Proceedings of the ASME 2014 Pressure Vessels & Piping Conference
PVP2014
July 20-24, 2014, Anaheim, California, USA

PVP2014-28174

Breech Lock Exchangers, Obtaining Leak Free Performance

David Reeves **David Clover**

ABSTRACT

High pressure screw plug (also called breech lock) exchangers are arguably the most complicated bolted connection in a refinery. After traveling around the world helping facilities turn these exchangers from chronic leakers (from internal tubesheet and external diaphragm and B style shell leaks) to one of the most reliable heat exchangers in the plant, it is very clear to me that manufacturers, engineers and mechanics struggle to correctly understand the interactions between gaskets, internal pressure, the two circles of external push bolts that are part of A style exchangers and all the internal parts.

Some manufactuers have given up on trying to make these connections operate leak free, and prefer to weld in the tubesheets and diaphragms. This significantly adds to the cost and time needed to open and close these exchangers, as special equipment is needed to machine out the parts and weld them back together again.

There are a handful of equipment manufacturers that build these exchangers, and while there are differences from one manufacturer to the next, once the basic design is understood, the reader should be able understand that they are all basically the same when it comes to the few critical steps that must be followed when opening and closing these exchangers.

In order to obtain reliable leak free performance, all the basic sealing tools must be employed, including:

1. A spreadsheet that will easily analyze the critical variables;
2. Correct gasket selection;
3. Correct assembly procedures that focus on important steps including proper spacing of internal parts and obtaining consistent thread friction;
4. And finally hot torqueing the exchangers after startup.

The goal of this paper to provide the end-user with a good understanding for how parts interact and how the bolted connections function, so that they might also achieve reliable, leak free performance. This paper will explain:

1. How the connections work;
2. Common misunderstanding about screw plug exchangers;
3. Pressure Testing Screw Plug Exchangers;
4. Analyzing the design and calculating the important variables;
5. The key assembly steps that must be included in all assembly procedures;
6. Important design considerations when building new exchangers.

INTRODUCTION

Screw plug exchangers come in two styles: An A style that has an internal tubesheet and contains high pressure on both the shell and tube sides (also called a High/High); And a B style which has either a removable shell or welded in tubesheets on both ends, both of which are high pressure on the tube side and low pressure on the shell side (also called a high/low). Since the B style design is simpler, this paper will focus on the A style. The same technical issues that we cover for the A style also apply to the B style.

Figure 1 at the end of this document is an exploded view of an A style screw plug exchanger. Most people are confused by the number of internal parts and have problems visualizing how they interact with each other. Here is an example: These exchangers are typically used in high pressure units (like Hydroprocessing units) which operate around 2,500 psi. The lock rings are screwed in using large ACME threads (typically between 1 and 1-1/4 inch), and then two circle of push bolts are used on the A style exchangers to load the tubesheet and diaphragm gaskets. (B style exchangers use a single circle of push bolts.) What forces are applied to the ACME threads?

1. Is it the internal pressure?
2. Inner circle of push bolts?
3. Outer circle of push bolts?
4. All of these forces?
5. Or what combination of them?

The answer below may surprise you!

Over the years there have been a number of papers written about how to make bolted connections operate leak free. Successful approaches generally include:

1. Some kind of analysis of the closure to establish the maximum stud stress that can be applied without damaging the closure. This sets an upper limit for the available force that can be applied.
2. A properly selected gasket that will tolerate process conditions, including being unaffected by the process, process temperature and any differential movement between sealing surfaces. The maximum load the gasket can handle without damage, and minimum needed to establish a seal must be understood, as well as the optimum gasket stress that should be targeted for reliable leak free performance under operating conditions.
3. The stud stress must be calculated to achieve this optimum gasket stress, and then the torque value must be calculated correctly to achieve this stud stress.
4. And finally a detailed assembly procedure that will assure that the connection is assembled correctly and that the correct stud stresses are applied. This procedure must also include any requirement to retighten the connection to make up for any relaxation that will occur during initial heating.

For most bolted connections that use conventional flanges, information is readily available to address the four points above. But for screw plug exchangers, each of these steps becomes even more important and each exchanger has to be analyzed separately based on its own unique design.

HOW THE CONNECTIONS WORK

In its simplest form, these exchangers are designed not only for their operating pressure, but to withstand their final test pressure. They have a tubesheet that is inserted into the shell, followed by an internal partition that controls the tube side process flows. A diaphragm is used to contain the process and pressure. A lock ring with a separate or integral channel cover screws onto the channel barrel, in most cases using large 1 inch or 1-1/4 inch ACME threads. Internal and external push bolts are used to either load the tubesheet or diaphragm gasket. An internal flange with push bolts may be present to hold the tubesheet in place and keep the tubesheet gasket loaded during a shell side pressure test. External push bolts in the lock ring or channel cover are generally used with push rods to mechanically pass the load that is applied from outside the pressure boundary through the diaphragm (which distorts slightly) to load either the diaphragm gasket (outer circle of push bolts) or tubesheet gasket (inner circle of push bolts).

Figure 2 is a cartoon of a typical A style screw plug exchanger. The red arrow at the top shows the load path for the outer circle of push bolts (19) that loads the diaphragm gasket. The load goes from the push bolts (19), through the push rods, into the outer compression ring (14), through the diaphragm (13) and onto the diaphragm gasket (12).

The inner circle of push bolts (17) ultimately loads the tubesheet gasket after the load passes through the push rods, inner compression ring (15) diaphragm (13), internal ring (11), internal flange (8), internal push bolts (10), internal partition assembly (3), tubesheet (2), and finally onto to the tubesheet gasket (1).

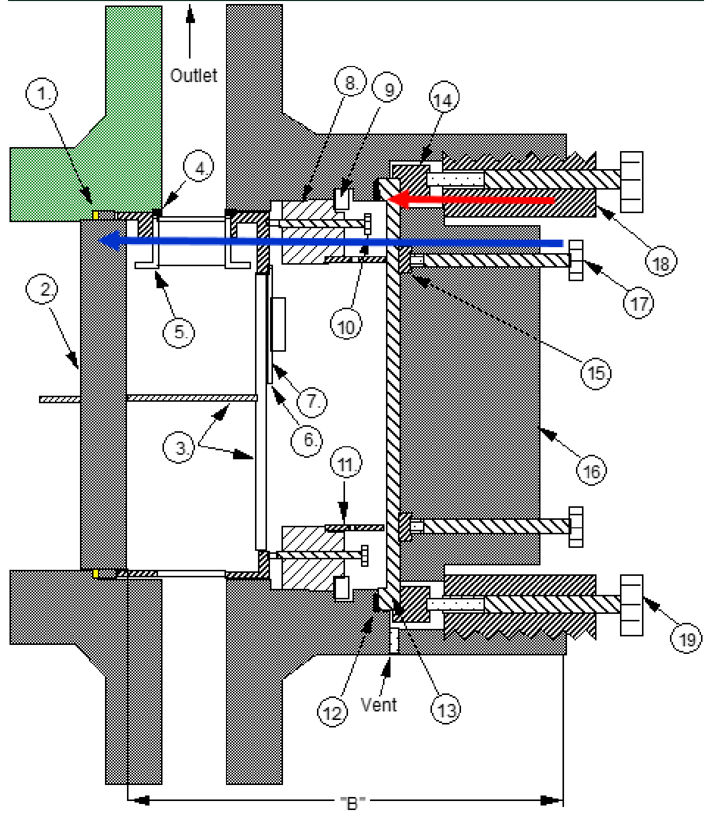


Figure 2

Remember, all screw plug A Style exchangers are basically the same, where force is applied from outside the pressure containing barrier, through mechanical parts that are in compression to load either the diaphragm or tubesheet gasket. Why build an exchanger this way? There are three primary reasons:

1) If a conventional flange was used it would be huge, as would be the studs, which can weigh approximately 150 pounds per stud for even a small exchanger. This not only adds material and fabrication costs, but creates lots of assembly problems including the setting of proper stud stresses.
2) Using a plug that screws into the channel is a very efficient and cost effective design to build.
3) The gasket stress changes very little between atmospheric and full pressure, eliminating the problem of huge swings in gasket stress between initial assembly and operating pressure.

A simple but clever design, right? Now if we go back and consider the question that was asked about what forces load the ACME threads, most people (including many manufactureres) believe that all three forces (both sets of push bolts and the internal pressure) load the ACME threads. However this is not the case, even though this is usually the design basis for the ACME threads. The lock ring connection is a combination of "springs in series" as well as "springs in parallel".

Let starts with the push bolts. Think about doing a standard pushup where both your arms are used to raise your body weight. As you push down with your arms your body rises. However if your left arm pushes harder, your right arm doesn't have to push as hard to raise your body weight, and vice versa. This is an example of springs in parallel and the same way the studs in a screw plug lock ring or channel cover works. If the load on the inner circle of push bolts goes up, the load on the outer circle will drop slightly, and vice versa. It is the total load generated by both the inner and outer circle of push bolts that gets transferred to the ACME threads. Figure 3 was generated by directly measuring push bolt load changes with strain gages. It shows how the stress drops on the outer circle of 1-1/2 inch push bolts from approximately 37ksi (2,600 Kg/Sq Cm) to 33ksi (2,320 Kg/Sq Cm) as the stress on the inner 1-1/8 inch push bolts were tightened up to appoximately 42ksi (2,950 Kg/Sq Cm).

