



FIXED EQUIPMENT NEWSLETTER

Volume 2020
October Issue

- Why Bolt Preload is Important?
- Lifting Lugs and Tailing Lugs
- Forces Acting on Bolted Flange Connection
- Heat Exchanger Fouling: Online Mitigation Methods (Part 1)
- Fastener Technical Data

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What lies ahead for the newsletter in 2021?



Every year as the newsletter nears the end of the year, I spend some time to review the articles that we have carried through the year to that point and through the previous years gone by. The objective of the review is to formulate a strategy for the newsletter for the coming year. For the newsletter issues in 2021, I would like to try out four new ideas as follows:

- 1) Continue to publish articles related to pressure vessels, heat exchangers and storage tanks; however, select articles for each issue around a theme specific to that issue. For example, articles in one issue could be about the design of heat exchanger tube sheets; while those in another issue could be about erection of pressure vessels, etc.
- 2) Starting next year, highlight one organization that works with pressure vessels, heat exchangers and storage tanks. It could be a fabricator, an erection company, a design and engineering organization, or an inspection agency, for example. This article will be provided by the organization being highlighted and will be followed by a small Q & A section with an official of the organization.
- 3) Search for new developments in this industry – whether it is introduction of new materials, a new cleaning technology for the heat exchanger, new technology in welding of pressure vessels, etc. I hope to include at least one article in every issue that will fit this description.
- 4) Finally, dedicate a corner in every newsletter issue to show sample calculations that demonstrate application of ASME/ API/ TEMA codes and standards in the design of components for pressure vessels, heat exchangers and storage tanks.

I eagerly look forward to comments on the above ideas for the newsletter in 2021; and also look forward to any new ideas you might want to share with me for future newsletter issues.

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WHY BOLT PRELOAD IS IMPORTANT?

Over the last fifty years great improvements have been made by the fastener industry in improving the design and reliability of their products. However, no matter how well designed and made the fastener itself is, it cannot alone make the joint more reliable. Fastener selection based upon an understanding of the mechanics of how a threaded fastener sustains loading and the influence that tightening procedures can play is also needed. This article provides an introduction to the basics of bolted joints and the major factors involved in the design of such joints.

It is not widely understood how a bolted joint carries a direct load. A fully tightened bolt can survive in an application that an untightened, or loose bolt, would fail in a matter of seconds. When a load is applied to a joint containing a tightened bolt it does not sustain the full effect of the load but usually only a small part of it. This seems, at first sight, to be somewhat contrary to common sense. Figure 1 shows a bolt and nut securing a bracket to a support plate.

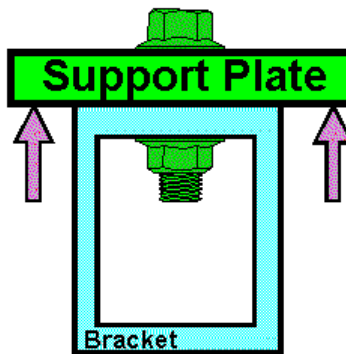


Figure 1: Bolt and Nut Securing Bracket to a Support Plate

With the nut loose on the bolt, if a weight of 1 unit is added to the bracket, as shown in Figure 2, then the force in the bolt shank will increase by 1. However, if the nut is now tightened and the weight applied, the force in the bolt shank will not increase by 1 but usually by only a small fraction of this amount. An understanding of why the bolt does not sustain the full effect of the applied load is fundamental to the subject.

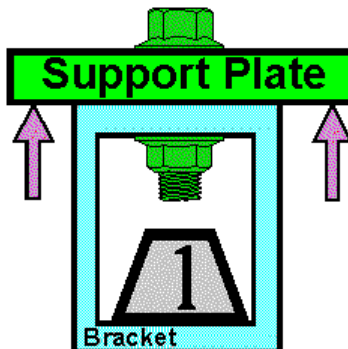


Figure 2: Weight Added to the Bracket

A model can often be of help in understanding why the bolt does not sustain the full effect of the applied load. Figure 2 is an attempt to illustrate the load transfer mechanism involved in a bolted joint by the use of a special

fastener. In the case of this fastener no significant load increase would be sustained by the fastener until the applied load exceeded the fastener's preload. (Preload is the term used for a bolt's clamp force)

With the special fastener shown, the bolt is free to move within its casing, a compression spring is included within the casing so that if the bolt is pulled down the spring will compress. A scale on the side of the casing indicates the force present in the spring and hence the force present in the shank of the bolt. Figure 3 illustrates this special fastener in its untightened condition.

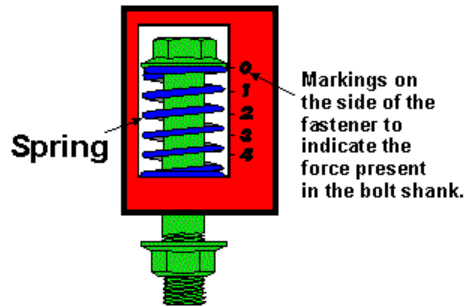


Figure 3: Special Fastener in Untightened Condition

The bolt is now inserted through a hole in a support plate and a bracket attached to the special fastener by securing a nut to the threaded shank. If the nut is now rotated so that the head of the bolt is pulled down, the spring will be compressed. If the nut is rotated so that 2 force units are indicated on the casing, the compressive force acting on the spring will be 2 and the tensile force in the bolt shank will also be 2. This is illustrated in Figure 4; this is like a tightened bolt without any working load applied.

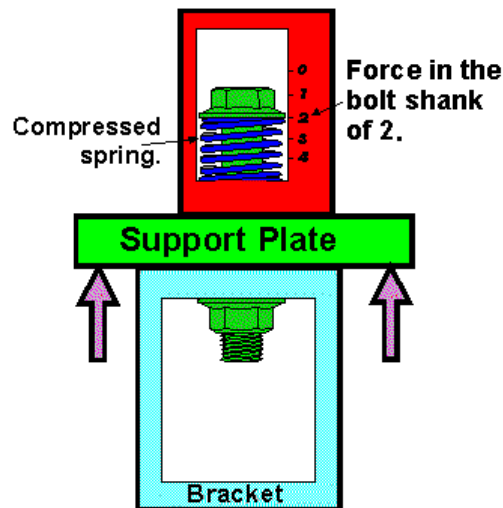


Figure 4: Tightened Bolt without any Working Load

If a weight is now added to the bracket (Figure 5) of value 1, then the initial reaction is to think that the load in the bolt must increase, otherwise what happens to the additional force? Surprisingly it will keep at its existing value of 2 - it will not 'feel' any of the additional force. To visualize why this is so - imagine what would happen if the load in the bolt did increase. To do this it would compress the spring more and a gap would be made between the bracket and the plate. If such a gap was to form then it would mean that there would be 2 units of force acting upwards - due to the spring, and 1 unit of force acting downwards from the applied weight. Clearly this force imbalance would not occur. What does happen is that the effect of the applied load is to decrease the clamp force

that exists between the plate and the bracket. With no load applied the clamp force is 2 units, with the load applied this decreases to 1 unit of force. The bolt would not actually 'feel' any of the applied force until it exceeded the bolts clamp force.

