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GUIDELINES FOR APPLICATION OF HIGH TEMPERATURE DUAL SEALS

by

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ABSTRACT

High temperature services are one of the more challenging applications for mechanical seals. Herein are recommendations and guidelines for selection and application of dual metal bellows seals for use in high temperature centrifugal pumps.

INTRODUCTION

In most refineries and chemical plants, high temperature pumps were among the last to be converted to mechanical seals. Today, although some high temperature pumps still use packing, mechanical seals are the norm even at process temperatures well above 700°F. In some high temperature applications, fluids are being pumped that are solids at ambient temperature. Achieving good reliability under such difficult operating conditions requires not only an appropriate seal design but close attention to application details, especially of the sealing system.

"High temperature" is a nebulous term. Typically, "high temperature" indicates that elastomers are not usually suitable for long-term use at that temperature. The seal standard, API 682 (ISO 21049, 2004), provides useful guidelines for the application of high temperature seals. In API 682 terminology, high temperature seals are referred to as *Type C* seals. The API 682 seal selection typically defaults to the Type C seal at 350°F. Type C seals are welded metal bellows seals using flexible graphite gaskets.

High temperature metal bellows seals like the API 682 (2004) Type C must be made from materials that are fully rated for elevated temperatures. Seal faces are typically carbon graphite, silicon carbide, or tungsten carbide. The default bellows material for Type C seals is Alloy 718. Low expansion alloys, such as Alloy 42, are strategically used to avoid thermal expansion incompatibilities. Adaptive hardware, such as sleeves and gland plates, is made of 316 stainless steel. When fitted with flexible graphite gaskets, high temperature metal bellows seals can be rated for up to 800°F.

Evolution of High Temperature Seals

Welded metal bellows have been used as sealing elements in mechanical seals, valve stems, and other equipment since the 1950s. These seals were originally developed for the aerospace industry, in particular for accessories and aero-engine main shaft seals. In these industries, welded metal bellows have been used for their integrity, reliability, toughness, and high temperature resistance. Operating conditions have ranged from −420°F to 1110°F.

In the 1960s, metal bellows derived from aerospace products were adapted for general industrial and process applications—mainly for use in pumps. High-temperature metal bellows seals have successfully sealed high-temperature fluids in the chemical and hydrocarbon processing industries for nearly 40 years.

There have been a number of major milestones in sealing hot pumps:

- Standardized products utilizing an optimized, tilt-edge welded metal bellows core
- Double ply bellows for high pressure
- Flexible graphite packings
- Silicon carbide seal face materials
- Corrosion resistant alloys
- Low expansion alloys
- Publication of API 682 standard (2004)

Modern high temperature seals have become very reliable through the evolution of both design and application techniques. Table 1 shows typical high temperature problems and how those problems have been addressed.

Table 1. High Temperature Sealing Problems and Solutions.

Problem	Design Fix	Application Fix
Temperature rating	High temperature seals	Cooling, external flush
Pressure rating	Double ply bellows	Pusher seal, cooling
Pump case distortion	Stationary bellows	New pump
Abrasive wear on face	Hard faces	External flush
Heat checked hard faces	Silicon carbide	External flush
Coking	Stationary bellows	Steam quench
Coking	Stationary bellows	External flush
Coking (with steam quench)	Steam distribution baffle	External flush
Steam contaminates bearing lube oil	Segmented throttle bushing with drain	Oil mist
Stress corrosion cracking of bellows	Alloy 718 bellows	External flush
Thermal distortion	Low expansion alloy	Cooling
Corrosion of low expansion alloy	Special designs using corrosion resistant metals	External flush
External flush expense	Dual seals	(accept reduced reliability)
Expensive external lubrication systems	Closed loop systems	(accept reduced reliability)

Selecting a High Temperature Seal Arrangement

High temperature seals are available in all the same arrangements as lower temperature seals:

- Single (API 682, 2004, Arrangement 1)

- Dual nonpressurized (classic tandem, API 682 Arrangement 2)
- Dual pressurized (classic double, API 682 Arrangement 3)

Each arrangement has advantages and disadvantages as summarized in Table 2.

Table 2. Comparison of Arrangements for High Temperature Services.

