

PUMP SPECIFICATIONS THAT IMPROVE MECHANICAL SEAL PERFORMANCE

by

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INTRODUCTION

As the chemical, petrochemical and petroleum refining industries continue to experience a slower growth rate than at any time in the last 10 years, emphasis on cost reduction, equipment reliability and "the way we do our business" have taken on new meaning. The former emphasis on big projects has refocused to cost reduction programs, revamps, energy balance programs connected with co-generation and debottlenecking projects. Although our business will continue to change throughout the 1980s, the challenges for more reliable, less costly to maintain and easier to repair and operate pumps will intensify. Thus, plant reliability and, more importantly, operating costs will continue to become more dependent on the performance of smaller rotating mechanical equipment.

To meet this challenge, users have voiced their concerns for better equipment through the API 610 Sixth Edition, and a new API 541 standard on electric motors that will be released in the near future. Work is already underway on the rewrite of API 610 (Seventh Edition), in which the manufacturers and engineering contractors will play an important role. Users are establishing sophisticated seal testing programs to test several standard designs against modifications of these designs, under controlled conditions, in both water and hydrocarbon service. These tests have yielded some important recommendations that will improve standard seal performance [1, 2]. Consulting companies, such as MTI, have ongoing research programs for an advanced design seal for general use. Many of the major seal

manufacturers have developed computer programs to study advanced design concepts.

Use of the pump specification to stress the necessary changes required in pumps for improved seal performance should be adopted by the user. A consistent bid award to those that comply with specifications will let those that ignore specifications and bid the "same old competitive bid" take notice.

How to Approach Pump Specifications

Advances in mechanical seal technology in the past three years have resulted in several major improvements to seal design, maintainability and materials. Computation has already produced better cartridge seal designs, an increased range of standard bellows seals, wider "standard material" selection, more training programs by the manufacturers and more emphasis on meeting the true service needs of the user.

In spite of these improvements, failures of mechanical seals are continuing to have major impact on pump and agitator repair costs and reliability. Bloch's detailed study showed that in a sample of 100 seal failures, 28 were design related, 40 operation-induced and 29 maintenance related [3]. For the last several years, management has imposed sizable cost cutting goals on the maintenance department to reduce pump expenditures by improving maintenance repair methods and field preventative maintenance programs. Our experience, and that of several other companies, indicates that approximately 55 percent to 70 percent of pumps coming in for repair have been "pulled" because of a reported mechanical seal failure. Pump repair costs in large plants will run from \$3.5M to as high as \$8M per year.

However, we sometimes overlook the fact that equipment reliability and cost reduction are not just a function of maintenance alone, but design engineering and maintenance combined. If detailed pump and seal system engineering and careful integration of the seal into the pump are not made by the project specialist, seal manufacturer or engineering contractor, a marginal pump and seal system may be purchased. The lack of careful selection often leaves the plant maintenance department with the job of correcting costly "built-in" deficiencies, which risk poor seal performance, short bearing life, coupling problems and increased safety hazards from fires and personnel exposure.

Since pumps are an integral part of overall mechanical seal reliability, thought and consideration must be given as to how the pump affects seal performance. Equally important is the manner in which the seal information and specification are passed from the pump vendor to the seal manufacturer.

Before these details are addressed, some groundwork must be done as to how one approaches the general specification for a pump. Most companies have local plant pump specification guidelines or corporate standards on pumps. These documents are a method of passing on the experience and judgment of other engineers. The first time a new engineer reads these

documents, little insight is gained. As experience is gained and pumps and seals are viewed in various stages of repair and installation, the standards begin to have meaning. Unfortunately, there are standards that repeat only pump vendor catalogs or are so general that little assistance can be gained. Hopefully, your company has documented these experiences and judgments in guidelines for use in future projects by engineers.

