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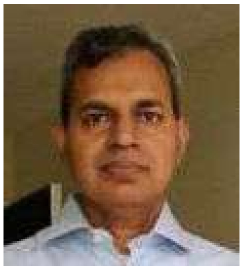
China – the next economic superpower?



Economists have a nasty habit of misreading the global trends in worldwide economic growth. For years, they have been pointing to China as the next economic superpower. However, the "one child" policy that China introduced in late 1970's to slow down the population growth seems to have created a new challenge for its economy. A decline in birth rate and increase in life expectancy means that there will soon be too few workers able to support an enormous and ageing population.

The fertility rate required to maintain population levels is 2.1 children per woman – a figure known as "replacement level fertility". China's fertility rate has dropped to 1.6 children per woman. Chinese women born during the years following the "one child" policy are now reaching or have passed their peak fertility rate. There are simply not enough of them to sustain the country's population level. The looming demographic crisis could be the Achilles heel of China's stunning economic performance over last 40 years.

The declining population would create an even greater burden on China's economy and its labor force, and with fewer workers in future, the country would struggle to pay for a population that is getting older and living longer. Fewer working-age population would also slow down the consumer spending, and thus have a spiraling downward impact on the economy.

This phenomena has occurred before in not-too-distant past – in the 1990's, the economic boom that was underway for many decades in Japan stalled, and the country's GDP growth has barely been in the positive territory ever since.

2018 is seen as the historic turning point in China's population – that is when it started to contract for the first time since the famines of 1961 and 1962. For last two years, China's population has been continuously declining and rapidly ageing. However, the decrease in the GDP growth began much earlier - in 2010. The downward trend in the GDP growth is expected to continue and even accelerate in the foreseeable future.

Not a recipe for a becoming a superpower.

Ramesh K Tiwari
ramesh.tiwari@codesignengg.com

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SEISMIC ANALYSIS OF PRESSURE VESSELS

Pressure vessels and their supports must be designed to resist the forces and loadings anticipated during a seismic event. Depending on where your pressure vessel will be installed, a seismic analysis may be required. While design standards for storage tanks such as API 650 provide details on how to calculate and evaluate for seismic loads, the ASME Boiler & Pressure Vessel Code does not. The most commonly specified standard for determining seismic loads in the United States is ASCE 7 ("Minimum Design Loads for Buildings and Other Structures").

A seismic event causes the pressure vessel to sway as a result of the ground motion. How much loading the vessel experiences will depend on many factors. A tall, thin, slender cylindrical tower mounted at grade, is relatively flexible and will therefore will have a long period and low frequency. By contrast a short, squat pressure vessel will have a short period and a higher frequency. Pressure vessels mounted in or on structures will be influenced by the relative stiffness of the structure.

CALCULATION OF THE MAGNITUDE OF THE SEISMIC LOADS

In ASCE 7, the magnitude of the seismic loads used in the design of pressure vessels are based on a number of factors. These include:

- Geographical location of the pressure vessel
- Type of soil the pressure vessel's support is on
- Fundamental period of the pressure vessel (and its contents)
- The importance of the pressure vessel with respect to health and safety
- Height of the base of the pressure vessel with respect to its support structure (if the pressure vessel is not on the ground)
- Ductility of the pressure vessel

GEOGRAPHICAL LOCATION OF THE PRESSURE VESSEL

Seismic ground accelerations used in determining the design seismic forces are based on the geographical location of the pressure vessel and its proximity to seismically active faults capable of producing large earthquakes. ASCE 7 provides maps of the U.S. with seismic ground accelerations. Alternately, the U.S. geological survey (USGS) website (www.usgs.gov) can be used to acquire this information.

SOIL

The seismic ground accelerations are then scaled by factors dependent on the type of soil the pressure vessel's support structure is on. There are six classifications of soil used by ASCE 7, which are referred to as the Site Class. The different Site Classes are:

A – Hard rock

B – Rock

C – Very dense soil and soft rock

D – Stiff soil

E – Soft clay soil

F – Soils vulnerable to potential failure or collapse under seismic loading such as liquefiable soils.

FUNAMENTAL PERIOD OF THE PRESSURE VESSEL

The fundamental period of pressure vessel can also affect the magnitude of the design seismic loads. While pressure vessels can usually be considered to act as a rigid body when subject to seismic loads, sometimes their construction is such that this is not the case. Additionally, if the pressure vessel is filled with liquid, calculation must be given to the effects of the movements of the contents when subject to seismic loads. With pressure vessels containing liquid, a portion of the liquid will move as a rigid mass (referred to as the impulsive mass), while the remainder will slosh back and forth (referred to as the convective mass). This behavior will affect the magnitude of the seismic loads on the pressure vessel.

IMPORTANCE FACTOR

The magnitude of the design seismic loads will also be affected by what is called an "Importance Factor". The Importance Factor is a scaling factor that accounts for the degree of risk to human life, health and welfare damage or loss of use of the pressure vessel. If the failure of pressure vessel could pose substantial risk to human life or is required in maintaining the operation of an essential facility (e.g. hospital, emergency services etc.), the design seismic loads will be scaled accordingly.

LOCATION OF THE BASE OF THE PRESSURE VESSEL

Another factor affecting the magnitude of the design seismic loads is the location of the base of the pressure vessel with respect to its support structure. If the pressure vessel is supported by another structure, such as located on the third floor of a building or platform, the seismic loads at the pressure vessel will be greater than those at the ground level.

PRESSURE VESSEL DUCTILITY

Lastly, the magnitude of the design seismic loads are adjusted by what is referred to as a Response Modification Coefficient. This factor is used to account for the ductility of the structure.

SEISMIC LOAD EVALUATION METHODS

There are two methods widely used to evaluate seismic loads:

1. Static Analysis Method
2. Dynamic Analysis Methods

STATIC ANALYSIS METHOD

Most of the time, pressure vessels can be evaluated for seismic loads using hand calculations. Specifically, this method is referred to as the Equivalent Lateral Force Analysis Procedure (ELFAP). EFLAP is appropriate for pressure vessels that respond to seismic load as a rigid mass, which is typically the case. With the ELFAP, seismic forces on the pressure vessel are calculated using equation provided in ASCE 7 (or similar standard). Most of the pressure vessel design software available have the capability of evaluating seismic loads using ELFAP. While hand calculations are generally used to evaluate the pressure vessel for the seismic loads, there may be times when finite element analysis is desirable. An example of this would be when FEA has been used to evaluate other loads which are combined with the seismic loads.

ELFAP approximates the effect that the ground displacements would have on the structure by applying an equivalent force to the structure itself. A seismic event is a time-dependent phenomena whereby the loading is not applied simultaneously, but over a period of time. However, ELFAP assumes that the entire earthquake force is applied instantaneously. This procedure is conservative, and has served the industry well for many years.

The procedure takes the total base shear and distributes it along the length of the column. Once the vertical distribution of the lateral seismic force is determined, the shear force and bending moment at each plane and the sum at the base of the column can be determined. The vertical component of the seismic design loads can be either added or subtracted to create the most stringent condition. These loads used with the corresponding load combinations are used to design all support components.

DYNAMIC ANALYSIS METHODS

While the Equivalent Lateral Force Analysis Procedure is often appropriate for evaluating pressure vessels for seismic loads, there will be circumstances where using a dynamic analysis is required. This typically happens when the pressure vessel or its support structure is such that it does not respond as a rigid mass to seismic loads, when the site soil for the pressure vessel is Site Class F, when failure of the pressure vessel could pose a substantial risk to human life, or if the operation of the pressure vessel after a seismic event is essential. Dynamic analysis more accurately depicts the response of the structure to the earthquake. It is mainly used for vertical vessels which are basically a cantilevered cylinder. Dynamic analysis frequently results in lower overturning moments that the ELFAP. Lower moments in turn translate into reduced thickness for skirt and baseplate and fewer anchor bolts.

ASCE 7 provides details on how to determine if a dynamic analysis is required. If a dynamic analysis is required, there are two types to choose from: the Modal Response Spectrum Analysis Procedure, and the Seismic Response History Procedure. Both of these methods typically require the use of finite element analysis.

