; and the inherent physical and chemical properties of the material from which the piping system is made. This significantly determines the type of fluid that flows through the piping system and the environmental and operating conditions to which it can be exposed. Therefore, "required throughput; origin and destination points; product properties, such as viscosity and specific gravity; topography of pipeline route; maximum allowable operating pressure; hydraulic calculations; pipeline diameter, wall thickness and required yield strengths; number of and distance between pump stations; and pump station horsepower required" are the general pipeline design considerations (Pharris and Kolpa 2007).

These form the main pipeline design fundamentals. It is important to note that pump stations encompass both incompressible flow (pumps) and compressible flow (compressors) though the emphasis is on the incompressible flow. Liu (2003) also noted that the design of pipelines generally involves “load determination; critical performance evaluation, such as determining the stress and/or deformation, of the pipe; comparison of performance with the limiting performance criteria established by the codes and standards; and final selection of the pipe and construction method based on the design”. More so, Casiglia (2000) outlined the more common potential pressure sources that must be considered in formulating design pressures to include “the hydrostatic head due to differences in elevation between the high and low point in the system, back-pressure effects, friction losses, the shut-off head of in-line pumps, frequently occurring pressure surges, and variation in control system performance”.

According to Moss (2004), Casiglia (2000) and Smith (2007), the design analyses here are based on the provisions of ASMEB31.2 (Fuel gas Piping); ASMEB31.3 (Process Piping); ASMEB31.4 (Liquid Transportation Systems for Hydrocarbon, Liquid Petroleum Gas, Anhydrous Ammonia and Alcohols Piping) and ASMEB31.8 (Gas Transmission and Distribution Piping Systems). These codes and standards are selected among all because of their predominant applications in the oil and gas facilities. They define specific design criteria and some of them that determine the bases for pressure tests will be discussed in details. Also Casiglia (2000) pointed out that “ASME BPVC section VIII, ASME B31.8 and ASME B31.2 do not provide rules to account for overpressure transients”; “ASME B31.4 allows pressure transients of up to ten percent over the system design pressure without restricting the amount of time that the transient may act”, and “ASME B31.3 provides rules that are about midway in relative complexity from the extremes indicated above”.

COMPONENT DESIGN ANALYSES

Design Analysis of Pipe Straight Section

Oil and gas piping and piping systems comprise, among other components, the pipes, pressure vessels, valves, machineries, pipe supports and instrumentations. The fundamental design analyses here will basically be concerned with the pipes and pressure vessels, which are the components often subjected to field pressure tests. The pipes consist of the straight section, the bend (elbow) section and the flange section. The flanges are used to create inline joints between one pipe and another pipe, between pipes and other piping system components. These joints are often susceptible to leakage and therefore are subjected to leak tests after plant piping system construction and mechanical completion. So the design analyses considered, firstly, the straight section of the pipes. A schematic representation of a straight section of pressure pipeline is shown in figure (1) below. This indicated the direction of the three principal stresses (hoop, circumferential or tangential stress, radial stress and longitudinal or axial stress) to which the wall of a typical pipeline is subjected due to internal pressure generated by the operating fluids.

Print ISSN: 2055-6551(Print)
Online ISSN: 2055-656X(Online)
Website: https://www.eajournals.org/
Publication of the European Centre for Research Training and Development-UK

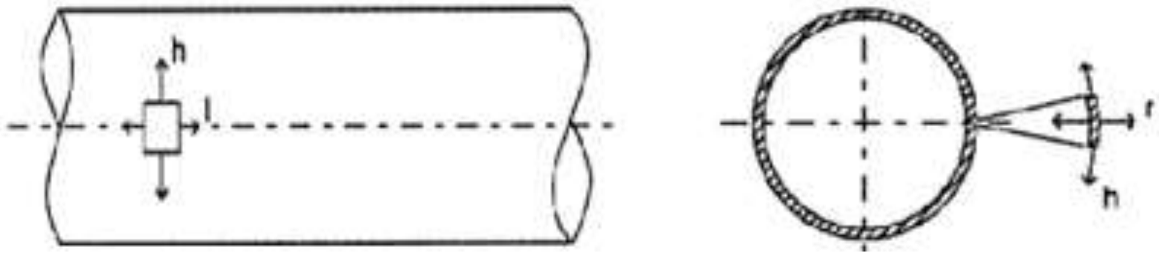

Fig. 1: A pipe section showing hoop (h), longitudinal (l) and radial stress (r) directions (Antaki 2003)

According to Khurmi and Gupta (2005) "the design of pipes involves the determination of inside diameter of the pipe and its wall thickness". Thus, the inside or internal diameter is given as a function of the fluid flow rate and flow velocity in equation (1):

$$D_i = 1.13\sqrt{\frac{Q}{v}} \tag{1}$$

The internal diameter is related with the outer diameter and mean diameter according to Liu (2003), thus:

$$D_i = 2D - D_o \tag{2}$$

$$D = \frac{D_i + D_o}{2} \tag{3}$$

Then thickness of the wall of the pipe depends on its internal diameters, internal fluid pressure and the allowable circumferential and longitudinal stresses. It may be obtained from the thin or thick cylindrical shell formula (depending on the ratios of the internal diameter to the wall thickness and internal fluid pressure to the allowable stress) since the pipe is cylindrical in shape (Ibid). For thin wall pipes, the internal diameter must be greater than 20 times the wall thickness (i.e. $\frac{D_i}{t} > 20$) and the internal fluid pressure must be less than $1/_6$ times the allowable hoop stress (i.e. $\frac{S}{P} > 6$); for thick wall pipes, the internal diameter must be less than or equal to 20 times the wall thickness (i.e. $\frac{D_i}{t} \leq 20$) and the internal fluid pressure must be greater than or equal to $1/_6$ times the allowable hoop stress (i.e. $\frac{S}{P} \leq 6$) (Khurmi and Gupta 2005 and Rajput 2006).