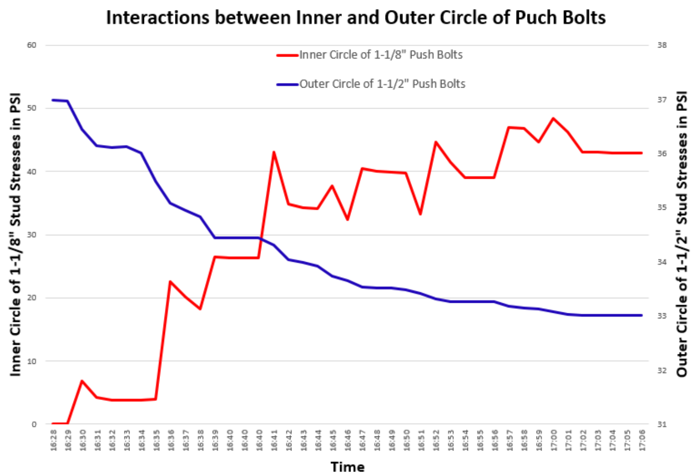


Figure 3

The pressure is in series with the load from the push bolts, and just like any other bolted connection, the push bolt stress will not change until it is overcome by the hydraulic end force. If the push bolt stress doesn't change, then the ACME thread stress can't change. It is not likely that the hydraulic end force will overcome the total force being applied by the push bolts as screw plug exchangers are design to contain the hydraulic end force as well as maintain a high enough gasket stress to remain sealed. Or put another way, if the hydraulic end force could overcome the total force from the push bolts, the gasket stress would drop and the connection would leak when the exchanger reached full pressure. Figure 4 is a cartoon that shows how these loads interact with the ACME threads.

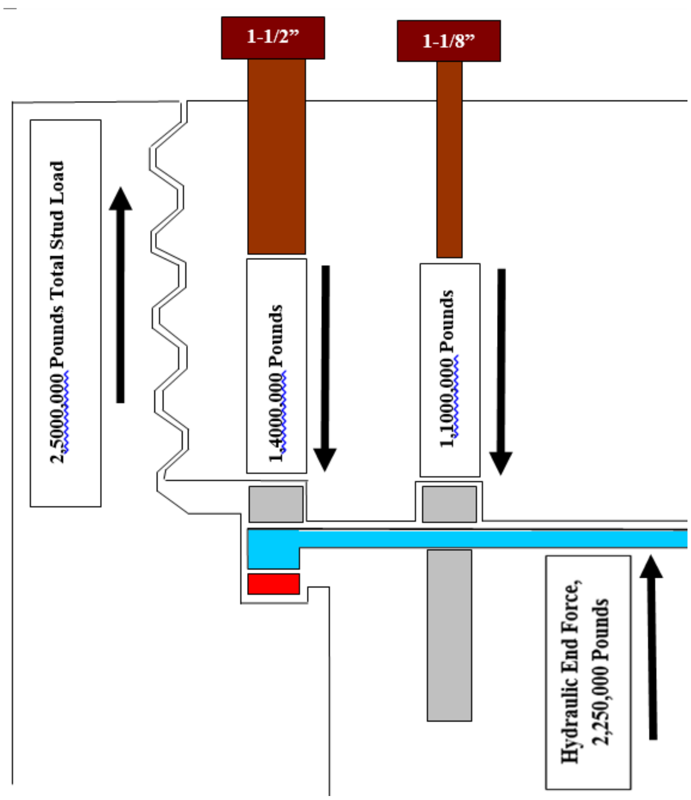


Figure 4

COMMON MISUNDERSTANDINGS ABOUT SCREW PLUG EXCHANGERS

1. Push bolts should be tightened in a star pattern.

In the paper “Common Misunderstandings About Gasket and Bolted Connection Interactions” [1] and “A Simple Recipe For Solving All Refinery Sealing Issues” [2], it was shown that tightening a stud on one side of a conventional heat exchanger flange had no impact on the stud stresses on the opposite side of the flange. This is because when gasket surfaces are machined flat there is no fulcrum in place to transfer the stud stress from one side of the flange to the other side.

Screw plug exchanger connections are significantly stiffer than standard ASME designed body flanges, so there is even less ability to impact push bolt stresses on one side versus the other.

In addition, the push bolts push on push rods, which push on either an inner or outer compression ring (Figure 2, items 14 and 15), which will conform to the flatness of the diaphragm or internal sleeve. This makes it impossible to impact push bolt stresses on one side of the flange by adjusting the push bolts on the other side.

Figure 5 shows push bolt stress changes at the top and bottom of the exchanger as four circular passes were made around the channel cover. The spike in push bolt stresses shows when the instrumented push bolt was tightened. The drop in push bolt stresses on either side of the spike is due to cross talk between push bolts just before and right after the instrumented stud was tightened. Cross talk was limited to within 3 push bolts either side of the instrumented push bolt. (The lock ring push bolts behave the same way.)

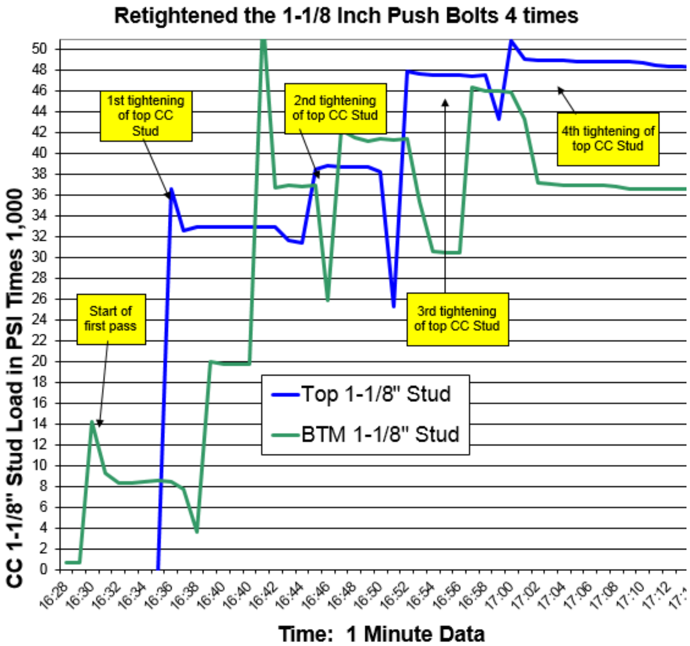


Figure 5

For the range of gaskets used in these kinds of closures, there is no impact on the ability of the gasket to seal if it is loaded all at once, or in stages.

Therefore, there is no advantage to using a star pattern.

<u>2. Spiral wound or double jacketed gaskets are appropriate for both the tubesheet and diaphragm gaskets if they are specified in the original design.</u>

As long as spiral wound and double jacketed gaskets continue to be installed in screw plug exchangers, end-users should expect both internal and external leakage. Both of these gasket designs cannot tolerate the radial shear that takes place between the tubesheet and shell, and channel and diaphragm sealing surfaces. Incorrect gasket selection is the biggest single cause of screw plug exchanger leaks. <u>Figure 6</u> is an example of the differential thermal movement between a 2-1/4 chrome channel and 347SS diaphragm over 4 days during the initial attempt to startup the plant. Note that the range of differential movement was between a plus .033 and minus .009 inches (+.838 and -.229 mm).

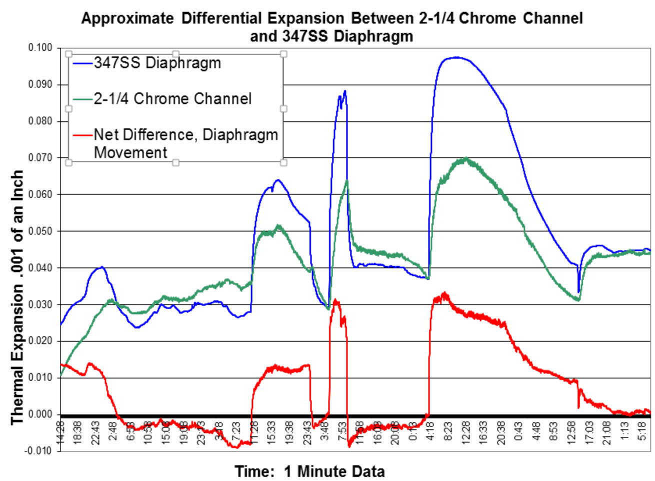


Figure 6

<u>Figure 7</u> shows a failed double jacketed gaskets, and <u>Figure 8</u> shows a failed spiral wound gasket.

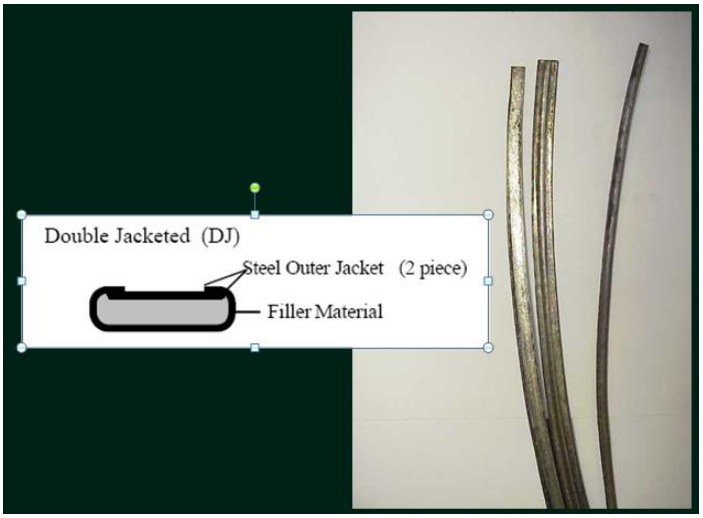


Figure 7

Figure 8

For more information on spiral wound gaskets, see the paper "The Influence of Winding Density in the Sealing Behavior of Spiral Wound Gaskets," [4].

The best gaskets to use are either graphite covered corrugated metal CGG type gaskets as show in <u>Figure 9</u>, or Kamprofile KAG type gaskets as shown in <u>Figure 10</u>.