It is a well known fact that if only physical and chemical data such as normal flow, temperatures, $NPSH_A$, viscosity, etc. are submitted to the pump vendor, a competitively priced piece of equipment for the flow and head will most likely be bid offered. Although not all competitively bid pumps affect seal performance, most have an influence. For example, a 60 gpm and 88 ft TDH (Total Dynamic Head) pump request could be covered by a 1½ in shaft pump at 3550 rpm or a 1¾ in shaft pump at 1750 rpm. Initial cost is higher for the larger model, but seal performance may show a marked decrease if the smaller model is selected and operated on a standard mechanical seal.

Equally important to the standard or specification guideline is a plant preferred vendor's list that is based on some of the following factors:

- Knowing the vendor's equipment capabilities.
- Reliable performance on previous installations.
- A history of vendor specifications compliance.
- Experience with various sizes in the pump line.
- Spare parts interchangeability with existing equipment.
- Relationship with local service personnel.
- The ability to deliver spare parts with reasonable price and response time.

The list should be an informal document that links plant experiences with pump names, models and sizes. To facilitate ease of use, the list should have a class heading for each type of pump such as ANSI End Suction, ANSI Vertical, API End Suction, Gear, Diaphragm, Multi-stage Horizontal Axial Split, etc. The list should be flexible to allow comments such as "vendor X's API end suction 4 in x 6 in—15 at 3550 rpm exhibits flow instabilities below 600 gpm." One should also continue to evaluate new pump projects or existing products not currently used. If no new evaluation is made, then there is no progress, but having five brands of ANSI pumps in the plant may be neither wise nor economical. The addition of vendors to the list can be made on a case-by-case basis after the general process data sheet is written for the pumps. It is not improper to consider only one vendor if a difficult and similar operation is successful. On API 610 pumps, 3 to 4 vendors are sufficient to provide a good cross-section of pricing and selection. If this is a pumping application where no company experience is available, ask the vendor for a user's list with names for possible contact. Contacting of these people can produce a surprising amount of useful information. Do not forget to inquire about

Table 1. API Pumps Shall Be Considered If Any Concern Is Exceeded. The Guideline Must Be Applied with Good Engineering Judgement and Experience.

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1. Head exceeds 350 ft. (106.6 m)
 2. Temperature exceeds 300°F to 350°F (depends on pipe size)
 3. Driver horsepower > 100 (74.6 kW)
 4. Suction pressure > 75 psig (516 kPa)
 5. Flow > BEP
 6. Speed > 3600 rpm
-

seal performance in similar applications.

To help define a practical application area for ANSI pumps, a guideline developed and upgraded during its 14 years of use is shown in Table 1. If any one of these particular concerns is exceeded, the application of an API 610 pump should be strongly considered. This particular set of guidelines has been cost-effective when weighted against reliability. It should be cautioned that this guideline requires good engineering judgment and experience. It must also be stressed that factors not listed in the guidelines, such as material of construction, exotic seal application, cooling or heating requirements, piping forces and moments, desired reliability and bearings are a few additional points to consider.

During the specification phase of the pump, several important points are often overlooked: 1) Information of the mechanical seal environment or possible alternate environments should be discussed. 2) Will the type of pump under consideration adequately handle the seal arrangement necessary for the process requirement? 3) Ask your preferred seal vendor if the proposed seal arrangement using *standard components* will fit into the pumps that are being considered in the bid phase. 4) There are several methods of selecting seals for pumps [4], the best method is for the user to completely involve the seal vendor in a "full disclosure" of process information related to the seal environment. 5) Contrary to current thinking, the mechanical seal should *not* be completely left in the hands of the pump vendor. Discussions on the seal application, installation requirements, materials, etc. should be between the user and the seal vendor, with the pump vendor being kept informed. Note, this may not be required on general applications, but is a must for critical pump and sealing applications. 6) One of the most important information resources, next to the seal vendor, for the seal specification is the chemical engineer responsible for that section of the plant or project.

Taking this a step further, some of the major reasons for seal failures are:

1. Wrong seal design of arrangement.
2. Wrong material of construction.
3. Shaft movement due to mechanical design and/or application and maintenance practices.
4. Seal environment—heat, dirt, pressure and gas.
5. Installation.
6. Operation.