Arrangement	Advantages	Disadvantages
Single API 1 Stationary bellows	Simple Lowest initial cost	Seal is directly in process fluid Environmental controls needed External flush may be expensive No leakage containment Best with steam quench
	Reliability is always best when the seal is cooled and quenched.	
Dual non-pressurized "Tandem" API 2 Rotating bellows Face-to-Back	Redundancy Leakage containment	More complex Higher initial cost Inner seal directly in process Environmental controls needed External flush may be expensive Physical size Buffer fluid decomposition
	Reliability is always best when both seals are cooled. The buffer fluid acts as a self-contained quench and must be cooled and circulated.	
Dual pressurized "Double" API 3 Rotating bellows Face-to-Back	Both seals in barrier fluid Least process fluid leakage Low operating cost	Most complex Highest initial cost Physical size Barrier fluid decomposition Barrier fluid leaks into process
	With proper system design and operation, offers the highest reliability. Proper barrier fluid, cooling and circulation are essential.	

As indicated in Table 2, there are many parameters to consider when selecting the arrangement for high temperature services. The arrangement that is "best" in one application may not even be acceptable in another application. Furthermore, some end users may emphasize initial cost whereas others may emphasize operating cost or reliability. It is necessary to consider and evaluate the details of the application.

Dual seals have gained popularity over the past few years primarily due to plant hazard/safety requirements and sometimes a need to reduce fugitive emissions due to plant environmental obligations. Dual seals may be specified for hazardous/toxic, dirty, abrasive, polymerizing processes and hydrocarbon liquids operating at a temperature above their auto-ignition temperature, and/or when the liquid is not allowed to enter the flare or atmosphere for any other reason. Naturally, dual seals are much more complex than single seals.

DUAL SEAL ARRANGEMENTS

Dual seals may be classified as *pressurized* or *nonpressurized*. In a pressurized dual seal, the fluid between the two seals is pressurized above the seal chamber pressure. In a nonpressurized seal, the fluid between the two seals is essentially at atmospheric pressure. The nonpressurized fluid is called a *buffer* fluid whereas the pressurized fluid is called a *barrier* fluid because it presents a barrier to process leakage.

Just as with single seals, the seal design and materials used in dual seals should be rated for the maximum pump operating temperature.

Nonpressurized Dual Seals

A good example of a dual nonpressurized seal design that meets the requirements of API 682 (2004) for high temperature services is illustrated in Figure 1. Flexible graphite gaskets are used throughout the cartridge. The inner, or process, seal (shown on the left) has buffer fluid on the inner diameter (ID) and process fluid on the outer diameter (OD). Buffer fluid pressure is essentially atmospheric. The outer, or atmospheric, seal (shown on the right) has buffer fluid on the OD and atmospheric air on the ID although sometimes steam is used as a quench. Since leakage is a function of pressure, the inner seal leaks process fluid into the buffer system. The outer seal leaks buffer fluid to the environment. This design utilizes an axial flow pumping ring to produce buffer fluid circulation in the Plan 52 system.

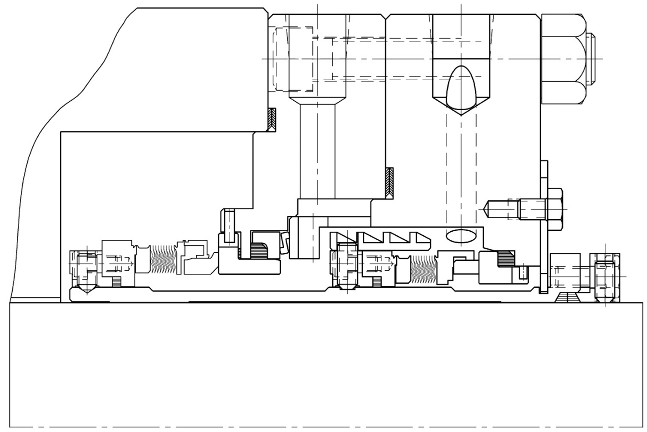

Figure 1. Dual Nonpressurized Seal.

