Pressure Vessel Newsletter

Volume 2014, October Issue

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***From The Editor's Desk*:**

In this space today, I would like to discuss the employability options for fresh engineering graduates in India. Today, India has more than 700 universities and 33,000 colleges which offer a large number of programs in arts, science, commerce, finance, engineering, technology, law and medicine. Together, they produce more than a million graduates each year, of which between 20-25% are engineers. However, only about 19% of the engineers are considered fit for employment in the workplace. The number reduces dramatically to 5% for other streams.

The "India Employability Report by Aspiring Minds", a research firm, paints an even bleaker picture. According to this report, Chennai has an employability rate of an awful 1%. Even the state with the highest employability percentage, the city-state of Delhi, is only at 13%. Bangalore, the celebrated "silicon valley of India" is at a staggering 3%.

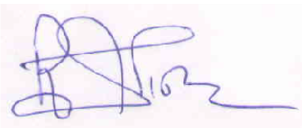
Clearly, something is horribly wrong with our education system. Most of the engineering colleges, including the better known ones, are understaffed and lack in qualified, competent and suitable faculty members. The course curriculum is mostly theoretical in nature and there is near-zero interaction with the industry. The assignments do not involve any research or innovation, thus making it very challenging to motivate and attract students to any serious learning.

There is too much focus on rote learning rather than critical thinking and comprehension. The problem starts early in the student life – in elementary school. The school projects that are intended to be done by the young ones are completed with disproportionate help from their parents; in the process depriving the young minds from developing their creative skills. By the time they show up at the college, they are simply incapable of any original thinking.

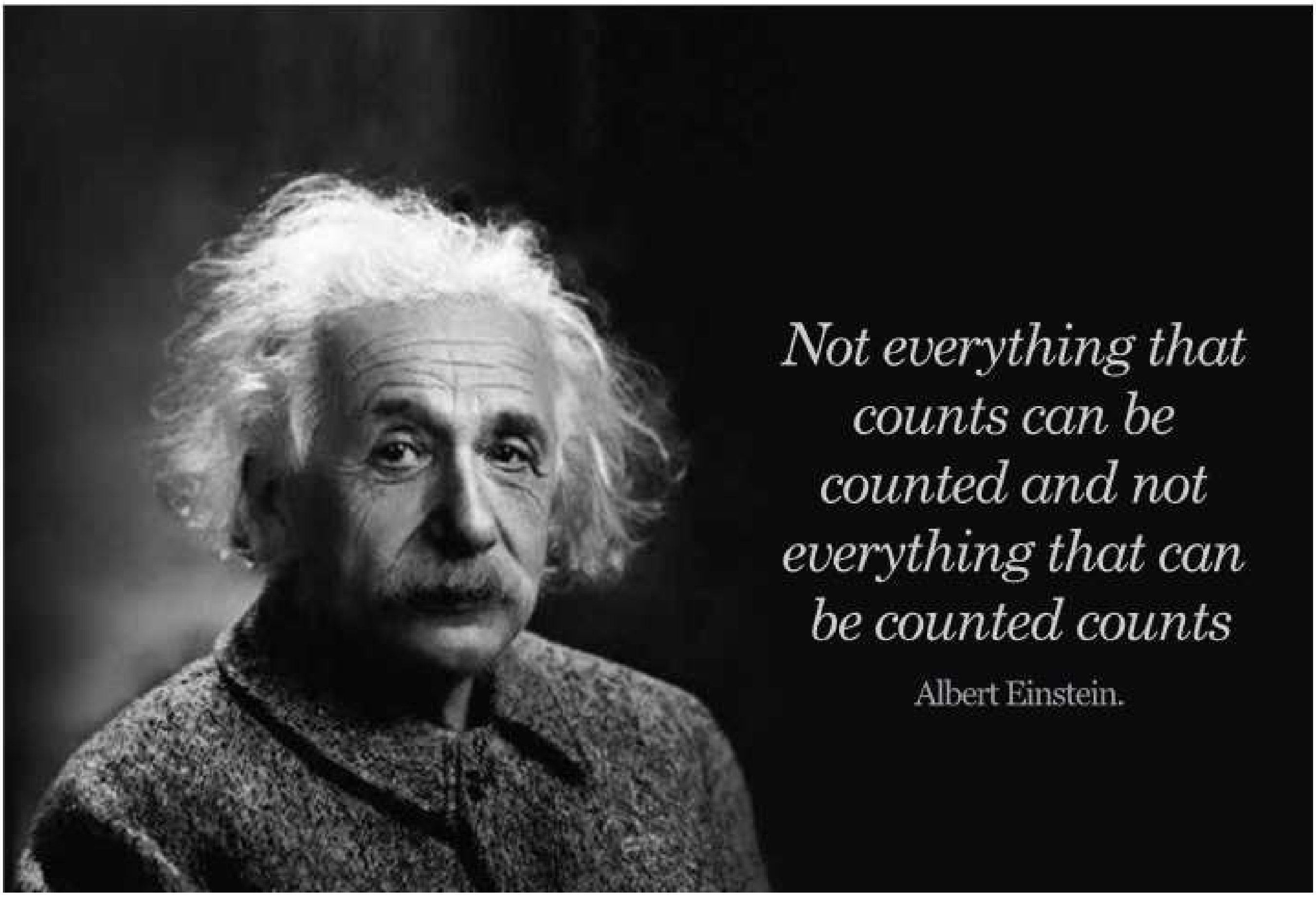
The current practice among the companies that employ pressure vessel and piping engineers (at least the larger ones) is to train all the fresh engineers for up to a year before using them productively on actual projects. Clearly, this is a huge drain on the company's finances, not to mention the time that some of the senior engineers have to allocate for providing the training. And there is no guarantee that these "freshers" will stick around in the company long enough to make this exercise worthwhile. The smaller companies simply don't have the necessary financial wherewithal and must provide "on-the-job training", in the process jeopardizing the performance of the projects.

