

FULL FLUID FILM FACE SEALS

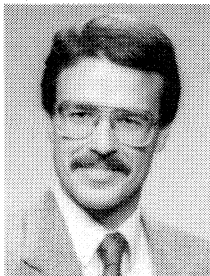
by

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Peter Lytwyn graduated from Concordia University in Montreal with a degree in Mechanical Engineering. After graduation, he joined Durametallic Corporation as a Design Engineer, where he was responsible for the design and development of high speed full fluid film face seals.

While working in this area, he did research at the Atomic Energy of Canada for one year in the special projects group, as attached staff. Presently, as Project Engineer, he oversees high technology seal applications primarily related to boiler feed pumps, boiler circulation pumps, high pressure pumps in light hydrocarbon service.

ABSTRACT

The basic types of seal face lubrication are described and the basic design concepts and deflection analysis are discussed. Present field installations are presented, along with operating conditions and common problems experienced by power plant operators. A full fluid film seal is described along with the potential advantages these seals have for high energy pumps.

INTRODUCTION

There are two fundamental types of seal face lubrication to consider when sealing pumps: boundary lubricated and full fluid film. When applying seals, it is important to consider the rubbing characteristics of the seal face materials and the load carrying and heat dissipation capabilities. These will govern seal type selection, especially when conventional seals are pushed beyond normal limits.

Pumps that were considered to be unsealable are being sealed today because the necessary analytical tools that did not exist ten or fifteen years ago exist today. These pumps can be found in any industry from power generation to petroleum. They can pump anything from water to heavy crude oil.

It is only through understanding basic seal design principles and theory that advanced concepts can be formulated and observed to attain successful seal designs. It requires much research to understand how seals react to changing operating conditions and to being pushed to their full potential.

FULL FLUID FILM LUBRICATION MECHANISMS

Full fluid film lubrication can be described as a constantly thick film of liquid between two sealing faces. Film thickness is dependant upon three basic mechanisms or modes:

- hydrostatic
- hydrodynamic

- face heat generation.

All of these modes are necessary for successful full fluid film design. The designer must understand how to control each mode. It is by being thoroughly familiar with the causes of each mode that film thickness can be controlled within tolerable limits.

Hydrostatic Lubrication

In this primary mode of lubrication, seal faces are fully separate, even though there is no relative rotation between the faces. Full hydrostatic lubrication is achieved only when there is a pressure differential across the face and the faces are deflecting convergently in the direction of leakage.

Proper face deflection produces the necessary forces in the seal gap which separate the faces. Hydrostatic lubrication can be controlled and even eliminated through careful design. It is the most predictable form of lubrication, since it relies on a pressure differential across the face.

Hydrodynamic Lubrication

This secondary mode of lubrication can be minimized by careful design but not eliminated. The main cause of hydrodynamic lubrication is the waviness of the faces. Waviness can be transmitted or can exist after the lapping procedure. Any type of drive mechanism such as pins or cap screws can transmit waviness. Moments and forces are set up at the drive slots, raising stress levels, thereby distorting the seal face. Seal faces, though lapped to very tight specifications, may have some waviness. This can be found in low modulus materials, such as carbon, but is not found to any high degree in high modulus materials, such as silicon carbide.

Face Heat Generation

A tertiary source of lubrication is face heat generation. This lubrication occurs as a result of fluid shear between the seal faces, as they rotate relative to each other. The shearing of the fluid sets up a heat flux condition whereby heat is conducted into the seal faces. As the heat conducts through the seal rings, high stresses are set up. The seal ring distorts under the influence of the stresses, usually favorably in the direction of leakage.

SEAL FACE DEFLECTION

Sufficient lubrication between the seal faces totally depends on adequate seal face deflection. It is possible, through proper analytical techniques to purposely cause the faces to deflect convergently, thereby introducing adequate lubrication. A properly designed seal will deflect under all loads as shown in Figure 1.

