Main-Coolant-Pump Shaft-Seal Guidelines

Volume 3: Specification Guidelines

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Main-Coolant-Pump Shaft-Seal Guidelines

Volume 3: Specification Guidelines

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EPRI PERSPECTIVE

PROJECT DESCRIPTION

This project (RP1556-1) was undertaken as a logical extension of earlier EPRI work to investigate the causes of failure and the state of the art in the design of nuclear main coolant pumps (MCPs). Both the failure history study reported in EPRI Final Report NP-1194 and the design study reported in EPRI Final Report NP-2458 concluded that problems with the mechanical face seal were major contributors to pump unavailability and to plant unavailability.

A project survey to update and augment this earlier work (reported in EPRI Interim Report NP-2611, Volumes 1 and 2) substantiated that a wide spectrum of reliability has been experienced in operating and maintaining "identical" shaft seal systems. The field survey responses were grouped into three general failure-cause categories: system-induced, maintenance-induced, and design-related. For each category, fault trees were constructed to describe how seven or eight events typically lead to the observed failure modes. This data analysis did not reveal a predominant event-failure mode relationship but rather pointed out that corrective actions in each of the three categories are necessary to improve seal and seal auxiliary-system reliability. These findings provided the bases for completing a comprehensive analysis of seal reliability and for developing guidelines with specific recommendations that would lead to improved MCP availability.

PROJECT OBJECTIVE

The overall goal was to develop a composite set of technical guidelines that can be used interactively by the utility, the nuclear steam systems supplier, the architect-engineer, and the pump manufacturer to increase the reliability of both the seal and seal auxiliary systems while at the same time to improve pump performance.

PROJECT RESULTS

This document is one part of the three-volume set of guidelines that has been developed to present the composite of required corrective actions. The volume titles are:

- Volume 1: Maintenance Manual Guidelines
- Volume 2: Operational Guidelines
- Volume 3: Specifications Guidelines

Woven through the specific details of each of these recommendations, a common problem-cause thread is apparent: the lack of an effective communication-response cycle between the pump seal supplier, the system designer, and the operational user. The data indicate that each of these parties has a contribution to add to the total corrective action. History indicates that successful mitigation of seal failure will only come about if these contributions are responded to in a spirit of mutual cooperation.

These guidelines are of interest to pump seal suppliers, system designers, and utility operations and maintenance staffs.

Floyd E. Gelhaus, Project Manager
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ABSTRACT

This report presents a set of guidelines and criteria to aid in the generation of procurement specifications for Main Coolant Pump Shaft Seals. The noted guidelines are developed from EPRI sponsored nuclear power plant seal operating experience studies, a review of pump and shaft seal literature and discussions with pump and seal designers. This report is preliminary in nature and could be expanded and finalized subsequent to completion of further design, test and evaluation efforts.

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SUMMARY

An investigation into main coolant pump (MCP) shaft seal failures in U.S. commercial nuclear power generating stations has been completed. The purpose of this project was to define the means to reduce high-cost, lost-power outages caused by MCP shaft seal failures. The initial effort consisted of a survey of U.S. commercial nuclear plants and led to the grouping of the observed failure modes into system/operational-related, maintenance-related, or design-related categories. A report (EPRI Interim Report NP-2611, Volumes 1 and 2, Main Coolant Pump Shaft Seal Reliability Investigation), containing the results of this survey was published in September 1982. The survey sample was representatively large (27% of total U.S. commercial plant population) and included the three industry seal suppliers (Bingham-Willamette, Byron Jackson, and Westinghouse Electric Corporation). Operationally incurred and/or induced problems and seal redesign parameters were identified. Failure hypotheses in the form of fault trees were developed to describe the failure mechanisms, and recommendations were made for seal reliability improvement.

The results of the survey reaffirm that the primary coolant pump shaft seals are complex and sophisticated devices. As a critical pressure-boundary component in the primary heat transport loop, the seal system is often taxed beyond design limits and forced into a failure mode. Experience shows that the seals have often been subjected to stress conditions exceeding their design capability because of improper operator procedures. In other instances, the overstresses were caused by seal auxiliary-system malfunctions or inadequacies. Problems during maintenance have been aggravated by a lack of appreciation of the component's sophistication and delicacy, and the findings show the severity and frequency of the "built-in" failures resulting from improper maintenance. Included, and synergistically interwoven amongst these field-induced problems, are the failures due to design shortcomings. These problems relate to the inherent parameters that require either a redesign for greater operating margins or alternate design mechanizations to improve the reliability of the shaft seal assembly.