Of these six categories listed, four relate directly or indirectly to the process environment of the seal. This is why it is important to understand the process and the service conditions the pump is expected to operate under. Ask the engineer some of the following questions:

- Does the product contain solids during an upset condition?
- Can it polymerize due to heat buildup? At what temperature?
- What can be expected during startup?
- Emergency shutdowns—flow rates, is the pump allowed to run dry?, other special demands.
- The possibility of gas—either dissolved or entrained.
- Shear sensitivity of the process fluid.
- Is it flammable, toxic or lethal?
- Any tendency towards coking?
- Corrosive properties (this area may need a review when the seals are selected because the seal environment may differ from the impeller environment).
- Does the pump experience a vacuum? If so, how long

and when in the process cycle? Are there provisions for filling the pump under vacuum? If the pump is not filled, the seal will start up dry.

- If in cold service, what is the lowest temperature possible? Have cool-down provisions been provided? The same concern applies for hot pumping operations.
- Anticipated flow rate at reduced plant throughput, during an economic swing, e.g., requirements for a bypass system.
- Viscosity—are there any dramatic changes?
- Lubricating properties of the liquid.
- Change in vapor pressure and TDH from winter to summer ambient temperatures, e.g., vapor pressure of propane at 25°F is 65 psia (447kPa abs.) versus 200 psia (137kPa abs.) at 100°F.
- If solids are present, are they abrasive, large, small, hard and/or sharp? Can the particles be separated in a cyclone separator?
- Possibility of changing to a non-pumpage seal flush is more probable if the question is addressed during the design phase of the project. If a good argument can be presented on possible poor seal performance, due to the seal environment, then an alternate seal environment stands a chance of being approved.
- Safety requirements for throttle bushing, vent and drain.
- Closed loop double seal provisions for containment.

ANSI Considerations

Some of the considerations that can affect seal performance directly or indirectly need to be reviewed when writing the pump specification. Maximum consideration for the pump's effect on seal and seal system performance weighted against the overall reliability desired in the installation need examination. Considerations most often addressed are listed below:

1. Not all ANSI manufacturers can supply quality in high alloys. One should be aware of the manufacturer's capabilities before considering the manufacturer to be a qualified bidder. Metallurgy integrity of the pump is related to seal performance, especially when dealing with Hastelloy B and C.
2. Seals to fit the conventional stuffing box—where $\frac{5}{16}$ in, $\frac{3}{8}$ in or $\frac{1}{2}$ in packing used to go—must be changed to meet the demands of needed reliability and cost cutting for today's petrochemical plants. Users and seal manufacturers should not continue to be forced to accept compromises in seal design that can only decrease reliability. Stuffing boxes must change! Many of our past, and some of our present standards, are written so that the mechanical seal is to be interchangeable with the packing if the seal should fail. In today's practice this is not the case; when a seal fails, it is replaced with another seal. The packing space constraint has forced seal designers to compromise on component thickness, length, radial clearance around the seal, as well as special on-standard components for tandem and double seal arrangements. Federal Emission Standards have further complicated the issue by forcing many exotic seal arrangements into ANSI chemical pumps that have many non-standard and compromised components. Seals need to be removed from the constraints of the stuffing box to a self-contained form, probably packaged by the seal manufacturer, to be installed on the pump. In the new concepts, the seal may fit the shaft or the "seal housing" both of which must be sized to accommodate the best seal.
3. Many of the ANSI manufacturers have realized the limitations of the stuffing box for tandem and some double seal arrangements. Some are modifying the box length to accept special glands. A few pump vendors have modifications availa-

ble that allow the seal manufacturer to bolt a complete "seal chamber" with gland to the backplate of the pump.

4. A design review should examine the method used to fit the thrust bearing into the bearing housing. Several different methods are used by the manufacturers, some of which provide too much looseness or too little support of the thrust bearing to the housing. Bloch has a concise discussion of this problem [5].