Seismic Response History Procedure

The Seismic Response History Procedure defines the seismic loads as a function of time. When this procedure is required by a customer, they will provide the seismic loads from a known (recorded) past seismic event. The design load will pretty much be what you would see from a seismograph, a rapidly changing load. Seismic response history analyses using finite element analysis require calculating solutions at very small time intervals in order to accurately capture the rapidly varying load and the response of the pressure vessel to that load. The computer run times for seismic response history analyses are typically long such that seismic response history analyses of pressure vessels are uncommon.

Modal Response Spectrum Analysis Procedure

The other, more common, dynamic analysis procedure is a Modal Response Spectrum Analysis. Unlike the Seismic Response History Procedure, Modal Response Spectrum Analyses use seismic loads that are defined with respect to period (or frequency). The magnitude of the loads for modal response spectrum analyses can be calculated using ASCE 7 or are sometimes provided by the customer. Performing a modal response spectrum analysis usually requires the use of finite element analysis. In a modal response spectrum analysis, the response of a pressure vessel at its natural frequencies are obtained from a modal analysis, scaled by seismic accelerations corresponding to the respective natural frequencies, and then combined using one of several methods such as the square root of the sum of the squares (SRSS).

PERIOD OF VIBRATION

Vessels will vibrate based on an exciting force such as wind or earthquake. Vessels subject to an external force or ground motion (earthquake) will deflect to a specific shape and then return to its original position once the applied force is dissipated or removed. The extent of deflection is proportional to the applied force. The vessel, or its support, will act as a spring. In the passage to equilibrium, the vessel will vibrate freely, through its various modes.

The period of vibration (POV) is the time it takes the vessel to deflect through one mode and return to its original position and is measured in seconds. The frequency, which is the inverse of POV, is the number of cycles per second. The POV of a pressure vessel is a function of the vessel weight, diameter, height, weight distribution, temperature, flexibility, type of support, damping mechanisms and location if supported in a structure. Typically when we are discussing the period of vibration for a vessel we are talking about the "first" period of vibration, or the first "natural" or "fundamental" period of vibration.

All vibrating systems, of which vessels are included, have multiple modes of vibration, known as the first mode, the second mode, etc. Each individual mode will have its own unique characteristics for that particular system. The deflected shape of a vessel for any single mode of vibration is always the same for that vessel, regardless of the magnitude. In other words, though the amplitude of displacement changes with time, the relation between displacements throughout the height remains constant. The mode with the lowest frequency (longest period) is called the first, or fundamental mode. The mode with the higher frequencies (shorter periods) are called the higher modes. Each mode would have a different POV and frequency.

The period of vibration is the inverse of the frequency of vibration. Typically, the symbol for POV is "T" and is given in seconds. The symbol for frequency is "f", and is given in hertz, which is cycles per second. $T = 1/f$ and $f = 1/T$. Generally, vessels with a POV less than 0.30 seconds ($f \geq 3.33$ Hz) are considered rigid. Vessels with a POV between 0.30 and 0.75 seconds ($1.33 \text{ Hz} < f < 3.33 \text{ Hz}$) are semi-rigid. Between 0.75 and 1.25 seconds ($0.8 \text{ Hz} < f < 1.33 \text{ Hz}$) are semi-flexible and vessels with a POV greater than 1.25 seconds (0.80 Hz) are flexible.

A vessel will have a different POV in the empty and full condition. It will have a different POV for the new and corroded condition. It will have a different POV for hot and cold conditions due to the modulus of elasticity of the steel at temperature. Vertical vessels on legs and skirts are the most flexible. Vessels on lugs and rings are normally supported in structures and therefore would be subject to the harmonics of the structure itself. Horizontal vessels vibrate with their supports as well and are dependent on pier deflection.

A vertical vessel is modeled as a cantilever beam whereas a horizontal vessel is modeled as a simply supported beam. A cantilever is a much more prone to vibration and deflection then a simply supported beam, therefore the POV is typically much higher. Guiding a vessel supported in a structure will greatly alter its POV because it changes the mode of vibration.

Wind and seismic design standards such as ASCE have base shear factors that are a function of the POV. This makes sense because the response of the vessel is dependent on the relative rigidity of the vessel. The more rigid the vessel (lower POV, high frequency) the higher the base shear will be. The more flexible (higher POV, lower frequency) vessels would have a lower base shear.

EVALUATION OF RESULTS

Whether ELFAP or one of the dynamic analysis procedures is used, the goal is to calculate stresses in the pressure vessel and its supports to compare to allowable stress limits from the ASME Code. Seismic events are short term loading conditions. As a result ASME Code Section VIII, Division 1 allows for an increase in the allowable stress of 1.2. ASME Code Section VIII, Division 2 uses load combinations and typically does not allow for increase in allowable stress; however, the seismic load is usually reduced when combined with other types of loads and so the effect is similar. The pressure vessel may only experience an earthquake several times during the life of equipment, though the vessel must be designed to withstand any seismic event.

Designing a vessel to be invulnerable to any earthquake would be both impractical and uneconomical. Building codes and design standards use the ability of the structure to yield and absorb energy in a ductile manner during a seismic event for design. This is part of the basis of the 'R' factor, which is used to reduce the design strength for a structure. As a result of the designed structure undergoing permanent deformation during an earthquake, some of the structure may be lightly or severely damaged. In the case of vessel support design, it should be understood that the anchor bolts provide a benefit to the vessel by yielding and absorbing energy that could otherwise have a greater impact on the support members.

There are other things as well that must be considered in seismic analysis. These include: the direction of seismic loads and whether they can be evaluated independently or should be combined, displacements of the pressure vessel, supports, and attachments, and flexibility of attached piping. ASCE 7 provides details on how to address and evaluate these topics.

In conclusion, there are a number of factors that go into determining the magnitude of the seismic loads to be used in the design of pressure vessels. Most pressure vessels can be evaluated for seismic loads using ELFAP and hand calculations. While pressure vessel design software often includes the capability of evaluating for seismic loads, the engineer should be conversant with the appropriate Code or standard to correctly define the input parameters used in ELFAP. For those circumstances where the ELFAP is not permitted, a Modal Response Spectrum Analysis or Seismic Response History Analysis should be performed by an engineer experienced with these types of analyses and FEA.

References:

Seismic Analysis of Pressure Vessels – Hedderman Consulting

Pressure Vessel Design Manual – Denis Moss and Michael Basic

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GASKET FUNDAMENTALS – PART 1

INTRODUCTION

“Zero leakage” condition is non-existent in the industrial gasket environment. For industrial applications, it is common to define as “zero leakage” the value that is lower than a limit set according to a certain criteria. For example, the Jet Propulsion Laboratory defines as zero leakage for gases a value lower than 1 cm^3/year when the pressure differential is one atmosphere. (For reference, we can say that a drop of liquid has an average volume of 0.05 cm^3. Thus 20 drops will be necessary to exceed 1 cm^3).

If it were technically and economically feasible to manufacture perfectly smooth and polished flanges, and if we could maintain these surfaces in permanent contact, there would be no need for gaskets. However, this is not possible, and therefore, gaskets are used as sealing elements. When gasket is seated against flange surfaces, it flows, filling the imperfections between them and providing the necessary sealing. Therefore, in order to obtain adequate sealing, we must consider four factors:

1. Gasket seating stress: We must provide an adequate way of seating the gasket, so that it will be able to flow and fill the flange imperfections. At the same time, this seating stress must be limited in order to prevent the crushing of the gasket by an excess of compression.
2. Sealing force: There must be residual pressure on the gasket in order to keep it in contact with the flange surfaces, thus avoiding leakage.
3. Material selection: The gasket material must resist the pressure as well as the fluid to which it is subjected.
4. Surface finish: There is a recommended flange surface finish for each style of gaskets.

