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***Development of high temperature,
high pressure rotating shaft seals:
Final report***

Heshmat, H.; Shapiro, W.

Jun 1987

Mechanical Technology, Inc., Latham, NY (USA)

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Development of High Temperature, High Pressure Rotating Shaft Seals

Final Report

DOE/MC/22074--2533

DE88 001029

**H. Heshmat
W. Shapiro**

June 1987

Work Performed Under Contract No.: DE-AC21-85MC22074

**For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
Morgantown, West Virginia**

**By
Mechanical Technology Incorporated
Latham, New York**

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Morgantown Energy Technology Center
P.O. Box 880
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Latham, New York 12110

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LIST OF ABBREVIATIONS AND SYMBOLS

B	Length of a contact in the direction of sliding
R_o	Outer radius
R_i	Inner radius, mean radius
R	$(R_o + R_i)/2$
E	Elastic modulus
V	Relative sliding velocity
T	Temperature at remote boundary of solid
F	Load
T_f	Flash temperature
A	Area
Q	Heat flow per unit time
Q_T	Total frictional heat generation per unit time
P	Interface pressure
t	Thermal resistance of solid
r_b	Bulk component of thermal resistance
K_1, K_2	Thermal conductivity of body 1 and 2 respectively
c	Specific heat
K	Thermal diffusivity ($K/\rho c$)
ρ	Density of sliding solid
Δ	Thermal boundary layer thickness
N	Rotational speed (rpm)

SUBSCRIPTS

1	Parameter of solid 1 (seal ring)
2	Parameter of solid 2 (seal runner)
amb	Ambient
i	Inner
o	Outer

SUMMARY

Rotating shaft seals used in coal gasification equipment are exposed to difficult environmental conditions including temperature from 430°C to 816°C (800°F to 1500°F), high pressure (to 10 MPa or 1500 psig), and high levels of particulate contamination (50,000 to 100,000 ppm). The program reported upon was designed to develop long-life seals that would not require external flushing or cooling.

The approach taken was to develop unbalanced face seals that would remain closed and prevent particulate entry into the interface. Wear resistant materials and coatings with low friction coefficients are required. Significant results of the program are enumerated below.

- The selected material combination was a rotating runner of Kentanium K162B mating against a non-rotating seal ring of Kentanium K162B. Kentanium is the trade name of a series of hard carbide alloys with pure titanium carbide as the principal ingredient. Nickel and nickel molybdenum are used as binder materials. Kentanium retains most of its strength at temperatures up to 1100°C (2000°F).
- Since hard materials are required to resist wear and withstand the temperatures, mechanical compliance must be built into the seal configuration for the opposed faces to follow runner nutations and remain closed. It was demonstrated that a completely rigid configuration performed poorly while a thin seal ring mounted on a compliant substrate performed very well.
- No matter how flat the opposed surfaces of a contact face seal are, thermal distortion will produce a saddle shape waviness in both the runner and seal ring. This phenomenon is well known in face seal technology. For liquid seals, the waviness allows hydrodynamic action which lubricates the seal and reduces friction. For the dry (non-lubricated) Kentanium faces, the saddle shape waviness was found to occur. Furthermore, environmental and frictional heating caused the formation of very small particles of titanium oxide which lodged in the face wave formations and acted as an interfacial lubricant. Typically, the coefficient of friction varied from 0.15 - 0.3, and the wear factor ϕ varied from 3-5 x

10^{-11} cm²/kg indicative of low wear. The self generation of the interfacial solid lubricant was the necessary ingredient to successful seal operation, and was the most profound discovery of the program.

- Limited chemical reaction tests were conducted by exposing material samples to a hot gas sulphur environment. The K162B produced some reaction products which may be beneficial for lubrication, although significantly more testing would be required to substantiate this theory. Surface and subsurface layers were produced. All phases analyzed contained various concentrations of sulphur. Only the thick, outer, single phase layers contained carbon.
- Since the test results demonstrated the advantages of powdered lubrication in dry contact face seals, further efforts should be conducted to exploit this phenomenon. The powder in the experiments conducted in this program was self-generating, but it may also be possible to externally introduce the powdered lubricant. Thus it is recommended that:
 1. Investigations be conducted to determine appropriate lubricant powders that could be introduced into the sealing interface. Characteristics of the powdered lubricant would include: temperature capability to 2000°F; lubricity at both cold and hot temperatures; particle sizes in the order of 1 μm; chemical compatibility with surrounding gases and seal materials.
 2. Means for external injection of the powdered lubricant should be investigated as this will be an important and integral part of any future seal system.

The results of this program provide encouragement and optimism that dry contact face seals can be developed for long unattended operation in the hostile environment of a coal gasification plant.

1.0 INTRODUCTION

The lack of high-temperature, high-pressure shaft seals is impeding technology development of coal conversion processes to produce gaseous and liquid fuels. The effluent temperatures of the various gasification processes available and now under development will vary from 800 to 2000°F, and the pressures may vary from 50 to 1500 psig. These processes include fixed, fluid, and entrained-bed reactor systems. The raw product gas will contain significant amounts of H₂, CO, CH₄, and higher hydrocarbons as desired products. It will also contain impurities such as H₂O, CO₂, coal and ash fines, tars, oils, phenols, and a variety of sulfur and nitrogen compounds. The typical commercially available mechanical seals utilize a double seal with a cooling fluid (usually steam) and even a flushing fluid to remove particle buildup. A requirement of the program was to establish seal configurations that would not include external flushing and cooling systems.