Some dual nonpressurized seals can also be used in a pressurized mode. Such seals are said to have *reverse pressure capability* or *dual balance ratio*. The seal shown in Figure 1 has those features. When applied as a dual pressurized seal, the flush plan is API Plan 53 or 54.

Pressurized Dual Seals

When the seal shown in Figure 1 is used as a pressurized dual seal, the inner (or process) seal has the higher pressure on the inside diameter of the bellows and seal face. In Figure 2, the components are configured such that the higher pressure is on the outside diameter of the bellows and seal face. Although there are advantages for each configuration, high pressure mechanical seals are usually pressurized from the outer diameter. In contrast to Figure 1, the particular seal shown in Figure 2 is not fitted with a pumping ring; however, this is not a requirement of such configurations. When not fitted with a pumping ring, the intention is to employ an external system to circulate the barrier fluid.

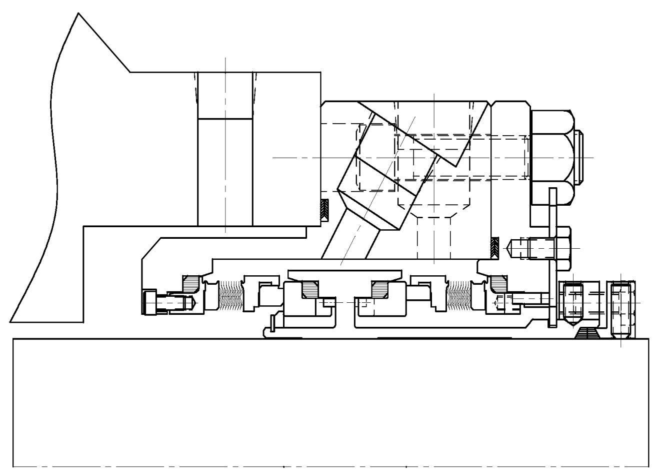

Figure 2. Dual Pressurized Seal, A.

Figure 3 is yet another example of a pressurized dual seal. At first glance, Figure 3 seems to be the same as Figure 1; however, Figure 3 does not have a pumping ring and is intended for use with an external lubrication system, Plan 54. Since an external lubrication system can produce higher flowrates than a pumping ring system, the seal in Figure 3 includes a flow diverter to direct barrier fluid beneath the inner seal. The combination of increased barrier fluid flow rate and additional cooling benefits of the flow diverter allow the seal of Figure 3 to be used at higher temperatures and pressures than the seal of Figure 1 when Figure 1 is used as a dual pressurized seal.

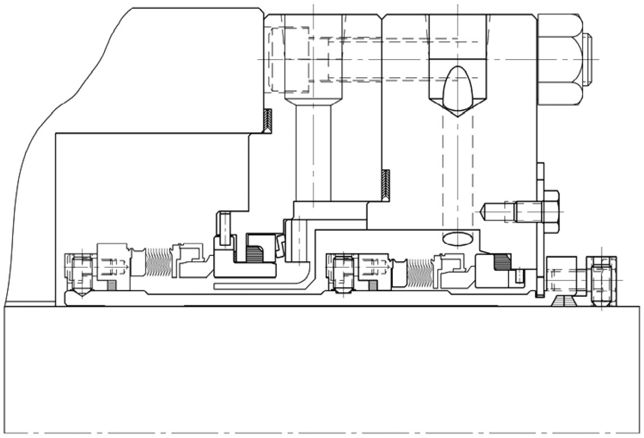

Figure 3. Dual Pressurized Seal, B.

In Figure 3, the inner, or process, seal (shown on the left) has barrier fluid on the ID and process fluid on the OD. API 682 (2004) recommends that the barrier fluid pressure be more than process fluid pressure by about 20 to 60 psi. The outer, or atmospheric, seal (shown on the right) has barrier fluid on the OD and atmospheric air on the ID although sometimes steam is used to quench this seal. The outer seal must be capable of operating at full barrier pressure. Since leakage is a function of pressure, the inner seal leaks barrier fluid into the pump. The outer seal leaks barrier fluid to the environment.