FORCES IN A FLANGED JOINT

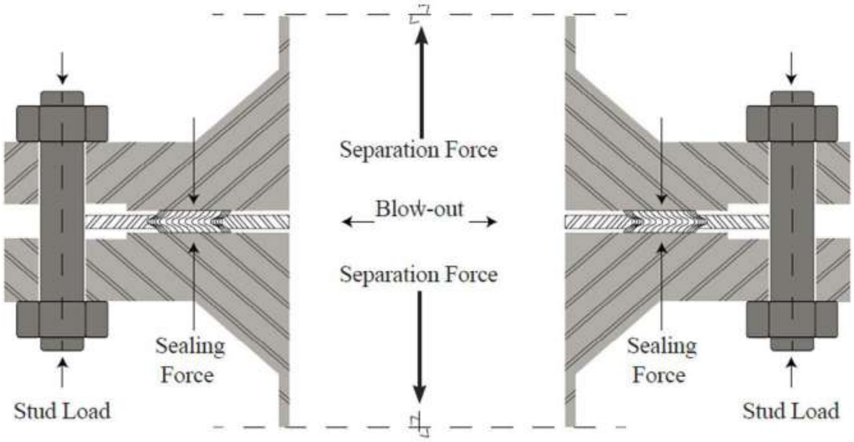


Figure 1: Forces in a Flanged Joint

Blowout force: Originated by internal pressure; it tends to blow out the gasket

Separation force: Also originated by internal pressure; it tends to separate the flanges. Also known as hydrostatic force

Bolt (Stud) force: It is the total load exercised by the bolts or studs

Sealing force: It is the force which compresses the flanges against the gasket

Initially, the sealing force is equal to the bolt (stud) force. After pressurizing the system, it is equal to the bolt (stud) force minus the separation force. The bolt (stud) force initially applied to the gasket, besides causing flow of gasket material, must:

- Compensate for the separation force caused by internal pressure
- Be sufficient to maintain a residual stress on the gasket, avoiding fluid leakage
- Compensate the relaxation of the flanged joint that will occur during service life

From a practical point, in order to maintain the sealing, the residual stress must be “x” times the fluid pressure. The minimum value of this force can be calculated by various methods; one of them being “Appendix 2 – Rules for Bolted Flange Connections with Ring Type Gaskets”. Appendix 2 recommends typical values for the characteristics of the gasket “m” and “y” that have not been updated in a very long time.

RELAXATION

Relaxation is the gradual loss of the bolt (stud) load applied when installing the gasket; and begins immediately after installing and tightening up the studs. The characteristic of the flanged joint must be considered to ensure their performance throughout the service life of the equipment. The relaxation can be divided into two phases: the initial phase, which occurs immediately after the installation, and then throughout the service of the gasket. The initial relaxation is caused mainly by the flow of the gasket when filling in the irregularities between the flanges.

In many critical applications, tightening at room temperature may not be sufficient to ensure sealability of the system. For these situations, startup retightening procedures are performed, as the system is set up. The chart below shows the relaxation of metallic gaskets in laboratory tests. It can be observed that there is a loss of up to 25% of the initial installation stress.

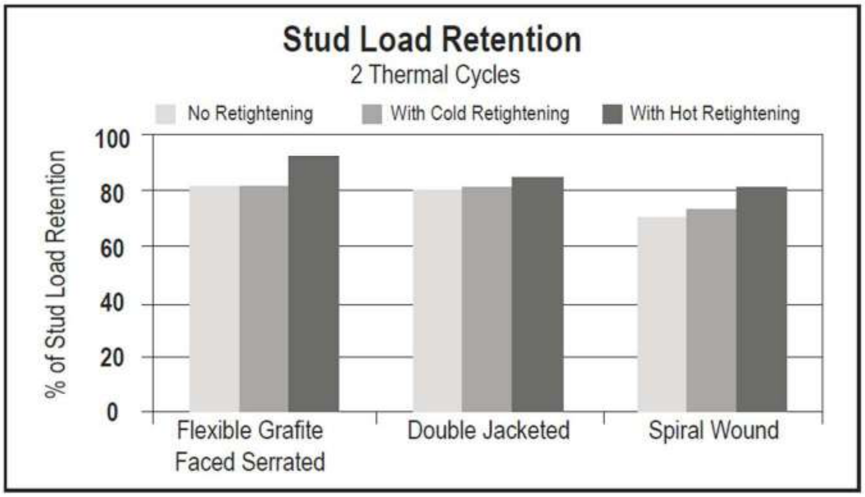


Figure 2: Relaxation of Metallic Gaskets in Laboratory Tests

The relaxation during the time of service of the gasket occurs primarily in systems operating at elevated temperatures with thermal cycles. Depending on the type of gasket, the operating temperature, and the frequency of the thermal cycles, the effects can take months or even years before there is leakage in the flange joint. The chart below shows the relaxation of a double jacketed gasket over a period of 17 months. In the initial part of the chart, we can observe the initial relaxation and the hot tightening right after system startup. The continued loss of bolt (stud) load can be observed until the disassembly of the gasket when bolts (studs) had only 45% of the initial tightening.

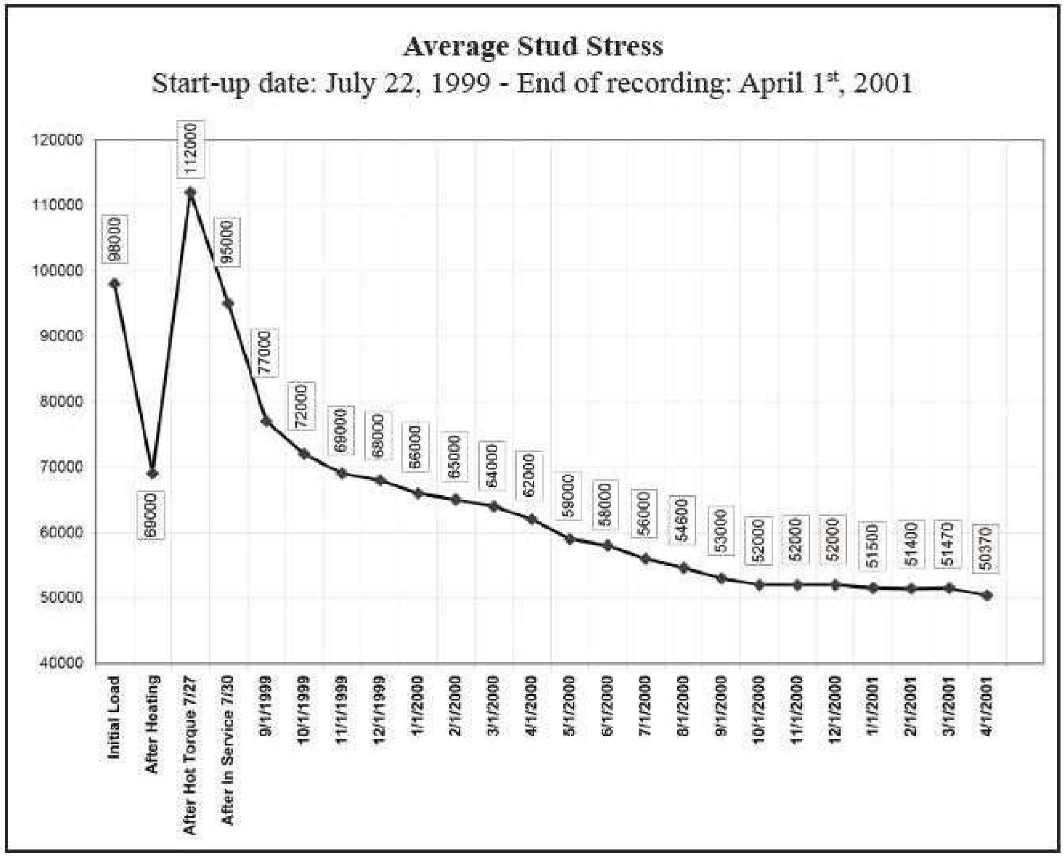


Figure 3: Relaxation of Double Jacketed Gasket over the Service Life

SURFACE FINISH

Table 1: Recommended Flange Surface Finish for Different Types of Gaskets

Flat non-metallic	
1/16” thick	125 – 250 μ in
> 1/16” thick	125 – 500 μ in
Faced corrugated metal	125 – 250 μ in
Spiral wound	80 – 250 μ in
Metal jacketed	1000 μ in (maximum)
Faced grooved metal	125 – 250 μ in
Ring joint	63 μ in

Following are recommendations to harmonize the flange surface finish with the gasket style:

- The surface finish has a great influence on the sealability.