The difficulties associated with high temperature and high pressure are compounded by high levels of contamination. An unbalanced face seal designed to remain closed and not permit contamination to enter the interface was considered a necessary requisite for a long life seal. Opposed contact face materials must resist wear, have low friction characteristics, and be able to withstand the environmental and friction-generated temperatures involved.

The general objectives of the program were to:

- establish unbalanced face seal concepts
- determine appropriate material combinations
- conduct simulated laboratory tests emphasizing interfacial phenomenon (friction, wear, etc.).

Face seal design is a combination of materials selection, balance ratio, preload, and interface compliance. Contemporary low temperature seals employ carbonaceous materials that have low modulus of elasticity and will compress several microns when hydraulically and mechanically preloaded. This initial compression must be released before face opening will occur. For high-temperature applications, carbons are not feasible; therefore ceramics, cermets, or superalloys that have a very high modulus of elasticity and that are not easily

compressed must be employed. Thus, other mechanical means must be devised to maintain interface closure.

Three principal types of seal configuration was designed and built for evaluation:

- Compliant surface, compliant mount
- Rigid surface, compliant mount
- Rigid surface, rigid mount

These three types of configurations are schematically shown on Figures 1-1, 1-2 and 1-3, respectively. The mean diameter of the sealing surface of the test seals is approximately 51 mm. The more compliant the seal interface, the better the chances of maintaining tight closure at the interface. The compliant surface can accommodate surface irregularities as well as gross motions of the seal ring relative to the seal runner.

For the compliant-surface, compliant-mounted seal, shown in Figure 1-1, the thickness of the surface components and structural support is approximately 0.305 mm (0.12 in.) with an axial stiffness of approximately 0.28×10^6 N/m (1b/in). This compares to a carbon-stiffness of 525×10^6 N/m (3×10^6 lb/in.). These values can be varied by changing the material thickness and the support structure configuration. The material of the compliant seal is Inconel X750, which has a temperature limitation of approximately 816°C (1500°F) where the yield strength falls off. The compliant seals are also pressure limited to approximately 698 kPa (100 psi) to avoid excessive deformation of the foil support members. The compliant surface seal can be difficult to balance because the contact position varies with load and wear.

The rigid-surface, compliant-mounted configuration, Figure 1-2, cannot easily accommodate surface irregularities; lapped surfaces are required. The major advantage of this seal configuration is its ability to accommodate misalignment and rigid body motions without losing contact. Ceramics or cermets, which increase the temperature capabilities, can be used for the surface material, rather than a superalloy material. The compliant mounting can also be stiff since surface irregularities are not accommodated. This seal has the capability for higher pressures (2756 to 3445 kPa, 400 to 500 psi). The temperature is

Fig. 1-1 Compliant-Surface, Compliant-Mount Seal

Fig. 1-2 Rigid-Surface, Compliant-Mount Seal

Fig. 1-3 Rigid-Surface, Rigid-Mount Seal

limited to that of the support structure, which will probably be about 816°C (1500°F), although some alloys are purported to withstand 1100°C (2000°F).

The rigid-surface, rigid-mount configuration, Figure 1-3, will require high hydraulic loading (or unbalance) to maintain closure. Important considerations to minimize friction and wear are surface finish, materials, and coatings.

The test seal rotor is shown on Figure 1-4. The mating surface is the inside surface. The taper on the outside surface is intended to centrifugate contaminant particles.

The work that was accomplished under the program and described in this report included the following:

- Design and construction of the three types of face seals described above with varying amounts of compliance incorporated.
- Selection and procurement of materials for the runner and mating rings.
- Modifications to a test rig to accommodate the seals for high temperature and contaminated testing.
- Testing of several seal configurations and material combinations.
- Establishing the effects of a sulfur environment on the materials selected.

The program emphasized interfacial phenomenon, in particular closure, friction and wear.

In addition to the above technical tasks, a commercialization plan was developed at the outset of the program [1]. The plan outlined potential applications and economic impact of developed seals and a sequence for making successful seals commercial products. Results of the plan are summarized in the following paragraphs.

Fig. 1-4 Seal Runner

For a coal gasification plant, the advanced seal can be applied to fuel-gas expanders, recycled-gas compressors, and sundry pumps. A conservatively estimated savings of \$350,000/yr/plant was predicted by implementation of a high temperature face seal. The savings derived from maintenance and life considerations.

Various sources were used to prognosticate the potential market for the advanced seals, including discussions with seal manufacturers and perusal of pertinent literature. The annual potential market for advanced seals, in a 1990 time frame including special applications, such as advanced diesel engine turbochargers, is \$128 million per year. Presently, the total face seal market is approximately \$600 million, and in the 1990s, it should grow to \$800 million. Thus, the advanced seal will represent approximately 16% of the total market.

It is clear, therefore, that a sufficient market exists to warrant interest by seal suppliers to promote and service a product line.

Commercialization involves the following sequential activities.

- Research and Development
- Technology Transfer
- Market Development
- Pilot Demonstrations
- Production

A 10-12 year program of concentrated effort would be required for a commercial product line to be made available.

2.0 MATERIAL SELECTION

An extensive review of materials for seal components and lubricants was performed. The review was conducted using tribological data bases, as well as personal communications with leading researchers and vendors (references 2 - 33). Information on the simple sliding mode of operation of ring on disc (sliding of two parallel plates) is sparse and dated.