From these results, user-oriented Maintenance Manual, Operational, and Specification Guidelines were generated. Each of the three volumes is written as a stand-alone

document. However, the solution to the seal failure problem will only come from the successful enactment of the recommendations in all three guidelines. These volumes are:

1. Volume 1: Maintenance Manual Guidelines. This volume represents a set of guidelines and a listing of information and data that should be included in maintenance manuals and procedures for MCP shaft seals. The maintenance-oriented results from the project's operating experience study are summarized. The shaft seal and its auxiliary supporting systems are discussed from both technical and maintenance-related viewpoints.
2. Volume 2: Operational Guidelines. This volume presents a set of guidelines and criteria for improving MCP shaft seal operational reliability. The data relating to usage procedures and practices and operational environmental influence on seal life and reliability from the project survey are summarized. The shaft seal and its auxiliary supporting systems are discussed from both technical and operational-related viewpoints.
3. Volume 3: Specification Guidelines. This volume presents a set of guidelines and criteria to aid in the generation of procurement specifications for MCP shaft seals. These guidelines were developed from EPRI-sponsored nuclear power plant seal operating experience studies, from a review of pump and shaft seal literature, and from discussions with pump and seal designers.

The recommendations in these three volumes of seal guidelines, if diligently applied, should enhance shaft seal procurement, operation, and maintenance, thus increasing equipment and plant availability.

1.0 INTRODUCTION

Numerous engineering considerations must be factored into the design of main coolant pump (MCP) shaft seals for nuclear power applications. These include:

- 1) critical seal design parameters;
- 2) primary and auxiliary coolant system interfaces;
- 3) motor and shaft coupling interface;
- 4) operating procedures and conditions; and
- 5) the maintenance environment.

Shaft seal design must be approached from pump and system standpoints. Correspondingly, specifications for shaft seals and pumps, must ensure that the seal and seal auxiliary system procurements are approached using equivalent system engineering disciplines. The utilities must identify, specify and control the critical factors which are required to attain and maintain the operational integrity of these shaft seals.

Reported experience to date with nuclear primary coolant pump seals has not been entirely satisfactory. Nor has seal reliability been consistent within the population of commercial nuclear power plants. Satisfactory seal performance involves functional and physical interfaces of equipment and designs provided by the pump vendor, the nuclear steam system supplier (NSSS), architect-engineer (A-E) and the utility. The pump vendor designs and supplies the pumps and seals. The NSSS designs the seal auxiliary systems and provides some of the component parts. The A-E provides the piping, component and instrumentation installation designs, and supplies the balance of the equipment. The utility operates and maintains the equipment. Thus, the pump/seal supplier, NSSS and A-E share in the interface responsibility but the overall system assurance effort must be borne by the utility. This report discusses factors which should aid utilities in the preparation of shaft seal procurement documents and specifications.

2.0 DESCRIPTION OF THE VERTICAL CENTRIFUGAL PUMP-SEAL AND THE MOTOR INTERFACE

Electric-motor-driven, single-stage, vertical centrifugal pumps provide the primary coolant circulation in U.S. commercial nuclear

power generating plants. The pump and motor shafts are rigidly coupled and are rotationally guided by two oil-lubricated motor bearings and one pump water bearing (see Figure 1). Axial thrust loads are supported by a thrust bearing in the motor assembly. The main coolant pump sealing at the pump drive shaft is accomplished by a seal assembly, sometimes referred to as a cartridge.

Establishing and maintaining the correct mechanical arrangement between the vertically mounted electric motor shaft and the pump shaft is critical to a successfully operating shaft seal. Because the coupling between the two vertical shafts is angularly rigid and axially solid, an improperly aligned pump-motor shaft can cause angular and lateral misalignment that may exceed the acceptable design limits set by the seal designer. For instance, the hydrodynamic type face seals are designed with opposing hydraulic areas that, along with the seal springs, create external seal forces that support a very thin fluid film, see Figure 2. Typically, these thin film separations are 20-40 microinches. Shaft induced forces operate on the seal tending to upset the force balance which controls the separation clearance. Static and dynamic hydraulic forces at the impeller may be large enough to cause sizeable affects on seal dynamics.

Prior to start up at low reactor pressure operation, the weight of the pump rotating element causes a downthrust. The effect of this downward force is relieved in some applications when pump rotation is started by engaging an oil lift system coupled to the motor thrust bearings. This raises the rotating mass typically .010 inches and thus decreases the drive motor starting torque load. Once the pump reaches rated speed conditiions, the oil lift-system (when applicable) is de-energized and the pump shaft seeks its equilibrium axial position. Total pump shaft axial motion, at full system pressure and operating temperature, of approximately 100 mils has been observed in operating pumps. Such motion results from motor bearing clearances, housing and bearing support deflections and thermal growth. Proper seal designs can accomodate typically 1/8 inch axial shaft motions without exerting excessive seal compressive loads. The 1/8 inch axial freedom can be lost however by improper installation of the seal assembly into the pump. This must be guarded against by the installers. It is the seal designer's responsibility to make the proper installation adjustments

FIGURE 1 SCHEMATIC PRESENTATION OF TYPICAL MAIN COOLANT PUMP

a. SEAL STAGE GEOMETRICAL ARRANGEMENT

b. CLOSING FORCE (F_c)

P_s = SEAL PRESSURE (PSI)

P_o = BLEEDOFF PRESSURE (PSI)

d_i = INSIDE FACE DIA. (IN)

d_b = BALANCE DIA. (IN)

c. OPENING FORCE (F_o)

F_s = SPRING FORCE (lbs)

ALTERNATE PRESSURE GRADIENTS

WHERE $0 \leq K \leq 1$

FIGURE 2 BASIC SEAL FACE FORCES

as simple and as easy to accomplish as possible.