- A minimum seating stress must be obtained in order to seat the gasket against the flange imperfections. A soft gasket (like rubber) requires a seating stress lower than a densergasket (like compressed fiber sheets).
- The gasket seating force is proportional to the flange contact area. Reducing the width of the gasket or its contact area with the flange will reduce the seating force.
- Regardless of the style of gasket or finish used, it is important that there are no scratches or radial tool marks on the sealing surfaces of the flanges. These radial markings are very difficult to seal, and with a metal gasket, it becomes almost impossible.
- The phonographic grooves are more difficult to seal than the concentric ones. The gaskets, when seated, must flow up to the bottom of the grooves to prevent any leak path to be formed from one end of the spiral to the other.

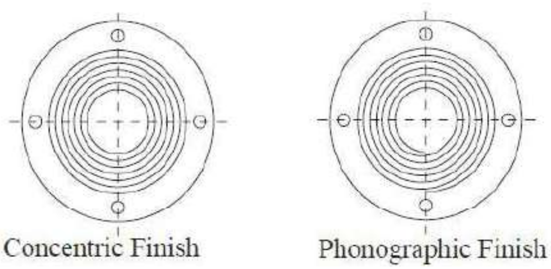


Figure 4: Concentric vs Phonographic Gasket Surface Finish

- Since the materials have different hardness and flow characteristics, the choice of a type of flange surface finish is going to depend basically on the gasket material.

STYLES OF FLANGES

FLAT FACE: These are non-confined gaskets. The contact surfaces of both flanges are flat. The gasket can be style RF which goes up to the bolts (studs), or style FF, covering the entire contact surface. Flat faces are normally used in flanges made out of fragile materials; for these reasons, it is not recommended to use RF gasket style (sketch on right) to avoid flange rotation.

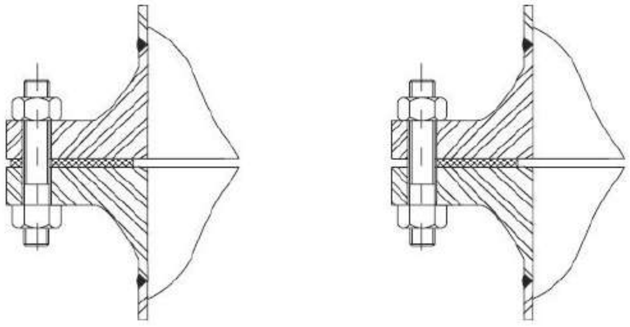


Figure 5: Flat Face Gaskets

RAISED FACE: These are non-confined gasket. Contact surfaces are raised about 1/16 in. or 1/4 in. Normally, the gasket outside diameter is up to the bolts (studs). RF flange are used more often in piping.

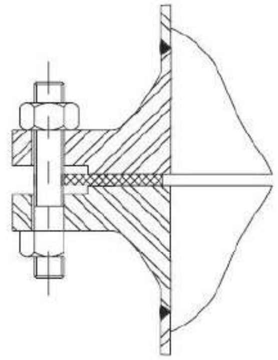


Figure 6: Raised Face Gasket

TONGUE AND GROOVE FACE: These are totally confined gaskets. The groove depth is equal to or greater than the tongue height. The gasket has, usually, the same width as the tongue. It is necessary to separate the flanges in order to change the gasket. Since this style of flange exerts high seating stress on the gasket, it is not recommended for non-metallic gaskets.

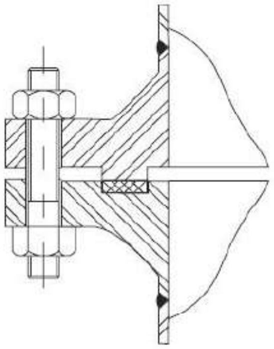


Figure 7: Tongue and Groove Gasket

MALE AND FEMALE FACE: These are semi-confined gaskets. The most common style is the one on the left. The depth of the female is equal to or less than the height of the male in order to avoid the possibility of direct contact of the flanges when the gasket is compressed. The female external diameter is up to 1/16" larger than the male. The flanges must be separated to change the gasket.

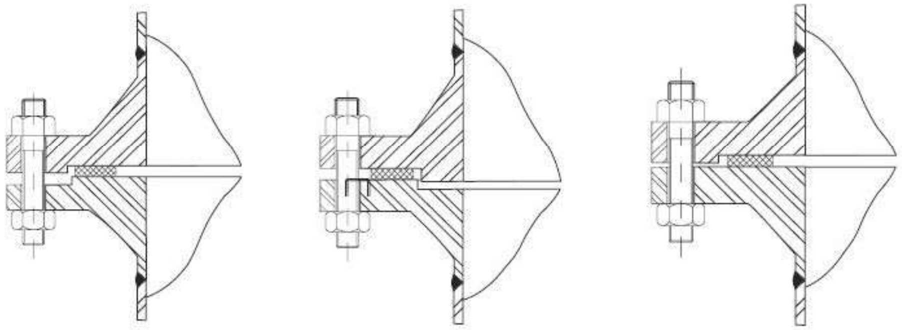


Figure 8: Male and Female Face Gaskets

FLAT FACE AND GROOVE: These are totally confined gaskets. The external face of one of the flanges is plain, and the other has a groove where the gasket is assembled. They are used in applications where the distance between the flanges must be precise. When the gasket is seated, the flanges touch each other. Only very resilient gaskets can be used in these types of flanges. Spiral-wound, O-rings, non-solid metallic, pressure activated and jacketed with metallic fillers are recommended.

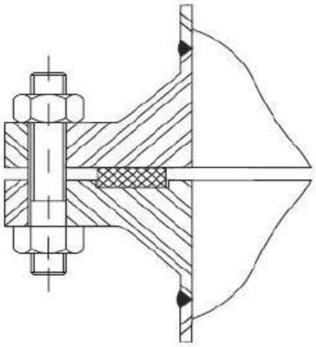


Figure 9: Flat Face and Groove Gasket

GASKET TYPES

Gaskets can be segregated into three (3) main categories:

- Non-metallic (soft)
- Semi-metallic
- Metallic

Gasket types will be discussed in part 2 of this article in an upcoming issue.

GASKET INSTALLATION

Even the best specified gasket might fail if joint assembly procedures are not followed. Basics such as lack of bolt and nut lubrication, excessive or insufficient tightening and gaskets not centered on the flange are major causes of leaks. ASME PCC-1 – Guidelines for Pressure Boundary Bolted Flange Joint Assembly provides guidelines for installation of gaskets in process piping and pressure vessels, as well as for training and qualification of gasket installation personnel.

BOLTS AND STUDS

In most applications, the material of the studs is the alloy steel ASTM A193-B7 – a high yield and tensile strength alloy. Figure 10 shows a typical graph of the bolt tightening stress according to the rotation angle of the nut (rotation angle is the angle of nut rotation once the nut face and bolt head contact the clamp surface). The usual stress range is between 40% and 75% of the yield strength of the bolt material. Tightening with values below 40% of yield strength do not stretch the bolt enough to allow for gasket relaxation; in this situation there is a risk of losing gasket stress and experience high leak rate or even gasket blowout. The widely used material for nuts is ASTM A194-2H. If the tightening is done with a torque wrench, it is recommended to use hardened washers to reduce friction between the nut and the flange surface. The most widely used material for washers is ASTM F436.

FRICTION FACTOR

Friction factor is primarily responsible for maintaining the clamping force of a bolt. This factor can vary widely depending on the type of lubricant, and the condition of studs or bolts and washers. ASME PCC-1 recommends a value of k = 0.20 for lubricated alloy steel bolts and a value of k = 0.16 for PTFE coated bolts.