Figure 9

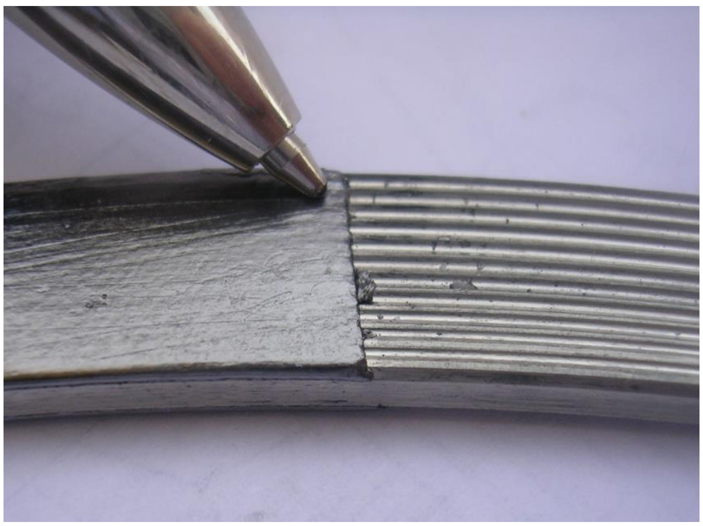


Figure 10

These gaskets are designed to handle radial shear. While some screw plug exchanger manufacturers advise that their originally supplied gaskets can only handle a limited number of thermal shocks (usually less than 10), these gaskets are used in Coke Drums which cycle once a day from 100F to 850F (38C to 450C) and will remain tight for 4 years and around 1400 cycles.

Also see the paper "Failure of Heat Exchanger Gaskets Due to Differential Radial Expansion of the Mating Flanges [5], and "Heat Exchanger Gasket Radial Shear Testing" [6].

Therefore spiral wound or double jacketed gaskets are ***NOT*** appropriate for both the tubesheet and diaphragm gaskets.

3. For screw plug exchangers with internal flanges, the inner circle of lock ring or channel cover push bolts should not be tightened unless there is an internal leak. This is required to prevent the ACME threads from becoming overloaded due to the thermal expansion of the internal partition assembly when the exchanger is heated up.

If the inner circle of lock ring or channel cover push bolts is not tightened during initial assembly, the exchanger is all but guaranteed to leak internally.

On a recent field application the internal flange push bolts were torqued on a screw plug exchanger to their final stress of 55,840 psi (3,926 Kg/Sq Cm), which resulted in a gasket stress of 19,589 psi (1,377 Kg/Sq Cm). After being heated up as part of the plant startup procedure, then shut down and reopened without ever have introduced feed into the exchanger, it was found the remaining stud stress was only 13,030 psi (916 Kg/Sq Cm), or a gasket stress of only 4,571 psi (321 Kg/Sq Cm), which is below the 6,000 psi (422 Kg/Sq Cm) minimum recommended gasket stress needed to maintain a seal with graphite faced gaskets.

Keep in mind the push bolts are loading all the internal parts in compression, and these connections are very stiff. Field measurements found that each degree of push bolt rotation resulted in a bolt stress increase of approximately 700 psi (49 Kg/Sq Cm). This means that if a 50,000 psi push bolt stress is targeted, 71 degrees of push bolt rotation would be required, or a flat plus 11 degrees.

During a recent shutdown, both circles of external push bolts were tightened to their final load and marked with horizontal lines. After the hot torque another picture was made of the push bolts. A grid was drawn on each push bolt head (see Figure 11), and the total number of degrees of rotation was estimated. It was found that the outer circle of push bolts that load the diaphragm gasket only relaxed an average of 11 percent, but the inner circle of push bolts that load the tubesheet gasket had relaxed 86 percent! Given that the same gasket was used in both connections, the significant difference in the amount of relaxation for each connection is likely due to greater thermal movement of the tubesheet during heating, thus generating more graphite flow and relaxation.

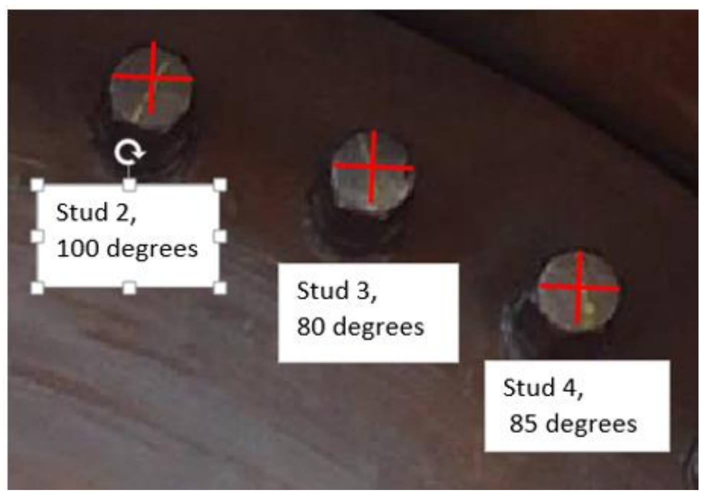


Figure 11

In addition, it is very common to be able to turn the internal flange push bolts with a hand wrench after the exchanger is reopened regardless of the original load that was applied. This is another indication that tubesheet gasket relaxation has caused these push bolts to completely unload.

Therefore the inner circle of lock ring push bolts ***MUST BE TIGHTENED*** or the tubesheet gasket will leak.

4. The internal partition assembly should be made from the same material as the channel to avoid overloading and suddenly yielding the ACME threads during a sudden rise in temperature.

The data below is from a screw plug exchanger with a 2-1/4 chrome channel and a 321 Stainless Steel partition assembly. Figure 12 and Figure 13 show that the temperatures of all the components that were measured at the top and bottom tend to move together. This is logical as they are all bathed in the same temperature process.

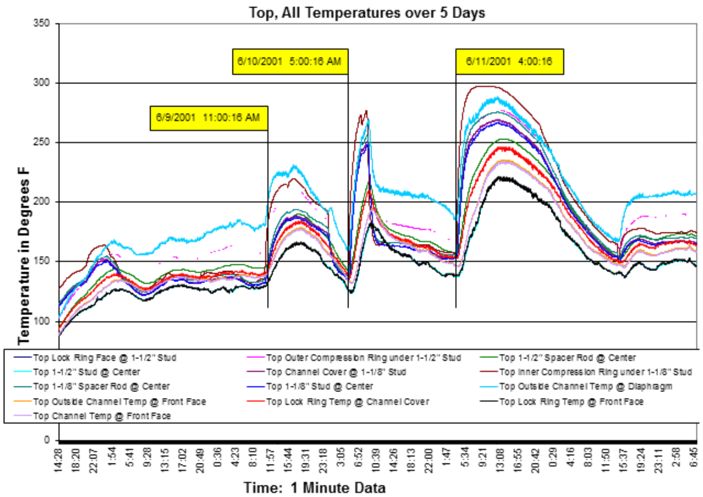


Figure 12

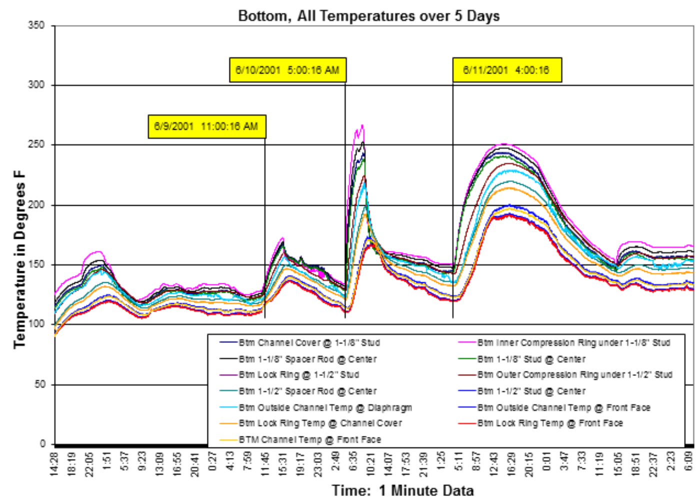


Figure 13

During the plant startup there was a problem, so the reactor was bottled up hot while the exchanger slowly cooled down. When flow restarted, the process temperature into the exchanger climbed 204F (95C) in 3 minutes, as can be seen in Figure 14.

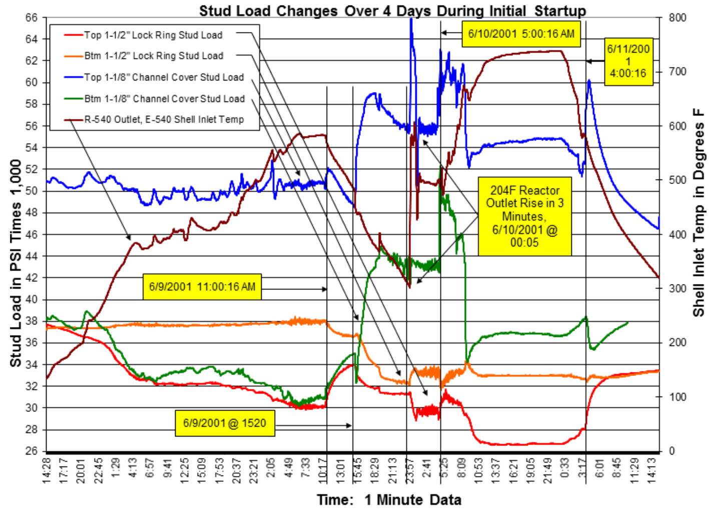


Figure 14

The net result as far as the increase to the ACME thread shear stress was in the range of only about four percent as can be seen in Figure 15.