Lateral shaft displacements of up to 20 mils peak to peak have been reported during steady state pump operation. This value may be exceeded during transients or periods not representing rated operating conditions. Obviously, initial alignment accuracy is critical towards minimizing the pump shaft run-out. Excluding this initial alignment, shaft runout is primarily a function of manufacturing tolerances, impeller/casing hydraulic characteristics, shaft stiffness, bearing stiffness, and degree of dynamic balance of the rotating assembly. Because the pump shaft is directly coupled to the motor and uses the motor bearing for guidance, shaft distances between bearing guide supports tend to be large. As a result, shaft orbital displacements may be greater than those of typical rotating machines. The situation at the seal is thus further aggravated because the seals are positioned near the point of maximum shaft deflection.

The seal designer must strive for a design tolerant of such axial and radial seal dynamic movements. The buyer should insure via seal and/or pump purchase specification/criteria that sufficient pump system analyses are performed to identify all shaft radial and axial motions for normal and off-normal operating conditions. When these analyses are performed, consideration should be given to the life cycle changes resulting from wear and maintenance. All such conditions need to be defined. Consistent with these analytical and design calculations, pump acceptance criteria should contain requirements for measurement of such shaft motions. Seal procurement documentation should define the maintenance parameters, manuals, procedures, required drawings, etc., that the buyer will need to properly maintain the seal. Reference 6 contains the guidelines/criteria for generating the required maintenance oriented documents.

3.0 HYDRODYNAMIC SEAL DESCRIPTION

This report deals only with balanced hydrodynamic seals which operate on a very thin fluid film separation between the faces of stationary and rotating sealing rings.

Hydrodynamic seals are comprised of two, three, or four tandem stages,

as shown in Figure 3. The number of stages depends on the primary coolant system pressure that must be contained. BWR installations operate at a nominal 1200-psi pressure, and two sealing stages, each subjected to a 600 psi differential pressure, are utilized. PWR primary coolant loops typically operate at about 2150 psi, and three sealing stages are used to divide differential pressure evenly across each stage to approximately 700 psi. The interstage pressure breakdown is accomplished by controlled leakage of cooled water through flow impedance paths to the seal cavity stages, which are series connected. Each such cavity contains the components that comprise a single mechanical seal stage (refer to Figure 4). It should be noted that a low-pressure fourth sealing stage (referred to as a vapor seal) is utilized in some hydrodynamic seal installations. This fourth stage operates similarly, in principle, to its upstream higher pressure stage counterparts but with a differential pressure of less than 50 psi. In the event of a failure of the preceeding stage, this sealing element is designed to contain full system pressure for a limited amount of time.

A typical hydrodynamic mechanical seal stage consists of a rotating flat face ring, driven by a seal sleeve, and a stationary mating face attached to the seal housing. The rotating face is provided freedom of axial and angular movement to allow for the formation of a thin fluid film of uniform cross section across the mating seal face area. This is accomplished by a spring-loaded flexible seal ring mechanism, as shown in Figure 4. An elastomer is used as a sliding secondary seal between the tandem differential pressure staging cavities. These secondary seals are in the form of O-rings, U-cups, and bellows. The return springs also serve to provide full seal closure during pump static conditions and aid the hydraulic seal face force balance when the pump is running.

In addition to the rotating and stationary seal rings mentioned above, there are other precision-lapped rings used to maintain squareness and rigidity of the faces/rings in running contact. These are referred to as "back-up rings."

The hydrodynamic seal, shown in Figure 4, depends upon the action of mechanical spring forces and fluid pressure on unbalanced areas of a seal ring to provide seal face closure. The fluid film separating

FIGURE 3 A TYPICAL 3 STAGE HYDRODYNAMIC SEAL ARRANGEMENT (WITH A 4TH VAPOR STAGE)

FIGURE 4 HYDRODYNAMIC SEAL STAGE TYPES

the sealing surfaces is developed by the hydrodynamic pumping action caused by the rotational velocity of one of the seal faces and the pressure gradient across the seal gap. Because of the thin-film seal face separation, hydrodynamic seal leakage is normally less than one gallon per hour (gph). These low-leakage seals are, however, sensitive to closing force levels. Therefore, balance ratios and spring load magnitudes are very important design controlled parameters. If the closing force is too high, the seal faces will contact during operation. The wear and heat generation between the rubbing surfaces varies directly with the speed of rotation and the magnitude of the closing force. Such wear and heat generation is very rapid and can become excessive leading to catastrophic seal failure. If the closing force is too small, the faces will readily separate with attendant high leakage.

The free body seal diagram noted in Figure 2 briefly describes the mechanical and hydraulic forces acting on a typically balanced hydrodynamic seal stage.