Figure 2-1, derived from the listed references, provides an indication of anticipated levels of friction and wear. The information was developed for pin-on-disk apparatus and is not directly applicable to two flat surfaces in rubbing contact, such as would occur in a seal. Debris formation and removal and its effect on friction and wear would be entirely different for the two types of rubbing configurations. The information is useful, however, for screening purposes and providing comparative data for initial selection.

The wear factor (ϕ , Figure 2-1) is defined as the ratio of the volume of material worn from a single mating contact surface to the distance slid and the normal load at the contact. Thus, ϕ , can be shown:

$$\phi = \xi A / (\ell \times F)$$

$$\ell = Vt$$

$$V = N \times \pi \times (R_o + R_i)$$

where

ξ = Wear depth, cm

A = Contact surface, cm²

ℓ = Sliding distance, cm

F = Normal load, kg

t = Sliding time, sec

V = Tangential velocity of contact surface, cm/sec

N = Shaft rotation velocity, rps

R_o = Outer contact radius, cm

R_i = Inner contact radius, cm

The physical interpretation of the numeric value of the ϕ factor is as follows:

Fig. 2-1 Friction and Wear of Pin-on-Disk Material Combinations

$\phi = 10^{-8} \text{ cm}^3/\text{cm-kg}$	high wear
$\phi = 10^{-9} \text{ to } 10^{-10} \text{ cm}^3/\text{cm-kg}$	moderate to low wear
$\phi = 10^{-11} \text{ cm}^3/\text{cm-kg}$	very low wear

The ordinate on Figure 2-1 indicates material combinations, where the numerator is the ball material and the denominator is the disk material. The spread in values is due to variations in temperature in accordance with the legend indicated.

The combination of PS200 on PS200 shows low wear and friction with little variation over a wide temperature range. As described in the testing section of this report, however, the PS 200 did not perform well when subjected to load. Many of the ceramic combinations show relatively high wear factors, but they may be significantly different when applied to seal-type rubbing.

The data shown for K162B vs K162B was developed under the present program for ring on disk type wear. The low values of friction and wear confirmed the selection as being close to the optimum available.

Materials fall into two categories: base materials and coatings. Coatings are applied to reduce friction and wear. Based on a perusal of the literature (see References 4 through 35) and on the experience of MTI personnel and consultants, several candidate materials were selected for testing.

Selected materials together with those materials that have potential for evaluations are as follows:

Seal Ring Materials

Inconel X750

XF818

Kentanium Grade K162B

Potential Candidate Materials

Inconel MA754

Silicon Nitride Si_3N_4

Carboloy (460 (Al_2O_3 plus additives)

α -Silicon Carbide

Exolon-Esk (α -Si C HD)

Carborundum Hexolloy (Silicon Carbide)

Kennametal-Sialon (Silicon Nitride plus additives)

Lucas Syalon 201 Si ALON

Norton (NC-132; NBD-100; NCX-34; NC-144)

Kentanium Grade KT150

(60% TiC/TiN + 30%WC/MoC + 10% Co/Ni as a binder)

Carborundum (Hexoloy-80T, Combat-BN with B_2O_3)

Ceradyne (Y_2O_3 - pressed AlN; Ceralloy 147Y/147Y-1) Al_2O_3 -TiC₃₀)

Seal Runner

Kentanium Grade K162B (Titanium Carbide Cermet)

K162B coated with PS200 (under development)

Potential Candidate Material

Babcock and Wilcox (BN/Si₃N₄; BN/AlN)

Union Carbide (BN/TiB₂)

Si₄N₄ + SiC

Hastelloy C-276

Carpenter Steel Alloy 20CB-3

Inconel 625

Type 316 Stainless

Compliant Mount Material

INC X 750

Potential Candidate Material

INC MA754

Carbon/Carbon Composites¹

Hastelloy C-276

Carpenter Steel Alloy 20 CB-3

Inconel 625
Type 316 Stainless

Secondary Seal - Piston Ring

The selected material for the piston ring was based on available data gathered from a Stirling Engine test using Stellite 6B. This material has shown low friction coefficient and low wear rate against XF818, the mating material. The rings that were used were made from Stellite 6B and coated with silver for additional lubricity.

An additional list of newly developed materials cited from a DOE report is presented in Table 2-1 for further consideration.

Two runners, Figure 1-4 were constructed of Kentanium 162B. Kentanium is the trade name of a series of hard carbide alloys with pure titanium carbide as the principal ingredient. Nickel and nickel-molybdenum are used as binder materials. Kentanium retains most of its strength at temperatures up to 1100°C (2000°F) and will resist even higher intermittent temperatures. Grade K162B has a nickel-molybdenum binder and is the best general-purpose Kentanium grade. It has the greatest resistance to oxidation, at temperatures in the 540 to 760°C (1000 to 1400°F) range. It is used in seal rings and bearings (can run against itself), as well as for valve parts, check valve balls, and hot flash trimming tools.

One of the runners was planned to be coated with PS-200, a chromium-carbide-based coating with additives of metallic silver and barium fluoride/calcium fluoride eutectic. It is plasma sprayed and then diamond ground to final thickness. Developed at NASA/LeRC, the coating has been found to produce low-friction coefficients and low wear over a wide range of temperatures. It can be applied to thicknesses of 180 to 200 microns, which is advan-

¹Carbon/Carbon Composites are very stable materials for a temperature in excess of 2000°F with density $\rho = 1.8 \text{ Kg/m}^3$. Flexural strength = 181 and 117 MPa.