Based on the foregoing, the wall thickness of the pipe is related to the internal pressure, pipe diameter and stresses on the wall as shown in figure (2) using the Barlow's pipe equation as follows (Menon 2005, Antaki 2003, Escoe 2006, McAllister 2009, Liu 2003, Khurmi and Gupta 2005 and Casiglia 2001):

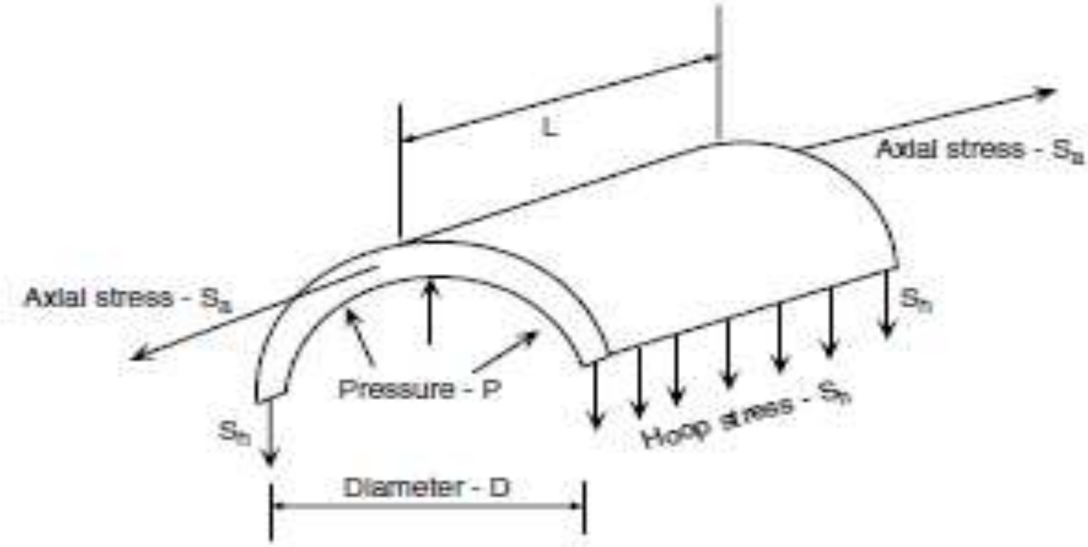

Figure 2: Stresses on the wall of a pipe due to internal pressure (Menon 2005)

$$t = \frac{PD_i}{2S_h} \quad (4)$$

$$t = \frac{PD_i}{2S_h\eta_a} \quad (5)$$

$$t = \frac{PD_i}{4S_a\eta_h} \quad (6)$$

$$t = \frac{PD_i}{2S_h} + C \quad (7)$$

The hoop or circumferential stress is given by the equation,

$$S_h = \frac{PD_i}{2t} \quad (8)$$

Where D_i= the internal diameter of the pipe (mm), D_o= the outer diameter of the pipe (mm), D= the mean diameter of the pipe (mm), Q= the fluid flow rate (m^3/s), v= the fluid flow velocity (m/s), t= the thickness of the pipe wall (mm), P= the internal pressure on the wall of the pipe (N/m^2), S_h= the hoop, tangential or circumferential stress (N/m^2), S_a= the axial or longitudinal stress (N/m^2), S= the allowable hoop stress (N/m^2), η_a= the axial or longitudinal joint efficiency, η_h= the hoop or circumferential joint efficiency, C= Weisback constant whose values for different pipe materials are given in table 3.1 below (Ibid);

Table 1: Values of constant (C)

Material	Cast iron	Mild steel	Zinc and Copper	Lead
Constant (C) in mm	9	3	4	5

For oil and gas pipeline, the hoop stress must be limited to a certain value of allowable stress (S) (Antaki 2003), hence;

$$S > \frac{PD_i}{2t} \quad (9)$$

$$S = 0.72(S_y)E \quad (10)$$

$$S = (S_y)FET \quad (11)$$

The axial or longitudinal stress is given by the equation;

$$S_a = \frac{PD_i}{4t} \tag{12}$$

Where S_y= the specific minimum yield stress (N/m^2), E= the welded joint factor, F is the design factor, T= the temperature derating factor.

Table 2: Examples of Yield and Ultimate Stress (Ibid)

Temperature (°F)	A 106 Gr. B S_y (ksi)	A 106 Gr. B S_u (ksi)	A 312 T. 304 S_v (ksi)	A 312 12 T. 304 S_u (ksi)
100	35.0	60.0	30.0	75.0
200	31.9	60.0	25.0	71.0
300	31.0	60.0	22.5	66.0
400	30.0	60.0	20.7	64.4
500	28.3	60.0	19.4	63.5

Table 3: Examples of Longitudinal Welded Joint Factors, E (Ibid)

Material	Pipe Class	E
ASTM A53, A106	Seamless	1.0
ASTM A53	ERW	1.0
ASTM A53	Furnace Butt Welded	0.6
ASTM A134	Electric Fusion Arc Welded	0.8
ASTM A135	Electric Resistance Welded (ERW)	1.0
API 5L	Seamless	1.0
API 5L	Submerged Arc Welded or ERW	1.0
API 5L	Furnace Butt Welded	0.6

Table 4: Location Design Factor, F (Ibid)

Location	F
Class 1 Div. 1: Deserts, Farm Land, Sparsely Populated, Etc.	0.80
Class 1 Div. 2: Class 1 with line tested to 110% design	0.72
Class 2: Industrial areas, town fringes, ranch, etc.	0.60
Class 3: Suburban housing, shopping centres, etc.	0.5
Class 4: Multi-storey buildings, heavy traffic, etc.	0.4

Table 5: Temperature Derating Factor, T (Ibid)

Temperature (°F)	T
250 or less	1.000
300	0.967
350	0.933
400	0.900
450	0.867

Applying the Lame's equation for thick cylindrical shell the hoop, radial and axial stresses at any radius, X (noting that hoop and radial stresses are maximum at $X = R_i$ and minimum at $X = R_o$, hoop stress is always tensile while radial stress is compressive) are calculated according to (Khurmi and Gupta 2006 and Antaki 2003) as:

$$S_h = \frac{PR_i^2}{R_o^2 - R_i^2}\left[1 + \frac{R_o^2}{X^2}\right] \tag{13}$$

$$S_r = \frac{PR_i^2}{R_o^2 - R_i^2}\left[1 - \frac{R_o^2}{X^2}\right] \quad (14)$$

$$S_h = \frac{PR_i^2}{R_o^2 - R_i^2} \quad (15)$$

where R_i and R_o are the inner and outer radii respectively. The maximum and minimum hoop stresses are respectively given as;

$$S_{h(\max)} = \frac{P[R_i^2 + R_o^2]}{R_o^2 - R_i^2} \quad (16)$$

$$S_{h(min)} = \frac{2PR_i^2}{R_o^2 - R_i^2} \quad (17)$$

$$S_{r(\max)} = -P \text{ (compressive)} \quad (18)$$

$$S_{r(\min)} = 0 \quad (19)$$

According to Khurmi and Gupta (2005), in designing pressure vessels and pipes of brittle materials, the maximum normal stress theory is applied, the maximum shear stress theory is applied for pressure vessels and pipes made of ductile materials; thus using the Lame's thick cylindrical shell formula, the wall thickness for brittle and ductile materials is respectively given as;