HIGH TEMPERATURE LUBRICATION SYSTEMS FOR DUAL SEALS

API 682 (2004) uses the following definitions:

- 3.3 Arrangement 2 seal: seal configuration having two seals per cartridge assembly with a containment seal chamber that is at a pressure lower than the seal chamber pressure
- 3.4 Arrangement 3 seal: seal configuration having two seals per cartridge assembly that utilizes an externally supplied barrier fluid
- 3.7 barrier fluid: externally supplied fluid at a pressure above the pump seal chamber pressure, introduced into an Arrangement 3 seal to completely isolate the process liquid from the environment
- 3.9 buffer fluid: externally supplied fluid, at a pressure lower than the pump seal chamber pressure, used as a lubricant and/or to provide a diluent in an Arrangement 2 seal
- A.4.12 Plan 52: Plan 52 is used with Arrangement 2 seals, with a contacting wet containment seal utilizing a liquid buffer system.
- A.4.13 Plan 53: A Plan 53 system consists of dual mechanical seals with a barrier fluid between them.
- A.4.14 Plan 54: Plan 54 systems are also pressurized dual-seal systems.

Certain terms are always used together and should not be mixed:

- Arrangement 2, buffer fluid, unpressurized, Plan 52
- Arrangement 3, barrier fluid, pressurized, Plan 53 or 54

Although discouraged as outdated terminology, “tandem seals” function as Arrangement 2, while “double seals” function as Arrangement 3.

Barrier and Buffer Fluids

As a practical matter, fluids used as buffer fluids are often used as barrier fluids and vice versa; however, it is important to match the fluid to the service and operating conditions. In general, both barrier fluids and buffer fluids have the following characteristics:

- Safe
- Clean
- A good seal face lubricant at the operating conditions
- Chemically compatible with the process fluid

Both buffer and barrier fluids are considered to provide a safety zone between the process and the atmosphere and must not create a hazard in the event of leakage.

Recommended barrier fluids for hot service include heat transfer fluids and synthetic commercial barrier/buffer fluids. Transmission fluid, mineral oils, and turbine oils are not recommended. It is usually best to get the barrier fluid viscosity between 1 cP and 5 cP at the barrier fluid operating temperature.

If not designed with care, the chamber for a dual seal can have local areas of poor, perhaps even zero, circulation. The temperature in a stagnant area could reach the pump temperature and cause the barrier fluid to decompose to form coke or similar solids. In recognition of potential decomposition problems, barrier fluids should be evaluated at the pump temperature as well as the normal system temperature.

The minimum circulation rate is usually based on a computed 30°F temperature rise in the fluid considering heat generated by both the inner and outer seals as well as the heat soak from the pump. A safety factor is sometimes applied depending on the accuracy of the available information and the nature of the pump service.

When using one of the more viscous barrier fluids, there can be problems in getting sufficient flow from a pumping ring system. In that case, an external pump might be used to provide adequate circulation. Alternately, an elevated barrier fluid temperature might be considered.

Heat Soak

Heat soak is heat transfer from the hot pump case to the fluid in the seal chamber. API 682 (2004) provides an equation for estimating heat soak in the form of:

$$H_s = UA\Delta T \qquad \textbf{(1)}$$

where:

H_s = Heat soak, Btu/hr
UA = 12S where S is the seal size in inches
ΔT = Pump temperature-seal chamber temperature, °F

Equation (1) is intended to be an estimate and used only in the absence of data. It seems to be approximately representative of the actual heat soak, especially for API 682 (2004) seals in water at shaft speeds of 3600 rpm. For oils and for slower shaft speeds, Equation (1) probably predicts a high value for heat soak.

In high temperature pumps, the heat load imposed on the lubrication system is mostly due to heat soak. Therefore, if Equation (1) is used to estimate the heat soak, then the barrier fluid flow rate requirements are nearly the same regardless of shaft speed. Notice that by allowing a high barrier fluid temperature, heat soak and, therefore, the computed required circulation rate are reduced.