Thermal conductivity = 16048 (W/KM)

TABLE 2-1

PIN-DISK COMBINATIONS TESTED IN DIESEL (O, N, CO₂, H₂O)
ENVIRONMENT AT 800°C CHOSEN FOR SURFACE ANALYSIS*

DISK [†]	PIN [†]	μ_F
1. Silicon Nitride Si ₃ N ₄ (Ti, Ni)	TiC	0.09
2. Silicon Nitride Si ₃ N ₄ (Ti, Ni)	TiC-5Ni-5Mo	0.22
3. Partially Stabilized Zirconia PSZ (ti, Ni)	TiC	0.25
4. Partially Stabilized Zirconia PSZ (Ti, Ni)	TiC-5Ni-5Mo	0.09
5. Partially Stabilized Zirconia PSZ (Co)	TiC-5Ni-5Mo	0.06
6. Partially Stabilized Zirconia PSZ (Co)	TiC	0.25
7. Silicon Nitride Si ₃ N ₄ (ni)	TiC	0.14
8. Silicon Nitride Si ₃ N ₄ (Ni) (Tested at room temperature)	TiC	0.28

* The elements given in parenthesis are those ion-implanted into the surfaces of the disks.

[†] Specimen Dimensions:

Disk Diam. = 7.62 cm
 Pin Diam. = 0.64 cm

Thickness = 0.95 cm
 Length = 1.27 cm

tageous because it will not easily wear through. Detailed information regarding PS200 compositions and particle size of plasma spray powders is given in Table 2-2. Because of significant differences in thermal coefficient of expansion between the Kentanium and PS200, a satisfactory bond could not be obtained and that particular combination was abandoned. Adherence of PS200 to XF818, however, was very good.

High-strength superalloys employed for the compliant seal can also be applied at temperatures exceeding 1000°C (2000°F). Such an alloy is Inconel MA 754. Submicron particles of inert oxides are incorporated into the material, and high strengths are achieved through a process of directional working and recrystallization, to develop a microstructure oriented along the direction of maximum stress of the intended component. The Inconel X-750 presently specified is satisfactory within the temperature range of the test rig (800°C) (1500°F). The top of the compliant seal, Figure 1-1, was planned to be tested with and without PS200 coatings.

High temperature ring housing components were made from XF818. XF818 is a cast material containing 55% iron, 18% nickel, 18% chromium, and 7.5% molybdenum. It is readily machinable and is presently used in some Stirling engine components. The rig has temperature capabilities to 816°C (1500°F) and will run at 377 rad/sec (3600 rpm). The PS200 coating adhered very well to the XF818 base material, which was used as one of the seal insert rings.

Candidate ceramic materials include silicon-nitride, α -silicon carbide, carboloy 460 (Al_2O_3 plus additives), Sialon, and partially stabilized zirconia. Ceramics, however, were not tested in this program. They were not top choice selections because of inferior friction and wear characteristics.

TABLE 2-2

COMPOSITIONS AND PARTICLE SIZES OF PLASMA SPRAY POWDERS (PS-200)

Component	Composition wt %	Particle Size
Bonded Chromium carbide		
Cr ₃ C ₂	58	-200 + 400 mesh
Ni	28	
Co	12	
Al	2	
Silver metal		
Ag	100	-100 + 125
Eutectic		
BaF ₂	62	-200 + 325
CaF ₂	38	

3.0 DESIGN AND PROCUREMENT OF TEST SEALS

The general seal configuration was designed to facilitate study of interface phenomena, and to be conveniently incorporated into an existing high temperature test rig. A general arrangement of the seal configuration is shown on Figure 3-1. The mating (rotating) ring is tapered at its outer periphery to help centrifugate contaminants back to the high pressure zone which is at the left of the crosssection. Normally, the seal is loaded by the hydraulic pressure being sealed. However, provisions were made to independently load the seal ring through an external connection. This permits varying seal loading without varying the pressure of the gas being sealed so that parametric studies of load could be made. The seal ring was designed to accept inserts for varying materials and compliance without requiring a completely new seal ring.

Figure 3-2 shows the major components of the seal assembly. The seal runner is shown at the left of the photograph; the insert is in the center of the picture and the seal ring to which the insert is placed is shown on the right. The piston ring secondary seal is shown surrounding the seal ring. The inserts which are relatively simple pieces are the principal test specimens and are made of varying materials. The same seal ring is used for all testing.

Figure 3-3 shows several of the mounts that support the inserts in addition to remaining parts. A compliant mount can be installed behind the insert to provide a rigid surface, compliant mounted seal or a rigid mount can be inserted to produce a rigid surface, rigid mounted seal. Not shown on Figure 3-3 is the compliant surface, that when combined with a compliant mount produces a compliant surface-compliant mounted seal. The compliant mount inserted in the seal ring is shown on Figure 3-4, along with the housing of the test rig to which it is inserted. The housing also contains the external connections for hydraulically loading the seal ring against the runner. Figure 3-5 shows the fully assembled seal ring with the insert installed.