$$t = R_i\left[\sqrt{\frac{S_h+P}{S_h+P}} - 1\right] \quad (20)$$

$$t = R_i\left[\sqrt{\frac{\tau}{\tau-P}} - 1\right] \quad (21)$$

And since shear stress (τ) is usually taking as half the tensile stress (S_h), equation (21) can then be written as (Ibid),

$$t = R_i\left[\sqrt{\frac{S_h}{S_h-2P}} - 1\right] \quad (22)$$

Also for high pressure oil and gas pipes, the thickness of the pipe wall can be calculated using the Barlow' equation (Ibid);

$$t = \frac{PR_o}{S_h} \quad (23)$$

Then for brittle materials,

$$S_h = 0.125S_u \quad (24)$$

And for ductile materials,

$$S_h = 0.8S_y \quad (25)$$

Table 6: Values of S_h for pipes of different materials (Ibid)

S/NO.	Pipes	Allowable tensile stress (S_h) in N/mm^2
1	Cast iron steam or water pipes	14
2	Cast iron steam engine cylinders	12.5
3	Lap welded wrought iron tubes	60
4	Solid drawn steel tubes	140
5	Copper steam pipes	25
6	Lead pipes	1.6

The maximum shear stress, according to Antaki (2003), is given by;

$$\tau_{max} = \frac{S_h - S_r}{K} \tag{26}$$

Considering the average principle stresses

$$\tau_{max} = \frac{P}{K}\left[\frac{D}{2t} - 0.5\right] \tag{27}$$

Where $K = 1.00$ for maximum shear stress (Tresca), $K = 1.15$ for maximum energy (Von Mises) and $K = 1.33$ for maximum strain energy (Saint Venant); 0.4 was recommended to replace 0.5 with $K = 1.00$ in equation (27), then;

$$\tau_{max} = P\left[\frac{D}{2t} - 0.4\right] \tag{28}$$

To accommodate wider range of operating temperature variations, equation (28) is written

$$\tau_{max} = P\left[\frac{D}{2t} - y\right] \tag{29}$$

where the values of y for different operating temperatures are given in table (7) below. Hence the minimum required wall thickness is calculated according to equation (31) or (32), thus;

$$t_{min} = \frac{PD_o}{2(SEW + Py)} \tag{30}$$

$$t_{min} = \frac{P(D_i + 2c)}{2[SEW - P((1 - y)]} \tag{31}$$

$$t_{min} = t + c \tag{32}$$

Where E= the joint efficiency factor, S= the maximum allowable stress in material (N/mm^2), y= the temperature coefficient, c = the sum of mechanical allowances (thread or groove depth) plus corrosion and erosion allowances, W= weld joint strength reduction factor (given in table 8 below). It should be noted that 0.5mm shall be assumed in addition to the specified depth of the cut for machined surfaces or grooves where the tolerance is not specified, while the nominal thread depth shall apply for threaded components. The coefficient, y, in equations (32) and (33) above is valid for $t < \frac{D_o}{6}$ and for the materials shown in table 2.8 below; but for $t \geq \frac{D_o}{6}$ or $\frac{P}{SE} > 0.385$,

$$y = \frac{D_i + 2c}{D_o + D_i + 2c} \tag{33}$$

Where CrMo = chromium – molybdenum alloy, CSEF = creep strength enhanced ferritic, N + T = normalizing + tempered PWHT, PWHT = post-weld heat treatment, AW = autogenous welds in austenitic stainless grade 3XX, and N088XX and N066XX nickel alloys, AS = austenitic stainless grade 3XX and N088XX nickel alloys.It should be noted that for carbon steel, $W = 1.0$ for all temperatures; but for other materials than carbon steel, CrMo, CSEF, and austenitic alloys listed in table (7) above, the following values of W shall be used; for $T_i \leq T_{cr}$, $W = 1.0$;for $T_{cr} < T_i \leq 1,500$°F,

$$W = 1 - 0.000909(T_i - T_{cr}) \tag{34}$$

Table 7: Coefficient y for $t < \frac{D}{6}$ for temperatures T (°F) (ASME 31.3 2016)

Materials	$T \leq 900$	950	1000	1050	1100	1150	1200	$T \geq 1150$
Ferritic steel	0.4	0.5	0.7	0.7	0.7	0.7	0.7	0.7
Austenitic steel	0.4	0.4	0.4	0.4	0.5	0.7	0.7	0.7

Nickel alloys	0.4	0.4	0.4	0.4	0.4	0.4	0.5	0.7
Gray iron	0.0	…	…	…	…	…	…	…
Other ductile materials	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4

Table 8: Weld joint strength reduction factor, W (ASME B31.3 2016)

Component Temperature, T_i, in °C (°F)															
Steel Group	427 (800)	454 (850)	482 (900)	510 (950)	538 (1000)	566 (1050)	593 (1100)	621 (1150)	649 (1200)	677 (1250)	704 (1300)	732 (1350)	760 (1400)	788 (1450)	816 (1500)
CrMo	1	0.95	0.91	0.86	0.82	0.77	0.73	0.68	0.64	…	…	…	…	…	…
CSEF(N+T)	…	…	…	1	0.95	0.91	0.86	0.82	0.77	…	…	…	…	…	…
CSEF	…	…	1	0.5	0.5	0.5	0.5	0.5	0.5	…	…	…	…	…	…
AW	…	…	…	1	1	1	1	1	1	1	1	1	1	1	1
AS	…	…	…	1	0.95	0.91	0.86	0.82	0.77	0.73	0.68	0.64	0.59	0.55	0.50
Other materials	…	…	…	…	…	…	…	…	…	…	…	…	…	…	…