Total seal face deflection is the sum of stator and rotor deflections. These in turn are composed of pressure, mechanical and thermal deflections, as indicated below:

$$D = D_s + D_r \quad (1)$$

$$D_s = D_p + D_m + D_{th} \quad (2)$$

$$D_r = D_p + D_m + D_{th} \quad (3)$$

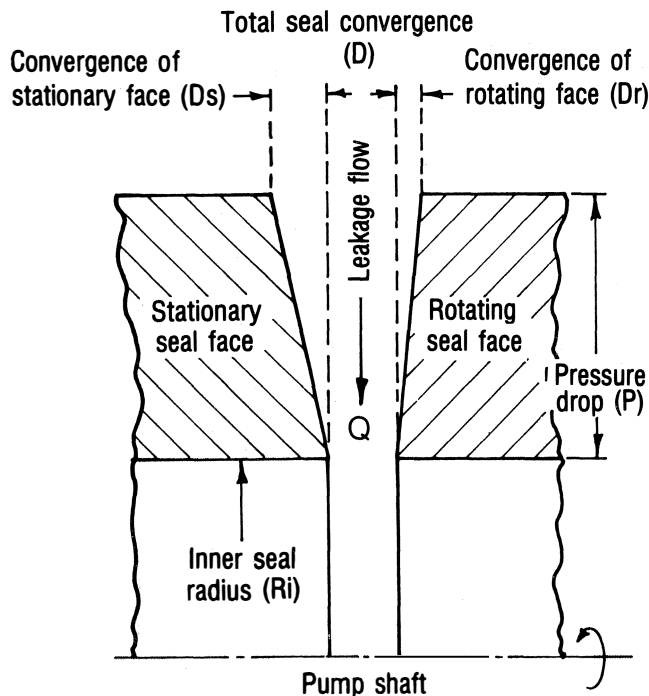


Figure 1. Seal Face Deflection.

where D is seal face deflection and the subscripts s and r represent stator and rotor deflection and p , m and th represent pressure, mechanical and thermal distortion components. Mechanically induced deflections result from spring loading.

Pressure loading

Deflection analysis under pressure is extremely important. It is essential to know how the seal face is deflecting hydrostatically. This will ultimately determine the final total face deflection and seal performance. Generally, the seal face must deflect slightly convergently or not at all. Zero deflection is important for applications where hot standby conditions are frequent, such as with boiler feed pumps. During hot standby, the seal cavity temperature may exceed 212°F . If this should happen with hydrostatically convergent seal faces, the water may flash to steam, cause excessive opening forces, and the faces will pop open. Hydrostatically flat faces will effectively seal off the water, keeping it contained in the pump. Hydrostatically divergent faces will also seal off the water, but would be detrimental for cycling performance. During cycling operation, faces that are running divergently wear along the seal face outer diameter. When the pressure decreases, the faces would start running convergently, and leakage would increase beyond acceptable limits.

Mechanical Loading

Seal faces will deflect somewhat under the influence of mechanical loading, such as spring forces. The magnitude of this type of deflection is small and is generally ignored.

Thermal Loading

As the pump rotates, there exists relative rotation between the stator and rotor. If there is no physical contact between the two faces, the only heat that is generated is due to fluid shear. Each seal face will conduct a constant proportion of heat. This heat will cause each seal face to deflect proportionally to the amount of heat conducted. As the speed of the pump varies, the amount of heat generated and hence conducted will vary.

Other Loadings

Other loadings that may be considered but do not have an appreciable influence are O-ring friction and centrifugal forces. When analyzing O-ring friction, the friction coefficient is assumed to be constant throughout the contact area. It can be assumed that friction will have a greater effect on low modulus materials such as carbon, but not appreciable on high modulus materials such as tungsten carbide.

Centrifugal forces can have an effect on ultra high speed applications, such as aircraft engines, but would not adversely affect most high speed application such as boiler feed pumps and compressors.

BOILER FEED PUMP SEALING

Power companies lose half to one billion dollars annually from unscheduled outages. Over the past decade, several technical papers have shown the main boiler feed pump to be a major cause [1]. Many pump failures were traced to bushing seals and the elaborate condensate injection system.

Bushing Seals

The two most common bushing seals used are fixed and floating bushings. These are shown in Figures 2 and 3.

Fixed bushings are the most common type of sealing device. They create a difficult leakage path parallel to the shaft. Radial running clearance between the shaft and bushing range typically between 0.010 in and 0.020 in. Although fixed bushings remain a fixed distance from the shaft, they are extremely susceptible to the effects of fluid stratification.