Two sets of seal ring inserts and seal runners were fabricated from Kentanium Grade K162B by Kennametal Inc., Greensburg, PA, and are shown in Figure 3-2. A seal ring with INC-718 and XF818 were both coated by Hohman Plating and Mfg., Inc., Dayton, OH as shown in Figure 3-6. The rings were coated with PS-200 to 12 mils thick then diamond ground after plasma spraying to obtain specified flat-

Fig. 3-1 Unbalanced Face Seal System Concept

Fig. 3-2 Photo of K162B Seal Component

Fig. 3-3 Photo of Seal Component, Rigid and Compliant Mount

Fig. 3-4 Photo of Compliant Mount Assembled on Seal Ring

Fig. 3-5 Photo of Fully Assembled Seal Ring (Rigid-Surface,
Compliant-Mount)

Fig. 3-6 Photo of XF818 Seal Ring Coated with PS-200

ness as finished (8 to 10 mil final thickness). The K162B seal runner and ring were finished by a lapping technique and a surface finish of 2-4 μ in. with a flatness of 2 light bands was produced. The coated ring had a final surface finish of 10-20 μ in. (CLA). Since the coating contains 10% metallic silver, lapping after diamond grinding is not done, because lapping causes depletion of silver from the base coating.

It was intended to also plasma spray the K162B seal material with the PS200 coating. Since NASA-LeRC has had most of its experience with nickel base alloys such as INC X-750 or 718 and stainless steels, samples of Kentanium K162B (1" x 1" x 0.5") were obtained which had diamond ground finish surfaces, for determining the integrity of the coating. NASA-LeRC made several runs in order to achieve adhesion of the coating to the substrate material. The results were unsatisfactory; the coating would not adhere to the K162B samples. It was postulated that differences in thermal expansion between the two materials was responsible for poor adhesion and also the two materials may not have been chemically compatible.

The photograph of plasma sprayed PS200 coating over K162B coupons is shown on Figure 3-7. It shows that the coating as sprayed on smooth surfaced K162B coupons are unattached and one of them is totally separated. Since application of this coating on K162B necessitated further development, it was decided to confine application of the PS200 to those materials that already had a proven record, such as XF818 and INC 718.

A drawing of the compliant surface member is shown on Figure 3-8. The design and construction of the compliant surface required extensive analysis, calibration and tooling.

The compliant structural support is shown in Figure 3-9 and a schematic of the coordinate system and the forces applied in the analysis of the deflection of the cross-section are shown in Figure 3-10. The simply supported trapezoidal cross-section is bowed at $X = R$. Sliding occurs at $X = R_1$ and R_0 at the simply supported ends for a uniform load F distribution over the contact portion. In a rigid surface concentrated loads and contact at the corners of the elastic support are applied. The deflection profile resulting from the global pressure is shown schematically in Figures 3-10 and 3-11.

L8697

Fig. 3-7 Photograph of Kentanium Coupons Coated with PS-200

Fig. 3-8 Drawing of Compliant Surface Foil

Fig. 3-9 View of Compliant Ring and Compliant Mount Seal Configuration

POST1 -INP-

ANSYS 4.2
AUG 15 1986
8:43:57
POST1 DISPL.
STEP=1
ITER=1

ZU=1
DIST=.385
XF=2.11
YF=.0375
DMAX=.0312
DSCA=1.24

DOE HIGH TEMPERATURE SEAL 50 PSI

Fig. 3-10 Schematic for Analysis of Compliant Support

DOE High Temperature Seal - ANSYS Summary

Material: INC-750

E = 22.0E6 ν = 0.3
G = 11.0E6

Case Description:

	Face 1	Face 2
#1)	5 psi	0 psi
#2)	10 psi	0 psi
#3)	50 psi	0 psi
#4)	50 psi	50 psi
#5)	5-50 psi	50 psi

Displacement Summary:

10ml & 20ml refer to foil thickness
all displacement values are in mils

Case	Point#1		Point#2		Point#3		Point#4	
	10ml	20ml	10ml	20ml	10ml	20ml	10ml	20ml
#1)	1.37	0.27	3.11	0.67	1.40	0.26	0.91	0.19
#2)	2.74	0.54	6.22	1.33	2.81	0.52	1.82	0.39
#3)	13.70	2.71	31.10	6.66	14.04	2.51	9.10	1.92
#4)	14.13	2.78	32.60	6.89	15.26	2.75	9.76	2.01
#5)	7.50	1.50	19.30	3.97	9.90	1.68	6.20	1.19

Stiffness:

all stiffness values are in lb/in

Case	Point#1		Point#2	
	10ml	20ml	10ml	20ml
#1)	3650	18520	1608	7508

Fig. 3-11 Dimensional Deflection and Stiffness for Various Compliant Support Elements at High Temperatures

On Figure 3-11, the rigid surface supported by the elastic trapazoidal member is exposed to a uniform pressure which was varied around the seal at face 1 and face 2. A total of five computer cases were studied considering various possible pressure distributions for 10 mil and 20 mil thick foils. The elastic deflection of the compliant support was solved via the finite element (ANSYS Computer Program) technique. The computed stiffness and deflection of various interest points are given in Figure 3-11. The displacement at point 4 is an indication for sizing the seal cartridge and displacements at points 1 and 3 is relevant to the support stiffness because it is assumed here that the top foil or seal ring surface will follow these locations under static or dynamic loadings.

Considering the elastic modulus of the compliant support made from Inconel X-750 at high temperatures to be 22 M psi, the calculated stiffness is 3650 lb/in and 18,520 lb/in for 10 and 20 mils thick foils, respectively. As can be seen from the results summarized in Figure 3-11, the axial stiffness of the seal is strongly a function of the compliant support foil thickness, T_h , where it denotes the flexural rigidity and is proportional to $1/T_h^3$.