The requirements for designing high pressure piping systems differ slightly from those reviewed above. High pressure is relative term in that its definition is based on application. Though, on grounds of personnel safety it can be defined as the pressure absolute value does not exceed the prevailing atmospheric pressure. However, ASME B31.3 (2016) considers high pressure as "pressure in excess of that allowed by the ASME B16.5 Class 2500 rating for the specified design temperature and material group". It recommends that the "allowable stress is values at design temperature for materials shall not exceed the lower of two-thirds of S_y at room temperature and two-thirds of S_{yt}", and S_{yt} can be determined from equation (35);

$$S_{yt} = S_y R_y \tag{35}$$

where R_y = ratio of the average temperature dependent trend curve value of yield strength to the room temperature yield strength,S_{yt} = yield strength at room temperature. The calculated displacement stress range shall not exceed the allowable displacement stress range given as;

$$S_A = 1.25S_c + 0.25S_h \tag{36}$$

where S_c = allowable stress at minimum metal temperature expected during the displacement cycle and analysis, S_h = allowable at maximum metal temperature expected during the displacement cycle and analysis. The appropriate equations for determining the required minimum thickness of the straight section of pipes reviewed above also applies in the high pressure pipes. However, the internal pressure design minimum wall thicknesses, according to ASME B31.3 (2016), for solution heat treated austenitic stainless steels and certain nickel alloys with similar stress – strain behaviour for pipes with specified outside and inside diameters are respectively given thus;

$$t = \frac{(D_o - 2c_o)}{2}\left[1 - e^{\left(\frac{-P}{S}\right)}\right] \tag{37}$$

$$t = \frac{(D_i + 2c_i)}{2}\left[e^{\left(\frac{P}{S}\right)} - 1\right] \tag{38}$$

The internal design pressure for the scenarios may be determined as follows;

$$P = S\ln\left[\frac{D_o - 2c_o}{D_o - 2(t - c_i)}\right] \tag{39}$$

$$P = S\ln\left[\frac{D_i + 2(t - c_o)}{D_i + 2c_i}\right] \tag{40}$$

Alternatively, the above internal pressure design minimum wall thicknesses and internal design pressures are calculated thus;

$$t = \frac{(D_o - 2c_o)}{2}\left[1 - e^{\left(\frac{-1.155P}{S}\right)}\right] \quad (41)$$

$$t = \frac{(D_i + 2c_i)}{2}\left[e^{\left(\frac{1.155P}{S}\right)} - 1\right] \quad (42)$$

$$P = \frac{S}{1.155}\ln\left[\frac{D_o - 2c_o}{D_o - 2(t - c_i)}\right] \quad (43)$$

$$P = \frac{S}{1.155}\ln\left[\frac{D_i + 2(t - c_o)}{D_i + 2c_i}\right] \quad (44)$$

Design Analysis of Pipe Bend Section

The above equations are suitable for designing the straight section of pipelines. However, Peng and Peng (2009) studied the effect of thermal expansion of the pipe and noted the need to provide enough flexibility in the piping design to cushion the effect of the axial stress due to thermal expansion of straight pipes arising from temperature fluctuations.

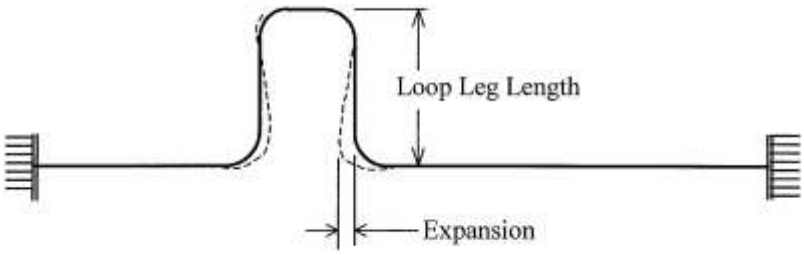

Fig. 3: Pipe expansion loop (Peng and Peng 2009)

The required flexibility was provided by creating expansion loop at a point along a straight pipe connecting two rigid terminals. The loop is a portion of the pipe that was bent at right angle to the straight piping (Peng and Peng 2009) as show in figure (3) above. They state that the expansion of an ideally anchored pipe and a really anchored pipe due to temperature variations is respectively given in equations (45) and (46), noting that both anchor and pipe receive the same expansion force (Ibid);

$$\Delta = \alpha(T_2 - T_1)L = eL = \frac{LS_a}{E} = \frac{LF_a}{EA} \quad (45)$$

$$\Delta = \alpha(T_2 - T_1)L = \Delta_1 + \Delta_2 = \frac{F_a}{k} + \frac{LF_a}{EA} = F_a\left(\frac{1}{k} + \frac{L}{EA}\right) \quad (46)$$

Thus the expansion force is given as;

$$F_a = \frac{\alpha(T_2 - T_1)L}{L\left(\frac{1}{k} + \frac{L}{EA}\right)} = \frac{\alpha(T_2 - T_1)EA}{\left(\frac{EA}{kL} + 1\right)} \quad (47)$$

The cantilever formula was also applied to determine the axial stress due to the thermal expansion, and hence the length of the loop leg as shown equations (48) and (49) respectively (Ibid);

$$S_a = \frac{M}{Z} = \frac{6EI\Delta}{ZL^2} = \frac{6E\pi R^3 t\Delta}{\pi R^2 tL^2} = \frac{6ER\Delta}{L^2} = \frac{3ED\Delta}{L^2} \quad (48)$$

$$L = \sqrt{\frac{3ED\Delta}{S_a}} = 66\sqrt{D\Delta} \tag{49}$$

given that $E = 29.0 \times 10^6 psi$ and $S_a = 20{,}000\ psi$.

Peng and Peng (2009) also noted that AMSE B31 piping code provided a criterion as a measure of adequate flexibility, as long as the requirements of the code that if "the piping system is of uniform size, has not more than two anchors and no intermediate restraints, is designed for essentially non-cyclic service (less than 7000 total cycles), and satisfy the following approximate criterion", $Dy/(L-U)^2 \leq 0.03$ (for imperial units) or $Dy/(L-U)^2 \leq 208.3$ (for SI units), are met, "no formal expansion flexibility analysis is required"; where D is nominal pipe size (mm), y is resultant of movement to be absorbed by piping system (mm), L is developed length of piping system between anchors (m) and U is anchor distance (i.e. length of straight line joining anchors) (m), $(L-U)$ is the leg length that is perpendicular to the line of expansion, e is the strain caused by the squeezing movement of the expanded piping system (m), α is thermal expansivity of the pipe (K^{-1}), Δ_1 is part of the pipe expansion absorbed by the real anchor (m), Δ_2 is part of the pipe expansion absorbed by the pipe itself (m) and k is the stiffness constant of the anchor (N/m).

More so, ASME B31.3 (2016), Hwang et al (2020) and Chen et al (2015) recommended that the minimum required wall thickness of the bend (figure 4), in its finished form, shall be determined with equations (32) and (33) above. Then t, for curved pipes is given by equation (50) below.

