AD-A138 434 MECHANICAL FACE SEAL TECHNOLOGY: A LITERATURE SURVEY 1/1
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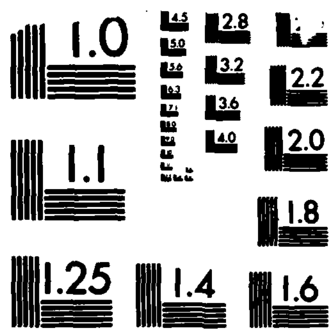

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MECHANICAL FACE SEAL TECHNOLOGY: A LITERATURE SURVEY

by

N.T. Sides

SHIP MATERIALS ENGINEERING DEPARTMENT
RESEARCH AND DEVELOPMENT REPORT

February 1984

DTNSRDC—84/004

MECHANICAL FACE SEAL TECHNOLOGY: A LITERATURE SURVEY

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER DTNSRDC-84/004	2. GOVT ACCESSION NO. A138434	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) MECHANICAL FACE SEAL TECHNOLOGY: A LITERATURE SURVEY		5. TYPE OF REPORT & PERIOD COVERED Research and Development 6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) N.T. Sides		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS David Taylor Naval Ship R&D Center Annapolis, MD 21402		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS Program Element 63569N Task Area S1257 SL001 Work Unit 2723-606
11. CONTROLLING OFFICE NAME AND ADDRESS David Taylor Naval Ship R&D Center Bethesda, MD 20084		12. REPORT DATE February 1984 13. NUMBER OF PAGES 22
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Seals
Mechanical Face Seals
Cavitation
Materials
Wear

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A literature survey was conducted to review and evaluate the state of the art in mechanical face seal technology. The survey is part of a study to develop the capability of designing improved seals for Navy use. Sixty-eight references are cited and discussed. The report is subdivided into several sections, each embracing a general physical aspect of face seal operation.

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A literature survey was conducted to review and evaluate the state of the art in mechanical face seal technology. The survey is part of a study to develop the capability of designing improved seals for Navy use. Sixty-eight references are cited and discussed. The report is subdivided into several sections, each embracing a general physical aspect of face seal operation. The difficulties in modeling seal performance are emphasized. Indeed, it is remarkable and encouraging that several of the models developed do predict face seal performance with some degree of accuracy.

ADMINISTRATIVE INFORMATION

This report covers work conducted under the SSN 688 Shaft Seal Improved Life Program under Program Element 63569N, Task Area S1257 SL001, and Work Unit 2723-606.

INTRODUCTION

Mechanical face seals find numerous applications in naval machinery, for example on the propeller shaft and in all the fluid-handling pumps. When a seal fails, repair is usually costly in terms of lost time and direct costs. Thus, improved seal life and reliability would be of significant benefit to the Navy.

The immediate objective of the present work was to review the state of the art in mechanical shaft seal technology. The ultimate goal is to develop the capability of designing seals with longer life and greater reliability.

Mechanical face seals consist of two annular rings rotating relative to each other and pressed together by spring and fluid pressures (see Figure 1). The idea is to maintain a small gap between the ring faces such that leakage is kept very low but a definite lubricant film is provided.

Although the fundamentals of operation of mechanical face seals have only been seriously addressed in the last two decades, important advances in the understanding of seal operation have been made during that time. Despite these advances, however, seal performance remains a hit-or-miss proposition. A seal may perform for a very long time with virtually no leakage or wear, while another will fail completely in very short order under seemingly identical working conditions. In general, seals for specific applications can be designed to perform reliably, although