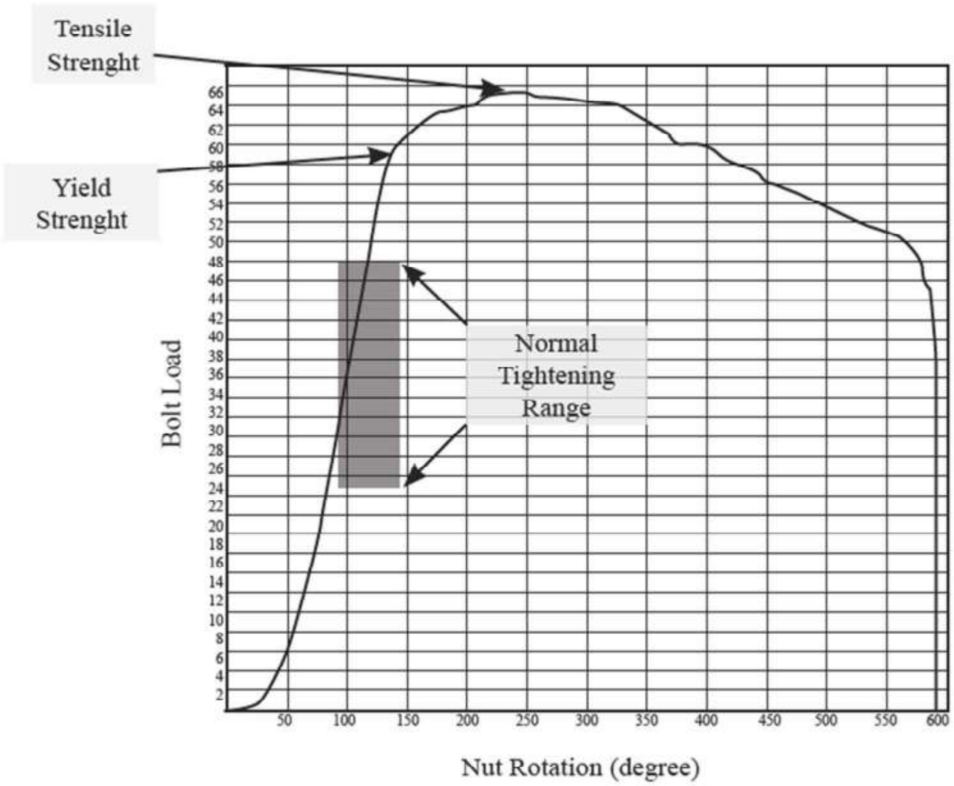


Figure 10: Bolt Tightening Stress according to Rotation Angle of Nut

FLATNESS TOLERANCE

ASME PCC-1 classifies gaskets into two groups:

1. Soft gaskets – those with greater capacity to compensate flange sealing surface irregularities. Examples: Spiral-wound gaskets and non-metallic gaskets like Compressed Fiber, PTFE or Flexible Graphite greater than 1/16 in. thick.
2. Hard gaskets – those with less capacity to compensate flange sealing surface irregularities. Examples: Camprofile, Double-jacketed, Ring Joints and non-metallic gaskets like Compressed Fiber, PTFE or Flexible Graphite less than 1/16 in. thick.

The flatness tolerance of the flange sealing surface recommended by ASME PCC-1 is shown in the table below:

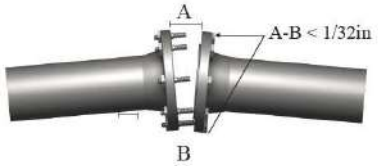
Table 2: Flange Seating Face Flatness Tolerance (inch)

Measurement	Hard Gaskets	Soft Gaskets
Acceptable variation in circumferential flange seating surface flatness	T1 < 0.006 in.	T1 < 0.01 in.
Acceptable variation in radial (across surface) flange seating surface flatness	T2 < 0.006 in.	T2 < 0.01 in.
Maximum acceptable pass-partition surface height vs flange face	-0.010 in. < P < 0.0 in.	-0.020 in. < P < 0.0 in.

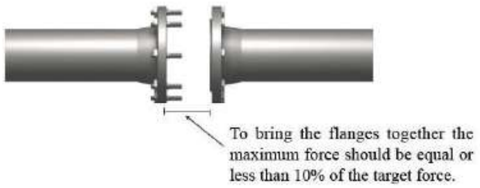
FLANGE MISALIGNMENT

Flanges must be aligned as shown in Figure 11.

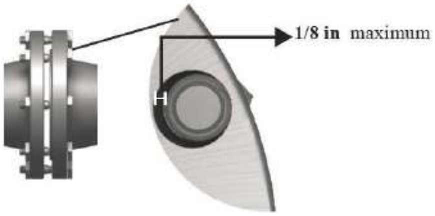
Centerline



Parallelism



Flange Gap



Bolt Hole Alignment

Figure 11: Flange Misalignment

INSTALLATION PROCEDURE

GASKET INSTALLATION

To obtain a satisfactory seal, it is necessary that basic procedures be followed during gasket installation. These procedures are of fundamental importance for a successful operation no matter what style of gasket is used.

a) Inspect the flange sealing surface. Check for tool marks, dents, scratches or corrosion. Radial tool marks on sealing surface are difficult to seal regardless of the style of gasket. Be sure that the flange finish is adequate for the style of gasket to be installed.

b) Inspect the gasket. Verify that the gasket material is compatible with the intended service. Check for defects and shipping or storage damage.

c) Inspect and clean bolts, nuts and washers.

d) Lubricate both threads and the nut contact surfaces. Do not install bolts and nuts without lubrication. A good lubricant will provide a better application of the torque and, consequently, higher precision of the bolt load. There are applications where bolts cannot be lubricated; for these situations, the friction factor, k, must be previously determined.

e) For Raised Face or Flat Faced flanges installed vertically, start installation by bolts on the lower part. Install the gasket, then the other bolts.

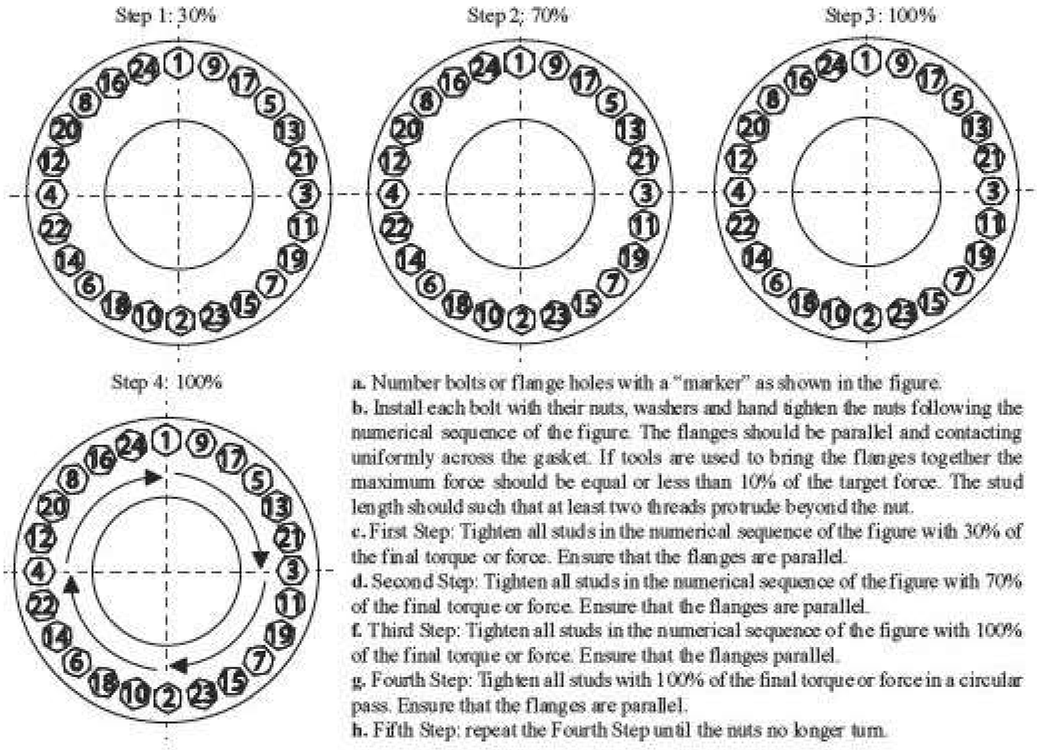
f) For Male and Female or Tongue and Groove flanges, the gaskets should be installed in the center of the groove.

g) Do not use adhesive or other fixing agent on the gasket or face flanges.

h) Install the bolts and hand tighten them.