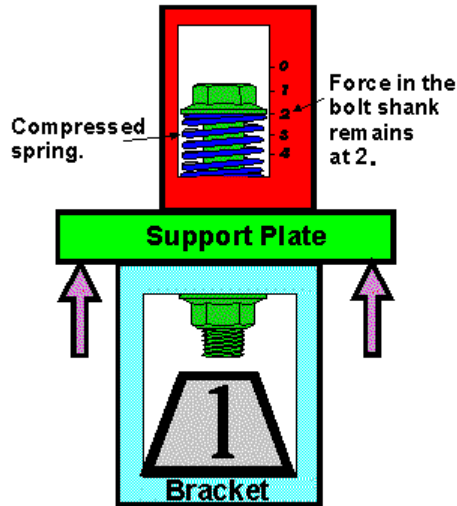


Figure 5: Weight Added to the Bracket

Older design procedures proposed calculation methods based upon the idea that the bolt will not 'feel' any of the applied load until it exceeds the bolts clamp force. That is, the bolt should be sized so that its clamp force is equal to the external load after a factor of safety has been included. With the special fastener used in this example the stiffness of the fastener is far smaller than the stiffness of the plate and bracket it clamps. Practical fasteners differ from that shown in Figure 3, 4 and 5 in that elongation of the fastener and compression of the clamped parts occurs upon tightening. This compression results in the bolt sustaining a proportion of the applied load. As the applied force reduces the clamp force existing within the joint an additional strain is felt by the bolt which increases the force it sustains. The amount of the additional force the bolt sustains is smaller than the applied force to the joint. The actual amount of force the bolt sustains depends upon the ratio of stiffness of the bolt to stiffness of the joint material.

The best way to understand and visualize how the force sustained by the bolt depends upon the joint stiffness is by the use of joint diagrams. These are the subject of the next page in this basics of bolted joints tutorial.

References:

Tutorial of Basics of Bolted Joint – Bolt Science

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LIFTING LUGS AND TAILING LUGS

Lifting lug is a plate with a hole in it, and is used for lifting pressure vessels. The hole is sized to fit a clevis pin. While lifting the pressure vessels, the lifting lugs must be able to carry the complete weight of the pressure vessel. Also, while transporting and installation, various kinds of forces depending on the mode of transportation may act on those lifting lugs.

Lifting lug can be attached to the pressure vessel directly; however it is common to weld a pad to the pressure vessel and then the lifting lug is attached to the pad. The benefit of the pad is to reduce the stress concentrations from the smaller area of the lifting lug exerted on the pressure boundary. Generally, the pad thickness should be equal to the pressure vessel component it is attached to. The load is transferred from the lifting lug to the pad and then to the pressure vessel.

Location and design of lifting lugs is particularly important for vertical vessels where the vessel must be rotated from the (typically) horizontal shipping position to the vertical lifting position. Common proven design methods include:

- Two lugs rotated 180° apart on the top tangent line of the head/shell – coupled with a tailing lug.
- Two trunnions located near the top head – coupled with a tailing lug.
- Top flange lug with a spreader – coupled with a tailing lug.

Most Lifting lugs for towers, reactors and other pressure vessels can be broadly classified into 1) ear-type, 2) trunnions, and 3) top nozzle blind type.

Ear-type: By far, the most common are the ear-type lifting lugs. These are usually installed at the top of the vessel and can be used for most vessels, especially large towers.

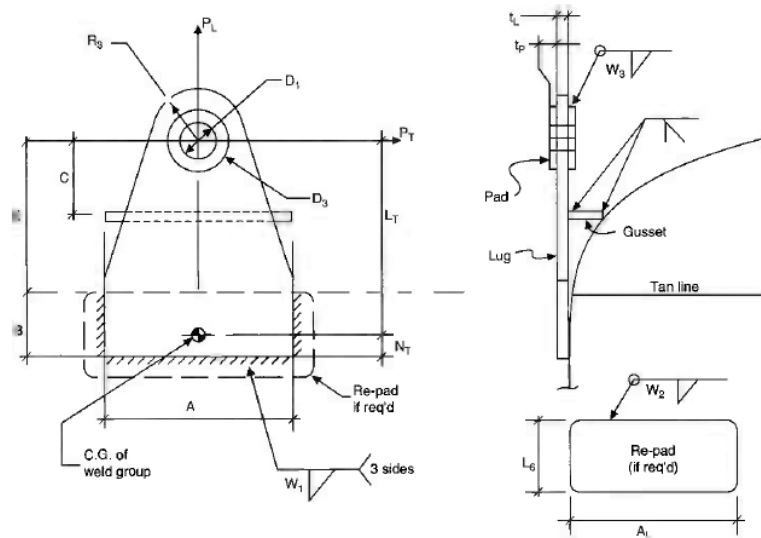


Figure 1: Ear-type Lifting Lug

Trunnions: When the tower is unusually tall, so that lifting it will require a large crane and/or a long boom, or will produce excessive bending stress in the vessel, use of trunnion type lifting lugs may be more suitable. These are attached to the vessel shell some distance down from the top head.

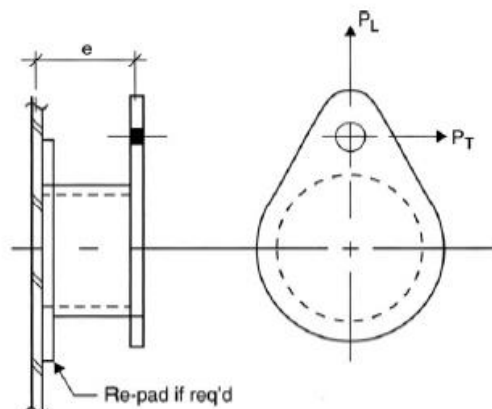


Figure 2: Trunnion

Top Nozzle Blind Type: Most heavy wall vessels have a large and strong nozzle located at the centre of their top head. With a special bolted attachment, this nozzle can be used for the erection of the vessel.