The mechanical balance ratio is defined as:

$$MB = \frac{d_o^2 - d_b^2}{d_o^2 - d_1^2} \times 100\%.$$

The closing force acting on the flexible seal member is:

$$F_c = F_s + P_s \pi/4 [d_o^2 - d_b^2] + P_o \pi/4 [d_b^2 - d_1^2].$$

The opposing or opening force acting on the same seal member is:

$$F_o = \pi/4 [d_o^2 - d_1^2] [P_o + K[P_s - P_o]].$$

The value of K (pressure gradient) is dependent upon whether the mating sealing faces are

- a. divergent in direction of fluid flow;
- b. parallel; or
- c. convergent in direction of fluid flow.

Another important design consideration is the seal face unit loading (P_L).

$$P_L = \frac{[F_C - F_O]}{\pi/4[d_o^2 - d_i^2]}$$

The hydrodynamic seals require:

- 1) accurate force balance on the sealing rings. This is to insure that closing forces are not excessive and that the working seal face be of sufficient area to support the unbalanced load resulting from seal fluid pressure and the mechanical loading springs.
- 2) that the design support the development of a thin fluid film (typically about 40 microinches) between the seal faces.

The safe operating domain which separates the high leakage and the high wear rate zones is characterized by this narrow film separation profile. The design parameters critical to establishing and maintaining such a film profile are:

- a) seal ring geometry;
- b) flatness and smoothness of mating faces;
- c) manufacturing tolerances of supporting rings and housing;
- d) spring loading characteristics;
- e) sleeve tolerances, finish, and concentricity;
- f) secondary seal radial stiffness (for centering);
- g) secondary seal axial and angular compliance for accomodating axial and angular shaft motion to allow for accurate tracking of the stationary seal face by the rotating seal ring;
- h) structural deformations caused by mechanical, hydraulic and thermally induced stresses; and
- i) a positive pressure gradient.

3.1 Carbon Seal Ring

A molded carbon composite material is utilized for one of the mating seal rings because of its low coefficient of friction and chemical, thermal and shape retaining stability. If kept within allowable surface loading, these carbon face seals will exhibit almost no measurable wear over extended operating periods.

3.1.1 Physical Design (Form and Size)

Although the carbon composite seal materials are exceptionally stable over a wide thermal range, the high operating pressures acting on the seal ring are high enough to cause structural deformations of magnitudes comparable to the developed fluid film clearances between the seal faces. Seal element shapes and cross-sectional areas must be such that these pressure loads can be sustained without causing excessive deformations which could destroy the lubricating film. The net closing force is dependent on the balance ratio and differential pressure across the seal stage and spring forces (see Section 3.0 above) whereas the compressive loading on the seal ring results from the absolute pressure on the exposed seal surfaces. The severest compressive loading occurs on the components in the high pressure stage.

High face velocities coupled with small running clearances and sizeable face loads may produce high heat concentrations in the sealing gap. This in turn can cause high temperature gradients in the sealing ring producing deformations destructive to the lubricating film profile. The superimposed affect of thermal, mechanical, and hydraulically induced stresses must be considered by the seal designer.

Another hydrodynamic seal parameter that should be considered during seal design is the seal face pressure-velocity (PV) value. Various reports, such as Reference 4, present a relationship between PV and seal life. That report indicates an upper limit PV value of 600,000 PSI-FPM for the present state-of-the-art.

PV values generated from research results and from successfully performing seal configurations can become a useful design parameter

for size scaling (up or down) mechanical face type shaft seals. The same type of parametric sizing is successfully done on hydrodynamic journals and thrust type bearings.

The PV number contains the necessary dimensional parameters for seal geometry factoring, i.e., shaft speed, face dimensions, face areas, balancing areas, film/pressure gradient profile, and center of face pressure radius.

The equation for PV is

$$PV = P_L [2\pi N] R_{cp}$$

where

P_L = Seal Face Unit Loading (PSI),

N = Shaft Speed (RPM), and

R_{cp} = Seal Face Center of Pressure Radius (FT)

As of this report, there has not been sufficient research work and categorization of PV value. Purchasers and users of hydrodynamic face type seals should be aware of this useful parameter and encourage its usage by seal designers and suppliers.

The following list of PV values is for two seal designs used in another current reliability survey, Reference 1.

<u>Stage</u>	<u>Design #1</u>	<u>Design #2</u>
1	395,500*	626,000
2	390,500	642,000
3	389,000	675,500

*PSI-FPM

No conclusion is made from the above tabulated data, but the information does represent a range of values for seals that have proven acceptable for nuclear MCP service.

Over the past several years, the literature reflecting the state of

the art on shaft seal design, suggests that the optimum separation profile between the seal faces be converging in the direction of the leakage flow. Such converging separation gaps have been shown to be stable in supporting the seal closing loads. Diverging gaps collapse under these closing loads and allow contact between the stationary and rotating seal faces. The result is increased heat concentration and higher thermal gradients in the seal rings which cause ever larger destructive stresses during this high wear rate mode. Divergent seal gaps must be avoided for all seal operating conditions. Figure 5 shows converging and diverging seal face separation profiles.