What is needed urgently in short term is emergence of organizations that act as a bridge between the engineering colleges and the industry. These organizations must provide sufficient training to the fresh graduates in the required fields so that they are productive from day one when they eventually gain employment. This will be a win-win situation for all parties involved – students as they will considerably increase their chances of getting good employment; companies because they will be able to put their hard-earned finances to productive use; and the colleges because it will improve their placement record.

However, in the long term we need better education system that includes extensive interaction with the industry so that the graduates have necessary skills to be productive from day one.



In this issue…



Math of real world seldom adds up due to intangible variables that cannot be easily captured. For final tally, we know we don't just have to win contracts, we must earn customer confidence too. Our main focus is customer delight achieved due to & through positive interactions, quality alertness, proactive involvement and personalized service for varying situations & requirements.

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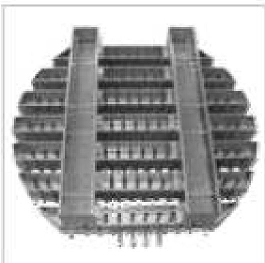
Random Packings


Tower Internals


Fractionation Trays


Structured Packings


Mist Eliminators

FRACTURE MECHANICS – A HISTORICL PERSPECTIVE

Designing structures to avoid fracture is not a new idea. Many structures commissioned by the Pharaohs of ancient Egypt and the Caesars of Rome, and numerous buildings and bridges constructed during the Renaissance Period (roughly between 14^{th} and 17^{th} Century) are still standing. These ancient structures represent successful designs. Undoubtedly, there were many unsuccessful designs with much shorter life spans also. Because the knowledge of mechanics was limited prior to the time of Isaac Newton (1643-1727), workable designs were achieved largely by trial and error.

The durability of ancient structures is particularly amazing when one considers that the choice of building materials prior to Industrial Revolution was rather limited. Metals could not be produced in sufficient quantity to be formed into lead-bearing members for buildings and bridges. This left timber, brick and mortar as the primary construction materials;of which only the latter two were usually practical for large structures because trees of sufficient size for support beams were rare.

Early Structures: Loaded in Compression

Brick and mortar are relatively brittle and unreliable for carrying tensile loads. Consequently, pre-Industrial Revolution structures were usually designed to be loaded in compression. Figure 1 schematically illustrates a Roman bridge design. The arch shape causes compressive rather than tensile stresses to be transmitted through the structure.

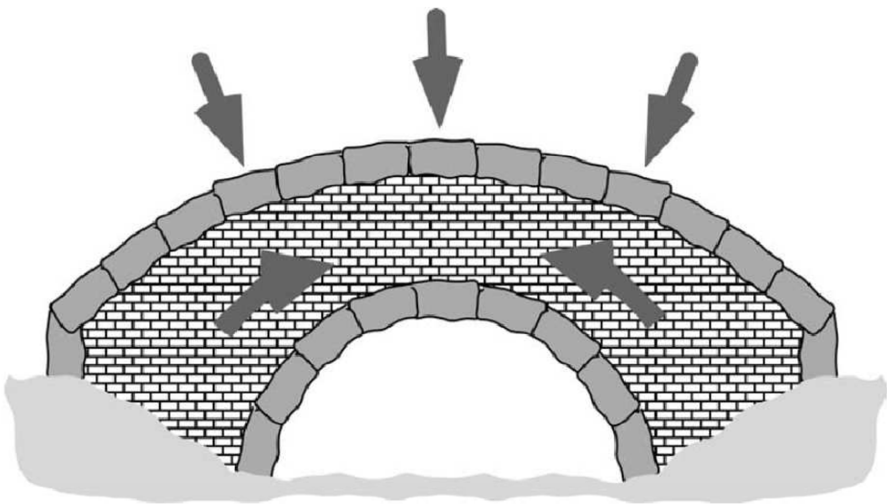


Figure 1: Schematic Roman Bridge Design

The arch is the predominant shape in pre-Industrial Revolution architecture. Windows and roofs were arched in order to maintain compressive loading. Compressive loaded structures are inherently stable; some have lasted for many centuries – the pyramids in Egypt are one good example.