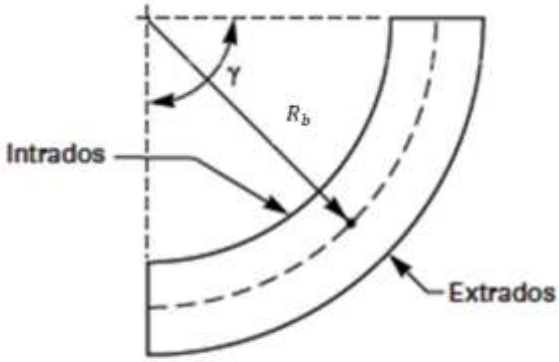

Figure 4: Nomenclature for pipe bends (ASME B31. 3 2016)

$$t = \frac{PD_o}{2\left[\left(\frac{SEW}{I}\right)+Py\right]} \tag{50}$$

where at the intrados;

$$I = \frac{4R_b - D_o}{4R_b - 2D_o} \tag{51}$$

and at the extrados;

$$I = \frac{4R_b + D_o}{4R_b + 2D_o} \tag{52}$$

$I = 1.0$ at the bend centreline on the sidewall, R_b = bend radius of welding elbow or pipe bend. In the analysis of the effect of pipe bending, Liu (2003) noted that the pipe buckles if the bend radius is less than the of equation (53), thus;

$$R_b = \frac{D^2}{1.12t} \tag{53}$$

Then considering the internal pressure loading of the pipe bend, Abdulhameed (2018) reviewed that for a smooth pipe bend that has constant wall thickness and initial cross section, the longitudinal and hoop stresses for a toroidal shell are given in equations (54) and (55) respectively;

$$S_a = \frac{PR_i}{2t} \tag{54}$$

$$S = \frac{PR_i}{2t}\left[\frac{2\rho+\sin\emptyset}{\rho+\sin\emptyset}\right] \tag{55}$$

where the radius ratio, ρ is given by,

$$\rho = \frac{R_b}{R_i} \tag{56}$$

and "the circumferential angle", $\emptyset$ "measured from the crown (i.e. $\emptyset = 0$) towards the extrados (i.e. $\emptyset = \pi/2$) and the intrados (i.e. $\emptyset = 3\pi/2$) according to figure (5) below.

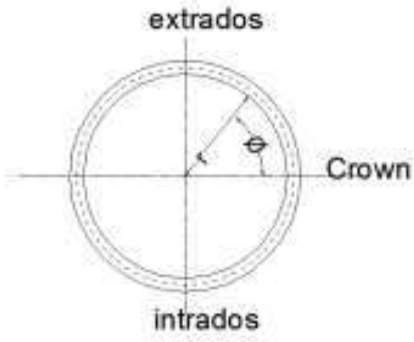

Fig. 5: Section of toroidal shell (Abdulhameed, 2018)

Christo et al (2017) pointed out that bending causes a to lose its circular section to ovular cross section, thereby increasing the thickness of the inner surface and reducing the thickness of the outer surface. Thus the consequent uneven distribution of stress at the bend weakens the piping system at the bend section as a result of cyclic accumulation of strain (i.e. "ratcheting or fatigue") at that point. It was noted that the above analysis considered only "the effect of toroidal shape on the hoop stress", but did not consider the deformation of the cross section due to bending and the stress due "the additional outward forces on the pipe bend" (Abdulhameed 2018). Then, since the curvature of the pipe affects only the hoop stress, the thin-walled theory of pipe bend considering elastic stress under internal pressure indicated that the hoop stress is given as (Ibid);

$$S = \frac{PR_i}{t}\left[\frac{2R_b+\mathrm{R}\cos\emptyset}{2R_b+2\mathrm{R}\cos\emptyset}\right] \tag{57}$$

where the circumferential angle was measured from the intrados (i.e. $\emptyset = 0$) towards the extrados (i.e. $\emptyset = \pi$). This change in the cross section of a pipe from circular to ovular due to bending is known as ovulization (figure 2.6); and it results to increase of flexibility, increase in longitudinal bending stress and creation of circumferential shell bending stress (Peng and Peng 2009, Polenta et al 2015 and Bhende and Tembhare 2013). The results of the theoretical analysis of these phenomena provided the models for the computation of bend flexibility factor (k_{bf}) and stress intensification factor (i) in each instance.

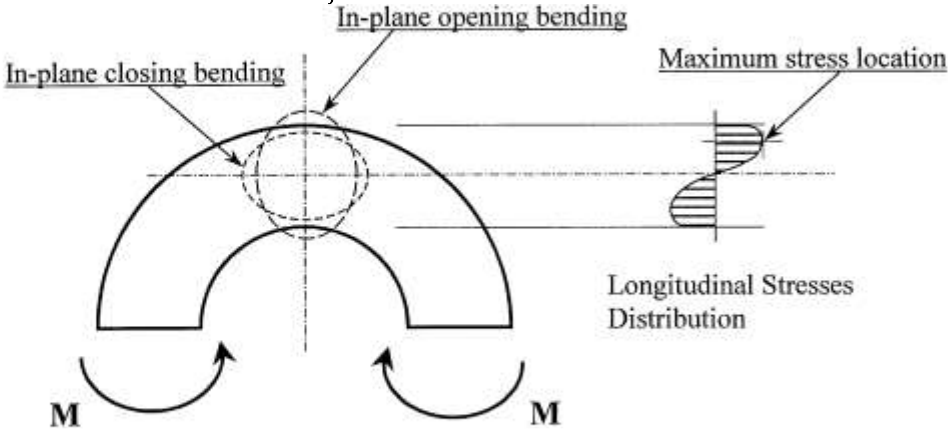

Fig.6: Ovulization of bend under external bending

Thus the bend flexibility and stress intensification factors are respectively given as;

$$k_{bf} = \frac{1.65}{h} \quad (58)$$

$$i_S = \frac{S_a}{M/Z} \quad (59)$$

where h is the bend flexibility characteristic given by,

$$h = \frac{tR_b}{R_m^2} \quad (60)$$

The theoretical longitudinal stress intensification factors (SIFs) for in-plane (i_{Li}) and out-plane (i_{Lo}) bending are respectively given as;

$$i_{Li} = \frac{0.84}{\sqrt[3]{h^2}} \quad (61)$$

$$i_{Lo} = \frac{1.08}{\sqrt[3]{h^2}} \quad (62)$$

Also the theoretical circumferential stress intensification factors (SIFs) for in-plane (i_{li}) and out-plane bending are respectively given as;

$$i_{Ci} = \frac{1.80}{\sqrt[3]{h^2}} \quad (63)$$

$$i_{Co} = \frac{1.50}{\sqrt[3]{h^2}} \quad (64)$$

where R_m is mean radius of the pipe. Note that the given flexibility factor applies to both in-plane and out-plane bending, while the theoretical longitudinal and circumferential stress intensification factors apply only to nuclear piping systems; and for other piping applications, only half of the theoretical values are used (Ibid). So the minimum recommended thickness before bending, for each bend radius is given in table (9). The design specification for miter bends and other components for piping flexibility under internal pressures are provided in the related American Society of Mechanical Engineers (ASME) B31 piping codes.