Current state of the art seal design includes computer aided finite element analysis to solve the thermal, hydraulic and mechanical stress-deformation problems associated with the specific shape, size, and material properties of the sealing ring. Seal procurement criteria should require sufficient analytic results to assure that the separation gap will tend towards convergence in the direction of leakage flow for all operating conditions. The example shown in Figure 6 is ideal since all external forces acting on the seal ring are assumed to produce a deforming moment that insures a converging gap in the direction of leakage flow. The same considerations must be given to the opposing seal face. Normally the carbon face is more susceptible to deformation because of its physical properties.

3.1.2 Material Composition

The carbon seal ring is made from graphite by furnace-curing the pressed powder with a binder. A homogeneous distribution of the carbon is essential to insure uniform strength of this seal material, since material density variations will compound problems associated with hydraulic, mechanical and thermal deformations previously noted. Furthermore, if material discontinuities exist (such as voids), they may give rise to stress concentrations which prevent a uniform distribution of deformation stresses.

The user should request that the seal manufacturer place sufficient emphasis on physical properties such as:

PUMP
 $\mathcal{E}_L$

FIGURE 5 THERMAL AND HYDRAULIC ELASTIC RING DISTORTION

FIGURE 6 TEMPERATURE AND PRESSURE EFFECTS ON FACE DEFORMATION

hardness;
compressive and transverse breaking stress;
modulus of elasticity;
density; and
porosity,

to insure carbon seal ring integrity and consistency.

Each batch produced should be sample tested and sample size and test methods shall be specified. The sample test data should be retained. If variations between these data and those used for computer aided design analyses are significant, an iteration of the seal deformation numerical solutions should be made using the data from the sample tests.

3.1.3 Carbon Ring Drive, Retaining and Protection

Industry sources and internal Byron Jackson field service reports indicate that fractured seal faces are commonly found in seals being refurbished. Responses from 55% of commercial nuclear power plants participating in the recent mechanical shaft experience survey (Reference 1) emphasize the occurrence of fractured-broken carbon face seals. It was noted that numerous severely damaged seal ring fractures typically emanate from the inside corners of axial grooves. The housing retention forces acting on the seal should be distributed as uniformly as possible over a large area to reduce unit loading. Good design will avoid sharp corners. However, damaging stress concentration may form even at filleted retaining corners such as those found peripherally around some carbon rings. Single point drive or retention design mechanizations should be avoided. Good design will therefore distribute driving and retaining loads uniformly over large areas. Providing multiple drive or retention locations is difficult because of the tight manufacturing and assembly tolerances required. These problems can be circumvented with use of more resilient drive pins or keys.

The carbon seal material is very brittle, and fractures without the formation of burrs. As a result, if the carbon ring can be physically contained by a shroud-type holding ring, a fractured seal may continue

to operate. Leakage through the seal in such instances will increase but such a fracture may not result in the total loss of sealing capability. Seals running without excessively high leakage rates have been disassembled for refurbishing and found to contain fractured carbon rings. The retaining shroud serves another purpose under these conditions. It retards the migration of carbon chips and pieces through the seal where they may plug staging flow passages, or damage elastomers or downstream mechanical seal faces. Although the shroud retainer does not prevent the initial seal failure, it may help extend operating time and allow a more orderly-planned pump shutdown. Procurement seal documents should request designs which inhibit the migration of carbon pieces if the carbon chips or fractures.

Because of the relatively low tensile strength of the carbon, design provisions must insure that the carbon seal ring is never subjected to tensile loads. The seal design must guard against reverse loads from being applied when secondary seals seep or fail. Improper sequencing and operation of vent valves during pump start-up may place the carbon ring under tension. There is evidence to suggest that such instances have occurred. Such procedural failure inducing modes are of concern for design criteria because appropriate design allowances can eliminate this failure inducing mode.

In many cases, venting procedures call for the partial opening of valves during the venting process. The only way to properly set the valve is by knowing the flow conditions through the valve. Instrumentation readouts (if any), however, are generally located in the control room away from the technician performing the function. As a result, human judgement is often substituted. A simple design fix is available to ease this situation through the incorporation of a rate limiting orifice in the venting port of hardware in current usage.

For new designs, appropriate design geometry for full self venting of the seal cavity and CBO/leakoff flow paths can be assured, greatly reducing the possibility of human error in the venting process. Vent holes connecting to the staging pressure reducing device should be at the high point of the seal cavity, at least 1 to 2 inches above the running seal surfaces.

Since dimensional stability can be a function of shelf life, the flatness of a stored carbon ring should be checked prior to utilizing it during rebuilding. Because of the low Young's Modulus and the exacting original manufacturing flatness requirements associated with the seal faces, the carbon ring may require touch-up relapping. This item should be included in both the design and maintenance documentation.