The results of the numerical computation were utilized to size and design compliant support elements.

The numerical analysis was used as a design guide. The maximum applied axial seal loading and stress associated with this loading was one of the important factors, beside level of desirable compliancy in the flexure design, and was calculated with the use of the computer code ANSYS. Various factors such as foil thickness, physical property of the material at a higher temperature and flexure geometry were varied independently in the computer program to obtain an optimal configuration with the Inconel X-750 foil material.

The flexure foils were formed on a precision steel die utilizing rubber pad forming techniques. Many foils of various thickness, 5, 10 and stacked 15, and also 20 mils were formed in an annealed condition and accurate measurements were made to check the dimensions. The formed foils were then heat treated inside special heat-treatment fixtures, these fixtures controlled the parallelism, flatness and heat-treatment environment. The results of using these fabrication

techniques were excellent; heat treated compliant foils have better conformity in the seal ring cartridge and better quality as well as strength.

Load deflection tests were performed on each of the assembled compliant mount seal rings by setting the rigid surface seal on the flexure. Static load was then gradually applied through the center of the seal up to 82 lbs. The standard dead weight was then gradually unloaded.

Figures 3-12 through 3-14 show experimental load vs deflection for various thicknesses of the compliant support. There is excellent agreement between theory and experiment, except for the level of hysteresis which is associated directly with the level of the friction coefficient between the foil and cartridge.

As can be seen from the plots of load/deflection data, the slope of the curve flattens out around higher loads. This particular behavior is more obvious for the thinner foil than the thicker foil, which is an indication of the nonlinearity of this compliant concept and their load capability margin.

Compliant surfaces as shown on Fig. 3-8 and compliant mount seal components were made from stacks of INCX-750 with various thickness foils by the wire EDM process. After forming compliant mount foils with the precision die, foils were heat treated in a nitrogen atmosphere.

Significant dimensions of the seal components are as follows:

Rotor, outside diameter	2.86 in.
Rotor, thickness	0.375 in.
RING CONTACT FACE:	
OD	2.20 in.
ID	1.53 in.
Thickness	0.105 in.
Compliant Mount, ID	1.76 in.
Compliant Mount, OD	2.46 in.
Compliant Surface Seal, ID	1.75 in.
Compliant Surface, OD	2.5 in.
Compliant Surface Thickness	0.20 in.

Fig. 3-12 Load Vs. Deflection for 20 Mil Inconel X-750

Fig. 3-13 Load Vs. Deflection for 15 Mil Inconel X-750

Fig. 3-14 Load Vs. Deflection For 10 Mil Inconel X-750

4.0 TEST FACILITY

4.1 Test Facility and Instrumentation

An existing MTI-owned test rig was upgraded and modified to meet the specific requirements of this program. A cross section of the test rig with a test seal in place is shown in Figure 4-1. A part of the modifications for high-temperature testing was the hot end components of the rig itself, which are made from XF818, a cast material containing 55% iron, 18% nickel, 18% chromium, and 7.5% molybdenum. It is readily machinable and is presently used in some Stirling engine components. The shaft extension and end spring washer are made from Rene 41.

The variable speed electric motor driven high speed spindle was supported on two preloaded angular contact ball bearings. Both support bearings were standard Class 7 bearings. The bearing at the test end was a 107H (35 mm bore), and at the drive turbine end a 105H (25 mm bore).

Oil for lubricating and cooling the support ball bearings was supplied through two oil jets 180° apart at each bearing. The jets were sized to produce a jet velocity of approximately 20 m/sec (65 ft/sec) at $4.4 \times 10^5 \text{ N/m}^2$ (64 psig) supply pressure with 2.27 kg/min (5 lb/min) total oil flow. A water-cooled heat exchanger in the oil supply loop removed heat from the oil.

A double labyrinth seal with pressurized air supplied between the seals prevent the oil from traveling down the test shaft into the test seal area. During the high temperature tests, eight 500 watt quartz heaters were used to heat the test seal. A 10 mm (.250 inch) thick Mycalex 500 disk acted as an insulator between the quartz heater box and support housing. A water jacket in the support housing assisted in removing heat from the support ball bearings.

The air supply was provided by a three-stage centrifugal air compressor having the following maximum output characteristics:

Pressure:	$6.89 \times 10^5 \text{ N/m}^2$ (100 psig)
Flow:	22.66 m ³ /min (800 scfm)
Temperature:	93.3°C (200°F)

Fig. 4-1 Test Rig Cross-Section

A higher supply air temperature was achieved via an 800 watt heat exchanger (Pure Flow Air Heater) where air was preheated prior to entering the secondary heat exchanger (coiled tube inside quartz heater). The preheated air was utilized to establish hydraulic loading pressure to load the seal ring against the seal runner. Figures 4-2 and 4-3 show the rig and heaters prior to testing.

A constant ambient temperature could be maintained utilizing this preheated dry filtered air system. The rig has temperature capabilities to 816°C (1500°F) and variable speeds to a maximum of 12,000 rpm. A fixed test speed could be accurately maintained utilizing a variable frequency control system and a digital magnetic speed pickup.

Rotor balancing was performed on a conventional low-speed, two-plane "Dynamic Balancer", Model MU-6, manufactured by Micro-Balancing, Inc. The test shaft was first balanced as a detail while supported on teflon vee-blocks. The rotor was then completely assembled, installed in the ball bearing cartridge, and balanced on its own support bearings at 1750 rpm. Residual unbalance levels were 0.018 gm.cm (0.000250 oz.in.) in each of two planes. The test rig design allowed for the rotor assembly to be installed into the support housing without disrupting the assembly after balancing.