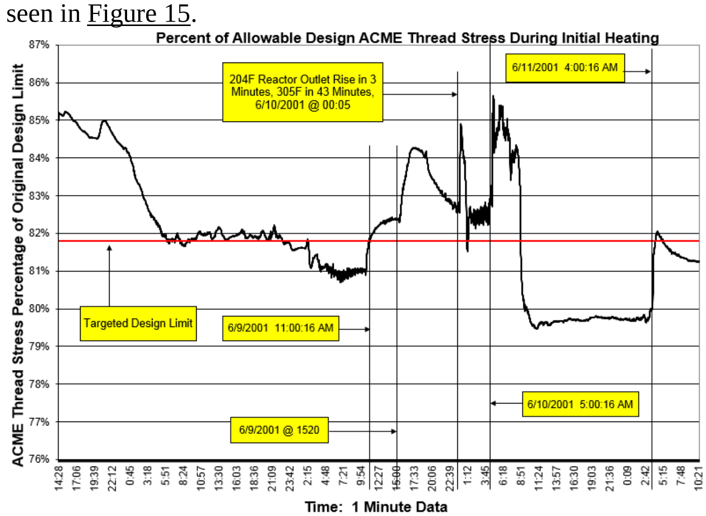


Figure 15

There are a number of reason why this makes sense. The hot process comes in on the shell side, which retards the impact on the channel side. The cold process comes in the bottom of the partition assembly which is open at the bottom, allowing the cooler process to keep the diaphragm, partition assembly and other internal parts cooler (see the blue arrows). On some screw plug exchangers the hot process may come into the top of the channel, but it has to pass through the tubes where it cools down before exiting out the bottom of the channel (red arrows). See the cartoon in Figure 16. Either way, the partition assembly is exposed to the cooler part of the process stream.

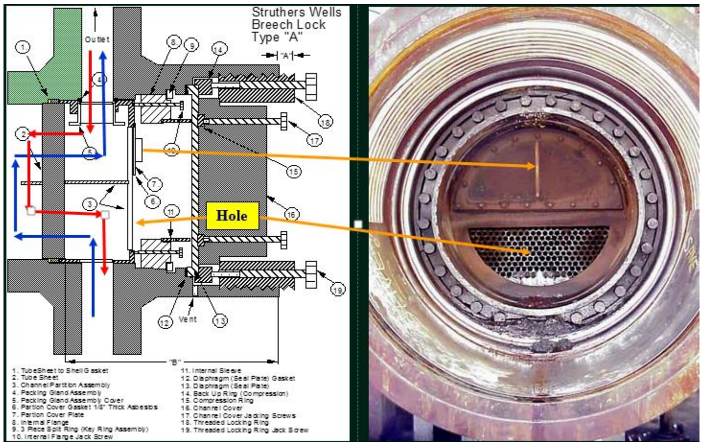


Figure 16

Screw plug exchangers where the channel has been designed to run hotter than the shell have the potential to be more problematic, but it really depends on the design. For instance is the partition assembly made from a stainless steel material that will thermally expand more than the channel? Does the exchanger has an internal flange? If so, this will limit the force to some degree that can be passed to the ACME threads.

Operators are given strict reactor temperature limits, and if these are violated the process has to be shutdown and the plant depressured. It is highly unlikely that a sudden increase in temperature would last long enough to cause damage to the ACME threads which are somewhat protected from process temperatures without causing a failure of the reactor or piping first. Piping failures are more likely because the piping is directly exposed to process temperatures, and the higher the temperature the less strenght the piping has.

But let's assume for a minute that for some reason there is a significant amount of disproportional thermal growth by the partition assembly, enough to significantly change the ACME thread stress. This would likely also be enough force to cause the diaphragm gasket stress to drop. Remember if the inner circle of push bolt loads increases, the outer circle of push bolt loads will drop. So the likely result will be a diaphragm leak before the ACME threads fail. A properly designed set of ACME threads will be able to tolerate 100% of the yield strength from both sets of push bolts without going over the yield strength of the ACME threads.

Also keep in mind that there are a couple of safety devices that should have been designed into the exchanger to keep the ACME threads from being overloaded. These include rounded ends on the push bolts and push rods. These not only prevent mushrooming, which can cause the push bolt or push rod to seize in the hole, but they also provide a weak point that should yield before these components become over loaded.

To my knowledge there has only been one screw plug exchanger where the lock ring blew out. This was the result of undersized ACME threads, extremely poor maintenance and assembly practices, and workers attempting to tighten the lock ring push bolts to stop a leak while the exchanger was on line at full temperature and pressure. Given the very limited number of failures to date, one could conclude that the designs tend to be fairly robust.

Therefore in most cases the internal partition assembly ***does not necessarily*** need to be made from the same material as the channel to avoid overloading and suddenly yielding the ACME threads during a sudden rise in temperature.

5. The internal flange studs should be replaced each time the exchanger is taken off line.

I only know of one facility that reports doing this on a regular basis. I am not sure why the practice exists because it is the external inner circle of lock ring or channel cover push bolts that loads the tubesheet gaskets. Testing has shown, given the stiffness of screw plug closures, that about 75% of the stress applied by the internal flange push bolts after a new gasket is installed is lost during initial heating.

Therefore there is no reason to replace the internal flange studs unless they are found to be damaged the next time the exchanger is opened.

PRESSURE TESTING SCREW PLUG EXCHANGERS

Most facilities want to do a full operating pressure hydro-test following assembly. Because of the unique design of the screw plug exchanger, the internal pressure has very little impact on the gasket stress. Therfore, it is not effective or necessary to do a full hydro test. It is simpler and faster to do a low pressure gas test (air or Nitrogen), which is just as effective.

All screw plug exchangers that I have been asked to work on have been previously tested with water a full pressure, and all have leaked on line, so what was the value of the pressure test? I would also suggest that the pressure testing misrepresents the ability of the exchangers to hold pressure, given there have been leaks on startup after pressure testing at other facilities. This was due to an improperly made gasket, and the fact that the gasket relaxes on first heating, where the gasket stress can drop significantly.

Screw plug exchangers are unique because the internal pressure causes very little change in gasket stress. See Figure 17. The red box is the only area where the pressure pushes back against the gasket. This is the area between the ID of the gasket and the ID of the outer compression ring. The rest of the pressure is taken up by the channel cover, lock ring and ACME threads.

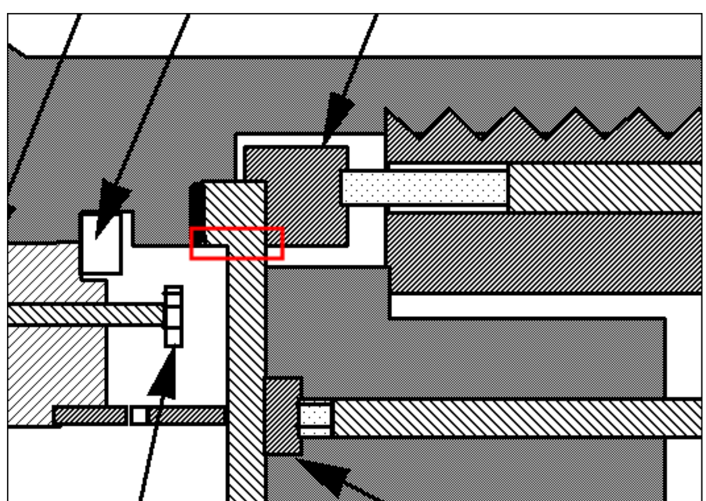


Figure 17

We will use an exchanger I just worked on as an example:

- The gasket ID for exchanger was 58.150 inch (1477 mm). The ID of the outer compression ring was 57.244 inches (1454 mm), meaning the radial width of the outer compression width that is exposed to the pressure is only 0.452 inches (11.5 mm).
- The 56, 1-1/4 inch push bolts were loaded to 61,840 psi or 700 Fp, (950 Nm)), which gave a gasket stress of 18,354 psi at assembly with no pressure.
- A full strength test was done and the final pressure was 2,500 psi (175 Kg/cm^2). The reason for the test was concern over past leaks, however this pressure only dropped the gasket stress by 1,165 psi to 17,189.

In order to really test the uniformity of thread friction and the accuracy with which the push bolt loads were set, which is the only way to really evaluate how well the contractor reconditioned the threads and how accurately they were torqued, the gasket stress really needs to be dropped to 6,000 psi. To put this into perspective, this would have required a pressure of 26,412 psi (1,857 Kg/cm^2), or 10.6 times the 2,500 psi (175 Kg/cm^2) pressure that was actually used. While this pressure would have clearly caused the shell to fail, the difference between the two numbers is intended to show how little the pressure test impacts the diaphragm gasket stress.

As an alternative, the same goal could have been accomplished by dropping the stud stress from 61,839 psi and 700 fp (950 Nm), to 20,373 psi and 230 fp (313 Nm), and using 100 psi (7 Kg/cm^2) air.

Hydro-testing with water will put a lot of water into the exchanger, especially the tubes, which cannot be completely drained out. The water either has to be removed during the plant dry-out, or it will have to wait to be displaced by the process. If water is used and any of the internal parts are stainless steel, the water must have very low levels of chlorides, or else chloride stress corrosion cracking can occur.

Figure 18 is an example of the water that can be contained in the tubes. It poured out like this for about 10 minutes, and this was after the exchanger had been drained and opened for several days.

Figure 18

There obviously needs to be clarity about the goals that any test is supposed to accomplish. While a hydro test done after the final torque value has been achieved might leave everyone confident that the connections won't leak, it is not a credible test to use in drawing any conclusions about the quality of the contractor's work or probable leakage performance over the next run cycle.

Many facilities have changed over to a simple "gross assembly error test" using air and either watching for pressure decay over an hour, or checking for leaks at joints with a soapy solution. Keep in mind that not every pipe flange that is opened during a shutdown is tested independently after assembly, so why do the connections in a screw plug exchanger need to be tested, especially since the entire plant will be pressure tested before feed is introduced as per normal startup procedures?