3.2 Hard Face Seal Ring and Ring Holder

The mating seal face material for the carbon seal is tungsten carbide or titanium carbide. The prime design requisites for these components are:

1. The carbide ring should have a low coefficient of sliding friction with carbon. To minimize the friction coefficient highly polished surface finishes are provided. Corrosion resistance of the face material is necessary to preserve the fine finish of the hard face;
2. The carbide ring holder geometric shape shall be such as to facilitate hand and machine lapping. The maintenance of the seal will depend on the ability to lap and mate these precise surfaces. The buyer should require that the seal supplier's design clearly provides the ease for performing such required lapping,
3. The carbide ring holder geometry and material selections, typically manufactured from 304 and 410 alloys, shall minimize stress concentrations; and
4. Frictional twisting forces should be minimized by design. The thermal growth of the carbide and its retainer differ because of their differing thermal coefficients of expansion causing these contacting faces to slide. Such sliding motion may also be produced by part deformation caused by hydraulic pressure loads on the ring holder. The pressure and mechanical forces pressing these parts together may cause high frictional forces at the mating surfaces which can twist the carbide ring and upset the seal separation

clearance. Therefore, the design should minimize the thermal growth differences between these two mating parts and design for minimum frictional forces at the mating interface.

Thermal, pressure and mechanical load deformation of the holder can be controlled by geometric form and size. Computer aided numerical solutions yielding carbide retainer deformations should be made to confirm the shape stability of this part.

3.3 Secondary Elastomer Seals

As noted earlier, elastomeric secondary seals form a critical sealing boundary at the shaft sleeve. This seal absorbs the shaft axial motion inherent within the pump and is therefore in sliding wear contact between the sleeve and the floating rotary seal element(s). These devices ("O"rings or "U" cups) are comprised of synthetic rubber compounds and are susceptible to damage primarily by extrusion between the annular gap being sealed. This is caused by excessive annular clearances, too high a temperature, abrasive wear and/or excessive flexure. At least 50% of the Reference 1 survey participants indicated a high incidence of visible elastomer damage when seals were being rebuilt. Byron Jackson field service reports substantiate the relative magnitude of these occurrences.

The general temperature effects on rubber materials are well established. Initially, rubber softens at elevated temperature at which condition it is most susceptible to gap extrusion. Sustained high temperature exposure causes the rubber to lose its resiliency and harden. A measure of the sealing ability of rubber seals is the compression set of the seal.

Compression set is the permanent distortion of an elastomer after compression at a specific temperature for a measured period of time. The importance of the compression set property of the elastomer is that the seal may fail to seal after taking a 100% compression set. Thermal changes, motion, or pressure changes may cause a break in the seal contact, creating a leak. Once this contact is broken, at 100% set, sealing can be restored only by replacement of the seal.

Figure 7 from Reference 2 illustrates the life vs. temperature characteristics of a nitrile (Buna N) compound which may be used in primary coolant pump shaft seals. Notice that in the area of interest for a seal (normally less than 150°F), each decrease of approximately 10°F will increase elastomer seal life (by the compression set definition) by one year. Temperature soak environment, which is totally dependent on appropriate and consistent seal auxiliary system interfaces and performance, is very critical to the reliability and longevity of these elastomers.

Another failure mode encountered in shaft seals is extrusion of the seal into the annular gap being sealed. The designer should limit clearance at gaskets to .010 inches or use retaining back-up rings to prevent extrusion. Seal purchaser specifications and criteria documents should require a dimensional tolerance stack-up check on the deliverable seal or seal parts to assure that these clearances are not exceeded. Maintenance procedures should identify this measurement as critical by using a cautionary flag.

Since the secondary seal is also used to center the seal ring and to provide the axial and angular tilt freedom, several other parameters associated with these functions are important. The elastomer should be of sufficient cross section for centering support (a requirement for rigidity), yet provide the low friction forces and elasticity for accommodating axial and angular motion (a requirement for compliance). These two opposing requirements must be properly balanced. One such technique is the "U"cup geometry which utilizes a large rubber cross section for radial centering while having the axial and angular flexibility required for tracking motion freedom.

The incorrect sequencing and operation of vent valves which may subject the carbon rings to a tension load may cause secondary seal damage as well. Improper vent valve operation can reverse the pressure on secondary sealing elements such as "U" cups, thus relieving the preload on the "U" cup follower, unseating the "U" cup and/or creating other problems such as drive lug loosening/dislocation. Although such problems have occurred, all were humanly induced and can be procedurally controlled to prevent occurrence. However, a self venting design as previously discussed would remove the human element and provide the

FIGURE 7 ESTIMATION OF SEAL USEFUL LIFE AT 80%, 90% and 100% COMPRESSION SET, BUNA N (FROM REFERENCE 2)

better deterrent solution for such failure modes.

Another secondary seal failure mode observed in the operational environment results from the dirt collecting tendency around concave shaped adjusting collars at the top of the shaft seal assembly. Such concave devices tend to trap and direct foreign materials to the low pressure elastomer where along with the rotary action of the shaft the debris may be ground into the rubber causing damage. Flat or convex adjusting collar configurations can greatly decrease failures from this type of failure mode. Purchase specification should require that the seal design should include guards against natural pockets and traps for dirt and prevent debris contact with critical sealing elements.