Pumping Rings

With either Plan 52 or 53A, a pumping ring is required according to API 682 (2004) and is necessary in order to generate the required barrier fluid flowrate. Analysis of the pumping ring performance is required in order to assure the adequacy of the pumping ring and system. Specific pumping ring curves should be used and compared to the system curve. Either axial or radial flow pumping rings can be satisfactory, especially at higher speeds and when using tangential outlets. Flowrates from such well-designed pumping rings in well-designed systems can be up to 2 gpm per inch of shaft size at 3600 rpm. However, flowrates are more typically less than 1 gpm per inch of shaft size at 3600 rpm and even less at lower shaft speeds, especially with radial flow pumping rings and nontangential outlets.

Unless the buffer fluid is flowing beneath the inner seal, that area will be stagnant and at elevated temperature. For these reasons, the buffer fluid should enter the dual seal chamber near the inner seal and flow toward the outer seal. In doing so, the seal chamber must not impose a "torturous flow path" on the buffer fluid flow.

In addition to the tangential outlet, large porting (½ inch minimum) and ¾ inch connecting tubing or pipe should be used (no 90 degree bends—only 45 degree fittings).

Again, a key requirement is that the pumping ring must produce the necessary flowrate. Otherwise, an external pump must be used.

Reservoirs

In general, the reservoir for dual seals in high temperature pumps should be designed according to API 682 (2004); however, some modifications are needed for high temperature service:

- Up to 10 gallon liquid volume may be required.
- Water cooling is definitely required.
- Additional cooling coils (in comparison to typical reservoirs) may be required.
- Removable head
- High temperature level gauge
- 316SS (not 316L)
- Mesh guard for personnel protection (if bulk temperature is high)
- Instruments rated for pump temperature
- Optional high temperature switch or transmitter

For some applications, the large reservoir as described above may not be necessary. However, special consideration should be given to the cooling coil area and reservoir volume. Typically, a 3 minute retention time is recommended. For example, if the fluid circulation rate is 2 gpm, then the reservoir should have at least 6 gallons of liquid volume.

Although some users require that reservoirs be designed, fabricated, inspected, and coded as ASME pressure vessels, this is not a requirement of API 682 (2004). For reservoirs built entirely of piping components, API 682 considers the reservoir to be part of the piping system. Therefore, API 682 reservoirs should be designed, fabricated, and inspected according to ASME B31.3 (ISO 15649) just as is the pump suction and discharge piping.

Plan 52

Plan 52 is used for nonpressurized dual seals. Fundamental issues affecting the reliability of seals when using Plan 52 in high temperature pumps include:

- Decomposition of barrier fluid.
- Heat transfer.
- Personnel protection.

In high temperature pumps, the buffer fluid of a Plan 52 system should be considered as a closed-system quench for the inner seal as well as a lubricant for the outer seal.

Plan 53A

Plan 53A is used with pressurized dual seals. In Plan 53A, pressurization is accomplished with pressurized gas in direct contact with the barrier fluid. The system pressure is usually 20 to 60 psi above the seal chamber pressure. Fundamental issues affecting the reliability of seals when using Plan 53A in high temperature pumps include:

- Absorption and liberation of gases (usually nitrogen).
- Decomposition of barrier fluid.
- Heat transfer.
- Personnel protection.

In Plan 53A, the pressurizing gas, usually nitrogen, is in direct contact with the barrier fluid. The reservoir temperature is less than the pump temperature. Therefore, the barrier fluid absorbs gas while inside the (cooler) reservoir and releases gas while in the (hotter) dual seal chamber. There are two significant problems associated with release of gases:

- A gas pocket can form around the seal face that might severely limit heat transfer, and
- The pumping ring could become vapor locked and the barrier fluid circulation would stop.

In consideration of these potential problems, conservative pressure and temperature limits have been traditionally used with Plan 53A.

For Plan 53A, API 682 (2004) recommends a maximum pressure of 150 psig maximum but does not comment on pump temperature or operating temperature. However, the following guidelines have been developed based on field experiences:

- Pump temperature < 500°F (or proven experience)
- Reservoir bulk temperature < 300°F
- Reservoir bulk temperature < pump temperature

For higher temperatures and/or pressures, Plan 53A is not recommended and Plan 54 should be considered.

Plan 54

Plan 54 provides clean pressurized barrier fluid to a dual pressurized seal from an external source. The external source is usually considered to be a self-contained lubrication system comprising a low pressure reservoir, a circulating pump, a cooler, filters, and various instrumentation and controls. Strictly speaking, there actually is no standard "Plan 54 System." Plan 54 means only that connections are provided in the seal glandplate.