Industrial Revolution:Onset of Structures Loaded in Tension

With the Industrial Revolution came the mass production of iron and steel (Or, conversely, we could say that the mass production of iron and steel fueled the Industrial Revolution). The availability of relatively ductile material removed the earlier restrictions on design. It was finally feasible to build structures that carried tensile stresses. Figure 2 shows the difference between the design of Tower Bridge in London (built 1886-1894) and the earlier bridge design (Figure 1).

The change from brick and mortar structures loaded in compression to steel structures loaded in tension brought problems, however. Occasionally, a steel structure would fail unexpectedly at stresses well below the anticipated tensile strength. One of the most famous of these failures was the rupture of molasses tank in Boston in January 1919. Over 2 million gallons of molasses were spilled, resulting in 12 deaths, 40 injuries, massive property damage, and several drowned horses. The cause of the failure of the molasses tank was largely a mystery at the time. In an effort to avoid these seemingly random failures, the designers typically applied safety factors of 10 or more (based on the tensile strength).

Figure 2: Tower Bridge in London, completed in 1894

Early Fracture Research: Griffith's Model

Experiments performed by Leonardo da Vinci several centuries earlier provided some clues as to the root cause of the failure in the steel structures. Leonardo measured the strength of the iron wires and found that the strength varied inversely with the wire length. These results implied that flaws in the material controlled the strength; a longer wire corresponded to a larger sample volume, and a higher probability of sampling a region containing a flaw. These results, however, were only qualitative.

A quantitative connection between fracture stress and flaw size came from the work of Griffith which was published in 1920. According to Griffith's theory, a flaw becomes unstable, and thus fracture occurs, when the strain energy change that results from an increment of crack growth is sufficient to overcome the surface energy of the material. Griffith's model correctly predicted the relationship between strength and flaw size in glass specimens. Because this model assumes that the work of fracture comes exclusively from the surface energy of the material, it applies only to the ideally brittle solids. A modification to Griffith's model, that made it applicable to metals, did not come until 1948.

The Liberty Ships

The mechanics of fracture progressed from being a scientific curiosity to an engineering discipline, primarily because of what happened to the Liberty ships during World War II. In the early days of the war, the US was supplying ships and planes to Great Britain under the Lend-Lease Act. Britain's greatest need at the time was for cargo ships to carry supplies. The German navy was sinking cargo ships at three times the rate at which they could be replaced with existing ship-building procedures.

The US had developed, under the guidance of Henry Kaiser, a revolutionary procedure for fabricating ships quickly. These new vessels, which became known as the Liberty ships, had an all-welded hull, as opposed to the riveted construction of traditional ship designs.The Liberty ship program was a resounding success, until one day in 1943, when one of the vessels broke completely in two while sailing between Siberia and Alaska. Subsequent fractures occurred in other Liberty ships. Of the roughly 2700 ships built during the war, approximately 400 sustained fractures, of which 90 were considered serious. In 20 ships, the failure was essentially total, and about half of these broke completely in two.

Investigations revealed that the Liberty ship failures were caused by combination of three factors:

1) The welds, which were produced by semi-skilled work force, contained crack-like flaws.

2) Most of the fractures initiated on the deck at square hatch corners, where there was a local stress concentration.
3) The steel from which the Liberty ships were made had poor toughness, as measured by Charpy impact tests.

The steel in question had always been adequate for riveted ships because the fracture could not propagate across panels that were joined by rivets. A welded structure, however, is a single piece of metal; propagating cracks in the Liberty ships encountered no significant barriers, and were sometimes able to traverse the entire hull.

Once the causes of failure were identified, the remaining Liberty ships were retrofitted with rounded reinforcements at the hatch corners. In addition, high toughness steel crack-arrester plates were riveted to the deck at strategic locations. These corrections prevented further serious fractures.

In the longer term, the structural steels were developed with vastly improved toughness, and weld quality control standards were developed. Also a group of researchers at the Naval Research Laboratory in Washington, DC, studied the fracture problem in detail. The field we now know as fracture mechanics was born in this lab during the decade following the war.