Table 9: Pipe thickness for bending (Peng and Peng 2009)

R_b	t
$\geq 6D$	$1.06t_m$
$= 5D$	$1.08t_m$
$= 4D$	$1.14t_m$
$= 3D$	$1.25t_m$

Design Analysis of Pressure Vessels

Fundamentally the design analysis of pressure vessels considers the type pressure vessels – cylindrical, spherical and ellipsoidal – inthe determination of the geometrical parameters.However, the cylindrical types with different end geometry (as shown in figure 2.7 below) are most popularly used in the oil and gas industries. When designing pressure vessels, the following factors should be considered (Toudehdehghan and Hong 2019): dimensional parameters; operating conditions; availability of materials in the market; corrosive nature of vessel contents; theories of failure; construction methods; fabrication methods; fatigue, creep and brittle fracture; and economic factors.

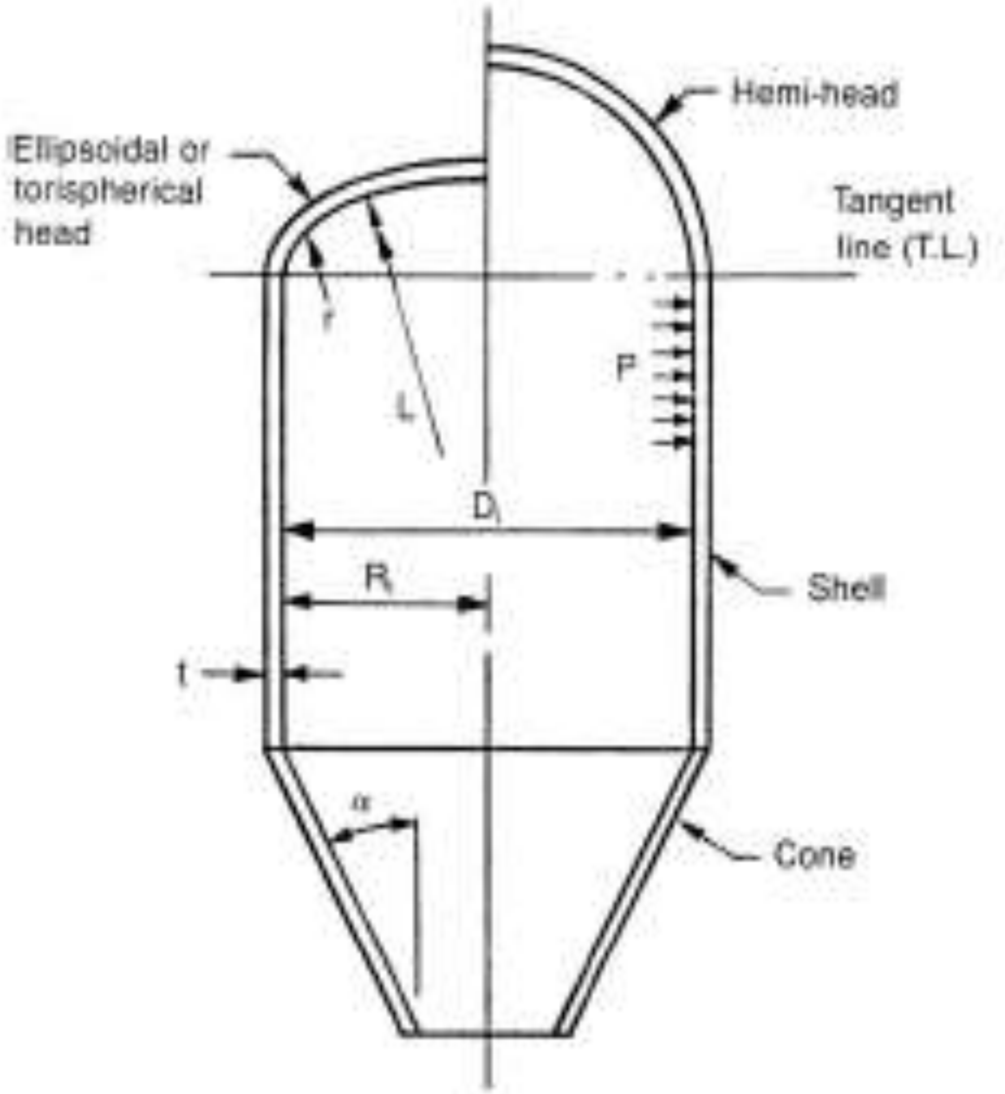

Fig. 7: General configuration and dimensional data for pressure vessel shells and heads

For the purpose of this study it is based on the rules specified in"rules for construction of pressure vessels" – ASME BPVC-VIII-1, ASME BPVC-VIII-2 and ASME BPVC-VIII-3 (Moss 2004, Moss and Basic 2013, Casiglia 2000 and Smith 2007).As in the case of pipes, Khurmi and Gupta (2006) applied the thin and thick cylindrical shells in the design of pressure vessels. Harvey (1985) noted that the applications of pressure vessels under unusual conditions of temperature, pressure and environment gives special emphasis to the analytical and experimental methods for determining their operating stresses. The values of the geometrical parameters of the vessels depend largely on their structural configurations. According to Thattil and Pany (2017);

a) the volume of cylindrical pressure vessels with tori-spherical head is given by:

Print ISSN: 2055-6551(Print)

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$$V = \frac{\pi {D_i}^2 (3H_c + 2D_{ic}K)}{12} \quad (59)$$

$$K = \frac{R_{id}}{R_i} - \sqrt{\left(\frac{R_{id}}{R_i} - 1\right)\left(\frac{R_{id}}{R_i} + 1 - \frac{2r}{R_i}\right)} \quad (60)$$

b) the volume of cylindrical pressure vessel with hemispherical head is given by:

$$V = \frac{\pi {D_i}^2 (3H_c + 2D_i)}{12} \quad (61)$$

where R_i = internal radius of cylindrical shell, R_{id} = internal radius of dome,r = radius of knuckle, D_i = internal diameter of cylindrical shell, H_c = height of cylindrical shell.