Each facility will have to decide if they want utilize the data, science, math and experience of other facilities, or continue to do things as they have been done before simply because that is what people are comfortable with. Change is hard, but it is only by making changes that performance can be improved and costs lowered.

ANALYZING THE DESIGN AND CALCULATING THE IMPORTANT VARIABLES

The problem with screw plug exchanger drawings is they are numbers on a page, which few end-users will easily understand. Drawings also do not answer the key questions, which include:

1. What push bolt stresses are being applied?
2. What is the resulting tubesheet gasket stress?
3. What is the resulting diaphragm gasket stress?
4. Did the manufacturer calculate the torque values correctly, using the correct K factor? (Since with push bolts only thread friction is involved),
5. How close are the ACME threads to their maximum allowable stress?
6. How close are the push bolts to their maximum allowed thread stress?

7. What happens if the torque value is increased, or decreased? How does this impact ACME and push bolt allowable strengths?

Without performing detailed calculations (loading the equations into a spreadsheet is highly recommended to reduce errors and add simplicity), the end-user is at a huge disadvantage in trying to understand the original design or the impact of any changes that might be made to gasket dimensions or torque values. This puts the engineer in a position of having to "guess" on what steps to take if problems arise, rather than being able to easily pull up the important design variables in a spreadsheet. By having the calculations in a spreadsheet it is easy to change inputs until the desired design is achieved, and standardize the analysis for each screw plug design.

The information needed for all the upcoming calculations would include:

1. Internal Flange Push Bolts that loads the Tubesheet Gasket. The size, quantity, material, code minimum yield strength, length of thread engagement, and the shear area of the threads.
2. Inner Circle of External Push Bolts that loads the Tubesheet Gasket. The size, quantity, material, code minimum yield strength, length of thread engagement, and the shear area of the threads.
3. Outer Circle of External Push Bolts that load the Diaphragm Gasket. The size, quantity, material, code minimum yield strength, length of thread engagement, and the shear area of the threads.
4. Material Strengths. The internal flange, lock ring, channel cover and channel section material and the code allowable material strength at operating temperature.
5. Recommended Push Bolt Stresses. The push bolt stress recommended by the manufacturer for the internal flange and both sets of external push bolts. This provides a good comparison to use against any changes that the engineer elects to make.
6. Gasket Type and Dimensions. The ID and OD of the tubesheet gasket's sealing area, and the gasket type that was used last. The ID and OD of the diaphragm gasket's sealing area, and the gasket type that was used last.
7. Compression Ring ID. The ID of the outer compression ring. This is used to calculate the pressure impact on the diaphragm gasket.
8. MAWP. The channel and shell side Maximum Allowable Working Pressure (MAWP) rating.
9. Bundle Weight. The bundle weight (so the correct crane can be selected).
10. Tubesheet Maximum Dp. The maximum differential pressure the tubesheet is rated for (so this pressure is not violated during any pressure testing).
11. ACME Threads. For the ACME threads on the lock ring (or channel cover): The root diameter of the thread, the mean diameter, the thread pitch, the angle of the thread and the length of the ACME threads, front to back, in either the channel section or lock ring, whichever is shortest.

With data in hand, these are the calculations that are needed (if you are good with spreadsheets, you can enter each measurement one time and then link it to all the equations where it is needed):

1. Gasket Area. Start by calculating the area of the gasket (the area of the circle formed by the OD of the gasket subtracted by area of the circle formed by the ID of the gasket). Do this for both gaskets.
2. Gasket Stress. Multiply the number of push bolts times the shear area times the stud stress to get the total force that is being applied, then subtract out the hydraulic end force and divide the remainder by the gasket area to get the gasket stress. For the tubesheet gasket, if the channel operates at a higher pressure use a negative number for the pressure because the pressure is actually adding additional gasket stress. For the diaphragm gasket, only consider the area between the gasket ID and the outer compression ring ID when calculating the hydraulic end force. The rest of the hydraulic end force is contained by the lock ring, channel cover and ACME threads.
3. Torque value. Calculate the torque value by multiplying the K factor (.117) times the push bolt stress, times the stud size divided by 12 (to convert inches into feet) times the shear area. (The K factor for push bolts is based on thread friction only as there is no nut friction against a bearing surface. This assumes a graphite and Molly based anti-seize is used like Jet-Lube 550 Extreme.)
4. ACME Thread Area. Next using the root diameter of the lock ring ACME thread, the mean diameter, the thread pitch, thread angle and length of thread engagement, calculate the area of the ACME thread that will be carrying the stress. This website provides the equation and an excellent explanation as to how to use it correctly. http://satyakhresna.files.wordpress.com/2011/04/acme_thread_designstd.pdf
5. ACME Thread Stress. Calculate the total hydraulic end force on the lock ring in pounds, the total force applied by the inner circle of push bolts in pounds, and the total force from the outer circle of push bolts in pounds. Add both sets of push bolt forces together and the total should be greater than the total hydraulic end force. Then add the total hydraulic end force to the total force from both sets of push bolts and divide this by the total area of the ACME threads to get the thread stress in psi. Divide this number by 80% of the allowable strength at operating temperature and you have the percent of allowable stress for the ACME threads. You can go to 100% of allowable because the actual yield strength is even higher, so this calculation is conservative.

For those readers that remember that the hydraulic end force is not added to the total push bolt force when calcualting the ACME thread stress (springs in series so it would be the higher of the two forces), you would be correct! We do it this way because it makes the calculation more conservative, and it follows the same rules most designers use.

6. Push Bolt Thread Stress. Finally calculate the shear stress the same way for the three sets of push bolts. Push bolts can be loaded to 100% of allowable.

THE KEY ASSEMBLY STEPS THAT MUST BE INCLUDED IN ALL ASSEMBLY PROCEDURES

Before even writting the instructions consider who will be doing the work. Working on screw plug exchangers is specialized work. I have worked with and trained a lot of contractors in the field. At best, most contractor employees will only work on a limited number of screw plug exchangers a few times in their careers. This means a lot of “on the job training” tends to take place during shutdowns. Contractors must be trained on the procedures or the work will be done very inefficently and mistakes will be made that will impact safety,cost, schedule and reliability. Given that these plants can cost $1.5 million a day to have down, and a simple internal gasket leak can result in an unscheduled shutdown, it is strongly recommended that a contractor be used that specializes in working on screw plug exchangers. They will have all the special equiment and experience necessary to get the job done quickly, reliably and at the lowest overall cost. The really good contractors have the resources to order all the material, write the opening and closing procedures, take the critical measurements and do the engineering needed to set push bolt stresses and evaluate shear stresses.

In terms of the time needed to open and close screw plug exchangers, schedules vary widely. Schedules are impacted not only by the scope of work, but also by how the equipment was assembled the last time. If lock rings were assembled correctly, they often can be unscrewed by hand. If not, it can take days to remove them.

Some contractors have taken a month to open and close screw plug exchangers where the work could have been done in a couple of weeks. I have seen other contractos that have taken on average 3-1/2 days per exchanger to complete opening and closing procedures, where others have taken two days. A really experienced contractor can do it in a day to a day and a half, depending on size.

The steps listed below are not only important for leak free performance, but for reliability and reduced maintenance costs.

Opening

1. Saturate the lock ring with a thread penetrating lubricant like WD-40, Kroil or Master Mechanic through the holes intended for grease fittings and around the OD of the ACME threads (Figure 19). Grease is too viscous and does a really poor job coating the ACME threads, which makes the lock rings harder to remove.

Figure 19

2. Air impact guns (Figure 20) can be used to remove the push bolts in a circular pattern, leaving every 5th one in place but loosened. This speeds the process up, leaves enough push bolts in place to connect a crane strap to, and eliminates the need to number the push bolts. There is no advantage to removing the push bolts in a star pattern.

Figure 20

3. Measure the distance the lock ring sticks in or out past the channel face and record this number (Figure 21). Use a paint marker to index mark the top of the channel to the lock ring. This provides a reference to use during assembly.

Figure 21

4. A strap can be placed around the push bolts and pulled up with a crane to break the lock ring loose (Figure 22).

Figure 22

5. Mark the exchanger number on all parts that are removed and keep all the parts together by exchanger number for later inspection or reuse (Figure 23).

Figure 23

6. After the diaphragm is removed, measure the dimples in the diaphragm to determine the relative spacing between internal ring and diaphragm sealing suface. The difference in the comparative depth should not be more than 0.1 inches (2.54 mm), with the inner dimple between zero and 0.1 inches (2.54 mm) deeper than the outer dimple. This is the most critical measurement to get right on a screw plug exchanger and ensures that the internal ring will rest just inside the internal face of the diaphragm before the inner cycle of push bolts are tightened. This eliminates the risk of the diaphragm being bent over the internal ring, which can prevent the diaphragm from properly contacting the diaphragm gasket. If the diaphragm is over bent it can crack and leak. If it comes out flat it can be reused.

Figure 24 shows the D1 versus D2 measurement that must be less than 0.1 inches (2.54 mm). Figure 25 shows how this measurement can be made using the back of the diaphragm as a reference. Figure 26 shows the measurement being made.