Dr. Irwin's Work

The fracture mechanics research group at the National Research Laboratory was led by Dr. G.R. Irwin.He understood that the basic tools needed to analyze fracture were already available, and his first major contribution was to extend the Griffith approach to metals by including the energy dissipated by local plastic flow. Irwin developed the energy release rate concept which was more useful for solving engineering problems He was able to show that the stresses and displacements near the crack-tip could be described by a single constant that was related to the energy release rate. This crack-tip characterizing parameter later became known as the "stress intensity factor". A number of successful early applications bolstered the standing of the field of fracture mechanics in the engineering community.

Non-linear material Behavior: J- Integral

Linear elastic fracture mechanics (LEFM) ceases to be valid when significant plastic deformation precedes failure. Such is the case with the stresses and displacements near the crack tip. Wells, who had earlier worked with Dr. Irwin, had attempted to apply LEFM to low- and medium-strengths structural steels at British Welding Research Association. These materials were too ductile for LEFM to apply, but Wells noticed that the crack faces moved apart with plastic deformation. This observation led to the development of the parameter now known as the crack-tip-opening displacement (CTOD).

In 1968, Rice developed another parameter to characterize nonlinear material behavior ahead of a crack. By idealizing plastic deformation as a nonlinear elastic deformation, rice was able to generalize the energy release rate to nonlinear materials. He showed that this nonlinear energy release rate can be expressed as a line integral, which he called the J integral, evaluated along an arbitrary contour around the crack. Rice, along with Hutchinson and Rosengren, was able to relate the J integral to crack-tip stress fields in nonlinear materials. These analyses showed that J integral can be viewed as a nonlinear, stress-intensity parameter as well as an energy release rate.

Rice's work might have been relegated to obscurity had it not been for the active research effort by the nuclear power industry in the US in the early 1970s. Because of the legitimate concerns for safety, as well as political and public relations considerations, the nuclear power industry endeavored to apply state-of-art technology, including fracture mechanics, to the design and construction of nuclear power plants. The difficulty in applying fracture mechanics in this instance was that most nuclear pressure vessel steels were too tough to be characterized with LEFM without resorting to enormous laboratory specimens. In 1971, research engineers at Westinghouse decided to characterize the fracture toughness of these steels with the J integral. Their experiments were very successful and led to the publication of a standard procedure for J testing of metals 10 years later.

Material toughness characterization is only one aspect of fracture mechanics. In order to apply fracture mechanics concepts to design, one must have a mathematical relationship between toughness, stress and flaw size. Although these relationships were well established for linear elastic problems, a fracture design analysis based on J-integral became available only in 1976. A fracture design handbook was published by Electric Power Research Institute (EPRI) a few years later.

While fracture research in the US was driven primarily by the nuclear power industry during the 1970s, fracture research in the UK was motivated largely by the development of oil resources in the North Sea. In the UK, Well's CTOD parameter was applied extensively to fracture analysis of welded structures. A CTOD design curve was developed in 1971 based on semi-empirical fracture mechanics methodology for welded steel structures. Since then it has been demonstrated that both CTOD and J parameters are equally valid for characterizing fracture. Both parameters are currently applied throughout the world to a range of materials.

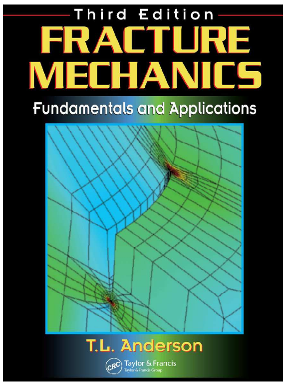
The Present State

The field of fracture mechanics has matured in the last two decades of the 20th century. The application of this technology to practical problems is so pervasive that fracture mechanics now is considered an established engineering discipline.

More sophisticated models for material behavior are being incorporated into fracture mechanics analyses. While plasticity was the important concern in 1960s, more recent work has gone a step further, incorporating time-dependent nonlinear material behavior such as viscoplasticity and viscoelasticity. The former is motivated by the need for tough, creep-resistant high temperature materials, while the latter reflects the increasing proportion of plastics in structural applications.

The continuing explosion in computer technology has aided both the development and application of fracture mechanics technology. An ordinary desktop computer is capable of performing complex three-dimensional finite element analyses of structural components that contain cracks. Computer technology has also spawned entirely new areas of fracture mechanics research. Problems encountered in the microelectronics industry have led to active research in interface fracture and nanoscale fracture.