Also the volume of cylindrical pressure vessel with ellipsoidal or torispherical head is given by:

$$V = \frac{\pi (3{D_i}^2 H_c + 8ABC)}{12} \quad (62)$$

where A, B and C are lengths of the principal axes of full ellipsoid.
The volume of cylindrical pressure vessel with flat end is given by:

$$V = \frac{\pi {D_i}^2 H_c}{4} \quad (63)$$

Then the internal diameter of the vessel can be calculated from any of the equations.

Therefore, the associated wall thickness, internal fluid pressure and the principal stresses are calculated according to Moss (2004), Moss and Basic (2013) and ASME BPVC Section VIII Div. 1 (2013):

a) considering cylindrical shell subjected to circumferential stress (longitudinal joints) due to internal fluid pressure for $P < 3000\, psi$, and $t \leq 0.25D_{ic}$ or $P \leq 0.385SE$;

$$t = \frac{PD_i}{2SE - 1.2P} \quad (64)$$

b) considering cylindrical shell subjected to longitudinal stress (circumferential joints) due to internal fluid pressure for $P < 3000\, psi$, and $t \leq 0.25D_{ic}$ or $P \leq 1.25SE$;

$$t = \frac{PD_i}{4SE + 0.8P} \quad (65)$$

c) considering wholly spherical (hemispherical) shell subjected to longitudinal (equal to circumferential) stress internal fluid pressure for $P < 3000\ psi$, and $t \leq 0.178D_i$ or $P \leq 0.665SE$;

$$t = \frac{PD_i}{4SE + 0.4P} \quad (66)$$

d) considering ellipsoidal head shell with $S_t > 80000\ psi$subjected to internal fluid pressure;

$$t = \frac{PD_iK}{2SE + 0.2P} \quad (67)$$

$$K = 0.167\left[2 + \left(\frac{D_i}{2H_c}\right)^2\right] \quad (68)$$

where P = internal pressure (psi),D_i = inside diameter of the cylindrical shell ($in.$), S = allowable or calculated stress (psi),E = joint efficiency,R_i = inside radius ($in.$), K = coefficient,t = thickness of shell ($in.$),r = radius of knuckle ($in.$).

European Journal of Mechanical Engineering Research, 11 (2),1-27, 2024
Print ISSN: 2055-6551(Print)
Online ISSN: 2055-656X(Online)
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Design Analysis of Pipe Flanged Joints

The design analysis of pipe flanges is based on the detailed design criteria and rules specified in ASME B16.5 (2009), ASME BPVC-VIII-1 (2013), ASME BPVC-VIII-2 (2013), ASME BPVC-VIII-3 (2013) and ASME B31.3 (2016). These codes and standards provide the dimensional specifications, pressure and temperature ratings for the different types of flanges and their applications. ASME BPVC-VIII-2 (2013) and Khurmi and Gupta (2005) identifies two major flange types based on the method of attachment to the pipe or vessel to include the integral flange types and the loose flange types. In the integral type of flange; the flange is integrally cast, forged, butt-welded or attached by any other method of welding with the pipe wall, vessel nozzle neck or vessel wall structurally together. The loose flange types are those without integral connection with the pipe wall, vessel nozzle neck or vessel wall structurally. These include flanges obviously welded or screwed to the pipe wall, vessel nozzle neck or vessel wall in which the mechanical strength of the welds is not equivalent to that of the integral attachment. Since the pipes and pressure vessels being discussed are cylindrical with circular cross-sections, the flanges being considered are also circular in cross-section. Figure (8) shows circular flanged pipe joints and the design mechanism is shown in figure (9) below.