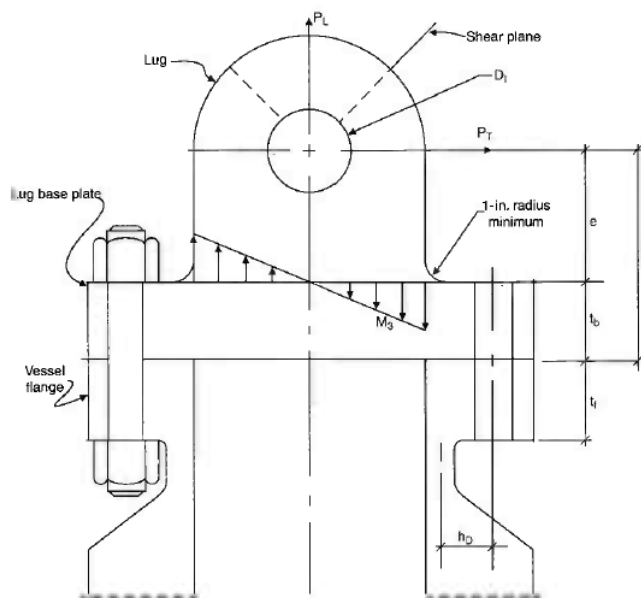


Figure 3: Top Nozzle Blind Type Lifting Lug

In lifting a tall pressure vessel, whether by lifting lugs attached near the top head, or by trunnions located lower down the shell, a tailing lug is usually required to lift the entire column off the ground in order to facilitate the uprighting of the vessel. Since most vertical vessels have skirt supports, the tailing lug is usually attached to the bottom of the vessel to take advantage of the stiffness of the base ring there. Unless the tailing load is unusually large, only one lug is required.

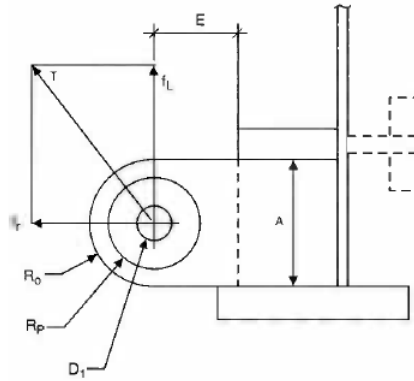


Figure 4: Tailing Lug

Lifting lugs and tailing lugs are structural elements that are used only for a short time during erection of vessels. For this reason, the most common structural steel, A-36 is adequate. However, parts of the lug that are welded directly to vessels which are constructed of alloy metals such as Cr-Mo steels, stainless steels, non-ferrous metals etc. should preferably be of the same type of material as the pressure vessel.

References:

Pressure Vessel Design Manual – Dennis Moss and Michael Basic

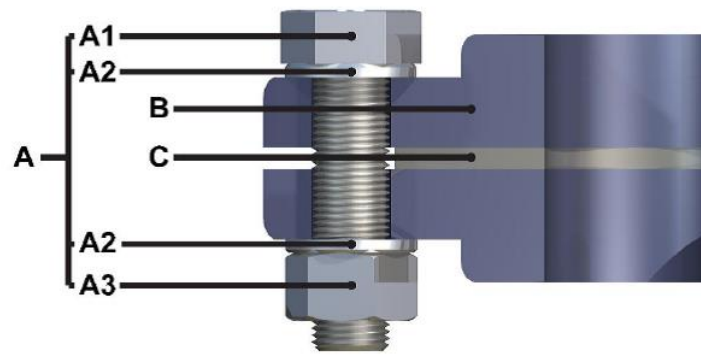
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FORCES ACTING ON THE BOLTED FLANGED CONNECTION

AN OVERVIEW

A bolted flange connection (Figure 1) is a complex mechanical system whose components must be selected and assembled properly to provide reliable sealing over a wide range of operating conditions. All of the various components of the assembled bolted flange connection are important to the proper operation of the joint. The components consist of the piping, or vessels, the flange(s), the gasket(s) and bolts. In addition to the components themselves, the joint design and assembly are critical to the long-term operation of the joint. This article will focus on the forces acting on the bolted flange connection.



A=Fastener (A1=Bolt, A2=Washers, A3=Nut); B=Flange; C=Gasket

Figure 1: The Bolted Flanged Connection

A gasket is used to create and retain a static seal between two flanges which connect a series of mechanical components while containing a wide variety of fluids (liquids and/or gases). These static seals aim to provide a complete physical barrier against the fluid contained within and block any potential leak paths through gasket material or between surfaces of flange and gasket, while still being capable of being disassembled and reassembled.

CREATING THE SEAL

To achieve a successful seal, the gasket must be resilient enough to conform to any irregularities in the mating surfaces. The gasket must also be sufficiently tough (rugged) enough to resist extrusion, creep and blowout under the operating conditions and unexpected pressure/temperature excursions. The seal is created by the clamping forces acting upon the gasket surface, compressing the gasket and causing the gasket to conform to flange imperfections. The conformance of the gasket material to the flange surface under the compressive load (contact pressure) fills any leak paths and prevents the escape of the contained fluid from the bolted flange connection while maintaining a specified leakage rate. Additional information on generating the initial seating stress can be found in Chapter 4 on Joint Assembly or in ASME PCC-1.

Once seated, a gasket must be capable of overcoming minor alignment issues, flange sealing face imperfections and operating variations such as, but not limited to:

- Non-parallel flange faces

- Misaligned flanges
- Distortions, troughs, or grooves
- Surface waviness (deviation)
- Surface scorings
- Other surface imperfections
- Flange rotation during assembly
- Operating Temperature / Design Temperature / Thermal Cycling
- Operating Pressure / Design Pressure / Pressure Cycling
- External environmental conditions
- Startup and shutdown processing variations
- Hydro test pressure during leak testing
- System internal cleaning/flushes

MAINTAINING THE SEAL

When the bolted flange connection is assembled, the gasket is subject to compressive load between the faces of the flanges. The compressive load is the force which is often generated by bolts under tension. The compressive load on the gasket must be high enough to compress the gasket into the surface finish of the flanges to fill any potential leak paths. In order to maintain the seal throughout the lifetime of the assembly, sufficiently high residual gasket load must remain on the gasket to minimize leakage. Under operating conditions the compressive forces on the gasket is reduced by the hydrostatic end load and influenced by other factors, such as thermal expansion behavior of flanges and bolts, lever arms, etc. The hydrostatic end load is the force generated by the internal fluid pressure acting to separate the flanges. The residual gasket load is the compressive load minus the hydrostatic end load. The gasket itself is also subjected to radial forces due to the internal pressure tending to extrude the gasket through the flange clearance space. See Figure 2. To maintain the seal integrity, the residual gasket load must be greater than the internal pressure, usually by some multiple. This multiple depends on the gasket material, gasket type and level of tightness required.