5. How the impeller design affects the stuffing box pressure should be considered. Generally, the suggested pump head limit in Table 1 should be lower than this guideline when using open impellers and an unbalanced seal. The clearance on the back side of an open impeller pump is an important factor in maintaining stuffing box pressure close to suction. As the open impeller is adjusted forward to provide proper pumping clearance, the back pumpout vane clearance to the head or backplate increases. When this clearance increases to $2\frac{1}{2}$ times the standard clearance (.015 in = .38 mm), the stuffing box pressure will rise to 50 to 60 percent of the discharge pressure (this assumes no balance holes in the impeller). This will also significantly increase thrust bearing loads.

6. Unqualified acceptance of the pump vendor's standard mechanical seal gland should be discouraged. For example: The use of very thin glands in the clamped stationary seal ring designs will distort from bolting loads so that a non-uniform pressure is applied at four points on the stationary seal ring, causing it to lose flatness. Many of these glands have no true pilot to the stuffing box (see API 610 requirements). A study of cost and design features should be made of the pump vendor's gland for all sizes of his pumps against several seal manufacturer's glands. Often the pump vendor will almost give the gland away to avoid the hassle. Glands should have an O.D. or I.D. pilot.

7. With today's stringent pollution abatement requirements on seal leakage, a rim type of baseplate or a stainless steel pan with a $\frac{3}{4}$ in connection under the suction nozzle could be a requirement for collecting leakage from the pump.

8. "Maximum capacity" or "filling slot" double row bearings should not be allowed in the thrust position.

9. The large ANSI pumps have been proven reliable with good seal life in intermittent loading (1500 to 2500 gpm) operations versus going to an API 610 pump. Again, bearing life calculations and shaft deflections should be considered along with other mechanical variables before final selection. Double volutes are preferred on large ANSI pumps, unless more than adequate size is selected to counteract deflection. This size of pump has proven to be cost effective and quite reliable in our experience with 26 installations over a 14-year span.

10. Designs using clamped inserts should be de-emphasized, due to the difficulty of the average craftsman applying uniform bolting loads. If this design of gland must be used: 1) Specify $\frac{1}{32}$ in or $\frac{1}{16}$ in thick (not $\frac{1}{8}$ in) gaskets for gland to stuffing box face makeup; 2) Dial-indicate the gland face during makeup; and 3) Require the gland and pump vendor to machine feeler gauge slots so the gland can be gauged as it is made up.

11. Balance holes in the eye of the impeller should be a part of the design even on impellers with wear rings. A rule of thumb is that the balance hole area should be at least two times the wear ring clearance area. If the balance hole area is found to be 50 percent or less than that of the wear ring area, pressure at the throat bushing could reach about 60 percent of the discharge pressure. This would be disastrous to the inboard seal of a double seal arrangement, since the seal can act as a relief valve when the pressure at the throat bushing exceeds the flush pressure.

12. Pumps should have shaft sleeves whenever possible

for ease of maintenance. Mechanical seals with dynamic Teflon shaft packing should have a hard coating or ceramic under this area. Teflon "frets" stainless steel with extreme ease, because the Teflon repeatedly wears the protective oxide which again reforms. The oxide also imbeds in the Teflon, further accelerating the oxidation and wear cycle.

13. A hard coating is also desirable under the throat bushing, whether it is a fixed Teflon lip seal or a close .002 in to .004 in radial close clearance carbon floating bushing. The main purpose of a throat bushing is to provide a controlled flush velocity to keep pumpage from entering the stuffing box. It also provides a back pressure that will elevate the stuffing box pressure significantly above suction pressure. This is helpful in vacuum service and with liquids near their flash point.

14. Use of the jacketed stuffing box on ANSI pumps to provide cooling is far less effective than flush cooling using cooling water exchangers or pumping rings used in combination with exchangers.