BOLT NUMBERING AND INSTALLATION

To help the installation and avoid errors, it is recommended to number bolts following the selected tightening sequence method. The star or legacy tightening pattern sequence is the best known and applied method of installing gaskets as shown in Figure 12.



a. Number bolts or flange holes with a "marker" as shown in the figure.
b. Install each bolt with their nuts, washers and hand tighten the nuts following the numerical sequence of the figure. The flanges should be parallel and contacting uniformly across the gasket. If tools are used to bring the flanges together the maximum force should be equal or less than 10% of the target force. The stud length should such that at least two threads protrude beyond the nut.
c. First Step: Tighten all studs in the numerical sequence of the figure with 30% of the final torque or force. Ensure that the flanges are parallel.
d. Second Step: Tighten all studs in the numerical sequence of the figure with 70% of the final torque or force. Ensure that the flanges are parallel.
f. Third Step: Tighten all studs in the numerical sequence of the figure with 100% of the final torque or force. Ensure that the flanges parallel.
g. Fourth Step: Tighten all studs with 100% of the final torque or force in a circular pass. Ensure that the flanges are parallel.
h. Fifth Step: repeat the Fourth Step until the nuts no longer turn.

Figure 12: Example of Star or Legacy Tightening Pattern for a 24-bolt Flange using a Single Tool

References:

Industrial Gaskets *by* Jose Carlos Veiga

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PRESSURE VESSEL MATERIALS

The vast majority of pressure vessels are made from ferrous and non-ferrous alloys. Ferrous alloys are defined as those having more than 50% iron and nonferrous alloys are defined as those having less than 50% iron. Ferrous alloys include carbon and low alloy steels, stainless steels, cast iron, wrought iron and quenched and tempered steels. Nonferrous alloys include aluminum, copper, nickel, titanium and zirconium alloys. ASTM designates all ferrous alloys by letter A, and all nonferrous alloys by B. ASME uses the prefix SA and SB respectively. In most cases, ASTM and ASME specifications are identical.

A summary of the cost of some frequently used pressure vessel materials is given in the Table below:

Type	**Cost ($/lb)**
Aluminum	2.00
Carbon Steel	0.80
Copper, Brass	8.50
Hastelloys	24.00
Incoloy	10.00
Inconel	18.00
Low Alloy Steel	1.60
Monel	15.00
Stainless Steel	2.70
Tantulum	80.00
Titanium	35.00
Zirconium	55.00

FERROUS ALLOYS

Iron alloys with carbon content of less than 2% are known as steels, and those with more than 2% are known as cast irons. Most steels used in pressure vessel applications have a carbon content of less than 0.4%. Steels with carbon content of over 0.4% are very brittle and hard to weld. ASME Section VIII, Division 1 divides steel alloys into three major categories:

1. Carbon Steels: These are widely used in pressure vessels. They have mainly silicon and manganese as the major alloying elements and are limited to application temperatures below about 1000°F.

2. Low-Alloy Steels: These are essentially chromium (up to 10%), molybdenum and nickel-alloy steels. These elements enhance the steel for high-temperature applications and in hydrogen service.
3. High-Alloy Steels: These are commonly referred to as stainless steels. They have mainly chromium (over 10%), nickel and molybdenum alloys. *Stainless steels will be discussed in depth later in this article*.

Some of the common alloying elements and their effects on steel are shown below:

Aluminum - Restricts grain growth.

Chromium - Increases resistance to corrosion and oxidation. Increases hardenability. Adds strength at high temperature.

Manganese - Counteracts sulfur brittleness.

Molybdenum - Increases hardenability. Raises grain-coarsening temperature. Counteracts tendency toward temperature brittleness. Enhances corrosion resistance.

Nickel - Strengthens annealed steels. Toughens steel.

Silicon - Improves oxidation resistance. Increases hardenability. Strengthens steel.

Titanium - Prevents formation of austenite in high-chromium steels. Prevents localized depletion of chromium in stainless steels during long heating.

Vanadium - Increases hardenability. Resists tempering.

NONFERROUS ALLOYS

ASME Section VIII, Division 1 lists five nonferrous for code construction – Aluminum, Copper, Nickel, Titanium and Zirconium. These alloys are normally used in corrosive environment or at elevated temperatures where ferrous alloys are unsuitable. Nonferrous alloys are nonmagnetic except for commercially pure nickel which is slightly magnetic.

Aluminum Alloys: Aluminum alloys are nonmagnetic, are light in weight, have good formability, and have an excellent weight-strength ratio. Aluminum surfaces exposed to the atmosphere form an invisible oxide skin that protects the metal from further oxidation.

Copper and Copper Alloys: Most copper alloys are used because of their good corrosion resistance and machinability. They are also homogenous as compared with steel or aluminum and thus not susceptible to heat treatment. Their strength may only be altered by cold working.

Nickel and High Nickel Alloys: Nickel and high nickel alloys have excellent corrosion and oxidation resistance which makes them ideal for high temperature applications and corrosive environment. Nickel products are normally called by their commercial names like Monel, Inconel, Incoloy and Hastelloy.

Titanium and Zirconium: These alloys are used in the process equipment subjected to severe environment. The modulus of elasticity for both Titanium and Zirconium is about half that of steel. Also, the coefficient of thermal expansion of both is about half that of steel. The density of Zirconium is slightly less than that of steel, and the density of Titanium is a little over half that of steel.

STAINLESS STEELS

Steels with a chromium content of 11% or more, but less than 30%, are known as stainless steels because of their excellent resistance to corrosion. The most important additional alloying element after chromium is nickel. Other alloying elements may be added including manganese, molybdenum, columbium, titanium, selenium, silicon, and sulfur, all of which result in properties required for special service. Stainless steels are frequently used for construction of petrochemical processing equipment and other applications to:

- Provide resistance to corrosive environment (thus increasing the service life of the equipment and safety of working personnel);
- Provide strength and oxidizing resistance at elevated temperatures and impact strength at low temperatures; and
- Facilitate the cleaning of the equipment.

Stainless steels become corrosion resistant (passive) because of the formation of an unreactive film which adheres tightly to the surface of the metal and acts as barrier protecting the metal against further attack in certain types of environment. If the chromium content is less than 11%, the film is discontinuous, and the corrosion resistance of such steels approaches the relatively poor corrosion resistance of ordinary steel. No protective coatings such as paints are applied to the surface of stainless steel parts, since they would only prevent oxygen penetration for formation of the passive layer. Based on the metallurgical microstructure, stainless steels can be classified as austenitic, ferritic, or martensitic.

Austenitic Stainless Steels: These are high chromium-nickel-iron alloys. Because of their large percentage of nickel, 300 series stainless steels retain their austenitic structure after cooling, with Cr, Ni and C in solid solution with iron. They are non-magnetic, highly corrosion resistant even at temperatures up to 1500°F, and hardenable only by cold working; and they possess high impact strength at low temperatures. The typical and most commonly used grades of stainless steel are grades 304 and 316. Higher Cr content austenitic stainless steels (grades 309 and 310) are resistant to oxidation and sulfur attack up to 2000°F.

The primary problem with austenitic stainless steels is grain boundary sensitization. At temperatures in the range of 800 to 1600°F, carbon molecules diffuse to the grain boundaries and precipitate out of the solid solution as chromium carbide at the boundaries. The effect is depletion of Cr content in the thin envelope surrounding each grain. The carbide formed is not as corrosion resistant as the metal from which it develops, and the corrosion resistance of the envelope depleted of Cr is drastically reduced. The stainless steel becomes susceptible to intergranular corrosion and is said to be sensitized. The corrosion properties of sensitized steel can be restored by *desensitization*, that is, heating above 1600°F to dissolve carbides and subsequent rapid cooling.