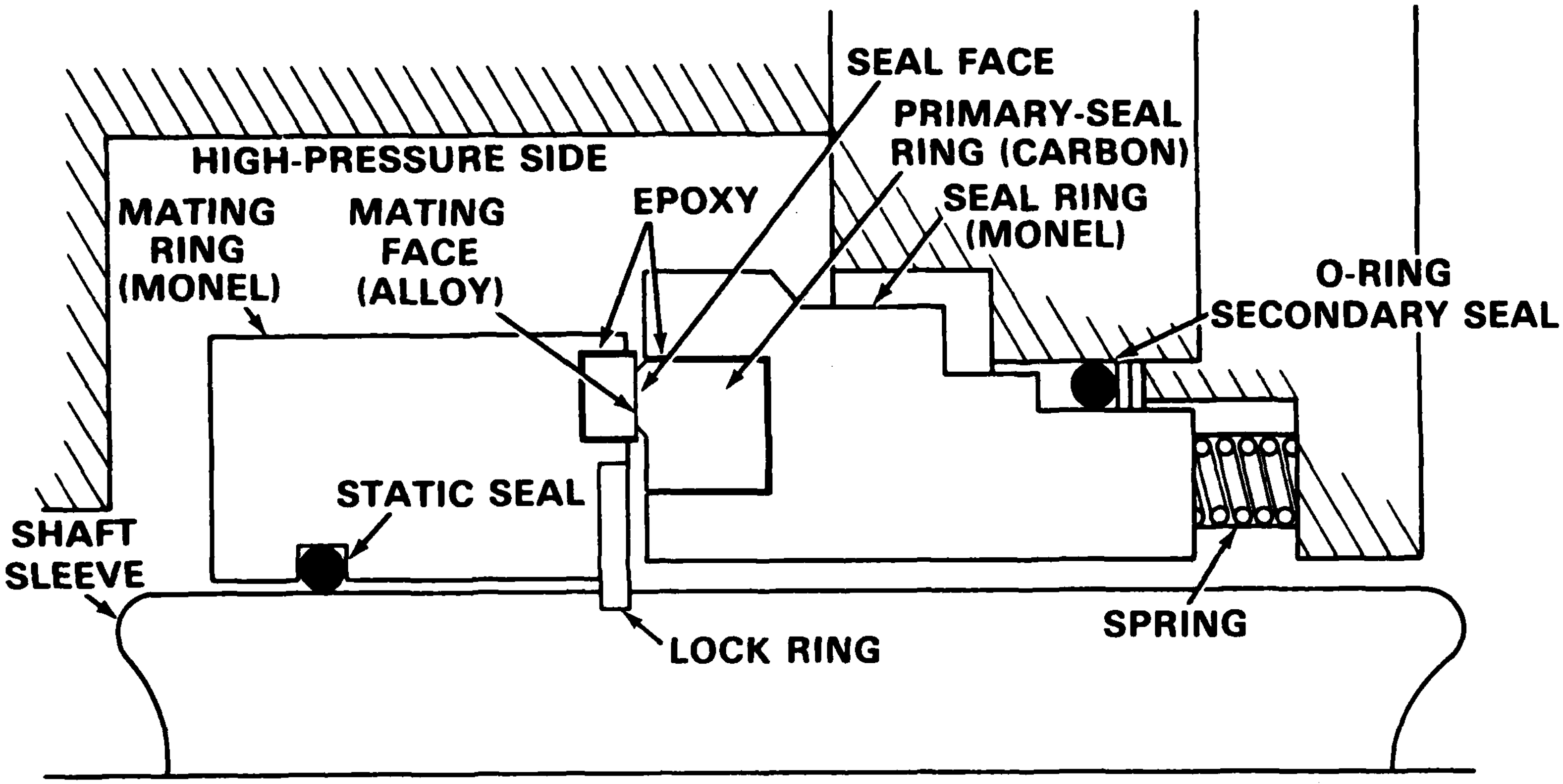

Figure 1 - Existing Propeller Shaft Seal Design

there are numerous applications where satisfactory operation has not been possible. The problem is that because the mechanics of seal operation are still not thoroughly under stood, the designer cannot anticipate the seal's performance as a function of its design parameters. In spite of the numerous theoretical models developed over the years, no fundamental theoretical basis is available to prescribe a seal design that will result in predictable performance. Improvements to date have resulted from the application of elementary theoretical models combined with trial and error from previous experience.

This report, which examines the literature available on the subject of mechanical face seals, was limited to work performed from 1970 to the present and focuses on actual seals or seal-like geometry. Thus, work on fluid-film bearings was excluded despite the similarities to seals in their theoretical treatment. Each section of this report loosely relates to a specific area of concentration, although significant overlap occurs in the subjects discussed. Many literature surveys have been written on seals, and they are referenced where applicable.

The literature search originated with the National Aeronautics and Space Administration (NASA) data banks and the DIALOG information retrieval services. The data sources queried were then expanded to include the major technical societies such as the American Society of Mechanical Engineers (ASME); the American Society of Lubrication Engineers (ASLE); the British Hydromechanics Research Association (BHRA); and the technical journals such as Wear, Lubrication Engineering, Tribology International, and others.