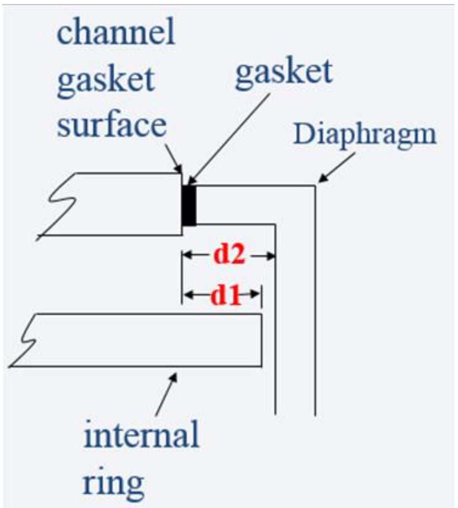


Figure 24

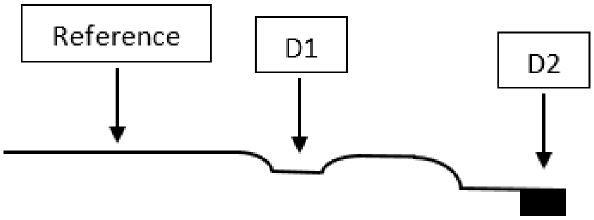


Figure 25

Figure 26

7. Forklifts, rather than special jigs, are much faster at removing internal parts provided they can be used safely in the area and can reach the exchanger parts (Figure 27).

Figure 27

8. The distance from the tubesheet to the front face of the channel must be measured and recorded before the bundle is removed. This measurement is repeated after the bundle has been reinserted to make sure it is in all the way and nothing has gotten stuck between the tubesheet and shell sealing surface (Figure 28).

Figure 28

9. Self-contained bundle pullers must be used to remove and replace the bundle (Figure 29).

Figure 29

Closing

1. Taps and dies must be used to chase all lock ring, internal flange and push bolt threads that are intended to be reused (Figure 30). When done properly, all push bolts should move freely using only the fingers. It has been well documented in numerous papers that accurate fastener stresses can not be set using used components unless the threads are reconditioned with taps and dies. If the right stress is not applied to the fastener, the gasket will not perform as designed. (See "A Simple Recipe For Solving All Refinery Sealing Issues," [2] for more information.)

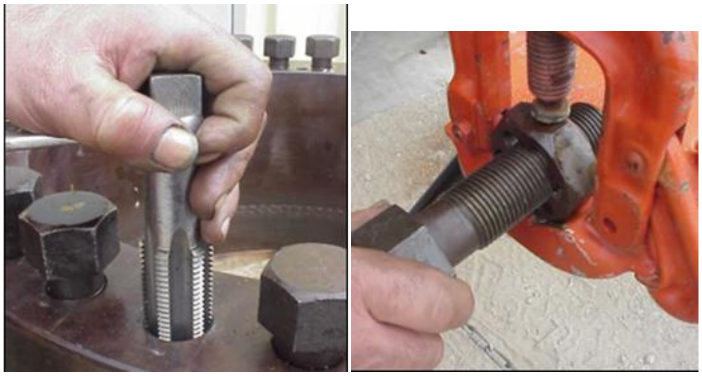


Figure 30

2. Coat all ACME threads with Jet-Lube 550 Extreme Anti-Seize (Figure 31). This anti-seize is currently the only one that contains APX-2 graphite, which has the highest oxidation temperature. This is the same graphite used to face KAG and CGG gaskets. One end-user reported that it took 2-1/2 days to remove a stuck lock ring where no anti-seize had been applied. After applying anti-seize during the assembly, the next time the lock ring was removed it took 45 minutes.

While most anti-seize materials and lubricants can be used effectively to set accurate stud and push bolt stresses, the primary job of a good anti-seize is to ensure that parts can be removed after being in service 10 years. If parts can be easily removed, equipment can be opened quickly. If not, it can take days! For more information on this topic, see "Anti-seize, friend or foe, the properties that really matter!" [3]

Figure 31

3. Heavily coat all push bolt and push rods with the same anti-seize (Figure 32).

Figure 32

4. Before the tubesheet is pushed into the channel, the ID of the channel and the tubesheet to shell gasket must be inspected to make sure they are free of damage or debris (Figure 33).

Figure 33

5. After the bundle has been completely reinserted, remeasure the distance from the tubesheet to channel face to make sure the bundle has been pushed all the way in (Figure 28 above).

6. It is recommended that a utility air pressure test be done on the shell(s) after the internal flange(s) have been torqued. Hold the pressure for an hour and watch for any pressure decay. This is a quick and easy way to make sure there is not some internal problem that will require the exchanger(s) to be reopened. This test is even more important if multiple exchangers are welded together because if there is a leak it is often impossible to tell which one the leak is coming from. This is also a good opportunity to make one final check that there are no tube leaks, or to locate a bad tube (Figure 34).

Figure 34

7. If any changes have been made to the internal parts, recheck the D1-D2 measurement. (See instruction 6 above under "Opening", Figure 24).)

8. Gaskets should only be held in place with a spray adhesive like 3M's Super 77 (Figure 35).

Figure 35

9. The inner compression ring should sit 1/4 inch (6.35 mm) inside the inside face of the lock ring or channel cover (Figure 36). This is to ensure that the push rods will actually move the compression ring inward, and that the compression ring will be supported by push bolts rather then bottoming out in the groove. If the exchanger is operated with the compression ring bottomed out in the groove, the push bolts will be unable to yield and absorb any unanticipated thermal growth by the internal parts.

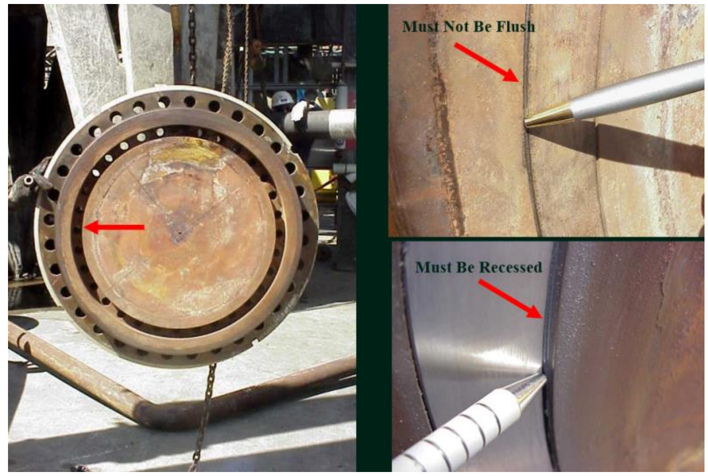


Figure 36

10. Push bolts can be run in using an air-impact gun, but must be turned by hand for the final turn to make sure the threads are running free. Using a calibrated torque wrench, tighten the push bolts in a circular pattern using two passes. Start with the inner circle of push bolts, then tighten the outer circle, then recheck the inner circle (Figure 37).

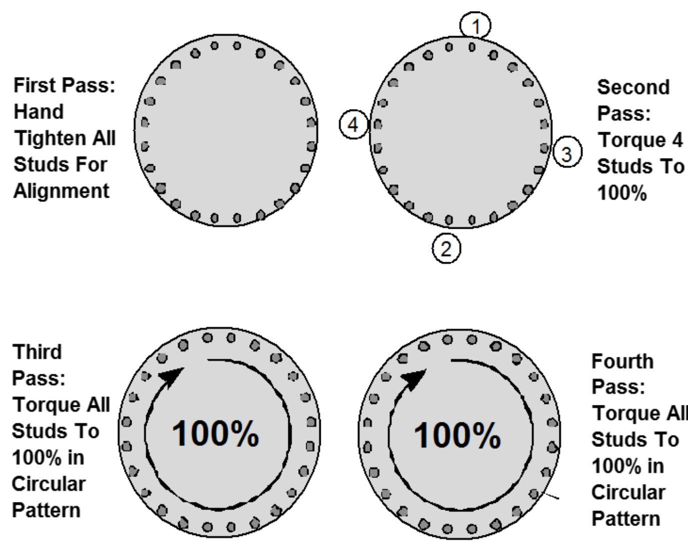


Figure 37

11. Using an air-impact gun, loosen all the external push bolts and retighten the lock ring. This is done to ensure that the lock ring and channel cover are up snug against the diaphragm. Usually the lock ring will only rotate 2 to 3 inches (50 to 76 mm). However about 10% of the time something will prevent the lock ring from screwing in all the way and it will rotate 90 to 180 decgrees during the second tightening (Figure 38). Given the one inch (25.4 mm) pitch on the ACME threads, this would mean that the lock ring was screwed in between 0.25 and 0.5 inches (6.35 and 12.7 mm) further. Had this not been done, the operating pressure would likely have cause the diaphragm to crack and leak.

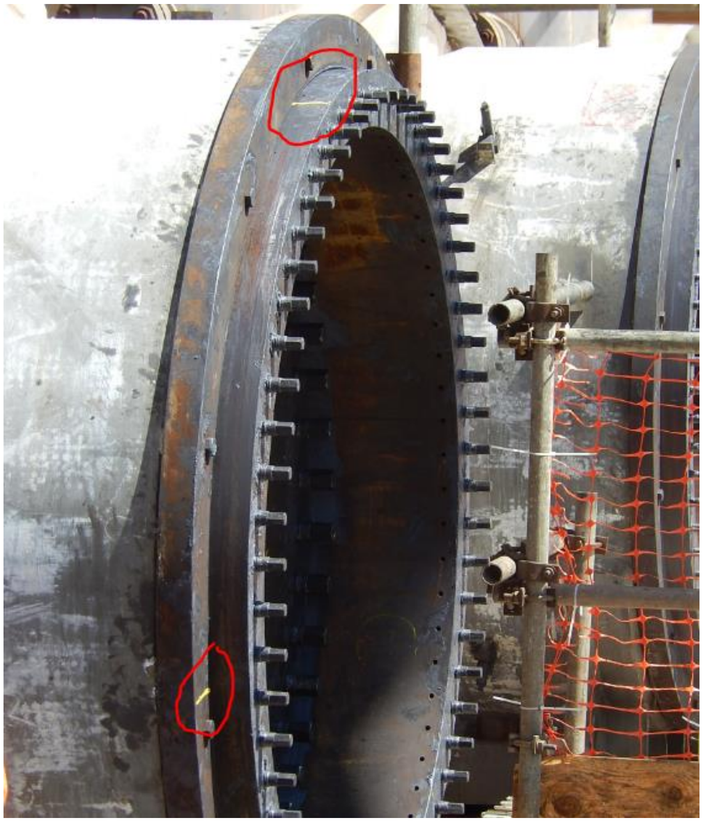


Figure 38

12.. Retighten the push bolts per step 10 and Figure 37.