In addition, safety factors are generally recommended to insure the residual gasket load is sufficiently higher than the load required to maintain the seal. Safety factors are generally applied to allow for any relaxation of the gasket compressive forces and to accommodate the application process involved.

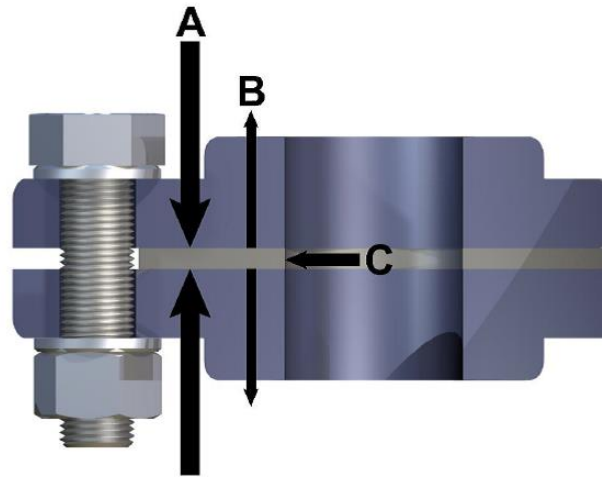
For non-metallic (soft) gaskets where the tensile strength of the gasket material may be less than the internal fluid pressure, there must be adequate friction (created by the compressive load) between the gasket and flange faces to prevent extrusion (blowout) of the gasket from the joint.

FORCES ACTING ON BOLTED FLANGED CONNECTION

The most common way to generate the compressive load required to seal a flanged connection, is through bolts. The bolts, as shown in Figure 3, when tightened act like springs, pulling the flanges together. These bolts need to be stretched enough to keep the load on the gasket as the system is pressurized (during startup) and pressure and temperature is cycled (during normal usage). Additional loading above the minimum load required to seal, will give the bolted joint flexibility to absorb these load changes and a safety margin to maintain the seal as these system forces fluctuate.

Bolt yield strength is a measure of the bolt load required to stretch the bolt and still allow it to spring back to its original length. If the bolt is overstretched and is loaded beyond its yield strength, the bolt will not "spring back" when the load is removed. Continued tightening of the bolts will not necessarily increase gasket load or stop a

gasketed joint leak and may lead to bolt failure. Caution should be used to avoid overloading bolts which can cause them to stretch beyond their yield strength and actually result in lower than expected loads exerted on the gasket.



A=Flange/Bolt Load; B=Hydrostatic End Load; C=Internal (Blow out) Pressure; G=Gasket

Figure 2: Forces in a Bolted Flanged Connection

The minimum bolt load should be at fifty percent (50%) of bolt yield strength to ensure the "spring" is stretched enough. The bolt load typically used is at eighty percent (80%) or even one hundred percent (100%) of bolt yield; if the calculation method and assembly is very accurate. Modern recommended calculation methods include tolerances on assembly methods for example, so the use of one hundred percent (100%) bolt yield can be acceptable. However, this recommendation should be tempered by the amount of gasket and flange stresses generated and ensuring that the applied load will not overload, damaging the gasket or flange. The flange connection will lose compressive load due to system relaxation. If the bolt is not stretched enough, the gasket residual load may drop below the load required to maintain a seal, thereby, causing a leak.

Bolts come in a variety of grades, each with individual yield strength and properties. Proper bolt selection is critical to the proper assembly of a bolted flanged joint.

Friction between the bolt, nut and the flange surface is a force often overlooked during gasketed joint assembly. When determining the torque required to properly tighten the bolts, a nut factor or friction factor needs to be taken into consideration. Using well lubricated nuts, bolts, and nut and bolt faces can help control the frictional load losses during assembly. The use of hardened steel washers between the nut and bolt heads and the flange is recommended to help control frictional losses from embedment of the bolt/nut face into the flange.

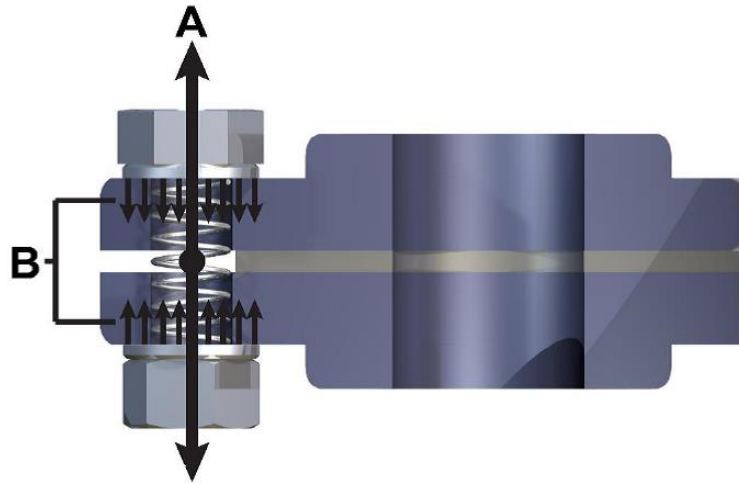
Bolt tightening should be controlled to insure proper bolted flanged connection assembly. Consult the ASME PCC-1 Guidelines for additional information.

OTHER FACTORS AFFECTING THE SEAL

After the initial assembly of a bolted joint, there are many other factors acting on the joint that work to compromise the initial seal (Cycling, Gasket Material, Bending moments).

Most applications undergo pressure and thermal cycles which work to decrease the compressive load on the gasket. Pressure cycles change the hydrostatic end load. Thermal cycles can change the stretch in the bolts through various mechanisms, including bolt material modulus changes due to temperature, differential thermal

expansion of the flange component materials and thermal transients, in heat-up and cool-down. These pressure and thermal cycles cause fluctuations on the compressive load of the gasket and may increase the potential for a leak.



A=Bolt (Spring) Stretch; B=Opposite Force Pulling Flanges (Spring) Together

Figure 3: Bolts Acting Like Spring

Temperature and time can affect gasket materials in a variety of ways. Gasket material degradation due to thermal or chemical exposure can result in a change of material properties, making them less resilient and reducing their ability to spring back under fluctuating loads. Temperature and compressive load can cause creep relaxation in the gasket material itself. Some gasket materials or styles have inherently less creep relaxation and are less susceptible to temperature or chemical exposure than other materials or styles.