Source: Fracture Mechanics – Fundamentals and Applications *by* T.L. Anderson



OPENINGS AND REINFORCEMENTS

There are three important points to keep in mind for design of efficient reinforcing pads for openings in pressure vessels:

1) DO NOT OVER-REINFORCE. Adding more material than required creates a too “hard” spot on the vessel, and large secondary stresses due to shell restraint can be produced as a consequence.
2) Place the reinforcing material adjacent to the opening for effectiveness. It is recommended to place two-third of the required reinforcement within a distance d/4 on each side of opening where “d” is the diameter of the opening.
3) Use generous transition radii (r_1 and r_2 in Figure 1 below) between the shell and the nozzle to minimize stress concentrations resulting from a grossly discontinuous junction under internal pressure.

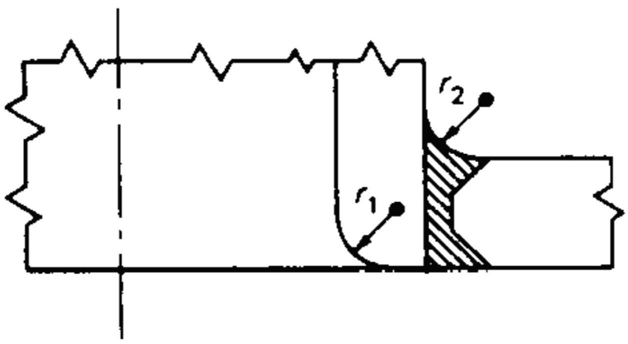


Figure 1: Transition Radii Between Shell and Nozzle

Source: *Pressure Vessel Design Handbook* by Henry H. Bednar

TRAINING ANNOUNCEMENT

DESIGN & FABRICATION OF PRESSURE VESSELS: ASME SECTION VIII, DIVISION 1

Pressure vessels, along with tanks, are the workhorses for storage and processing applications in the chemical, petroleum, petrochemical, power, pharmaceutical, food and paper industries. ASME BPV, Section VIII, Div. 1 Code is used as a standard for the design and fabrication of pressure vessels by most companies across the world.

Wewould like to announcetraining course for "Design and Fabrication of Pressure Vessels: ASME Section VIII, Div. 1" on **November 6-8, 2014 at Mumbai** and **December 11-13, 2014** at Chennai. This course provides the information that will help you understand the ASME requirements for the design and fabrication of pressure vessels. The course material follows the contents of 2013 edition of the code, and is replete with worked examples covering important aspects of pressure vessel construction. This hands-on learning will allow you to master in 3 days what would otherwise take up to a year or more of on-job training.

The contents of the training course will be as follows:

- Introduction to Boiler and Pressure Vessel Code
- Materials of Construction
- Low Temperature Operation
- Joint Efficiencies
- Welding Requirements
- Design of Components
- Openings and Reinforcements
- Design of Flanges
- Fabrication, Inspection and Tests
- Markings and Reports
- Tall Towers and Pressure Vessel Supports
- Nozzle Loads
- Fatigue Analysis
- Introduction to ASME Section VIII, Division 2

The instructor, Ramesh Tiwari, is internationally recognized specialist in the area of pressure vessels, heat exchangers, materials, and codes and standards. He holds Bachelor's and Master's degrees in mechanical engineering from universities in India and United States. He is also a registered Professional Engineer in the State of Maryland in the United States. Mr. Tiwari is a member of ASME Boiler & Pressure Vessel, Section VIII Subgroup on Heat Transfer Equipment, and a member of ASME International Working Group on B31.1 for Power Piping in India. In this capacity, he has made invaluable contribution in resolving technical issues in compliance with the ASME codes for Code users. Mr. Tiwari has over 24 years of design engineering experience on a variety of projects spanning industries such as oil & gas, power, nuclear, chemical, petrochemical, pharmaceutical, food etc. He has provided engineering advice and code interpretations to senior management and guidance to several companies he has worked for in the US, India and Germany. He has initiated and implemented numerous innovative ideas to improve working process and quality, and developed and conducted training programs for peers as well as clients. Mr. Tiwari is an approved pressure vessel instructor at NTPC, a premier thermal power generating company in India and at several other companies, both public and private.

Registration fee for the training course is Rs. 25,300 for professionals and Rs 16,000 for students (inclusive of all applicable taxes). Discount of 15% is available for group registration of 2 or more participants. Additionally, early bird discount of 15% is available if registration is done on or before 24th October (Mumbai) and 26th November (Chennai). Registration fee includes training, a collection of articles on design and fabrication of pressurevessels, electronic copy of the presentation, certificate from CoDesign Engineering, and beverages and lunch on all days. It excludes travel to and from Mumbai, accommodation, and meals and beverages other than those provided during the course. We invite you to make nominations.

In case of any queries, including the registration process, please email at learning@codesignengg.com, or call at +91 98109 33550.