13. Complete either a hydrotest, or the recommended gross assembly error test using air. Another advantage of a lower pressure test is the avoidance of brittle fracture concerns. As an example, the ASME pressure vessel code has used 40% of MAWP as the point where minimum pressurizing temperatures have to be met. Different design codes have different thresholds below which brittle fracture is not credible.

14. After the plant starts to heat up and the minimum temperature on the lock ring reaches 250F (120C), re-torque both sets of lock ring push bolts per step 10 and Figure 37. ***<u>This is the most important step!</u>*** ***<u>If this is not done the exchanger will likely leak in 12 to 15 months!</u>*** All gaskets relax. As shown above, the inner circle of push bolts can lose up to 86% of their load after inital heating.

IMPORTANT DESIGN CONSIDERATIONS WHEN BUILDING NEW EXCHANGERS

When screw plug exchangers are built correctly they provide the highest level of reliability and leak free service at the lowest total cost of ownership. When they are not built correctly they will leak, requiring additional plant shutdowns to correct at a pretty substantial cost in lost production. In addition, there is a pretty significant cost when facilities have to go back to the manufacturer to fix and correct mistakes made by the manufacturer in the original design, or to make modification to improve operation or maintainability.

When a facility or company does not have their own detailed screw plug specification, there is no way bids can be compared except by the final dollar amount. There is also no way to standardize designs across a facility or company, or make sure that important design requirements get included.

As an example, there is a major oil company that has bought screw plug exchangers from just about every current manufacturer based on the lowest price at the time of the bid. This guarantees that the total population of screw plug exchangers will include some of the designs that need improvement. The company is now spending a significant amount of money to made modification to correct design defects, like internal parts that are buckling under assembly loads.

While not a complete list, the items below should be considered for inclusion in a written specification for building new screw plug exchangers:

1. Start by defining when screw plug exchangers should be used. This might be based on the maximum MAWP, a ratio of diameter to pressure, or wall thickness.
2. All internal parts, including the partition assembly, internal flange and internal ring, shall be designed to withstand the total force from the push bolts that directly load the component without causing any buckling or distortion, even if the push bolts are tightened to their yield strength. I have seen internal partitions buckle under the recommended push bolt loads.
3. Generally a 1 inch to 1-1/4 inch ACME thread is recommended. At a minimum, the ACME threads should be long enough so if all push bolts are tightened to their yield strength, the ACME threads would not exceed 75% their allowable strength after the total force of the push bolts is added to hydraulic end force. This ensures that the ACME threads are designed appropriately.
4. Only graphite covered CGG (corrugate metal) or KAG (serrated metal) gaskets shall be used for tubesheet and diaphragm gaskets. Spiral wound, clad and double jacketed gaskets are specifically prohibited from use in screw plug exchangers. The minimum radial gasket width shall be 1/2 inch (12.7 mm).
5. Gaskets and push bolt shall be sized to obtain a 20,000 psi (137.9 MPa) gasket stress at full operating pressure when loaded to between 50% and 70% of their yield strength. The number or size of the push bolts can be increased to meet this target.
6. All push bolt and push rod ends shall be rounded or chamfered to prevent mushrooming of the ends. A 3/16", 45 degree bevel is recommended (Figure 39).

Figure 39

7. The push rods should be slightly smaller than the push bolts so they can be inserted and removed through the push bolt holes.
8. Fixed long baffle bars shall be slotted to make it easier to attach the long baffle seal strips.
9. F-436 hardened washers shall be included under the nut on B style shell studs. The length of stud shall be limited to not more than 2 to 4 threads past the top of the nut.
10. Thread engagement for push bolts shall ensure that the internal threads will fail when the push bolts are loaded to 90% of their yield strength.
11. Screw plug exchangers shall have internal flanges made from one piece construction and held in place with a 3 piece split ring. Enough push bolt load has to be present to achieve a 12,500 psi tubesheet gasket stress when the shell is pressured up to the maximum differential pressure that the tubesheet is rated for.
12. The internal flange, partition and diaphragm support ring faces (if provided) shall all be flat and not keyed into each other like the drawing below (Figure 40). If the partition or internal flange yields or distorts, the internal flange can become jammed and blocked from sliding into the shell so the split rings can be removed. "Days" were spent at one facility trying to jack the internal flange in before an in-place machining contractor was brought in to machine in greater clearances before the internal flange could be pushed past the lip of the partition (Figure 41).

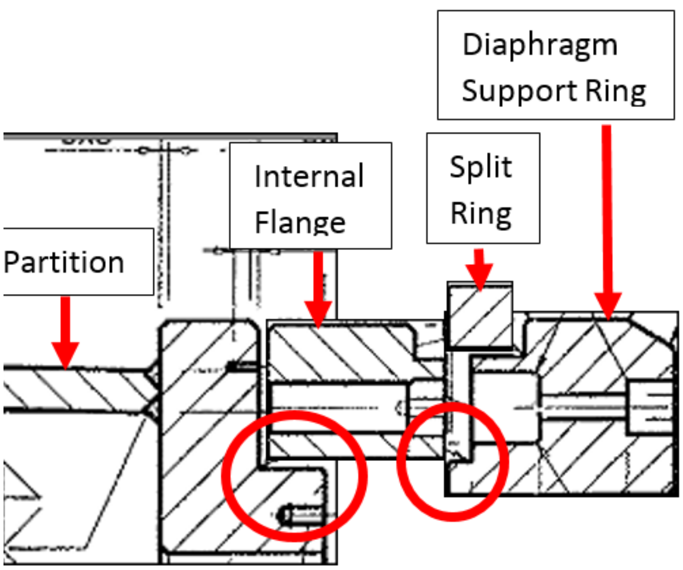
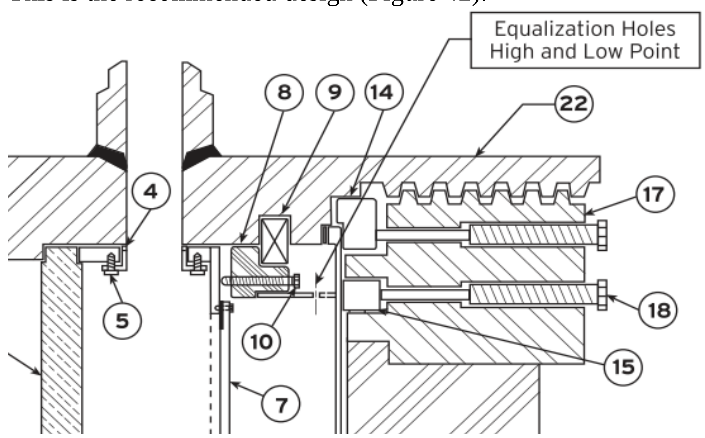


Figure 40

Figure 41

This is the recommended design (Figure 42).



Components	
1. Tube Sheet to Shell Gasket	12. Diaphragm Plate Gasket
2. Tube Sheet	13. Diaphragm Plate
3. Internal Partition Assembly	14. Outer Compression Ring
4. Packing Gland Assembly	15. Inner Compression Ring
5. Packing Gland Assembly Cover	16. Channel Cover
6. Partition Cover Gasket	17. Threaded Lock Ring
7. Partition Cover Plate	18. Inner Jack Screw
8. Internal Flange	19. Outer Jack Screw
9. 3 Piece Split Ring (Key Ring Assembly)	20. Inner Push Rod
10. Internal Flange Jack Screw	21. Outer Push Rod
11. Internal Sleeve	22. Channel Body

Figure 42

13. When the internal flange is pushed all the way in so it contacts the partition, there should be a gap of 3/4 inch (10 mm) between outside face of the internal flange and the inside surface of the split ring groove.
14. Threaded rod shall be threaded into the inner and outer compression rings at four locations evenly spaced around the compression rings and extend out through the lock ring or channel cover as appropriate. These rods will hold the compression rings in place as well as containing the push rods. They can also be used to pull the push rods back when tightened.
15. Assembly instruction shall be provided with pictures. The instructions shall not advise to leave the inner circle of lock ring push bolts loose.

16. The recommended push bolt stresses, torque values, resulting gasket stresses and ACME thread shear stresses as a percentage of allowable shall be included on the drawings.
17. The partition assembly shall not be bolted to the tubesheet.
18. The inlet nozzle seal at the top of the partition assembly shall be made from square packing and held in place by an appropriate pocket built into the follower (Figure 43). The follower shall be built so it can be inserted into the partition assembly before it is installed in the channel, and then the bolts tightened after the partition assembly is in place.

Figure 43

19. Exchangers should be designed whenever possible so the higher pressure and coldest process stream is on the channel side. This is usually the feed coming into the plant for feed effluent exchangers. This helps to protect critical parts from sudden changes in temperature. The hottest part of this stream should be contained in the channel box, leaving the coolest part free to contact the diaphragm through the hole in the bottom of the partition.
20. Grooves should be machined into the lock ring ACME thread at 5 locations evenly spaced around the circumference (Figure 44). The grooves should be rounded and extend across the full width of the of the ACME threads so spray lubricant can be injected across the width of the ACME before the lock ring is removed.