External bending moments or forces can also affect an assembled joint. Bending moments tend to unload one side of the flanged joint and increase load on the other side creating load changes that can result in leakage. Misalignment of piping and flanges should be minimized. Flange misalignment creates additional loads that the bolts have to overcome before the bolt loads can be applied to the gasket. Minimizing alignment issues will help to minimize bending issues, but external forces such as weight of fluid in the piping, or thermal expansion, can create external bending forces. Proper piping supports and piping design can help minimize these external bending forces.

References:

Gasket Handbook – Fluid Sealing Association

HEAT EXCHANGER FOULING: ONLINE MITIGATION METHODS (PART 1)

The following article provides an overview of the broad categories of mitigation methods, and describes some general approaches. This is the first part in a two part series – this part addresses chemical fouling mitigation methods. The second part in the next issue of the newsletter will address mechanical fouling mitigation methods.

START-UP PROCEDURES

The use of constant fouling resistances in the design of heat exchangers leads to initially oversized equipment. Heat duties in new or cleaned heat exchangers can, therefore, be considerably higher than the design specifications. In most chemical processes, however, product inlet and outlet temperatures, product flow rate, and cooling water inlet temperatures are specified. If this is the case, the heat exchanger is usually controlled via the flow rate of the cooling water. To reduce the heat duty, the water flow velocity must be reduced. Figure 1 shows that this procedure may cause a considerable increase of fouling as compared to fouling under design operating conditions.

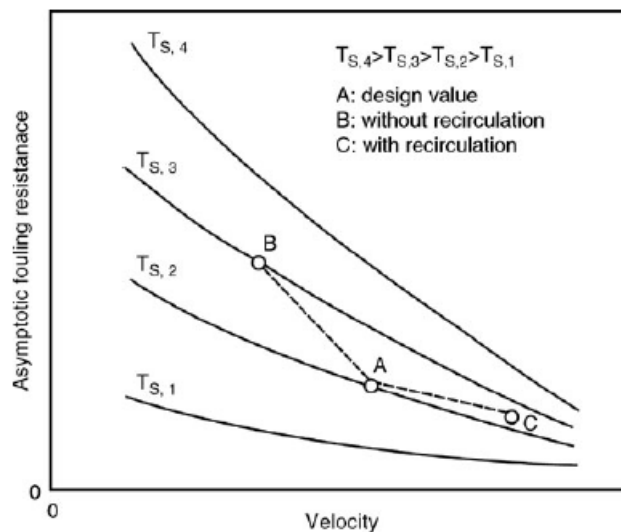


Figure 1: Startup of a New or a Cleaned Heat Exchanger

Point “A” refers to the design values of flow velocity and heat transfer surface temperature. As the heat exchanger is initially oversized, the cooling water flow velocity is throttled which also causes an increase of the heat transfer surface temperature, see point “B”. However, fouling at point “B” is considerably worse and deposits created during this part of the operation may not be removed completely even if the flow velocity is increased later. Therefore, by specifying high fouling resistances, fouling may become a self-fulfilling prophecy. If part of the cooling water is recirculated, as shown in Figure 2, the flow velocity and the cooling water inlet temperature can be increased to meet the required heat duty. The anticipated fouling (“C”) will be similar to the design value (“A”), but a price is to be paid to provide the higher flow velocity.

CHEMICAL FOULING MITIGATION METHODS

In the following subsection, a small selection of methods to reduce fouling by chemical means is discussed. Commercial anti-foulants usually contain a number of components. These polyfunctional anti-foulants are more versatile and effective since they can be designed to combat various types of fouling that can be present in any given system. Anti-foulants are designed to prevent equipment surfaces from fouling, but they are not designed to

clean up existing deposits. Therefore, anti-foulant addition should be started immediately after equipment is cleaned

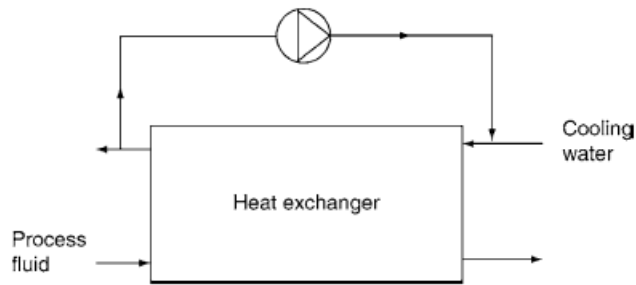


Figure 2: Recirculation of Cooling Water

Scale Formation

In general, there are three alternatives available to mitigate or to prevent scale deposition due to high concentration of scale forming ions in aqueous solutions:

1. Removal of scaling species
2. pH control
3. Scale inhibitors

REMOVAL OF SCALING SPECIES

Scaling species may be removed by ion exchange and by chemical treatment. In the latter treatment, carbonic acid, and calcium hardness are removed by the addition of chemicals.

During slow decarburization (1-3 h reactor residence time), the calcium carbonate precipitates as silt; during fast decarburization (5-10 min reactor residence time), it precipitates in the form of particles. With the exception of installations with high calcium hardness or large throughput, chemical removal of scaling species is not used anymore.

Instead, ion exchangers are used in which “harmful” scaling species in the fluid are replaced by “harmless” ions (for example, Ca^{++} or Mg^{++} by Na^+). The so-called cation exchangers contain weak and strong acids; the anion exchangers contain weak and strong alkaline groups. With these two variations, all cations and anions can be removed from the fluid. Ion exchangers have to be regenerated regularly with the appropriate salt solution. Chemical decarburization leaves a residual hardness of 17-30 ppm as CaCO_3 , ion exchange can reduce the hardness down to 2 ppm as CaCO_3 . Both methods of fluid treatment have high capital and operating costs.

In the oil industry, desalters are installed at the beginning of crude oil heat exchanger train to replace salty water, which may otherwise cause scale formation at high temperatures.

pH CONTROL

The solubility of scale-forming constituents increases with decreasing pH. Many treatment programs, therefore, involve the addition of acid (usually H_2SO_4) to the system to maintain a pH in the region of 6.5 to 7.5. If the system contains corrosion-resistant materials, a pH may be selected at which no scaling will occur.

SCALE INHIBITORS

Growth of crystals or the nucleation of crystals can be inhibited by the addition of scale inhibitors, and many proprietary compounds are available for scale control. Processes, which are based on physical rather than on chemical reactions are those that stabilize supersaturated solutions by adsorption at the crystal nuclei or that

modify or weaken the crystalline structure. Table 1 below shows the ability of additives to maintain CaCO₃ in solution.