15. Statistics have shown that a larger number of seal failures exist in the 1½ in x 1 in - 8 pumps with impellers of 7 to 8 inch diameter at 3350 rpm. Also, a higher percentage of problems with bearings and mechanical seals exists when the 4 in x 3 in - 13 or 6 in x 4 in 10 pump is used at 3550 rpm.

16. Some pump vendors require the seal manufacturer to use an axial anti-rotation pin on the back side of the mating ring when the pump vendor's gland is used. When the mating ring is made from carbon, breakage and installation problems are common. If the pump vendor's gland is used, require radial anti-rotation pins. Use of a proper radius on the anti-rotation slot is often overlooked (Figure 1).

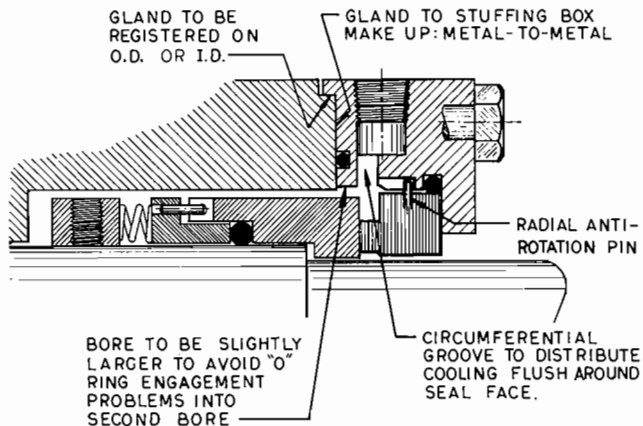


Figure 1. Balanced, Single Inside Seal with Several Desired Features.

17. Use of rubber element couplings will help dampen vibrations due to misalignment that are frequently found in ANSI pumps. Improved seal performance with rubber element couplings can be seen over the use of a gear or grid-type coupling. The use of couplings weighing more than 25 pounds is questionable on ANSI pumps.

18. Use of an external flush and normal ANSI throat bushing clearances of .030 in to .050 in would require a flush rate of 7-9 gpm to maintain a 10 ft/sec velocity through the clearance for proper barrier flow.

19. Stuffing box configurations on non-metallic pumps can and do cause poor seal life. Lack of proper registers, use of envelope gaskets and stuffing box surfaces that are not always square with the shaft have caused early failure of the Teflon bellows seal. The non-metallic shaft may also allow sliding of Teflon bellows seal (Figure 2).

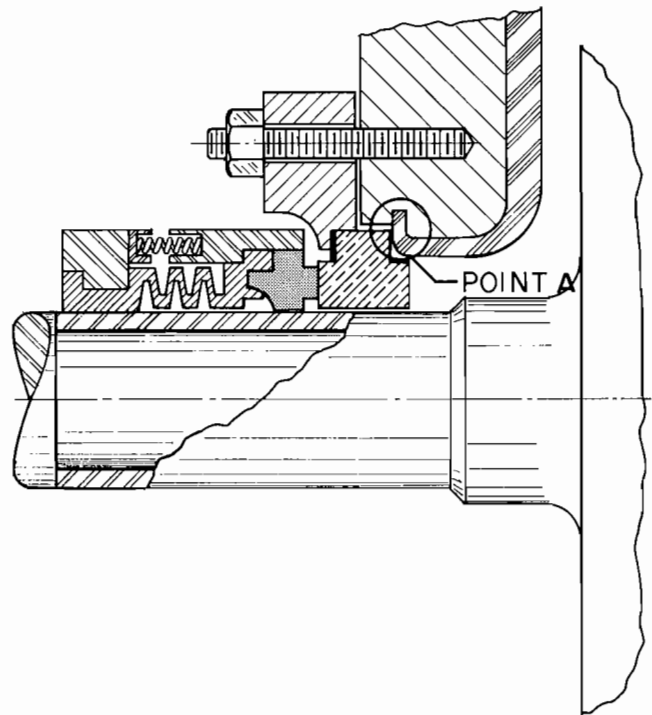


Figure 2. A Teflon Bellows Seal Mounted on a Non-Metallic Pump. The Flexibility of the Lines at Point A and the Lack of Registers Promotes the Stationary Seal Ring to Go Out of Square with the Shaft.