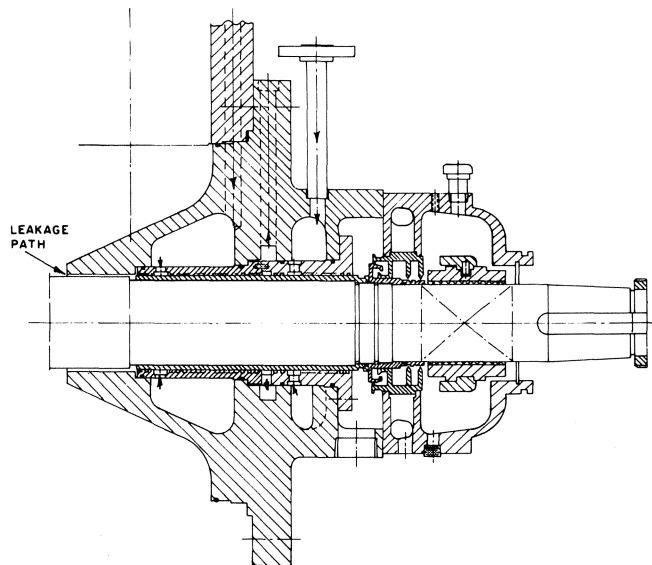


Figure 2. Boiler Feed Pump with Fixed Bushing.

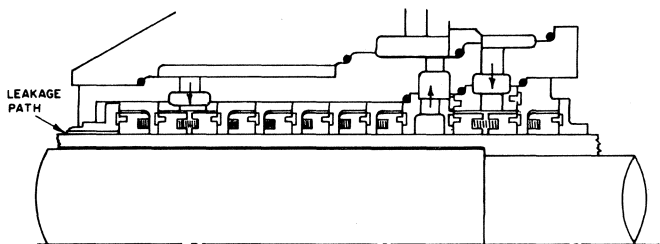


Figure 3. Boiler Feed Pump with Floating Bushing.

Floating bushings, which are a series of short, spring loaded segments, center themselves along the shaft during normal operation. Self centering is accomplished only during shaft rotation when a film support is generated. During standby, the bushing segments come into physical contact with the shaft.

Fluid Stratification

Every pump in every power plant today must operate with speeds below its best efficiency point. This occurs daily, as plants increase and decrease electrical supply to meet demand. Pumps that were not designed to undergo this type of swing operation are constantly failing. One primary mode of failure results from fluid stratification.

Stratification, as shown in Figure 4, is the separation of hot and cold fluids in the pump. This separation sets up a temperature differential between the top and bottom of the pump casing and shaft, forcing the casing and shaft to bow, as shown in Figure 5.

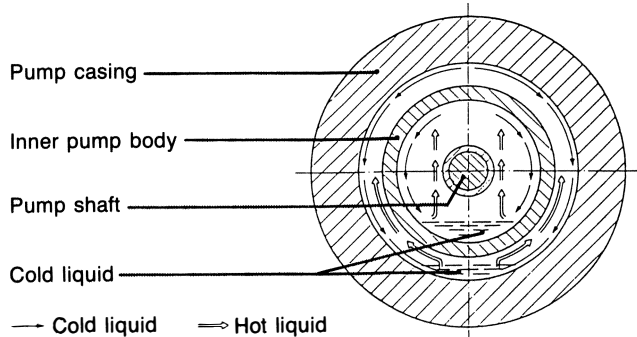


Figure 4. Fluid Stratification.

Stratification effects are most severe during hot standby when the hot water is sitting stagnant in the pump. Research done by Simon, Vach, and Pace [2] showed that separation occurs during both normal operation and standby. The pump can experience a temperature differential of 144°F within two hours of shut-down. Dynamically, the pump can experience a differential of 37°F.

Fixed bushings are extremely susceptible to stratification. The bushing area is the longest, closest clearance area in the pump. Bowing during standby will cause the bushing to contact the shaft. Contact will inevitably result in galling, fusing of the bushing and shaft together leading to outage. Floating bushings, which automatically center themselves as soon as the shaft starts rotating, are not as susceptible to fluid stratification effects.

They are, however extremely susceptible to loss of condensate injection.

Condensate Injection

There are two types of injection systems used on boiler feed pumps to ensure the hot water does not leak to atmosphere. These are pressure controlled and temperature controlled systems.

Pressure Controlled System

Cold condensate is injected into the bushing area at a pressure approximately 20 psi higher than suction pressure. Most of the injection drains to atmosphere with a small percentage flowing into the pump. This not only reduces thermal efficiency but enhances fluid stratification during hot standby.