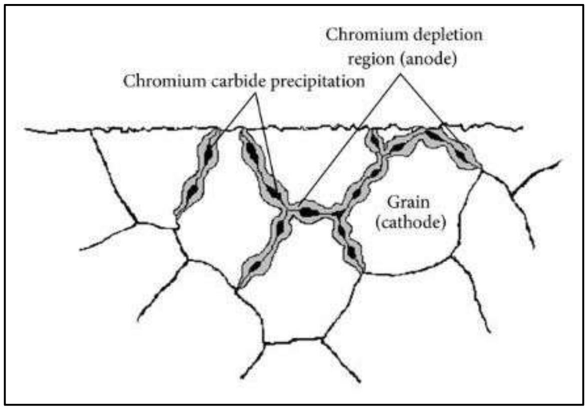

Figure 1: Grain Structure in Type 300 Sensitized Stainless Steel

Ferritic Stainless Steels: Ferritic stainless steels usually include straight chromium stainless steels with 16 to 30% chromium. They are non-hardenable by heat treatment. A typical stainless steel of this group is Type 430. The grade quite often used for corrosion-resistance cladding or lining is Type 405 which contains only 12% chromium; however, addition of aluminum renders it ferritic and non-hardenable.

Ferritic stainless steels are sensitized by heating to a temperature of 1700°F and then air cooled at normal rates. If they are cooled slowly (in a furnace), their resistance to intergranular corrosion is preserved. Sensitized ferritic stainless steel is much less corrosion resistant than sensitized austenitic stainless steel. When ferritic stainless steels are heated into the 750 to 900°F range for a prolonged period of time, notch toughness is reduced. This has been termed 885°F embrittlement.

Ferritic stainless steels exhibit lower ductility at low temperatures which limits their use in low temperature applications. In general, ferritic stainless steels are seldom used in pressure vessel construction, except for the corrosion-resistant lining or cladding (Types 405 or 410S), heat exchanger tubing, and vessel internal hardware (trays) for less corrosive environments, since they are not as expensive as austenitic stainless steels.

Martensitic Stainless Steels: Martensitic stainless steels include straight chromium steels, usually with 11 to 16 percent chromium as alloying element. They are hardenable by heat treatment, that is, their strength and hardness can be increased at the expense of ductility. Type 410 is typical of this group. In the annealed condition at room temperature, it has ferritic structure. When heated from 1500°F to 1850°F, its microstructure changes to austenitic. If the steel is then cooled suddenly, for instance as in deposited weld metal with adjacent base metal zones in air, part of austenite changes into martensite, a hard and brittle material.

Martenistic stainless steels are only rarely used as construction material for pressure vessel parts. They are the least corrosion resistant of the stainless steel grades, and if welding is used in fabrication, heat treatment is required.

DESIGN AND FABRICATION PROCEDURE FOR STAINLESS STEELS

Design procedures for austenitic stainless steels will be the same as for carbon steel vessels, modified by the technical properties of austenitic stainless steels. Since austenitic stainless steels are used mainly for highly severe environmental service conditions, they are more often subject to stress-corrosion cracking than are carbon steels. Under such conditions, they are sensitive to stress concentrations, particularly at higher temperatures (above 650°F) or at normal temperatures under cyclic conditions. Surface conditions have a great effect on fatigue strength and on corrosion resistance.

Weld surfaces in contact with the operating fluid are often ground smooth. Welded connections should be designed with minimum stress concentrations. Abrupt changes, e.g. fillet welds, should be avoided, and butt welds should preferably be used and fully radiographed. The edge of the weld deposits should merge smoothly into the base metal without undercuts or abrupt transitions. Sound, uncontaminated welds are important. The welds should be located away from any structural discontinuities. Weld deposition sequences should be used that will minimize the residual stresses. Austenitic stainless steels conduct heat more slowly than carbon steel, and expand and contract to a greater degree for the same temperature range. Thermal differentials, larger than for carbon steels, should be minimized by construction which permits free movement. These steels tend to warp and crack due to thermal stresses. All stainless steel parts should be designed to facilitate cleaning and self-draining without any crevices or spots, where dirt could accumulate and obstruct the access of oxygen to form the protective layer.

Fabrication and Handling - Since stainless steels exhibit maximum resistance to corrosion only when thoroughly clean, preventive measures to protect cleaned surfaces should be taken and maintained during fabrication, storage and shipping. Special efforts should be made at all times to keep stainless steel surfaces from coming into contact with other metals. For cleaning, only clean stainless steel wool and brushes should be used. If flame cutting is used, additional metals should be removed by mechanical means to provide clean, weldable edges. All grinding of stainless steels should be performed with aluminum oxide or silicon carbide grinding wheels bonded with resin or rubber, and not previously used on other metal. Proper identification and correct marking of the types of the material is important.

LOOK IN THIS SPACE IN A FUTURE ISSUE OF THE NEWSLETTER FOR A COMPREHENSIVE DISCUSSION ON FATIGUE OF METALS.

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Structural Analysis and Design of Process Equipment by Maan Jawad and James Farr

Pressure Vessel Design Handbook by Henry Bednar

TWELVE GREATEST CHALLENGES FOR SPACE EXPLORATION

HUMANITY BEGAN IN Africa. But we didn't stay there, not all of us - over thousands of years our ancestors walked all over the continent, then out of it. And when they came to the sea, they built boats and sailed tremendous distances to islands they could not have known were there. WHY?

Probably for the same reason we look up at the moon and the stars and say, "What's up there? Could we go there? Maybe we could go there." Because it's something human beings do.

Space is, of course, infinitely more hostile to human life than the surface of the sea; escaping Earth's gravity entails a good deal more work and expense than shoving off from the shore. But those boats were the cutting-edge technology of their time. Voyagers carefully planned their expensive, dangerous journeys, and many of them died trying to find out what was beyond the horizon. So why keep doing it? There are many reasons:

- Spin-off technologies - ranging from small products of convenience to discoveries that might feed millions or prevent deadly accidents or save the lives of the sick and injured.
- We shouldn't keep all our eggs in this increasingly fragile basket—one good meteor strike and we all join the non-avian dinosaurs.
- It might be good for us to unite behind a project that doesn't involve killing one another, that does involve understanding our home planet and the ways we survive on it and what things are crucial to our continuing to survive on it.
- Moving farther out into the solar system might be a good plan, if humanity is lucky enough to survive the next 5.5 billion years and the sun expands enough to fry the Earth.

Space is not only hostile to human life – getting to space is incredibly hard also. Listed below are 12 challenges that must be overcome in any space journey.

DEPARTURE - Gravity's a Drag

You want to get off the earth quickly and with as little extra weight as possible. But powerful forces conspire against you - specifically, gravity. If an object on Earth's surface wants to fly free, it needs to shoot up and out at speeds exceeding 25,000 mph.

That takes serious money. It cost nearly $200 million just to *launch* the Mars Curiosity rover, about a tenth of the mission's budget, and any crewed mission would be weighed down by the stuff needed to sustain life. Composite materials like exotic-metal alloys and fibered sheets could reduce the weight; combine that with more efficient, more powerful fuel mixtures and you get a bigger bang for your booster.

But the ultimate money saver will be reusability. SpaceX's Falcon 9, for example, was designed to relaunch time and again. The more you go to space, the cheaper it gets.

PROPULSION - Our Ships Are Way Too Slow

Hurtling through space is easy. It's a vacuum, after all; nothing to slow you down. But getting started is a bear. The larger an object's mass, the more force it takes to move it - and rockets are kind of massive. Propulsion needs a radical new method.

SPACE JUNK - It's a Minefield up There

Congratulations! You've successfully launched a rocket into orbit. But before you break into outer space, a rogue bit of broke-ass satellite comes from out of nowhere and caps your second-stage fuel tank. No more rocket.

Space debris is very real. The US Space Surveillance Network has eyes on 17,000 objects - each at least the size of a softball—hurtling around Earth at speeds of more than 17,500 mph. Launch adapters, lens covers, even a fleck of paint can punch a crater in critical systems. Whipple shields - layers of metal and Kevlar - can protect against the bitsy pieces, but nothing can save you from a whole satellite. Some 4,000 satellites orbit Earth, most dead in the air. Mission control avoids dangerous paths, but tracking isn't perfect.