20. Because of the small stuffing boxes in ANSI pumps, it is very difficult to install a tandem seal without using many non-standard seal components. Attempts to resolve this problem using cartridge seals are still faced with component length limits and compromises.

API Considerations

Pumps bid against an API 610 Specification can include single stage overhung, double suction, high speed-high head flow, vertical multi-stage canned, horizontal multi-stage, etc. Due to the large variety of pumps, influence on the seal from each type of pump cannot be totally addressed. The considerations listed below are those that most often have the greatest impact on seal performance. Further information on writing pump specifications and bid analyses can be found in [6].

- The API Standard's purpose is to provide the petroleum industry with a document that specifies the *minimum* acceptable level of desired design features, material and mechanical quality, testing procedures, documentation, etc. Thus, this standard should be used as a basic document from which specifications for special requirements, materials, seals and testing can be issued. The special requirements come from the local plant, corporate and contractor standards developed for centrifugal pumps. Remember, specification of just API 610 Sixth Edition Standard will probably not fulfill all of your true requirements.

- Stable flows in a pump are a *basic* requirement for good seal performance. Pumps operating in their recirculation range have shown a history of high maintenance costs [7] and poor seal performance, regardless of the seal. Pumps with Suction Specific Speeds exceeding 12,000 heads above 650 ft TDH and horsepower above 300 hp merit close attention if the pump is operated more than 20 percent away from BEP. Some users are using *only* Suction Specific Speed as a bench mark for flow

stability in the impeller. Fraser points out that other parameters are involved [8]. Sloteman, et al., has an excellent description of what is happening during recirculation [9].

- High frequency vane pass vibrations can bring seal performance to its knees. Until the Sixth Edition of API 610, little could be done by the user when this problem was found on the test stand. The standard now allows a maximum vibration of 0.2 in/sec peak in the vane pass frequency region. The distance from the impeller O.D. to the volute is called gap "B" (Figure 3). It controls the *strength* and *amplitude* of the hydraulic shock created by the impeller passing the volute cutwaters. When placing limits on gap "B", it is defined as a percentage by:

$$\frac{D_3 - D_2}{D_2} \times 100 = \text{percent}$$

where D_3 is the diameter from cutwater to cutwater and D_2 is the impeller diameter [10]. These vane tips generated forces are a parabolic function. A change in the gap "B" percentage from 6 to 3 percent will double peak-to-peak pulsation, while a change from 6 to 1 percent will increase peak-to-peak pulsation forces by 6 times.

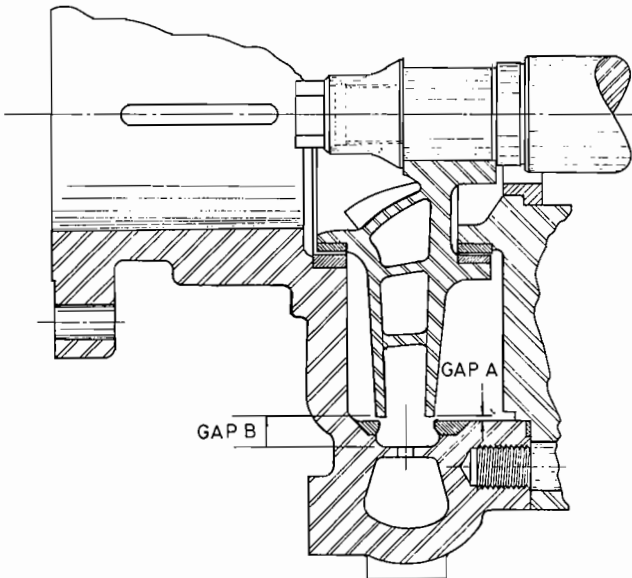


Figure 3. Modification of the Shroud Impeller Side Clearance to Volute Wall Produced a Significant Reduction in 3X and 6X Vibration and Eliminated Shroud Failures.