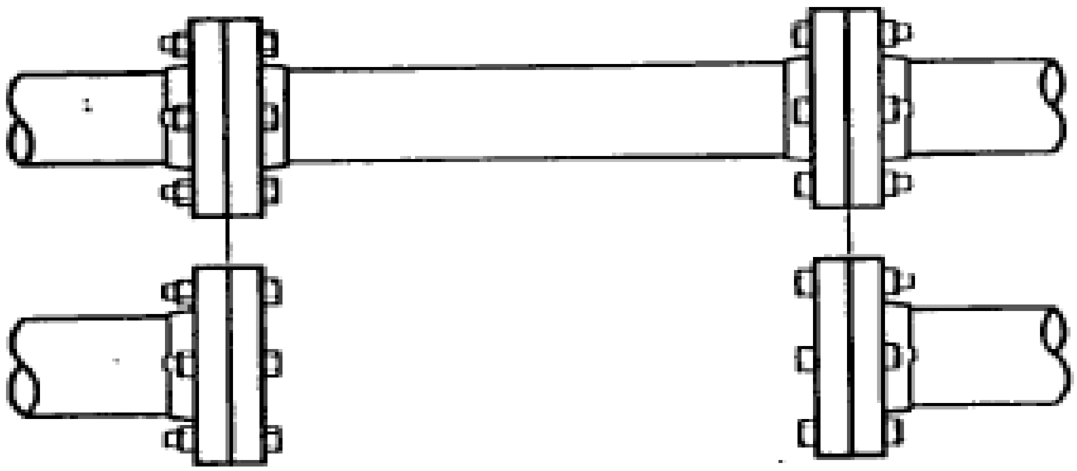

Figure 8: Section of a pipeline showing circular flanged joints

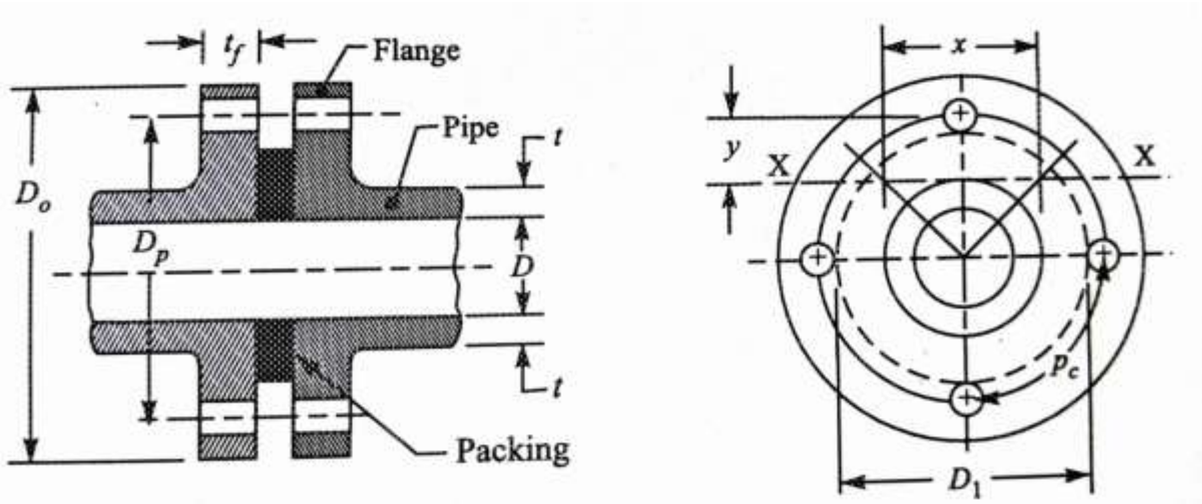

Figure 9: Circular flanged joint showing design parameters (Khurmi and Gupta 2005).

According to Khurmi and Gupta (2005) the fluid pressure that acts in between the flanges and tends to separate them with a pressure existing at the point of leaking; and the bolts are required to take up tensile stress in order to keep the flanges togetherare considered in designing circular flanged pipe joints. If D_e is the diameter of the circle touching the bolt holes (i.e. the effective diameter of the flange on which the fluid pressure acts at the point of leaking), d_b is the diameter of bolt holes, D_p is the pitch circle diameter (i.e. diameter of the circle that passes through the centre of the bolts); then:

$$D_e = D_p - d_h \qquad (69)$$

The force that tends to push the two flanges apart is given by,

$$F = \frac{\pi(D_e)^2 P}{4} \qquad (70)$$

The force resisting the tearing of the bolts is given by,

$$F_r = \frac{\pi n(d_c)^2 S_t}{4} \qquad (71)$$

The circumferential pitch of the bolts is given by,

$$p_c = \frac{\pi D_p}{n} \qquad (72)$$

And $20\sqrt{d_b} \le p_c \le 30\sqrt{d_b}$ for a leak-proof joint.

As F tends to push the flanges apart, the sections of the flange tend to bend; thus the resisting moment of the flange is given by,

$$M_r = S_b Z \qquad (73)$$

$$Z = \frac{x(t_f)^2}{6} \qquad (74)$$

The nominal diameter of the bots is given by,

$$d_b = 0.75t + 10mm \qquad (54)$$

The number of bolts is given by,

$$n = 0.0275D_i + 1.6 \qquad (75)$$

Note that the number of bolts should be even to ensure symmetry of the flange section.

Thickness of the flange is given by,

$$t_f = 1.5t + 3mm \qquad (76)$$

Width of the flange is given by,

$$B = 2.3d_b \tag{77}$$

Outside diameter of the flange is given by,

$$D_f = D_i + 2t + 2B \tag{78}$$

Pitch circle diameter of the bolts is given b,

$$D_p = D_i + 2t + 2d_b + 12mm \tag{79}$$

Thickness of the pipe near the flange is given by,

$$t_{nf} = \frac{t + t_f}{2} \tag{80}$$

Where S_t = the permissible tensile stress for the material of the bolts,S_b = bending or tensile stress for the material of the flange, Z = section modulus of the cross-section of the flange,x = the width of the segment of the flange created by a sectional line tangential to the outside diameter of the pipe.This analysis is fundamentally generic, other and further design specifics are available in Kirkemo (2002), Bouzid and Beghoul (2003), Abid and Nash (2004),Schaaf and Bartonicek (2003), Sawa et al (1991), Brown et al (2008), Estrada (2015), Moss (2004), Moss and Basic (2013) and Omiya et al (2014).

Table 10: Analysis of hoop tension (Liu 2003)

European Journal of Mechanical Engineering Research, 11 (2),1-27, 2024
Print ISSN: 2055-6551(Print)
Online ISSN: 2055-656X(Online)
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Facility	Location Class 1 Div. 1	Location Class 1 Div. 2	2	3	4
Pipelines, mains, and service lines [see para. 840.21(b)]	0.80	0.72	0.60	0.50	0.40
Crossings of roads, railroads without casing:					
(a) Private roads	0.80	0.72	0.60	0.50	0.40
(b) Unimproved public roads	0.60	0.60	0.60	0.50	0.40
(c) Roads, highways, or public streets, with hard surface and railroads	0.60	0.60	0.50	0.50	0.40
Crossings of roads, railroads with casing:					
(a) Private roads	0.80	0.72	0.60	0.50	0.40
(b) Unimproved public roads	0.72	0.72	0.60	0.50	0.40
(c) Roads, highways, or public streets, with hard surface and railroads	0.72	0.72	0.60	0.50	0.40
Parallel encroachment of pipelines and mains on roads and railroads:					
(a) Private roads	0.80	0.72	0.60	0.50	0.40
(b) Unimproved public roads	0.80	0.72	0.60	0.50	0.40
(c) Roads, highways, or public streets, with hard surface and railroads	0.60	0.60	0.60	0.50	0.40
Fabricated assemblies (see para. 841.121)	0.60	0.60	0.60	0.50	0.40
Pipelines on bridges (see para. 841.122)	0.60	0.60	0.60	0.50	0.40
Pressure/Flow Control and Metering Facilities (see para. 841.123)	0.60	0.60	0.60	0.50	0.40
Compressor station piping	0.50	0.50	0.50	0.50	0.40
Near concentration of people in Location Classes 1 and 2 [see para. 840.3(b)]	0.50	0.50	0.50	0.50	0.40

Table 2.11: Design factors for steel pipe construction (ASME B31.8 2005)

Temperature, °F	Temperature Derating Factor, T
250 or less	1.000
300	0.967
350	0.933
400	0.900
450	0.867

CONCLUSION

This paper presents an overview the principles and mechanical design analysis of oil and gas piping systems. The piping systems considered are the pipes (including the straight, flanged and curved sections) and pressure vessels of circular cross sections. Design formulations were drawn from various literatures and presented. More so the historical overview of piping systems with particular applications to the oil and gas production, transportation, transmission and storages facilities were presented. This study shows the bases for hydrostatic and pneumatic pressure tests and makes the formulation of pressure tests procedures easy.

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