In addition to the heat loads from the seal and heat soak from the pump, heat loads for Plan 54 include the inefficiencies of the pumping system. On low pressure/flow systems this is minimal, but can become significant on larger systems operating at high pressures and flows.

The complexity of the Plan 54 system should be inline with the importance of the equipment to the overall process and the associated hazards of the pumped fluid. When the Plan 54 system is supplying multiple seal chambers, precautions should be taken so a failure of one seal will not drain the entire system causing a chain reaction. Precautions should also be taken to prevent contamination of the barrier fluid should one seal fail.

EXAMPLE AND CALCULATIONS

A 3.5 inch high temperature pressurized dual seal similar to Figure 1 (or Figure 3), is to be used in a 500°F pump at 3600 rpm. The pump seal chamber pressure is 100 psig. Evaluate this application for Plan 53A or Plan 54 barrier system.

Whether for Plan 53A or Plan 54, the barrier pressure would be set at about 140 psig. Assume a reservoir bulk temperature of 150°F. Select a synthetic barrier fluid that is rated for high temperature service and estimate the required flowrate. (Notice that the assumed operating conditions meet the general guidelines for Plan 53A that were previously recommended.)

A typical set of physical properties for the barrier fluid at 150°F might be:

- Specific gravity, sg = 0.77
- Specific heat, C_p = 0.55 Btu/lbm °F
- Thermal conductivity, k = 0.12 Btu/hr ft °F
- Viscosity, μ = 6 cP

Without going into detail, for purposes of this example, assume that the inner seal generates 4000 Btu/hr and the outer seal generates 5000 Btu/hr. (The differential pressure on the inner seal is 40 psi; differential pressure on the outer seal is 140 psi. Inner and outer seals may have different materials, face designs, spring loads, balance ratios, etc.)

The heat soak can be estimated from Equation (1). If the bulk temperature of the barrier fluid is 150°F and the pump temperature is 500°F, then the heat soak is:

$$\begin{aligned} H_s &= 12\,(3.5)\,(500-150) \\ H_s &= 14{,}700\ Btu\,/\,hr \end{aligned} \tag{2}$$

The total heat load on the reservoir is the heat load from the seals plus the heat soak.

$$\begin{aligned} H_t &= 4000+5000+14{,}700 \\ H_t &= 23{,}700\ Btu\,/\,hr \end{aligned} \tag{3}$$

The total heat load must be transferred to and from the barrier fluid. An energy balance on the barrier fluid is:

$$H_t = m\,C_p\,\Delta T \tag{4}$$

where:

m = Mass flowrate of the barrier fluid, lbm/hr

C_p = Specific heat of the barrier fluid, Btu/lbm °F

ΔT = Differential temperature of the barrier fluid, °F

The required circulation rate can be determined using Equation B3 and the recommended guideline of a 30°F temperature rise in the barrier fluid.

$$\begin{aligned} m &= 23{,}700\,/\,(0.55\times 30) \\ m &= 1436.4\ lbm\,/\,hr\ (= 3.73\ gpm) \end{aligned} \tag{5}$$

It is important to recognize that this 3.73 gpm is the recommended flowrate; the actual flowrate might be different. The actual flowrate will be a function of the pumping ring and system design (Plan 53A) or the external pump and system design (Plan 54).

To determine the circulation rate for a Plan 53A system, it is necessary to have a performance curve for the pumping ring that can be superimposed on the system curve for the Plan 53A. Often, the available performance curves are typical and may not necessarily match the barrier system. A typical set of curves is shown in Figure 4.