Figure 44

21. Tubesheet gasket surfaces must be flat in both the radial and circumferential directions and perpendicular to the tubesheet.
22. Gasket surface finishes shall be 125 RMS.
23. Jet-Lube 550 Extreme shall be used for assembly of all threaded fasteners including all ACME threads (lock ring and channel), push rods and push bolts. A heavy coat shall be applied.
24. An appropriate K factor like 0.117 shall be used to calculate all push bolt torque values. (The K factor for push bolts is based on thread friction only as there is no nut friction against a bearing surface. This assumes a graphite and Molly based anti-seize is used like Jet-Lube 550 Extreme, and the threads are new or reconditioned.)
25. A standardized hole pattern shall be used on the channel cover or lock ring so existing jigs can be used on multiple exchangers. The best jig I have found to date is shown below (Figures 45 and 46).

Figure 45

Figure 46

26. For B style exchangers, the shell studs that are threaded into the integral tubesheet have to be threaded in far enough that the stud will break before the blind-hole threads strip out.

A quick note on B style exchangers which are similar to A style on the high pressure side but have a removable shell with a conventional flange on the low pressure side (Figure 47).

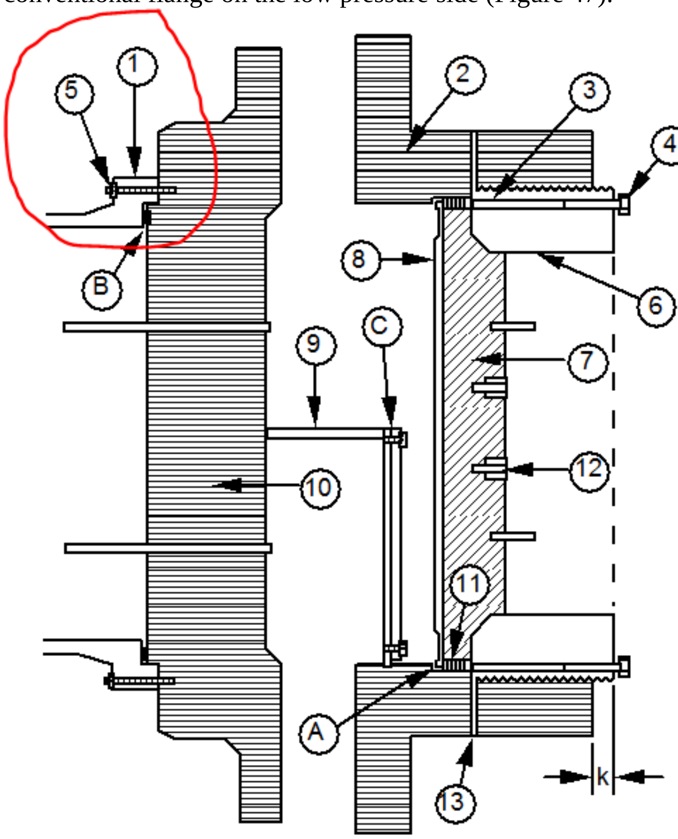


Figure 47

The only problem with these exchangers is when the channel runs hotter than the shell, which can happen in some plants during startup, shutdown or upset conditions. This causes the shell studs to thermally grow, which can drop the gasket stress and cause a leak. There are only a handful of exchangers with this problem around the world. The solution is to put 3 Inconel 718 spring washers stacked in series on each stud to increase the spring in the stud and dampen out the temperature effects.

CONCLUSIONS

Screw plug exchanger designs might seem complicated, but like any mechanical device, once they are understood and a person gains some experience, they are pretty easy to open and assemble back together again so they will operate leak free. While they have a number of internal parts that require special handling, the gasketed connections are really not that much different than those in standard flanges as far as selecting the right gasket, installing it correctly and applying the correct load.

Given the financial impact to a facility if the exchangers do leak either internally or externally, the goal and focus must be on assembling the exchangers so they will run leak free regardless of the number of upsets or unplanned shutdowns. This is an achievable goal that has already been achieved at different facilities around the world. Facilities will need to decide if they want to develop the required level of expertise and experience internally, and maintain this important level of competency, or if they would rather contract it out to a specialty company.

When new equipment is ordered, the facility will have to operate and maintain it for years to come. Good designs that are reliable and easier to work on can significantly reduce the total cost of ownership including plant down time.

REFERENCES

[1] D Reeves, W. Brown, “Common Misunderstandings About Gasket and Bolted Connection Interactions,” Proceedings of the ASME 2012, Pressure Vessel and Piping Division Conference, Toronto, Canada, ASME PVP2012-78702

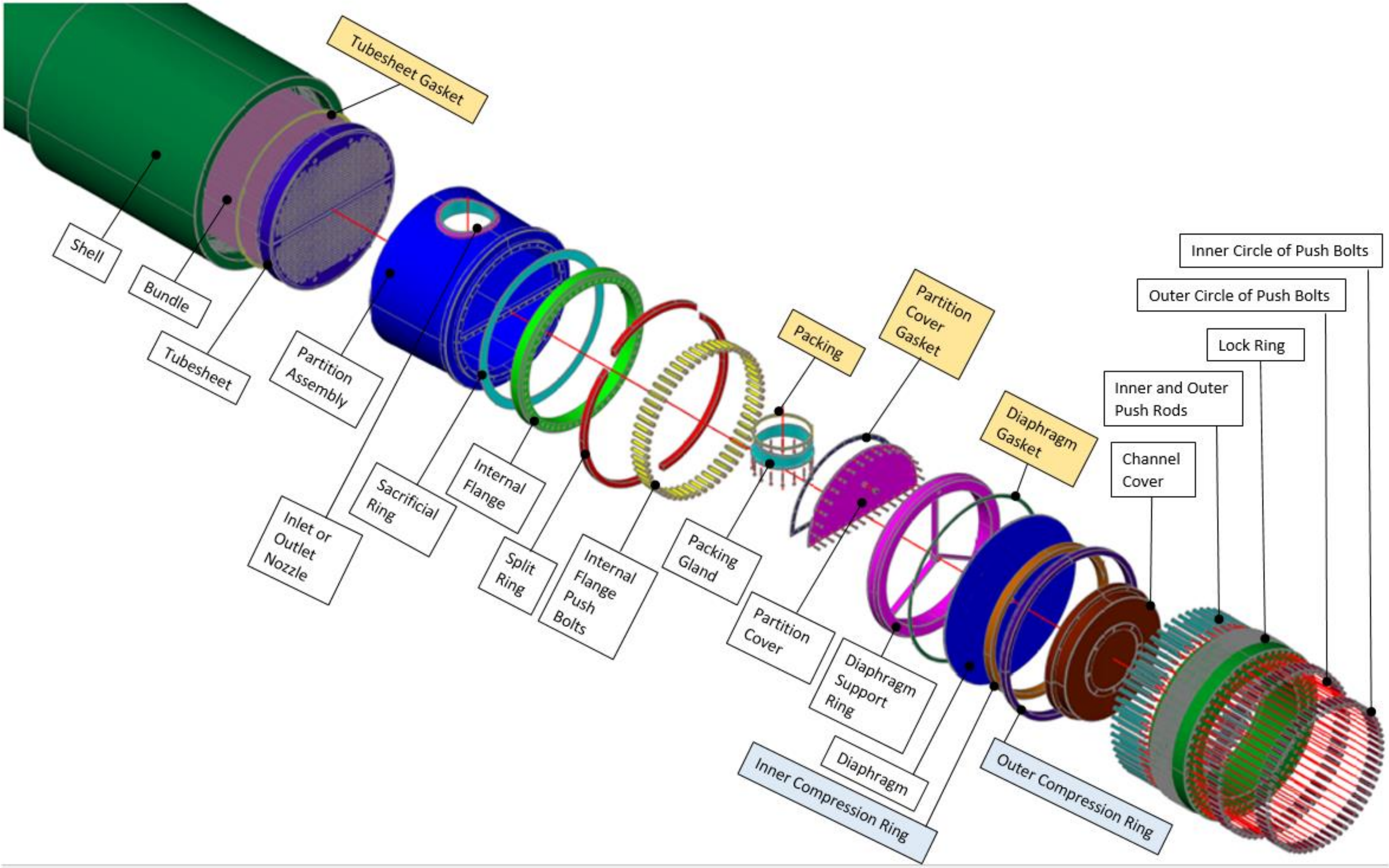
[2] D Reeves, W. Brown, “A Simple Recipe For Solving All Refinery Sealing Issues,” Proceedings of the ASME 2012, Pressure Vessel and Piping Division Conference, Toronto, Canada, ASME PVP2012-78701

[3] D. Oldiges, D. Reeves, W. Garrison, “Anti-seize, friend or foe, the properties that really matter!”, Proceedings of the ASME 2011, Pressure Vessel and Piping Division Conference, Baltimore, Maryland, ASME PVP2011-57406

[4] J. Veiga, C. Cipolatti, N. Kavangh, D. Reeves, “The Influence of Winding Density in the Sealing Behavior of Spiral Wound Gaskets,” Proceedings of the ASME PVP 2011, ASME, Pressure and Piping Conference, Baltimore Maryland, ASME PVP2011-57556

[5] W. Brown, D. Reeves, “Failure of Heat Exchanger Gaskets Due to Differential Radial Expansion of the Mating Flanges.”, ASME- PVP 2001, Atlanta GA.,

[6] J. Veiga, D. Reeves, “Heat Exchanger Gasket Radial Shear Testing”, Proceedings of the ASME PVP 2008, ASME, Pressure and Piping Conference, Chicago, Illinois, ASME PVP2008-61121



Tubesheet Gasket
Shell
Bundle
Tubesheet
Partition Assembly
Inlet or Outlet Nozzle
Sacrificial Ring
Internal Flange
Split Ring
Internal Flange Push Bolts
Packing
Packing Gland
Partition Cover Gasket
Partition Cover
Diaphragm Gasket
Diaphragm Support Ring
Diaphragm
Inner Compression Ring
Outer Compression Ring
Channel Cover
Inner and Outer Push Rods
Lock Ring
Outer Circle of Push Bolts
Inner Circle of Push Bolts

Figure 1