BOLTED FLANGE CONNECTIONS

Bolted flange connections perform two important functions in a pressure vessel connection:

a) Maintenance of the structural integrity of the connection itself, and
b) Prevention of leakage through the gaskets preloaded by bolts.

A representative bolted flange joint is shown in Figure 1. This is typically comprised of a flange ring and a tapered hub, which is welded to the pressure vessel. The flange is a seat for the gasket, and the cover (along with the gasket) is bolted to the flange by a number of bolts. The preload on the bolts is extremely important for the successful performance of the connection. The preload must be sufficiently large to seat the gasket and at the same time not excessive enough to crush it. The bolts should be designed to contain the pressure and for the preload required to prevent leakage through the gasket. The flange region should be designed to resist bending that occurs in the spacing between the bolt locations.

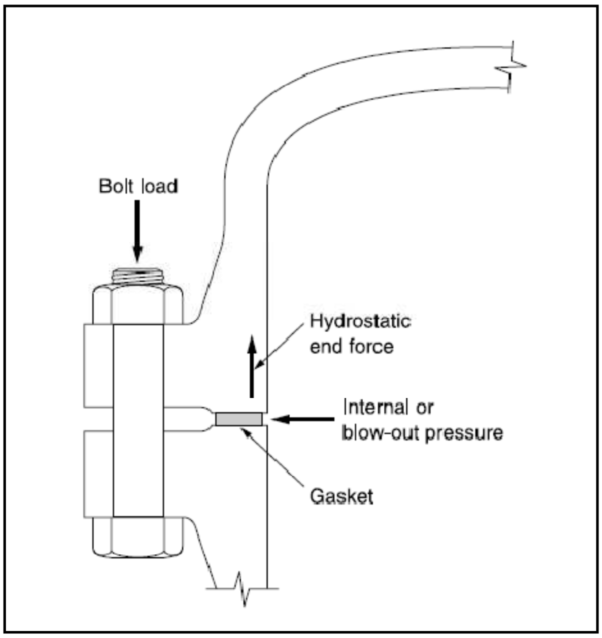


Figure 1: Bolted Flange Joint

The gasket, which is really the focal point of the bolted flange connection, is subject to compressive forces by the bolts. The flange stiffness in conjunction with the bolt preload provides the necessary surface constraint and the compressive force to prevent the leakage of the fluid contained in the pressure vessel. The fluid pressure tends to reduce the bolt preload, which reduces gasket compression and tends to separate the flange faces. The gaskets are therefore required to expand to maintain the leak-proof boundary. Gaskets are made of nonmetallic material with composite construction. The serrated surfaces of the flange faces help to maintain the leak-proof joint as the material expands to fill up the irregularities on the face of the flanges.

Gasket Joint Behavior

The flange and bolts must meet two design requirements:

a) The gasket seating condition, and
b) The operating condition.

The design calculations use two gasket factors, the gasket seating stress y, and the gasket factor at operating conditions m. A higher gasket seating stress ensures better sealing performance. When the joint is in service, the fluid pressure unloads the joint, resulting in a reduced gasket stress (see Figure 2). Under operating conditions, it is advantageous to have a residual gasket stress greater than the fluid pressure. For good sealing performance, the ASME Code recommends the residual stress at operating conditions to be at least two to three times the contained pressure. This is the so-called m-factor. The relationship between the initial seating stress and the residual seating stress is indicated by the gasket stress vs. deflection plot shown in the Figure 3. S_{G2} is the initial seating stress and S_{G1} is the operating gasket stress. During assembly, the gasket follows the nonlinear portion of the curve from zero to S_{G2}. When the operating pressure unloads the gasket, the gasket follows the unloading curve from S_{G2} to S_{G1}.