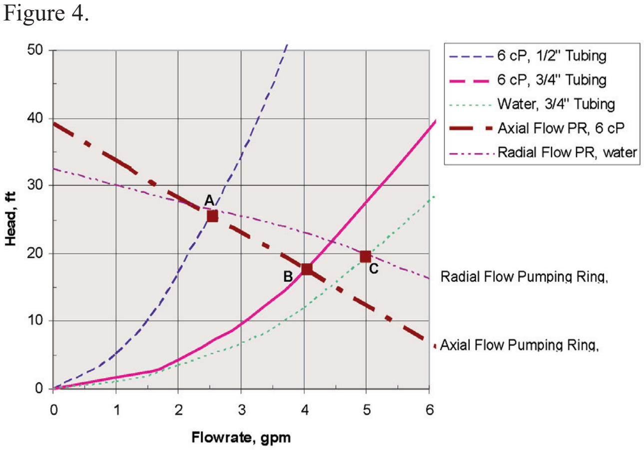

Figure 4. Pumping Ring and System Curve for 3.5 Inch Seal at 3600 RPM with Typical Plan 53A.

Figure 4 illustrates the performance of both an axial flow pumping ring and a radial flow pumping ring. However, the radial flow pumping ring performance is shown for water, whereas the axial flow pumping ring performance is shown for oil. Also, Figure 4 illustrates three system curves: one for water in ¾ inch tubing and the others for oil in ½ inch and ¾ inch tubing. In an overlay of pump and system curves such as Figure 4, the intersections represent the actual performance. That is, Point A (2.6 gpm, 26 ft of head) represents the actual flowrate of oil when an axial flow pumping ring is used with ½ inch tubing. Point B (4 gpm, 18 ft of head) represents the actual flowrate of oil when an axial flow pumping ring is used with ¾ inch tubing. Point C is for water and not applicable to this example.

It is essential to realize that Figure 4 includes many design parameters for the pumping ring and system. A comparison of Points A and B quickly shows the effect of tubing size but there are other important parameters: radial clearance, inlet/outlet port size, tangential versus radial outlet, relationship of inlet connection to outlet connection, viscosity, length of tubing, tubing valves and fittings, etc., etc.

At this point, a recommended circulation rate of 3.73 gpm has been determined based on the general guideline of a 30°F rise in the barrier fluid. Figure 4 indicates that a circulation rate of about 4 gpm can be obtained with a pumping ring in a Plan 53A system. Therefore, Plan 53A can be considered for this application.

For 3.73 gpm circulation rate and a 3 minute retention time, the barrier fluid volume in the reservoir would be about 11 gallons. If made of 6 inch pipe, such a reservoir would have to have about 8 ft of liquid filled length; total length would probably be about 10 or 12 feet. If 8 inch pipe is used, the reservoir would have about 5 ft of liquid filled length; for 10 inch pipe, about 3 ft. The point is that these reservoirs can be physically large. For reference, API 682 (2004) only considers 3 or 5 gallon reservoirs.

Assumed is a bulk fluid average temperature of 150°F and a fluid temperature rise of 30°F. The reservoir must have the cooling capacity to operate at these temperatures. Similar to any heat exchanger, the necessary heat transfer area can be estimated from Equation (6).

$$H_t = UA\overline{\Delta T} \tag{6}$$

where:

U = Overall heat transfer coefficient, Btu/hr ft^2 °F

A = Heat transfer area, ft^2

$\overline{\Delta T}$ = Log mean temperature difference, LMTD, °F

Some additional assumptions are now needed.

Checking various handbooks, the overall heat transfer coefficient for oil to water, U, varies from about 20 to 60 Btu/hr ft^2 °F. The low numbers will match with higher viscosity oils and the higher numbers with lower viscosity oils. Of course, the overall heat transfer coefficient will vary with flowrate and the design of the reservoir as well. For purposes of this example, an average value of 40 Btu/hr ft^2 °F will be used.

The log mean temperature difference, LMTD, could be computed based on either parallel or counterflow temperatures. Neither exactly applies for typical reservoirs but choose parallel flow as the basis unless more specific information is available. Actually, because inlet to outlet temperature differences are small, the LMTD is nearly the same as the average temperature difference. Assuming an inlet cooling water temperature of 80°F and an outlet of 100°F, the average temperature difference is 60°F (LMTD is 56°F).

From Equation (6), for a heat load of 23,700 Btu/hr, overall heat transfer coefficient of 40 Btu/hr ft^2 °F and average temperature difference of 60°F, the required heat transfer area is 9.9 ft^2. As a point of reference, this might be a cooling coil made of 50 ft of ¾ inch tubing or 76 ft of ½ inch tubing. This is about 4× the amount of the cooling coil area in a typical seal pot.