3.4 Pressure Breakdown Device (Staging Pressure)

As an integral part of the cooling water circulation path through the seal, the pressure breakdown device provides equal differential staging pressure between adjacent seal cavities. When this flow is restricted or totally blocked, all but the last sealing stage run without seal face film lubrication, and all run without the aid of interstage cooling flow. Without the aid of the lubricating water film, the carbon seal face friction coefficient may increase by a factor of 10. The additional frictional heat load, coupled with the decrease or absence of cooling water supply, is catastrophic to the seal.

Figure 1 shows the typical series arrangement of the pressure breakdown devices and the seal cavities. This type of arrangement provides equal staging flow to each seal cavity in a simple manner, but with a potential for seal failure in the remaining stages if one breakdown device becomes inoperative.

The pressure breakdown flow path is susceptible to blockage. The Reference 1 and 3 seal reliability surveys have noted the existence of corrosion products (crud) in seals disassembled for repair and refurbishment. Such internally generated contaminants, along with foreign material introduced into the primary coolant system during maintenance, have been reported to cause flow blockages within the pressure breakdown coils. The numerous reports citing the presence of foreign material in damaged seals is indicative of the importance of internal cooling

water cleanliness for seal reliability. Some literature sources note that seals operating in several injectionless systems are of comparable reliability to those which are injection fed. Such observations have led to conclusions which question the value of an independent seal water injection supply. Primary water cleanliness varies from one system to another and, within a single system, may vary at different system locations. Thus, such water quality can vary from pump to pump and/or from time to time at the same pump. For these reasons, it is understandable that a seal will yield satisfactory life performance characteristics in some injectionless installations and have a high mortality incidence in another. The goal is a constant supply of clean cooling water, and this goal is best reached through the use of an independent injection supply.

Additionally, the cooling water flow path through the seal should be designed for optimum distribution of temperatures. Such optimum condition is defined as one that results in the lowest thermal gradient for steady state and transient operating conditions. The buyer should require the seal supplier to perform sufficient thermal-hydraulic analyses to assure that the cooling flow heat transfer design has been optimized.

It may be prudent for the seal/pump designer to consider a parallel pressure breakdown arrangement where each seal cavity is independently provided a staging flow from a common pressurized manifold and then with each seal cavity (stage) return into a common low pressure outlet header. The parallel flow arrangement may be slightly more complex but can provide more uniform temperatures throughout the seal cartridge and is potentially less sensitive to contaminants. Additionally, a single point failure in the staging flow path would not necessarily affect every seal stage as severely as it does in the series flow design.

3.5 Instrumentation Provisions

Proper shaft seal performance is dependent on the ability to measure and control critical seal operating parameters. To enhance the reliability of seal operations, it is important to track the time variant behavior of: a) leakage flow rate, b) seal cavity operating pressures,

c) temperatures, and d) controlled bleedoff rate. Design provisions for redundant measurement of these parameters in each sealing stage are recommended.

Accurate, dependable flow measurement capability is the most difficult to implement. This is particularly true with the low leakage flow associated with the mechanical seal faces. Development of unique measuring techniques should be highly encouraged.

3.6 Dynamic Balance

Because of the possible benefit to seal reliability and life, maintenance procedures should include seal assembly dynamic balancing prior to installation. Correspondingly, the specification for new seals should require such balancing. In this regard, design studies should be conducted to define the benefits of keeping the spring-mass assembly stationary, since this is the most massive rotating seal part in many current seals.

A stationary spring assembly may offer additional benefits, such as allowing for lighter spring-loading forces because of the absence of inertia effects.

3.7 Maintainability

Numerous maintenance induced failures have been identified and documented. Figure 8, from the Reference 1 seal experience survey, shows the high incidence of faulty seal assembly and installation problems that users experienced in maintaining the primary coolant pump shaft seals.

Note that out of the twenty stations surveyed:

- (1) Fifteen reported a high correlation of seal failures with defective/improper parts usage in user seal refurbishment;
- (2) Fifteen had a high incidence of faulty assembly/installation-induced seal failures. It should be noted that the combination of defective/improper parts usage during refurbishment and faulty assembly affected 100% of the survey-sampled power

FIGURE 8 MAINTENANCE-INDUCED SEAL FAILURES

stations; and

- (3) Eight pointed to difficult or improper installation/adjustment/misalignment as a major source of seal problems. It should be noted that seven of these stations use seals that require partial of seal assembly at the pump site. The single-cartridge installation, which precludes seal assembly at the pump, does enhance the reliability of the seal.

A properly trained cadre of maintenance personnel would greatly reduce the incidence rate of the above noted failure types. However, the maintenance burden may be lightened by inherent seal design features which:

1. maximize parts interchangeability between the various seal cavities;
2. avoid using hard to distinguish different parts such as "O"rings of small differing cross section or size;
3. prevent the incorrect insertion of parts such as the seal ring which may require specific face orientation. Where this is not practical, the pieces can be clearly and visibly marked;
4. make it very difficult to assemble the seal incorrectly by prominent size, form-fit design aspects;
5. make insertion of sensitive or easily damaged parts easy;
6. identify shelf time limits on time limited parts such as elastomers and possibly the carbon ring (for checking flatness stability). These markings should be either on the parts or their wrapping and be highly visible;
7. provide case markings to indicate proper assembly stack height at various stages of assembly; and
8. provide for easy adjustment once the seal is fully installed

in the pump.