- Gap "A" controls the severity of the pressure pulsations behind the impeller shroud, which will cause shroud failures and high dynamic axial forces as a function of vane pass frequencies. For example, a 15½ inch maximum diameter impeller experienced 13 shroud failures in one service at a number of different plant locations. The original gap "B" clearance was ⅛ inch. Modifications made that eliminated the shroud failures and reduced the 3X and 6X vibrations are shown in Figure 3 (impeller is three vanned and the case is double voluted).

- All pump thrust bearing designs with ball bearings should have 40° angular contact duplex ball bearings, according to the new requirements of the Sixth Edition of API 610. Pump rotors with Kingsbury type thrust bearings should have a positive hydraulic method, to hold the rotor against one side of thrust bearing. Rotor "shuttling" is the primary reason for poor seal performance on some models of double suction pumps.

- Beware of mechanical seal applications on double suc-

tion pumps that were designed for packing. The excessive shaft deflection causes early seal failures. Most of the two stage overhung and double suction overhung with excessive shaft deflection have left the market, but one can still find the old double suction pumps that require packing for shaft support being bid.

- Ease of seal removal on pumps with bearings outboard of both seals is a consideration.

- Some manufacturers have optional designs on their vertical multistage pumps where the mechanical seal is not exposed to discharge pressure. This is a viable option when the discharge approaches 44 psig.

- The sleeve plays an important sealing function with the shaft packing. Make sure the sleeve is resistant to fretting and corrosion. Remember, a coating is only as good as the base metal.

- Require the manufacturer to keep sleeve diameters in ⅛ in increments for better interchangeability of spare parts.

- For some seal arrangements, stuffing box dimensions of API pumps force seal manufacturers to use non-standard seal components or small Teflon stationary packing members. You do not have to get stuck with stuffing box dimensions designed for packing! Have the stuffing box bored, if the gland bolt circle allows, or specify a smaller bore to accommodate seals mounted on the shaft. In one case, a tandem seal under 400 psi suction and 290°F has a persistent failure mode, because the force on the inside stationary seal caused the small Teflon packing members to cold flow and fail in a few days. On high suction pressure applications, calculate the forces on the seal members.

- Piping forces and the resulting movement in hot services can change a normally well performing seal design to one with a high failure rate. The Sixth Edition of API 610 has done a lot to alert the pump vendors that a heavier design is required to minimize the effects of piping loads on mechanical seal, bearing and coupling performance. Some problems exist with this section of the standard that are being addressed in the writing of the Seventh Edition. Steiger presents an excellent documentation of revision required by one pump vendor to meet Sixth Edition requirements [11]. Based on the material Steiger presents and knowing that some pump models have been manufactured since the early 1960s, a verification of compliance under API Sixth Edition, paragraph 2.4.7 may be advisable for critical applications.

- Pumps that are subjected to high piping loads and develop leaking seals when switched should merit extra attention. Verify that the gland is resistant to distortion (gland is mounted metal-to-metal against stuffing box) and ascertain that the stationary seal ring is flexibly mounted. This is a special concern with high suction pressures.

- Gland material should generally be the same material as the pump wetted parts specification. This avoids corrosion problems that are often overlooked as the cause of a seal failure.

- Some single inside seal arrangements and gland designs locate the mating ring face back in a close side-clearance part of the gland where the seal flush cannot provide proper cooling to the faces. Use of circumferential groove around the seal face will provide proper cooling (Figure 1).

- Gland designs that have the flush orifice drilling in the gland directed at the seal faces tend to promote a liquid jet at the seal faces, causing a lightly loaded seal face to open. In case of a discharge flush, the pump vendor can make the orifice an integral part of the discharge nozzle takeoff. This type of orifice is never lost in the middle of the night.