NAVIGATION - There's No GPS for Space

The Deep Space Network, a collection of antenna arrays in California, Australia, and Spain, is the only navigation tool for space. Everything from student-project satellites to the New Horizons probe meandering through the Kuiper Belt depends on it to stay oriented. But as more and more missions take flight, the network is getting congested. The switchboard is often busy. So in the near term, NASA is working to lighten the load. Atomic clocks on the crafts themselves will cut transmission time in half, allowing distance calculations with a single downlink. And higher-bandwidth lasers will handle big data packages, like photos or video messages.

The farther rockets go from Earth, however, the less reliable this method becomes. For future missions, we would have to design an autonomous system that would collect images of targets and nearby objects and use their relative location to triangulate a spaceship's coordinates - no ground control required. A deep-space positioning system - DPS for short.

RADIATION - Space Turns You into a Bag of Cancer

Outside the safe cocoon of Earth's atmosphere and magnetic field, subatomic particles zip around at close to the speed of light. This is space radiation, and it's deadly. Aside from cancer, it can also cause cataracts and possibly Alzheimer's. When these particles knock into the atoms of aluminum that make up a spacecraft hull, their nuclei blow up, emitting yet more superfast particles called secondary radiation. You're actually making the problem worse.

A better solution? One word: plastics. They're light and strong, and they're full of hydrogen atoms, whose small nuclei don't produce much secondary radiation. NASA is testing plastics that can mitigate radiation in spaceships or space suits. Or how about this word: magnets. Scientists on the Space Radiation Superconducting Shield project are working on a magnesium diboride superconductor that would deflect charged particles away from a ship. It works at -263° C, which is balmy for superconductors, but it helps that space is already so damn cold.

FOOD AND WATER - Mars Has No Supermarkets

Large-scale gardening in zero gravity is tricky. Water wants to float around in bubbles instead of trickling through soil, so engineers have devised ceramic tubes that wick it down to the plants' roots. Also, existing vehicles are cramped. Some veggies are already pretty space-efficient, but scientists are working on a genetically modified dwarf plum tree that's just 2 feet tall. Proteins, fats, and carbs could come from a more diverse harvest - like potatoes and peanuts.

All that's for naught, though, if you run out of water. GMOs could help here too. NASA is working on a water filter made of genetically modified bacteria. It is basically a water recycling system with a useful life of 75 or 80 years. This filter would continually replenish itself, just like your innards do.

BONE AND MUSCLE WASTING - Zero Gravity Will Transform You into Mush

Weightlessness wrecks the body: It makes certain immune cells unable to do their jobs, and red blood cells explode. Astronauts on the ISS exercise to combat muscle wasting and bone loss, but they still lose bone mass in space, and those zero-g spin cycles don't help the other problems. Artificial gravity would fix all that.

In his lab at MIT, former astronaut Laurence Young is testing a human centrifuge: Victims lie on their side on a platform and pedal a stationary wheel as the whole contraption spins around. The resulting force tugs their feet - just like gravity, but awkward.

Young's machine is too cramped to use for more than an hour or two a day though; so for 24/7 gravity, the whole spacecraft will have to become a centrifuge. A spinning spaceship could be shaped like a dumbbell, with two chambers connected by a truss. As it gets easier to send more mass into space, designers could become more ambitious - but they don't have to reinvent the wheel. Remember the station in *2001: A Space Odyssey*? The design has been around since 1903.

MENTAL HEALTH - Interplanetary Voyages Are a Direct Flight to Space Madness

When physicians treat stroke or heart attack, they sometimes bring the patient's temperature way down, slowing their metabolism to reduce the damage from lack of oxygen. It's a trick that might work for astronauts too. Which is good, because to sign up for interplanetary travel is to sign up for a year (at least) of living in a cramped spacecraft with bad food and zero privacy - a recipe for space madness. That's why it may be better to sleep through it. Cold storage would be a twofer: It cuts down on the amount of food, water, and air a crew would need *and* keeps them sane.

TOUCHDOWN - Crashing Is Not an Option

When you're careening through frictionless space at 200,000 mph (assuming you've cracked fusion) for months or even years, and a formerly distant world is finally filling up your viewport, all you have to do is land. But there is the planet's gravity to worry about. You don't want your touchdown to be remembered as one small leap for a human and one giant splat for humankind.

RESOURCES - You Can't Take a Mountain of Aluminum Ore with You

When space caravans embark from Earth, they'll leave full of supplies. But you can't take *everything* with you. Seeds, oxygen generators, maybe a few machines for building infrastructure. But settlers will have to harvest or make everything else.

Luckily, space is far from barren. Every planet has every chemical element in it, though concentrations differ. The moon has lots of aluminum. Mars has silica and iron oxide. Nearby asteroids are a great source of carbon and platinum ores - and water, once pioneers figure out how to mine the stuff. If blasters and drillers are too heavy to ship, they'll have to extract those riches with gentler techniques: melting, magnets, or metal-digesting microbes. And NASA is looking into a process that can 3-D-print whole buildings - no need to import special equipment.

In the end, a destination's resources will shape settlements, which makes surveying the drop zone critical. Just think of the moon's far side. It's been pummeled by asteroids for billions of years; whole new materials could be out there.

EXPLORATION - We Can't Do Everything by Ourselves

To spread out on a new world, we'll need a new best friend: a robot. Settling takes a lot of grunt work, and robots can dig all day without having to eat or breathe. Current prototypes - bulky, bipedal bots that mimic human physiognomy - can barely walk on Earth. So automatons will have to be everything we aren't - like, say, a lightweight tracked bot with backhoe claws for arms. That's the shape of one NASA machine designed to dig for ice on Mars: Its two appendages spin in opposite directions, keeping it from flipping over as it works.

Still, humans have a big leg up when it comes to fingers. If a job requires dexterity and precision, you want people doing it. Today's space suit is designed for weightlessness, not hiking on exoplanets. NASA's prototype Z-2 model has flexible joints and a helmet that gives a clear view of whatever delicate wiring needs fixing. When the job's done, just hop on an autonomous transporter to get home.

SPACE IS BIG - Warp Drives Don't Exist... Yet

The fastest thing humans have ever built is a probe called Helios 2. It's dead now, but if sound traveled in space, you'd hear it screaming as it whips around the sun at speeds of more than 157,000 miles per hour. That's almost 100 times faster than a bullet, but even at that velocity it would take some 19,000 years to reach Earth's first stellar neighbor, Alpha Centauri. It'd be a multigenerational ship, and nobody dreams of going to space because it's a nice place to die of old age.

To beat the clock, you need power - and lots of it. Maybe you could mine Jupiter for enough helium-3 to fuel nuclear fusion. After you've figured out fusion engines. Matter-antimatter annihilation is more scalable, but smashing those particles together is dangerous. You'd never want to do that on Earth. You do that in deep space, so if you have an accident, you don't destroy a continent.

Humanity will need a few more Einsteins working at places like the Large Hadron Collider to untangle all the theoretical knots. It's entirely possible that we'll make some discovery that changes everything. But you can't count on that breakthrough to save the day. If you want eureka moments, you need to budget for them. That means more cash for NASA - and the particle physicists. Until then, Earth's space ambitions will look a lot like Helios 2: stuck in a futile race around the same old star.

THERE IS ONLY ONE EARTH - Let's Not Boldly Go… Let's Boldly Stay

Why should we go to space? The need to explore is built into our souls, goes one argument - the pioneer spirit and manifest destiny. But scientists don't talk about pioneers anymore. Since the New Horizons probe passed by Pluto last July, we've explored every type of environment in the solar system at least once. Humans could still go dig in the dirt to study distant geology - but when robots can do it, well, maybe not.

Of course, Earth's impending destruction could provide some incentive. Deplete the planet's resources and asteroid-belt mining suddenly seems reasonable. Change the climate and space provides room for humanity (and everything else).

But that's a dangerous line of thinking. It creates a moral hazard. People think if we screw up here on Earth we can always go to Mars or the stars. However, as far as anyone knows, Earth is the only habitable place in the universe. If we're going to leave this planet, let's go because we want to - not because we have to.

References:

Article from March 2016 Issue of the WIRED magazine.

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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, and can offer a range of services related to pressure vessels, tanks and heat exchangers.

- Pressure Vessels
- Heat Exchangers
- Storage Tanks
- Oil & Gas
- Petrochemical
- Chemical
- Power
- Fertilizer

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