3.8 Seal Reliability

Seal reliability objectives must be directly factored into purchase specifications since failures will cause reduced plant operating capacity or force a total plant outage. It is not the intent of this paper to develop even a brief foundation of reliability concepts. However, some mention of reliability objectives is required because design considerations are reliability associated.

Shaft seals should be reliability qualified by some combination of test, analysis, and operating experience of identical-similar designs in comparable operating conditions. Useful life requirements should be at least the time interval between one scheduled maintenance or refueling outage and preferably multiples thereof. The reliability goal (expressed in Mean Time Between Failure) should be at least three times the useful life expectancy of the seal.

3.9 Seal Auxiliary Systems

All pump-seal and auxiliary system-seal operating and static interface conditions should be identified to the seal designer.

3.9.1 Injection

Temperature controlled filtered injection water must be provided to the seal. Injection water circulation for a specified time prior to pump start should be used to temperature stabilize the seal. This could be accomplished by a simultaneous limit on the portion of the injection flow entering the primary coolant system and on the seal injection portion exiting in the controlled bleedoff (CBO) line. The downward pump flow can be restricted through a fixed orifice and the CBO flow can be throttled by an in-line valve.

Since seal ring deformation load is a function of injection supply pressure, pressure control and/or overpressure protection should be provided. To assure cleanliness of the injection supply, redundant, parallel filters should be considered. These filters should have by-

pass provisions based on the differential pressure developed across the filter element. Such protection will assure that the seal will not be water starved because of clogged filters or that the filter element will collapse for the same reason. Means to isolate each filter should be provided so that filter element cleansing or replacement will not require the curtailment of water injection to the seal.

If the injection supply is interrupted and subsequently reestablished, the injection water flowrate and/or temperature should be controlled so as to minimize the thermal downshock on the seal. Such flow-temperature requirements on the injection flow as a function of seal operating temperature should be analyzed and the results of this analysis be reflected in operational procedures. The injection system instruments and controls should be capable of providing these features.

3.9.2 Component Cooling Water (CCW)

An independent CCW supply should be provided for seal cooling if the injection water source is interrupted. The capacity of the CCW supply shall be sized to maintain safe temperature levels, minimize thermal up-shock in the event of the loss of seal injection, and also to provide cooling for hot standby conditions when the internal recirculating pump is not operating. The CCW flow capacity to the seal shall be sized to cool primary coolant water within the seal manufacturer's temperature and supply flow requirements.

Automatic closure of the CCW supply due to instruments and/or controls should be prevented. Any supply line automatically controlled valve should be of the fail-open type and contain physical limit stops to preclude full closure. CCW system isolation for maintenance purposes can be accomplished with manual shut-off valves. Numerous reported failures have been caused by inadvertent closure of the CCW supply. The aforementioned provisions will prevent such occurrences.

3.9.3 Bleedoff Return

The controlled bleedoff (CBO) return path should provide the means for primary coolant system and seal cavity venting. Visible means to determine the sufficiency of the venting process is highly recommended.

In some systems, the CBO return flow path is closed when the sum of the staging flow and leakage flow exceeds the cooling capacity of the CCW to cool the seal. For injectionless systems this has been typical procedure. For systems with injection, such procedure would be invoked only when that condition is incurred simultaneously with loss of injection. At times, such closure of the CBO flow path was based on false premises because of instrument anomalies or failures. Such erroneous closure of the CBO line places a good seal into a dry operating mode, which rapidly deteriorates into a failure. Since flow, differential pressure, and temperature sensors are utilized, and since these parameters have a definite interrelationship, CBO valve closure should not be based on a single parameter readout. System design specifications should guard against such occurrences by requiring parameter correlations to substantiate acceptable readings.

3.10 Instrumentation

The reader is referred to Reference 5 for a summary discussion on instrumentation controls and readout requirements associated with the operation of seal and seal auxiliary equipment.

3.11 Maintenance Provisions

The reader is referred to Reference 6 for a detailed discussion and identification of maintenance provision, criteria and requirements for shaft seals.

4.0 REFERENCES

1. EPRI NP-2611, Volume 1
Main Coolant Pump Shaft Seal Reliability Investigations,
Borg-Warner Corp.; Byron Jackson Pump Division
2. HEDL-TI-707-810-013
Cover Gas Seal Development Recommendations for Seal
Types, Seal Materials and Seal Gland Sizes and Finishes
for Use in FFTF-Dec. 1975, Hanford Engineering Develop-
ment Laboratory

3. EPRI NP-361, Final Report, Volume 1, Recirculating Pump Seal Investigation, MPR Associates
4. EPRI NP-1194, Final Report, Sept. 1979, Operation and Design Evaluation of Main Coolant Pumps for PWR and BWR Service, Energy Research and Consultants Corporation
5. Volume 2, "Operational Guidelines," of this document
6. Volume 1, "Maintenance Manual Guidelines," of this document