Rotational speed was measured by an MTI Fotonic SensorTM fiber optic probe which responds to the six times per revolution passing of dark bands established on the face of the drive end coupling. A Monsanto digital frequency counter Model #103A measured the output of the Fotonic Sensor speed pickup and provided a continuous digital display of rotor speed.

Radial and axial motions of the seal runner were measured using capacitance type displacement probes. Two sets of x-y probes were mounted on the base plate of the test rig to measure test runner motion relative to ground. The output of each probe was displayed on a cathode-ray oscilloscope. A fine tuning of the rotor resulted in 0.1 mil axial runout and a maximum of 0.08 mil radial orbital motion. This had been established prior to any testing.

A gas-tight enclosure surrounds the test seal and circulating fan. Pressurized air is introduced into this enclosure. The flow required to maintain pressure indicates the quantity of leakage through the seal. An additional source of

HIGH-TEMPERATURE, ROTATING-SHAFT TEST RIG

Fig. 4-2

CLOSE-UP OF HIGH-TEMPERATURE, ROTATING-SHAFT SEAL TEST RIG

Fig. 4-3

pressure could be applied behind the seal so that the axial load on the seal could be measured.

Pressure and flow into the gas-tight enclosure and to the underside of the seal ring, shaft speed, and horsepower input were the quantities measured. Approximately 689 kPa (100 psi) was available to be applied to both the gas-tight enclosure and the hydraulic loading region behind the seal ring.

Temperatures were measured by using thermocouples mounted at the following location:

- Two on heater chamber about 1" away from rotor (ambient chamber).
- One 2 mils away from outer edge of the seal ring and 20 mils from contact surface of runner.
- One at inlet air to hydraulic loading region.

The thermocouples were Type J, Iron-Constantan and temperatures were recorded on a Honeywell multipoint chart recorder. Standard Fisher-Porter flowmeters were used to measure the air flow entering the seal chamber and into the hydraulic loading region. The frictional torque measurement of the seal component was measured via a Watt transducer made by Scientific Columbus Model XL5C5-A2-1. Total and differential input power to the drive motor was displayed on an x-y plotter, against a time base.

The input power to the drive motor was recorded for various rotor speeds and seal loads. An increase in power intake was considered an indication of frictional power loss of the seal under load, speed and ambient temperature. All equipment used for the acquisition of test data was calibrated at the start of the test program. The equipment is periodically checked to insure its accuracy, stability, and repeatability at intervals recommended by the equipment's manufacturer.

The heater temperature control was automated to turn the heaters on and off. The temperature control unit was used to maintain the seal environment temperature to a desired level. A separate control panel monitored the variables such

as the rotor speed, sealed air pressure, flow, hydraulic loading pressure and air flow, temperature at the seal ring tip, temperature at the seal chamber, and room temperature. Heater temperature, and input power to the motor were measured and conveniently recorded on strip charts. The rotor speed and pressure differential across the seal as well as pressure in the hydraulic loading chamber were manually chosen at the control panel. Due to the limitations of the pressure regulator used, the sealed air pressure was variable up to 85 psig.

Figure 4-4 shows the instrumentation of the test facility and Figure 4-5 shows various components of the rigs and seals on the granite base block.

TEST RIG INSTRUMENTATION

Fig. 4-4

Fig. 4-5 Photo of High Temperature Rotating Shaft Seal Component and Test Rig

5.0 DYNAMIC TESTING

5.1 Test Preparation and Assembly

In preparation for the seal tests, the fully assembled rotor was balanced to a 0.002 oz-in accuracy. The seal parts were marked for assembly orientation. Balancing was done for the disk runner assembly since the first seal chosen to be tested was the two K162B seal runner arrangement.

Final run-out, squareness and displacement tests were performed after balancing to determine the level of dynamic misalignment. Interfacial contact surfaces of the seal runner and seal ring were checked and topographical as well as surface measurements were conducted and recorded, before and after each test. A maximum axial run-out was approximately 0.0002 in. displacement during operation (at room temperature). These tests established a baseline for the compliant mount operation and measurement of wear depth (wear factor) and other parameters.

5.2 Test Program

Seal performance is highly dependent on the mechanical, thermal, topographical, and tribological properties of the mating surfaces and sliding conditions. In dry friction operation, wear can be rapid, and wear is not always indicated by the measurement of the temperature and friction level. Face loading can also have dramatic effects. In some instances high face loading can cause very rapid wear and destroy the test specimen. When testing began, it soon became apparent that it would be necessary to proceed with caution and that frequent teardown and inspections would be required to determine wear rate and seal condition. A test plan that designated hours of testing could not be followed with any accuracy and was thus discarded early in the program.

The overall intent of the testing program was to try each compliance category of seal with various material combinations. This is indicated on Table 5-1, which was an early test plan. It is included to indicate the material combinations and seal configurations to be tested.