GENERAL FACE SEAL DESIGN AND FIELD EXPERIENCE

The British Hydromechanics Research Association published several bibliographies containing citations from their data base on the general and specific aspects of seal design. The performance of dynamic seals was discussed in terms of frictional forces, as were the methods of lubrication and wear resistance and the procedures to test and measure them.[1*] Consideration was given to antifriction behavior, improved materials, lubricant effects, accelerated model wear testing, and waviness effects of tilt and coning.[2] General applications for seals and patent literature were discussed[3] and extremely high-pressure applications were reviewed.[4]

*A complete list of references appears on page 11.

Metcalfe[5] examined seal failures in Canadian nuclear reactor pumps, and he described the experience gained from these failures relating to inadequate lapping, inelasticity, erosion, corrosion, abrasion, adhesion, surface fatigue, cavitation, thermocracking, seal ring breakage, and static seal failure. Ludwig and Greiner[6,7] described basic assembly configurations, their advantages and shortcomings, and their applications. They related lubrication mechanisms to seal operation and discussed hydrodynamic and hydrostatic effects. Nau and Rowles[8] wrote a computer program to aid in seal design. The program accounts for surface topography, interfacial fluid dynamics, cavitation, self-generated heat, and thermal and mechnical distortion. Buck[9] developed a methodology to evaluate seal designs or applications based on observations of maintenance data representing over 130 pump years of service. Seal life was correlated to three dimensionless parameters.

Martinson[10] discussed the development of a large diameter, high-pressure seal for nuclear reactor service. He also described the final seal design configuration in detail and included computer modeling techniques, materials, seal design and operating parameters, and results of special development tests. Recently, Metcalfe[11] reviewed seal-related equipment used in the nuclear power industry, highlighting its novel or useful features and the understanding gained from its use. Summers-Smith[12] analyzed maintenance data for centrifugal pump seals in a petrochemical plant. Results suggested that pressure is generated in the lubricant film as a consequence of surface waviness. A major factor of early seal failure was the inability to generate the correct face profile during the initial stages of running.

VIBRATION AND DYNAMICS

An extensive literature review was conducted recently by Etsion.[13] He divided the review into three sections. The first relates to experimental observations of seal vibration, the second to investigations of dynamic properties of seal elements, and the third to theoretical dynamic analyses of mechanical face seals. Several relevant papers have been published since that time. Green and Etsion[14] calculated stiffness and damping coefficients of the seal fluid film for three degrees of freedom using small perturbation of the ring from its equilibrium position and compared the results with those of other analyses. Childs[15] also derived dynamic coefficients for high-pressure seals and compared them to other published results. In a companion paper,[16] he developed a finite length solution procedure based on Hirs' turbulent lubrication model. The model gave reasonable results with no swirl present, but failed when swirl was introduced. Dhagat et al.[17] studied the response of a hydrodynamic seal to a specified initial disturbance. They numerically integrated the linearized and the nonlinear equation of motion for lateral oscillations of the seal.

CAVITATION

The subject of cavitation in face seals was recently reviewed by Nau.[18] He summarized past work and presented an analytical model to calculate film cavitation boundaries. Kistler et al.[19] conducted an experiment simulating cavitation in submarine stern-tube seals. Cavitation bubbles were observed and several analytical models were studied. The physical parameters influencing bubble growth and collapse were identified. Ikeuchi and Mori[20] adopted a different numerical technique, whereby the mass continuity is preserved in a ruptured film, thus allowing the cavitation boundary to be determined. They showed that load capacity rises with cavitation. In a companion paper,[21] the authors showed that inward pumping was induced by cavitation and film thickness variation. They concluded that leakage is independent of cavitation unless radial forces are applied to the cavities.