Table 1: Ability of Various Additives to Maintain CaCO₃ in Solution

| Additive | % Inhibition at Dose Level | | | |
|------------------------|----------------------------|---------|---------|--------|
| | 2.5 ppm | 5.0 ppm | 7.5 ppm | 10 ppm |
| Polyphosphate | 98% | 98% | 99% | 100% |
| Aminophosphonic Acid | 97% | 96% | 95% | 94% |
| Acetodiphosphonic Acid | 83% | 82% | 83% | 90% |
| Polyacrylate | 30% | 65% | 84% | 93% |
| Polymaleic Acid | 26% | 35% | 44% | 56% |
| EDTA | 15% | 20% | 20% | 20% |

Particulate Fouling

Particulate fouling is usually mitigated by the addition of surfactants or dispersants. If the surface tension is reduced, large particle agglomerates can break down into smaller particles, which tend less to sedimentation. Dispersants impart like charges to both the heat transfer surface and the particles and reduce deposition. For cooling water applications polyacrylates or polysulfonates are used with molecular weights between 2000 and 3000 g/mol. The addition of polyphosphates to reduce scaling may cause a slight reduction of the dispersion of particulates.

Chemical Reaction Fouling

Chemical reaction fouling increases exponentially with increasing heat transfer surface temperature. As activation energies E are fairly high for chemical processes, even a modest reduction of the heat transfer surface temperature due to process or design modifications may already cause a considerable reduction of fouling. Particles suspended in the fluid (e.g., from upstream corrosion) can act as catalysts. Reaction fouling may be mitigated by removing these particles.

Especially for oil refining processes a number of chemical additives to reduce reaction fouling have been developed. Most anti-foulants have several functions. Generally they are oxygen scavengers, metal deactivators and dispersants.

ANTIOXIDANTS

Even very small amounts of oxygen can cause or accelerate polymerization. Accordingly, antioxidant type anti-foulants have been developed to prevent oxygen from initiating polymerization. Antioxidants act as chain-stoppers by forming inert molecules with the oxidized free radical hydrocarbons.

METAL DEACTIVATORS

Traces of metals are invariably present in hydrocarbon streams, which may catalyze polymerization reactions. For example transition metal ions, such as Cu, Fe, Zn, and Mn are powerful hydroperoxide decomposers and provide a steady source of free radicals for oxidation chain initiation. By complexing the metal ion, it can be prevented from participating. Thus chelating compounds are used as metal deactivators.

DISPERSANTS

Dispersants or stabilizers prevent insoluble polymers, coke, and other particulate matter from agglomerating into large particles which can settle out of the process stream and adhere to the metal surfaces of process equipment. They also modify the particle surface so that polymerization cannot readily take place. Dispersants generally play an important role in anti-foulant programs. The feedstock or hydrocarbon stream may already contain polymerized materials which, if allowed to agglomerate, would deposit. In most applications, it is not possible to fully eliminate oxygen- or metal-induced reactions and dispersants are necessary to prevent the polymerized materials from agglomerating and depositing on heat transfer surfaces.

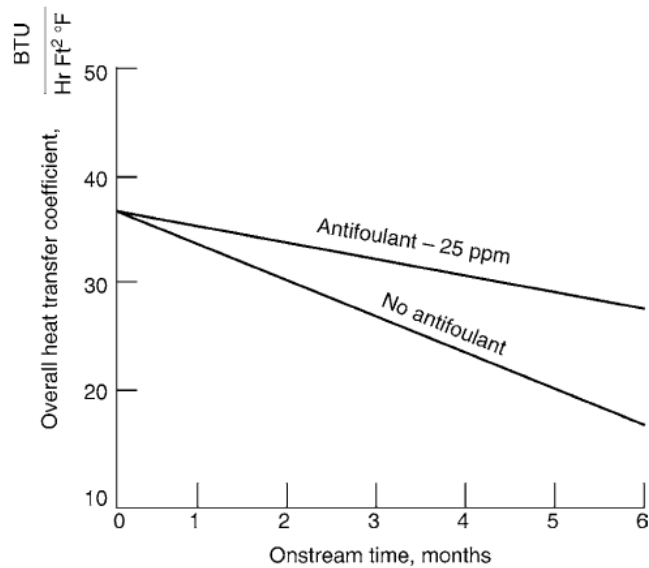


Figure 3: Reformer Feed/Effluent Exchanger Performance

Figure 3 shows the performance of the feed/effluent heat exchanger in an oil refinery with and without chemical treatment.

Biofouling

The environment in cooling towers and cooling systems is particularly conducive to the growth of microorganisms in water and on surfaces of the system. Microorganisms attach and grow on surfaces, and produce polysaccharides, which increase the stickability of suspended matter and hence promote further deposition. Biological growth is usually controlled by addition of biocides. In recent years, chlorine has most widely been used, which reacts with water to form hydrochloric and hypochlorous acid:



Hypochlorous acid is an extremely powerful oxidant that easily diffuses through the cellular walls of microorganisms. Continuous application of chlorine at concentrations between 0.1 and 0.5 ppm has shown to be a reliable but costly method to avoid deposition. Cheaper but less effective is a dosage of 1–10 ppm for 15 min in intervals of 4 h.

Biological fouling control with chlorination has the disadvantage that chlorine has to be added continuously, since it does not only react with microbes but also with process contaminants such as H₂S or NH₃. Chlorine concentration in water exceeding 0.5 ppm may give rise to corrosion problems, especially for stainless steel equipment. Due to the biocidal action of chlorine, there are increasing restrictions on the effluent chlorine concentration. For these reasons, chlorine is increasingly replaced by other chemicals such as methylene-thiocyanate or chlorophenols.

Because of the toxic effect of copper ions on biological matter, another method to reduce bacterial growth is the use of piping with a copper content above 60% or the addition of copper sulfate to the water. For potable water, the copper concentration must be below 1 ppm.

Corrosion Fouling

Generally, it is desirable to have a thin, passivating oxide layer on the heat transfer pipes. This oxide layer is removed if the flow velocity exceeds 30 m/s. Excessive corrosion can be controlled by the addition of corrosion inhibitors (chromate or polyphosphate based) or by control of the pH. Chromate is a highly efficient and cost effective inhibitor. However, the toxicity of chromates in the environment has restricted their use. This also holds for zinc based inhibitors. Under some circumstances, corrosion inhibitors (such as phosphates) themselves can be the source of fouling in heat exchangers as they increase the total salt content of the water. However, this can be mitigated by careful control of parameters such as inhibitor concentration, flow velocity and surface temperature.