Another major problem is feedback resonance experienced during pressure transients. This directly affects the stability of the pressure control valve causing fluctuating injection flow rates. As a result, flashing to atmosphere may result. Flashing will not affect fixed bushings, but floating bushings will lose all bearing support and contact the shaft, resulting in bushing failure and forced outage. Secondary effects are water contamination of the bearing lube oil system, leading to bearing failure if the water is not centrifuged.

Temperature Controlled System

Condensate injection is controlled by two temperature sensors located on the atmospheric drain line. The signal is read by a controller and the flow of injection is maintained so that the leakage does not exceed 150°F.

The major disadvantage is failure of the control system which will allow the boiler feed water to escape to atmosphere, flash and contaminate the bearing lube oil system. Other disadvantages are a higher initial cost than the pressure controlled system and during hot standby cold water will enter the pump.

BOILER FEED PUMP INSTALLATIONS

There are three installations in the U.S. where full fluid film lubricated seals are used to successfully seal high energy boiler feed pumps. The sites and conditions are as follows:

| SITE | Pressure psig | Temp F | RPM | Pressure- Velocity psi-ft/min | Startup |
|-------|------------------|-----------|------|-------------------------------------|---------|
| LCRA | 425 | 420 | 5900 | 916000 | 6/85 |
| PSE&G | 265 | 375 | 8310 | 568000 | 7/87 |
| HL&P | 385 | 375 | 5700 | 614000 | 5/88 |

LCRA — Lower Colorado River Authority, Marble Falls, TX
PSE&G—Public Service Electric & Gas, Linden, NJ
HL&P —Houston Lighting & Power, Houston, TX

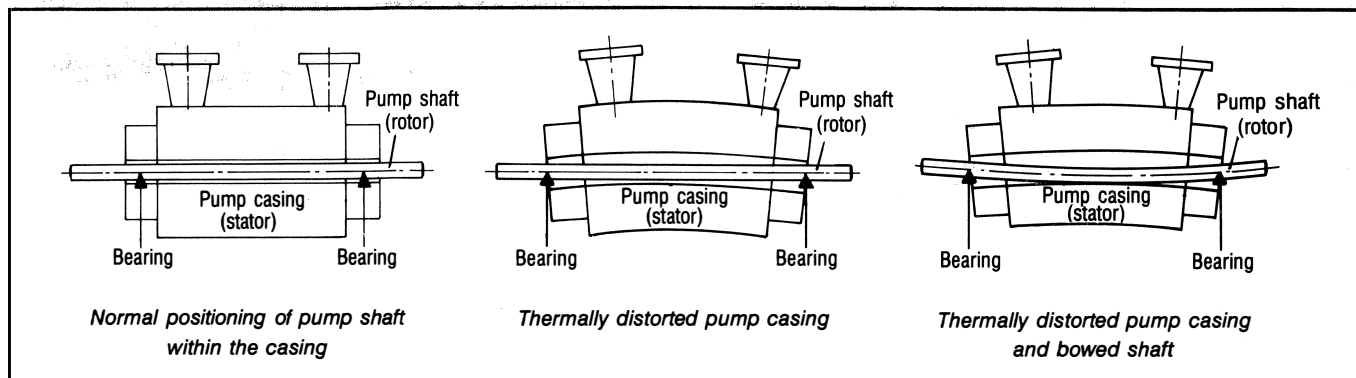


Figure 5. Pump Shaft and Casing Distortion.

Although all seals were designed with finite element analysis to control face deflection, much effort also went into designing the seal cooling system. It is important that the seals operate in water below the boiling point to ensure stable performance. Full fluid film seals can operate at temperatures 50 °F above the boiling point, but only for a short time before permanent damage occurs. A description of seal cooling systems is presented in the next section.

DESCRIPTION OF A HIGH TECHNOLOGY FULL FLUID FILM FACE SEAL

A properly designed full fluid film face seal, as shown in Figure 6, incorporates the latest design concepts, not only in analysis, but in face material combinations and seal cooling systems. An acceptable full fluid film seal package as described by Adams and Lytwyn [3] must meet the following criteria:

- *The seal must be a cartridge.* The seal, since it contains many small parts, must be preassembled before installation into the pump. Cartridge installation also aids pump alignment as the shaft may be moved axially without concern that the seal will come apart.