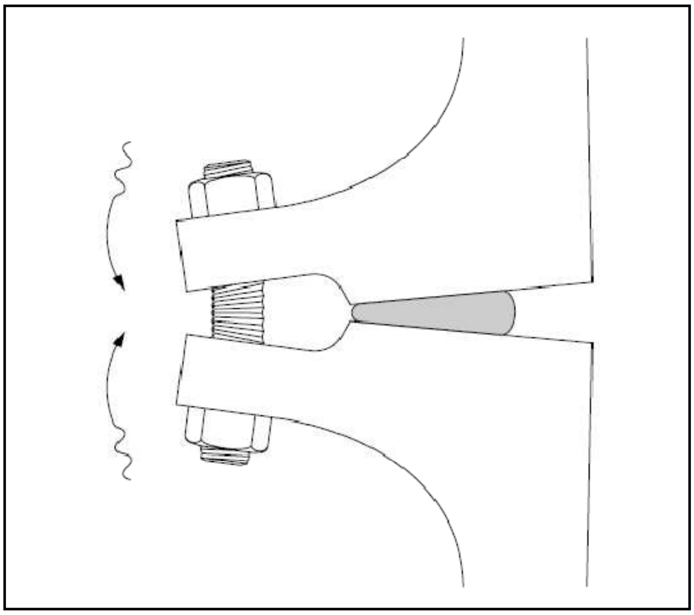


Figure 2: Flange Bending

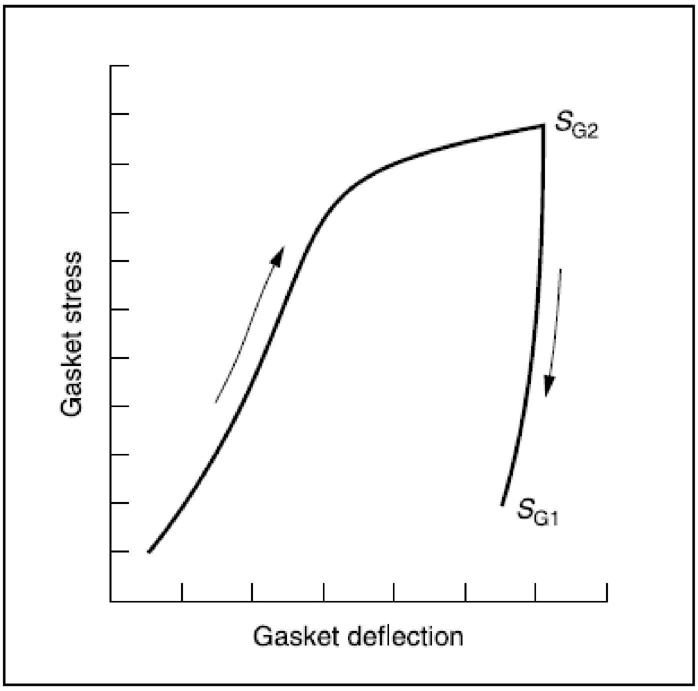


Figure 3: Gasket Stress vs. Deflection

In the absence of the fluid pressure, the required bolt load to seat the gasket is given by (The notations are from ASME Section VIII, Division 1, Appendix 2):

$$W_{m2} = \pi bGy$$

Here W_{m2} is the gasket seating load, b is the effective gasket seating width, and G is the minimum gasket diameter. When the joint is pressurized by the fluid, the operating bolt load W_{m1} is given by:

$$W_{m1} = \frac{\pi}{4}G^2p + (2b\pi Gmp)$$

Here p is the design pressure. The first term in the previous equation is the joint end load due to fluid pressure. The second term is the joint contact load.

Design of Bolts

The chosen bolt material should be compatible with the flange material. There must not be any chemical or galvanic reaction between the materials to produce the possibility of thread seizure. The total minimum required cross-sectional area of bolts should be the greater of the following areas:

$$\frac{W_{m2}}{S_a} \text{and} \frac{W_{m1}}{S_b}$$

Here S_a is the allowable bolt stress at room temperature, and S_b is the allowable bolt stress at design temperature.

Example:

The data for a bolted flange connection for a pressure vessel is given below. Should twelve (12) 50-mm diameter bolts be adequate to ensure a leak-proof joint?

Design pressure, p	=	17 MPa
Gasket diameter, G	=	382 mm
Gasket width, b	=	9 mm
Gasket factor, m	=	3
Gasket seating stress, y	=	69 MPa
Bolt allowable stress, S_a	=	132 MPa
Bolt allowable stress, S_b	=	132 MPa

Gasket loading, W_{m2} = $\pi(9)(382)(69)$

= 745.255 kN

Gasket loading, W_{m2} = $(\pi/4)(382)^2(17) + (2)(9)(\pi)(382)(3)(17)$

= (1948.344 + 1101.681) kN

= 3050 kN

The required bolt area is the greater of (3050 x 1000 / 132) and (745 x 1000 / 132), and is equal to 23,106 mm^2. Therefore, twelve (12) bolts of diameter 50 mm (total bolt area = 23,562 mm^2) should be adequate.