Gas-Side Fouling

Removal of contaminants which promote fouling, such as sodium, sulfur, or vanadium, from fuels prior to combustion and contaminant removal from combustion gases are two approaches to mitigate gas-side fouling. Water washing has helped to overcome some of the fouling problems experienced with residual oils in marine applications by removing sodium and sediment. Inorganic sulfur can be removed from coal by gravity settling or by froth floatation if the mineral particles are well above micron size.

Electrostatic precipitators, mechanical collectors, fabric filters or wet scrubbers can be used to remove particles from combustion gas streams. Removal of gaseous constituents, which is considerably more difficult than particle removal, may involve limestone addition, wet scrubbing without sulfur recovery, MgO systems with sulfur recovery or use of dry sorbent systems.

Under certain conditions, chemical fuel additives or chemical flue gas additives can mitigate the effects of gas-side fouling and corrosion. Many proprietary additives have been marketed, with varying degrees of success in mitigating fouling by companies such as Betz, Dearborn, Drew and Nalco.

For oil-fired boilers, additives are used to control SO₃ related problems, high temperature fouling, high temperature corrosion.

References:

VDI Heat Atlas

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FASTENER TECHNICAL DATA

INDUSTRY STANDARDS

Most industrial fasteners are covered by two basic standards: one for materials and properties; the other for dimensions and tolerances. Specifications for materials and properties are published by the American Society for Testing and Materials (ASTM), although other groups such as the Society of Automotive Engineers (SAE) also publish specifications covering these requirements. Standards for dimensions and tolerances are issued by the American National Standards Institute (ANSI) in cooperation with the American Society of Mechanical Engineers (ASME) and the Industrial Fasteners Institute (IFI).

Standard fasteners are basic industrial fasteners - square and hex bolts, cap screws, carriage bolts, plow bolts, lag screws, studs, nuts, rivets and others - which have been standardized over the years as to type, style, usage, properties, dimensions and tolerances. These include a vast range of sizes and types stocked by distributors and manufacturers for an almost limitless range of applications: the assembly and maintenance of vehicles, appliances, farm equipment, construction equipment, industrial and plant machinery of all kinds, furniture and toys.

Wherever there's a need for holding parts together, holding them apart, holding them up, or holding them down, a standard fastener can usually be found to do the job efficiently and economically. Modern industrial fasteners are manufactured to a variety of standards covering dimensions, tolerances, materials, mechanical properties, testing procedures, etc.

BASIC FASTENER TYPES

A *Bolt* is a device with a head on one end of a shank or body and a thread on the other end (See Figure 1). Designed for insertion through holes in assembly parts, it is mated with a tapped nut. Tension is normally induced in the bolt to compress the assembly by rotating the nut. This may also be done by rotation of the bolt head.

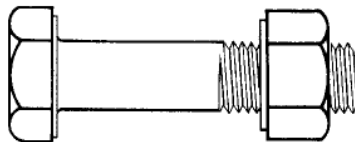


Figure 1: Bolt

A *Screw* is a headed and threaded bolt used without a nut (See Figure 2). It is inserted into an internally tapped hole and tension is induced by rotation of the screw head.

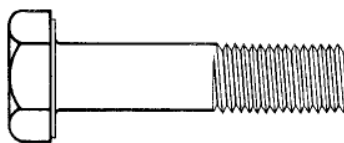


Figure 2: Screw

A *Stud* is a fastener with no head but it has threads at both ends of the shank (See Figure 3). It, like a screw, has one end that screws into a tapped hole. A nut is used on the other end to create tension.

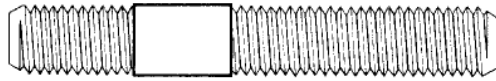


Figure 3: Stud

If a stud is threaded its entire shank length and a nut used on both ends to create tension, it serves the function of a bolt and is then classified as a *Stud Bolt* (See Figure 4).

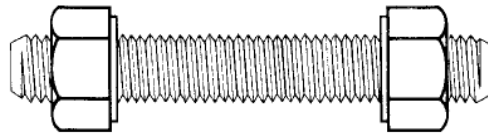


Figure 4: Stud Bolt

TERMINOLOGY AND MEASUREMENTS

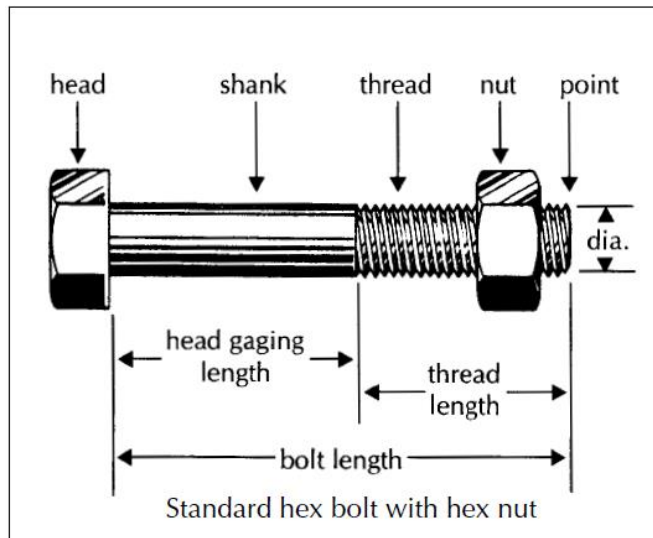


Figure 5: Fastener Terminology

The diameter of all bolts is measured as the outside of major diameter of the thread. The length of a headed bolt is measured from the largest diameter of the bearing surface of the head to the extreme end of the point in a line parallel to the axis of the bolt. For example, square or hex head bolts are measured from under the head to the end of the bolt; a bolt with a countersunk head is measured overall. The point of a bolt is always included in the measured length. Headless fasteners such as studs are measured overall, including points, except for continuous thread alloy studs made to ASTM Specification A193. This type is measured from first thread to first thread.

Bolts have various styles of heads. Some of the more popular styles are illustrated in Figure 6.

Heavy Heads and Nuts

Bolt users may require a bolt head or nut of greater width to compensate for wide clearance in bolt holes or unusually heavy loading. For such needs we offer heavy bolt heads and nut sizes. Heavy nuts are quite common and readily available in various types and sizes but heavy head bolts have more limited availability.