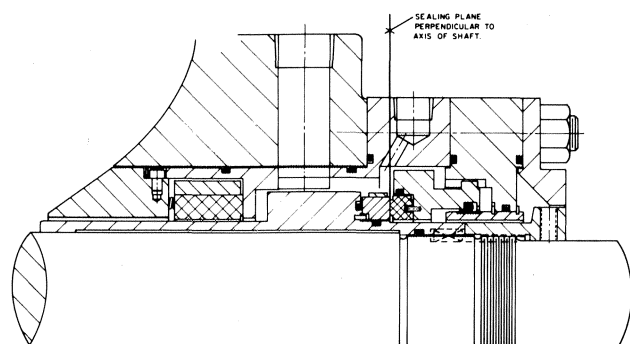


Figure 6. High Technology Fluid Film Face Seals.

- *Finite element designed seal rings.* Finite element analysis is performed for each special application. This is a necessary procedure to ensure the seal is designed with the proper face dimensions, hence proper balancing and face deflections. The stator must be balanced and flexibly mounted. The seal face and balance dimensions must be on the stator for optimum performance. This design is better able to compensate for eccentricity, out-of-perpendicularity, and shaft runout than balanced rotors. Flexibly mounting the seal ring will isolate it from deleterious effects caused by warpage of the stator. The stationary face is normally a low modulus material such as carbon and the rotor is a carbide, such as silicon carbide. High performance seals are normally constructed with two material composite stators for better control of distortion. Full fluid film seals for lesser duty pumps, compressors and high pressure pumps can be constructed with homogeneous materials, such as solid carbon or solid tungsten carbide.

- *Low friction surfaces.* Wherever there are any sliding O-rings, it is necessary to coat the undersurface with a hard finish to ensure a low coefficient of friction and thusly a freely sliding O-ring.

- *Auxiliaries.* The seal is not complete without a properly designed cooling system. It is essential to maintain low operating temperature so that the seal will be stable in operation. The de-

signer must accurately predict heat loads and then choose a properly sized heat exchanger or external cooling flow. Heat loads are a combination of heat soak through the pump cover, and shaft and frictional heat due to the rotation of the seal components. Frictional heat is due to seal face generated heat and turbulence from the rotating seal hardware. When a heat exchanger is used, the seal must be equipped with an integral pumping feature. It must be able to pump a sufficient amount of liquid at all operating speeds to ensure a cool running seal. During hot standby, it must not interfere with thermal convection flow through the heat exchanger.

- *Chemically compatible metal parts.* Full fluid film seals can be applied to many liquids, from water to oil. Each application must be reviewed for possible corrosion and the proper materials used. Normally, metal parts are stainless steel. In corrosive environments, it may be necessary to use Monel or Hastelloy [4].

FULL FLUID FILM ADVANTAGES

The full fluid film seal has the following advantages over bushing seals:

- *Large diametrical clearances.* Mechanical seals are able to compensate for any shaft movement and bowing that bushings are not designed to.

- *Low leakage to drain.* While bushings will leak anywhere from 1.0 to 25.0 gallons per minute to drain, the mechanical seal will leak less than 0.1 gallons per minute. This results in a much reduced chance of bearing lube oil contamination.

- *Energy savings.* Energy wastage is reduced to practically nothing because of the reduced leakage to drain and elimination of condensate injection during normal operation and hot standby.

- *Reduced fluid stratification.* A properly designed seal and sealing system will eliminate condensate injection and hence the main cause of stratification and pump/shaft bowing.

- *Greater pump availability.* The greatest benefit of the mechanical seal is the ability to compensate for pump related distortion during cyclic operation. This is achieved by careful control of seal face deflection and wider diametrical clearances than bushing seals.

CONCLUSION

It is possible, as present field installations prove, to design a non-contacting seal for excessively high pressure-velocity relationships where conventional seals cannot operate. By exploring new sealing concepts difficult applications can be effectively and successfully conquered.

REFERENCES

1. Survey of Feed Pump Outages, Electrical Power Research Institute. Palo Alto, California, EPRI FP-754 (April 1978).
2. Simon, A., Vach, R., and Pace, S., "Cycling Effects on Boiler Feed Pumps," presented at Fossil Plant Cycling Workshop, EPRI, Miami Beach, Florida (1985).
3. Adams, W. V., and Lytwyn, P., "Retrofit of an Unspared Main Boiler Feed Pump to End Face Mechanical Seals," ASME 86-JPGC-pwr-52 (1986).
4. Monel is a registered trademark of Huntington Alloys, Incorporated. Hastelloy is a registered trademark of Cabot Corporation.