Source: *Pressure Vessels: Design and Practice* by SomnathChattopadhyay

TRAINING CALENDAR ANNOUNCEMENT

2014	
October 6-17 ***New***	Ten (10) Day Workshop: Pressure Vessels and Heat Exchangers New Delhi
November 6-8	Design and Fabrication of Pressure Vessels: ASME Section VIII, Division 1 Mumbai
December 11-13	Design and Fabrication of Pressure Vessels: ASME Section VIII, Division 1 Chennai
2015	
TBA	One (1) Day Pressure Vessel Workshop Vadodara
TBA ***New***	Introduction to API 510: Pressure Vessel Inspection New Delhi
TBA	Introduction to ASME Section VIII, Division 2 Bengaluru
TBA	Design and Fabrication of Pressure Vessels: ASME Section VIII, Division 1 Vadodara
TBA ***New***	Introduction to European Code EN 13445 for Unfired Pressure Vessels Mumbai
TBA	Design and Fabrication of Pressure Vessels: ASME Section VIII, Division 1 Chennai
July 16-18	Introduction to ASME Section VIII, Division 2 New Delhi
August 20-22	Design and Fabrication of Pressure Vessels: ASME Section VIII, Division 1 Mumbai
September 7-18	Ten (10) Day Workshop: Pressure Vessels and Heat Exchangers New Delhi
October 15-17 ***New***	Thermal and Mechanical Design for Shell & Tube Heat Exchangers Pune
November 12-14	Design and Fabrication of Pressure Vessels: ASME Section VIII, Division 1 Bengaluru
December 10-12 ***New***	Introduction to Fitness for Service Using API 579/ ASME FFS Mumbai

TBA: Date to be announced at a later date.

NEWS AND EVENTS

2014 Abu Dhabi International Petroleum Exhibition & Conference

November 10-13, 2014 | Abu Dhabi, UAE

This event is an opportunity for like-minded professionals to join and contribute to one of the largest industry shows in the Middle East. Providing a first-rate platform for exchanging knowledge and best practices, the conference brings together renowned international speakers, researchers, and experts with a carefully selected mix of technical presentations, executive plenary session, and panel discussions.

FABTECH

November 11-13, 2014 | Georgia World Congress Center, Atlanta, GA USA

The exposition will host over 27,000 attendees and 1,400 exhibiting companies. FABTECH is the United States' largest metal forming, fabricating, welding and finishing trade show. People in the pressure vessel manufacturing and fabrication industry along with folks in the steel processing and fabrication industry who attend FABTECH will learn about metal forming, fabricating, welding and finishing products and developments.

September 16, 2014: Technip was awarded by The Bahrain Petroleum Company (BAPCO) a significant contract on a reimbursable basis to develop the Front-End Engineering Design (FEED) of the refinery located in the Kingdom of Bahrain. The FEED contract covers four main work packages that include units aimed at processing the "bottom of the barrel" components to high value products, and all associated offsites and utilities to provide seamless integration with existing refinery facilities earmarked for retention post this major modernization. The project aims at enhancing the refinery configuration, by increasing the throughput from 267,000 to 360,000 barrel per day as well as improving the product slate and profitability. The project is scheduled to be completed at the end of 2015.

September 16, 2014: Samsung Engineering has officially announced that it received a contract from PEMEX, a Mexican state-owned petroleum company, for the PEMEX Salamanca ULSD Project. The project will be executed in two phases. Phase I will include detail engineering and procurement of long-lead items and Phase II will comprise the rest of detail engineering, procurement, construction and commissioning. The $80 million contract awarded to Samsung Engineering is for Phase I. The project site is located in Salamanca 250 km northwest of Mexico's capital, Mexico City. Samsung Engineering's design will include the new HDS (hydrodesulfurization) unit with the capacity of 38,000 BPSD and the revamping of an existing HDS unit with the capacity of 53,000 BPSD. The initial engineering phase of the project is expected to be completed in September of 2015.

September 16, 2014: TécnicasReunidas, TR, has been selected by Pemex Refinación for the execution of the ultra-low sulfur diesel project at the General Refinery Lazaro Cardenas in Minatitlan, Mexico. This contract includes EPC and commissioning of three new refining units: diesel hydrodesulphurisation unit (30,000 bpd), hydrogen production plant (25 Mcfd) and sulphur recovery plant (150 tpd); as well as modifications to an existing hydrodesulphurisation unit and the integration of the facilities beyond the battery limits for these plants. The contract will be implemented in two phases. The first phase, costing approximately $50 million, includes the execution of a basic design (FEED), a detailed estimation of the investment cost and the purchase of long-term delivery equipment. This first phase will take 12 months. The second phase is project implementation, including detailed engineering, procurement of equipment and materials, construction and commissioning, under the turnkey model, with an estimated impact exceeding $500 million, and an execution period of 27 months.

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BUILDING A BETTER TOMMORROW

It is becoming less practical for many companies to maintain in-house engineering staff. That is where we come in – whenever you need us, either for one-time projects, or for recurring engineering services. We understand the codes and standards for pressure vessels, and can offer a range of services related to them.

Training & Development

Consulting Services

CoDesign Engineering

Pressure Vessels ● Heat Exchangers ● Piping Systems ● Welding

Oil & Gas ● Power ● Chemical ● Petrochemical ● Fertilizer ●Solar ● Biogas