Without going into the details, the cooling water flowrate can be easily computed from the total heat load and the assumed 20°F temperature rise (Equation [4], except with properties of the cooling water). The cooling water flowrate would be 2.4 gpm.

These calculations, although simplistic, illustrate that Plan 53A is a reasonable candidate for the support system for this dual pressurized seal. The barrier pressure is less than the suggested maximum in API 682 (2004), the pump temperature is the same as the API 682 qualification test temperature (500°F), a good barrier fluid is available, and the pumping ring performs as needed provided the system uses ¾ inch tubing. The calculations also show that a rather large reservoir with a significant size cooling coil is needed. In fact, the API 682 qualification test parameters are similar to this service; therefore, an API 682 qualified seal can be found and applied.

In a simplified evaluation such as this example, the question of safety factors often comes up. In the example above, there is no apparent application of safety factors. The general philosophy of applying safety factors should be to use safety factors when the available information is suspect as well as when the consequences of error or failure become increasingly significant. A major consideration is where to apply the safety factor and when to apply it in the evaluation.

Frequently, the heat load is one of the first calculations and there is a great temptation to apply a safety factor to the estimated heat load. However, keep in mind that applying a safety factor to the heat load will have a significant effect on almost every parameter and probably will dictate whether the flush plan is 53A or 54. Also, the API 682 (2004) correlation for heat soak is believed to be conservative; that is, the actual heat soak is probably less than estimated anyway.

The circulation rate (3.73 gpm in this example) is determined from the total heat load and a guideline of 30°F temperature rise for a good synthetic oil barrier fluid. Computational examples in API 682 (2004) for flush rate are based on a 5 to 10°F temperature rise and a design factor of two. It is important to distinguish that the API 682 example is for the flush injection rate to a single seal on water and not a high temperature dual seal with oil barrier fluid. Whereas it is relatively easy to obtain high flush rates with injection flush plans such as Plan 11, it is very difficult to get similar flowrates with a pumping ring. For example, if the allowable temperature rise in this example had been limited to 5°F instead of 30EF then the computed circulation rate would have been 22.4 gpm instead of 3.73 gpm. After application of the design factor of 2, the recommended flush rate would have been 44.8 gpm. These flush rates (44.8 and 22.4 gpm) are unnecessarily high.

A better method for application of a safety factor to the circulation rate is to select a pumping ring (Plan 53A) or external pump (Plan 54) with a higher capacity than the recommended/computed capacity. In this example, a pumping ring was available that could produce 4 gpm of barrier fluid flow in a Plan 53A. This is a safety factor of 1.07 as compared to the computed circulation rate of 3.73 gpm. If an increased safety factor is needed, then Plan 54 would be required because there was no higher capacity pumping ring available. For example, to get a safety factor of 2 for circulation rate, an external pump with a capacity of 7.5 gpm is necessary.

Based on a 3 minute retention time and 3.7 gpm circulation rate, the reservoir was sized as 11 gallons. The concept of average retention time is widely applied to lubrication systems having a large low pressure reservoir for cooling and degassing. Retention time is a less significant variable for Plan 53A than for Plan 54 systems. In particular, for Plan 53A systems, there is no degassing in the reservoir; in fact, gas is absorbed in the reservoir. For these reasons, there is no particular reason to apply a significant safety factor to the volume of the Plan 53A reservoir. The reservoir volume for the Plan 54 system should be based on the retention time for the actual flow rate of the external pump.

Of all the estimates, the required cooling coil area for the reservoir is probably the least accurate calculation. Since heat transfer is a function of the barrier fluid properties, barrier fluid flowrate and cooling water flowrate, as well as the physical design and arrangement, this calculation will probably always be best considered as an estimate. This is a good place to use a safety factor. When selecting a reservoir, select one that has more than the estimated required amount of cooling coil area.

CONCLUSION

Pressurized dual seals are becoming increasingly popular in high temperature pumps. Although these are complex seals and systems, good reliability can be achieved by paying close attention to the details. Some of those details have been presented and discussed.

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