WAVINESS

Lebeck[22] found that concentrated drive forces are the largest contributors to waviness. He developed general equations for ring deflection based on curved-beam theory and found that second and odd value harmonics are generally predominant. He also developed a mixed lubrication model[23] to predict the effects of longitudinal roughness and waviness on lubricant behavior and found that most of the load was supported by hydrodynamic pressure. The model was refined[24] to consider the contribution of mechanical and hydrostatic pressures, and of thermal rotation. The results showed a strong influence of thermal rotation on initial seal performance and wear. Subsequently, a test machine capable of imposing waviness on a seal ring was described.[25] The waviness was moved slowly relative to the ring under rotation to preclude localized wear, which greatly reduced friction and wear. Waviness and tilt then were varied[26] to optimize friction and wear for a given leakage. The geometry of the seal is therefore dictated by the optimum design developed. Lebeck's mixed lubrication model was further refined[27] to consider waviness amplitude, surface roughness, asperity contact, hydrodynamic and hydrostatic pressures, and fluid cavitation.

A numerical method of solution was presented by Young and Lebeck,[28] who obtained experimental data and compared it to the mixed lubrication model. Good agreement was reported for initial torque and leakage as a function of initial taper. While predicted equilibrium thermal taper (as a function of torque for a balance ratio of 1.0) was in good agreement with experimental results, the agreement was not so good for a balance ratio of 0.75. Based on previous tests,[25] Lebeck proposed a wavy seal design for a submarine application.[29] Ruddy et al.[30] analyzed the effects of a two-period waviness on both the face and the seat. They obtained a closed-loop cyclic solution of a two-dimensional Reynold's equation including squeeze film effects. The waviness amplitude was found to be an order of magnitude less than that measured on seals that have successfully operated in service.[31] Peng and Cheng[32] investigated flow between a rotating and a stationary disk with concentric waviness. They solved the Navier-Stokes equations using a perturbation scheme and concluded that the disk with single negative sine waviness results in minimum leakage and torque at a given wave amplitude.

PHASE CHANGE

The hypothesis that a phase change may take place as the liquid flows between the mating seal rings was first proposed in the work of Orcutt.[33] Hughes et al.[34] studied the phenomenon almost a decade later. They concluded that boiling occurs, particularly when the liquid nears saturation conditions; that the main reason for boiling is the flashing which occurs as the pressure decreases across the seal's radius; and that stability criteria exist under boiling conditions. The model was extended to isothermal and adiabatic cases and solved numerically in a follow-up paper.[35] Turbulent two-phase flow, with heat addition due to rotational dissipation and the effects of centrifugal inertia, was considered next.[36] A general theory was presented for quasi-steady operation. Lebeck[37] emphasized a mixed friction hydrostatic seal model where both mechanical contact and phase change effects were considered. He showed that phase change resulted in a greater load support by fluid-film pressure than either an all-liquid or an all-gas seal and concluded that operation at a higher temperature reduces seal wear rate and friction.

Will[38] conducted experimental tests that supported Lebeck's theoretical work. Kuzma[39] analyzed a grooved seal to determine the location and shape of the liquid-gas interface. Barnard and Weir[40] examined seals that operated successfully in an oil refinery. They observed three concentric rings across the seal faces and hypothesized that the gas-band conditions approach the critical temperature and pressure of the sealed fluid. They correlated actual and computed track dimensions.

THERMAL EFFECTS

Black[41] solved the equations of thermoelasticity in a three-dimensional, axisymmetric space. He obtained a pressure-velocity relation based on the analytical stress values. Chin-Hsiu[42] solved the temperature fields and deformations numerically for both the seal and mating rings. He found the mechanical deformation to be small compared to the thermal deformation. Lebeck[43] developed deflection equations which included both thermal and mechanical loading and combined them with a wear equation. He formulated a mathematical model that predicted the limits of stable operation and later modified some of the assumptions made in the model[44] such that results agreed with those of Burton et al.[45] Banerjee and Burton[46] conducted experiments on hydrodynamic seals at thermal equilibrium conditions, comparing changes in waviness and mean film thickness to theoretical predictions of thermal growth of waviness. Wu and Burton[47] developed a model to explain the coupled inertial/ thermoelastic instability observed experimentally by Banerjee.[48] Netzel[49] reviewed field and laboratory results with respect to the development of thermoelastic instabilities at the seal interface and reported the effect that interface cooling has on the surface disturbances. Kilaparti[50] studied the thermoelastic instability problem by breaking it into subproblems and obtaining a solution for each case. Effects of material properties on critical speed and of critical speed at the onset of instability were presented analytically. Suganami et al.[51] formulated a model to predict the oval deformation of a seal ring due to hot spots and the rotation of the oval shape relative to the hot spots. Experimental results were presented to confirm the model predictions. Kennedy et al.[52,53] investigated thermocracking on a cobalt-based superalloy seal ring and proposed a mechanism for initiation and propagation of microcracks that agreed with microscopic observations. A contact probe was built to determine the number, size, location, and surface temperature of contact patches over a range of seal velocities. The size and temperature of the contact was found to be influenced by the operating velocity and by the thermal, elastic, and wear properties of the seal materials.