- Frequent dry seal failures on loading pumps indicate the

need for an automatic shutdown of the motor with a loss in flow (pump is cavitating or the pump lost suction). This can be accomplished by a low level signal from the level indicator, or by a low motor current cut-off set for a value which corresponds to cavitation or loss of suction. Calibrating the shutdown controls with an uncoupled motor will not result in a shutdown on loss of flow.

- Most multi-stage pumps use a breakdown bushing and a return line back to suction to reduce the pressure against the seal on the high pressure end of the pump. If abrasives or corrosion/erosion change this bushing clearance significantly, the seal will fail from overpressure. A pressure gage reading seal cavity pressure is often a useful diagnostic tool. Special coatings and a metallurgical change in the bushing material can be tried next. Multistage pumps with maximum or near maximum impeller diameters can have mechanical problems if all impellers are aligned to allow each impeller vane to pass the cutwater at the same time.

- Care should be taken to insure that the seal cavity is as self-venting as possible. This is of particular concern in vertical pumps since vapors in the seal cavity will rise around the seal and cause a dry seal failure. A good way to avoid the problem is to use a reserve flow flush from the stuffing box to the pump suction per API Plan 13.

- Request the manufacturer to state the required minimum flow to maintain stable flow in the impeller. This should not be based on temperature rise alone! If the flow value is 10 to 15 percent of the BEP, it should be rejected since it is probably temperature rise. Fraser and Heald [8,12] provide a method for calculating the onset of recirculation in the impeller. If operations are sometimes required in this range, a by-pass control system should be specified. The control should be set to bypass the amount required to keep the pump operating above the recirculation/unstable flow region.

- Some pump sizes have two power frames (bearing housing and shaft) available. Evaluate the larger frame against costs, shaft deflection, bearing life and increased seal PV.

- Critical pumps and pumps with special or critical sealing requirements should have a witness test. This test should not be a typical witnessed performance test, but a complete test of the seal, seal hardware and a mechanical integrity test using vibration readings as a basis. Draw up a testing plan and have the pump vendor agree to it before testing. If the testing equipment requested for the test is not present, hold the test until it arrives. Do not let the pace of the test be set by how fast the testing group can take data (most can do a performance test in 5 to 10 minutes). Remember, the test is your test on your pump. Problems addressed at the factory can be resolved from 4 to 6 times faster than in the field.

- The use of an "O" ring between the stuffing box face and gland assures trueness and is simple and positive (Figure 1).

- For suction pressures above 150 psig, request the pump vendor to supply the maximum thrust developed over the full curve range. State the L-10 life at that thrust with the thrust bearing proposed for this pump.

- In vertical sump pump applications, request the shaft thrust value to assure that it is down and of a magnitude that the thrust bearing balls are loaded against the race. Otherwise, axial vibration will cause failure bearings and seals.

- If tungsten carbide is run against tungsten carbide, a distortion of the seal faces by 5 helium light bands will cause unacceptable seal leakage. Distortion in the tungsten could be from heavy piping loads transmitted to the stuffing box face and seal gland.

- Sleeves shall be chamfered on both ends to aid in seal

assembly (this is often forgotten with double seals).

- If a dead-ended stuffing box must be used in a horizontal pump, have the pump vendor provide a 1/8 in hole at the top of the throat bushing to help prevent gas from accumulating in the stuffing box.

- For cold light hydrocarbon services, the basics for success are: exclusion of moisture, verify the adequacy of the seal system, required pump parts for cold service, vents, etc., design execution by one person and a good startup procedure. Studying a typical startup procedure and system schematic will help construct a good specification [13,14,15].

- When a gland has a flush feed groove for better circulation of the flush around the seal face, (Figure 1), make the initial step at least .060 in larger in diameter to keep the "O" ring from being cut upon entry into the second bore. This feed groove is preferred if there is concern about heat removal around the seal faces.

CONCLUSION

The underlying theme of this paper has been to demonstrate that by attention to details in the specification of the pump and the pumping system a more reliable seal and seal system are selected. The person most likely to enhance seal performance through good pump specification is the *seal user*.

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