Bolt and Nut Bearing Surfaces

Cold-upset bolts made on a “boltmaker” usually are washer faced but large or very long bolts may have a flat face (not machined). Nuts made on cold “nut-formers” may have double chamfered or washer faces. Large size nuts have hot-forged washer bearing faces, not machined, unless specially ordered.

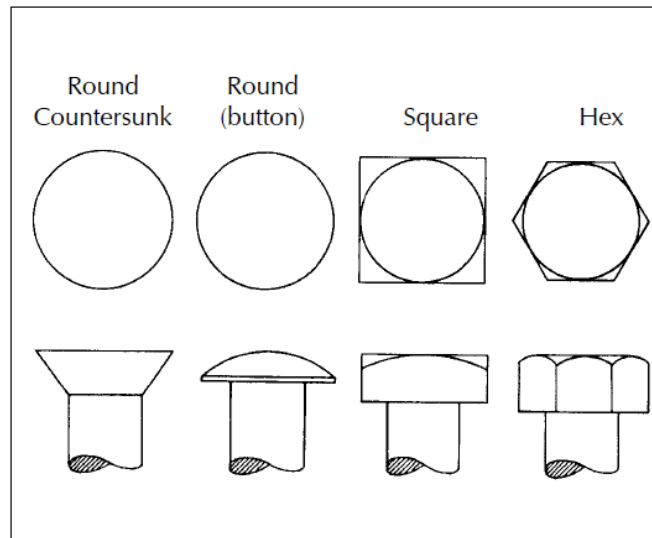


Figure 6: Styles of Heads

HEAT TREATMENT

Heat treatment covers various techniques that may be used to develop certain end-product characteristics. Customary procedures for fasteners include annealing, stress relieving, case hardening, direct quench and temper, and carbon restoration.

Annealing

A thermal cycle involving heating to, and holding at a suitable temperature and then cooling at a suitable rate, for such purposes as reducing hardness, improving machinability, facilitating cold working, producing a desired microstructure, or obtaining desired mechanical or other properties.

Stress Relieving

A thermal cycle involving heating to a suitable temperature, usually 1000/1200°F, holding long enough to reduce residual stresses from either cold deformation or thermal treatment, and then cooling slowly enough to minimize the development of new residual stresses.

Case Hardening

A term descriptive of one or more processes of hardening steel in which the outer portion, or case, is made substantially harder than the inner portion, or core. Most of the processes involve either enriching the surface layer with carbon and/or nitrogen, usually followed by quenching and tempering, or the selective hardening of the surface layer by means of flame or induction hardening.

Quenching and Tempering

A thermal process used to increase the hardness and strength of steel. It consists of austenitizing, then cooling at a rate sufficient to achieve partial or complete transformation to martensite. Tempering should follow immediately, and involves reheating to a temperature below the transformation range and then cooling at any rate desired. Tempering improves ductility and toughness, but reduces the quenched hardness by an amount determined by the tempering temperature used.

MECHANICAL PROPERTIES

Standard industrial fasteners are manufactured from either carbon or alloy steels. When strength requirements are moderate, low-carbon steel is used. High-strength fasteners are made from medium-high carbon or alloy steels and are heat treated to develop desired properties. Most fasteners are covered by specifications that define required mechanical properties such as tensile strength, yield strength, proof load, and hardness.

Tensile Strength

The maximum tensile stress in psi which a material is capable of sustaining, as developed by a tension test.

Yield Strength

The stress at which a material exhibits a specified deviation from the proportionality of stress to strain. The deviation is expressed in terms of strain, and in the offset method, usually a strain of 0.2 % is specified.

Proof Load

The point to which a material may be stressed without evidence of permanent deformation.

Hardness

The resistance of a material to plastic deformation. Usually measured in steels by the Brinell, Rockwell, or Vickers indentation-hardness test methods.

SCREW THREADS

Thread Forms

Screw threads are spiral grooves produced on the outside of a bolt (external threads) or machined inside a nut (internal threads). Four thread forms have been established by the ANSI: UN (internal and external threads), UNR (external threads only), UNK (external threads only), UNJ (internal and external threads).

Cut vs. Rolled Threads

The terms “rolled thread” and “cut thread” refer solely to the physical characteristics of commercial bolts or studs, not to the method of thread production. A cut-thread bolt has a shank diameter equal to the threaded diameter; a rolled-thread bolt has a shank diameter less than the threaded diameter.

Classes of Thread Fit

Classes of thread fit are distinguished from each other by the amounts of tolerance specified. Classes 1A, 2A, and 3A apply to external threads; Classes 1B, 2B and 3B, to internal threads.

Classes 1A and 1B are shown in ANSI standards but are rarely used. Most standard fasteners are produced with a Class 2A fit for bolts and a Class 2B fit for nuts. The Class 2A allowance assures easier assembly of mating parts, minimizes galling and seizing in high-cycle wrench assembly, and can be used to accommodate commercial electroplated finishes. Classes 3A and 3B afford no allowance or clearance for mating parts and are used chiefly for applications where a close tolerance fit is important.

Class 5 is a special interference fit which results in an actual overlap of dimensions when threaded parts are fitted together. It is normally used only on the tap-end of studs. Threading requirements are designated (1) by the number of threads per inch applied to a specific diameter; (2) by the initial letters of the thread standard (UNR or UN); (3) by the letters C, F, or numeral 8 to indicate coarse-, fine-, or 8-thread series; and (4) by the thread fit. The following example illustrates the method of designating a screw thread:

$$\begin{aligned} & 1/2 \text{ in.} - 13 \text{ UNRC-2A, where} \\ & 1/2 \text{ in.} = \text{Nominal diameter of thread} \end{aligned}$$

| | | |
|-----|---|---|
| 13 | = | Number of threads per inch |
| UNR | = | Thread form standard (Unified National Radius-Root) |
| C | = | Coarse-thread series |
| 2A | = | Class of thread fit |

The coarse-thread series (UNRC or UNC) is used on the vast majority of bolts and nuts. Number of threads per inch ranges from 20 threads for a 1/4-in.- diameter bolt to 4 threads for a 4-in.-diameter bolt. The fine-thread series (UNRF or UNF) is found mostly in automotive and aeronautical work. Threads per inch range from 28 threads for a 1/4-in.-diameter bolt to 12 threads for a 1-1/2-in.-diameter bolt. There is no fine-thread standard for fasteners over 1-1/2 in.

The 8-thread series (8UNR or 8UN) is used only for sizes over 1 in. in diameter, usually in high-temperature, high pressure service.

References:

Fastener Technical Data and Charts Section *by* K.L. JACK

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