MATERIALS

Merrick[54] characterized interfacial wear reactions of pure nickel and nickel 20 atomic percent molybdenum. Hard facing alloys were weld-deposited on Inconel 625 coupons in a study by Vreeland et al.[55] A special cermet, developed for possible use as a wearing surface, showed excellent wear and corrosion-resistance properties. Lohou[56] discussed a new version of silicon-carbide (SiC-S) as applied to water-injection pumps. Echtenkamp[57] compared the properties of tungsten and titanium carbides with other materials. Lashway[58] reviewed the commercial forms of silicon-carbide with respect to the manufacturing processes, resultant properties, and impact on seal performance. Lagerquist and Sandvick[59] developed a new type of corrosion-resistant cemented carbide that maintained the physical and mechanical properties of the base alloy by adding chromium and molybdenum to tungsten carbide/nickel grades. These additives stabilize the metallic binder phase both mechanically and electromechanically without affecting the carbide phase.

Effertz and Fichte[60] examined seal failures in large reactor water pumps and concluded that material loss from the tungsten-carbide rings with nickel or cobalt binders followed the dissolution of these binders. They recommended the use of a NiCrMo binder, the use of SiC rings, or the use of synthetic resins in place of Sb impregnation. Dziedzic[61] presented experimental data to compare the corrosion, thermal shock, and wear resistance of siliconized graphite and SiC with that of carbon graphite for nuclear service pumps. Paxton et al.[62] ran a carbon seal ring against hard carbon, fused alumina, tungsten carbide, silicon-carbide, and silicon-carbide-coated graphite in water. They found that SiC run against hard carbon produced the most stable coefficient of friction. Earlier, Paxton and Strugala[63] ranked carbon impregnated with silver, babbitt, bronze, copper, antimony, and resin against 85% sintered aluminum oxide. The resin-impregnated carbon had the lowest wear rate. Ya[64] examined the phenomenon of selective material transfer in which specific components of one element of a friction pair are transferred to the other when in contact with a liquid dielectric. Beck[65] presented a theory of electrochemical corrosion caused by accelerating fluid through a gap. The flow generates electrokinetic streaming current, which causes the corrosion. Hirano and Sakai[66] demonstrated reduced friction, wear, and leakage by chain matching. This effect, which is considered to play a predominant role in thin film lubrication, appears when numbers of carbon atoms of a solvent and an additive coincide.

Tribe[67] characterized and assessed a wide variety of seal materials and discussed a variety of failure mechanisms. Labus[68] investigated the following seal materials: carbon, aluminae ceramics, siliconized carbons, silicon carbide, cast iron, and tungsten carbide. He established that speed was the dominant factor, with pressure and temperature having secondary effects.

CONCLUSIONS

The difficulties in formulating an encompassing (Universal) seal operating model become obvious. The mechanical face seal is a complex system in which many physical phenomena coexist and interact such that their contributions are impossible to consider separately. Mathematically, the large number of coupled equations is unwieldy in any kind of closed-form solution. Combining these equations into a single model without solving them would be a formidable task in itself. All of the authors cited addressed one or more physical aspect. Some formulated an approximate model that either neglected other physical phenomena or made simplifying assumptions about them, which renders the outcome predictably incomplete. However, the models are valid for a specific range of operation under specific constraints. Others attempted to combine one or more existing models, with mixed success, but experimental data are still lacking. Most of the existing data are generated from an attempt to highlight the theoretical model developed. As such, these data find little applicability in supporting another author's model, which may require parameters not measured by the original researcher. Difficulty also arises because seal performance cannot be scaled. A small seal can operate successfully while a larger one, using the same geometry and materials and operating under identical conditions and loads, can fail. This makes comparisons between different sizes of seals difficult.

It is a remarkable achievement nonetheless to find that several models were developed that do indeed predict seal operation with some degree of accuracy. Steady progress is being made in understanding the underlying principles of seal operation, as evidenced by the increased sophistication of the most recent models.

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