TABLE 5-1

ORIGINAL SEAL TEST PLAN

Configuration	T	M	H	M	H	M	H	C	I	
RS-RM	72	A	8	B	8			N	Y	
	750	↓	8	↓	8			N	Y	
	1300	↓	8	↓	8			N	Y	
RS-CM	72	A	8	B	8	C	8	N	Y	
	750	↓	8	↓	8	↓	8	N	Y	
	1300	↓	8	↓	8	↓	8	N	Y	
CS-CM						C	8	N	Y	144 hrs.
						↓	8	N	Y	
						↓	8	N	Y	
<hr/>										
RS-RM	72	A or	40					Y	Y (every 8 hrs.)	
	1300	B	40					Y	Y	
RS-CM	72	A, B	40					Y	Y	240 hrs.
	1300	or C	40					Y	Y	
CS-CM	72	C	40					Y	Y	344 hrs.
	1300	↓	40					Y	Y	Total Test Time

Legend:

T = Temperature °F

M = Material Combination

H = Hours

C = Contamination

I = Inspection

N = No

Y = Yes

RS-RM = Rigid Surface, Rigid Mount

RS-CM = Rigid Surface, Compliant Mount

CS-CM = Compliant Surface, Compliant Mount

A = K162B/K162B Runner / Seal Ring

B = K162B/PS200

C = K162B/Inconel X750

5.3 Test Results and Discussions

Rigid Surface, Rigid Mount Test

A seal test was initiated with a rigid surface, rigid mount (RS-RM) concept with K162B vs. K162B. The seal pressure was increased up to 35 psi and immeasurable leakage at static and room temperature was recorded. This was achieved via 80 psi air pressure acting on the seal ring where the face loading F of 124 lbs was determined from the pressure balancing equation given below:

$$F = 12.5684 P_h - 2.3273 P_s$$

where

P_h = hydraulic loading pressure (psig)

P_s = differential pressure across the seal
(seal chamber pressure) (psig)

$$F = 12.5684 (80) - 2.3273 (35)$$

$$F = 124 \text{ lbs}$$

A 10 minute run was conducted at room temperature with a rotor speed of 1810 rpm and a face load of about 75 lbs. to 76 lbs. Insignificant leakage at the running condition was experienced, but a ringing sound and some grinding noise suggested a need to disassemble the seal for examination. Visual examination of the seal ring showed considerable wear of both the runner and seal ring. Optical examination of the seal ring contact zone (wear area mostly penetrated towards ID zone rather than the OD area) indicated severe contact stresses, signs of delamination, heat check and microcracks oriented radially as shown in Figures 5-1 and 5-2. The wear pattern on the seal ring was also uneven, covering 50% of the circumferential length, which is an indication of misalignment of the ring relative to the runner. As can be seen from the photo of the seal runner on Figure 5-3, there is a transfer film and indication of severe wear marks. However, there was no indication of delamination nor microcracks but the wear could be identified as adhesion type wear. It should be mentioned that prior to test the mating surfaces were very clean and free of oxide film, smooth (CLA = 2-4 μ in.),

30X

Fig. 5-1 Photo of Seal Ring Surface (x30) after Test at RT at a Load of 124 lb, N = 1810 rpm (K162B)

Fig. 5-2 Photo of Seal Ring Surface (x45) After Test @ RT at Load of 124 lbs and $N = 1810$ rpm Showing Microcracks on O.D. Area Oriented Radially (K162B)

Fig. 5-3 Photo of Seal Runner Surface (x30) After Test @ RT
at Load of 124 lbs and N = 1810 rpm Showing
Transfer Film (K162B)

and the flatness was on the order of 2 light bands. After-test measurements indicated that thermal distortion caused dishing of both parts and was severe in the seal ring. Misalignment and radial crowning was due to thermal dishing which reduces the actual contact area drastically. The estimated hertzian maximum pressure was 138 ksi and the corresponding flash temperature was calculated to be 3490°F (flash temperature analysis in Appendix A). The degree of misalignment of the mating surfaces is a matter of perfection in manufacture and assembly of the entire test rig and parts as well as maintaining aligned surfaces at operating conditions. This appears to be not only a costly process but nearly impossible to achieve with the RS-RM seal concept which utilizes a hard material such as K162B. The inability of the RS-RM configuration to maintain even contact over the seal area makes it unattractive for use as an unlubricated seal. It was decided not to continue further testing of the RS-RM configuration.

Rigid Surface, Compliant Mount Tests

In the next run significant changes were made to enhance the interfacial condition by preoxidizing the seal runner and utilizing an INC 718 seal ring coated with MoS₂ in an annealed condition which is a soft material by comparison to K162B. A thin layer of oxide film was generated on the K162B runner by heating the specimen in an open oven for 4 hours at 1500°F. This was done to minimize adhesion wear and also to reduce the transitional coefficient of friction η , thus enhancing wear characteristics of the seal. During initial room temperature testing the seal chamber was not assembled. This test was done in order to be able to diagnose any erratic behavior and perhaps to prevent catastrophic failure of the rubbing parts at operating conditions by observing and closely monitoring the rubbing noise.

The rigid spacer was replaced with a 10 mil compliant support in order to achieve a rigid surface, compliant mount assembly.

At room temperature the rotor speed was brought up to 910 rpm then face loads of 2.5, 5, 7.5, 10 and 12.5 lbs were applied over a period of 30 minutes. Throughout the test operation, smooth rubbing noise could be heard. The rotor was stopped and the seal components were disassembled for inspection. The seal runner showed no wear marks, but the seal ring contact face had some smooth

burnished signs unevenly placed indicating some thermal distortion together with polishing of uneven MoS₂ coating (i.e., initial variation in coating thickness).

The test rig was reassembled and preparations were made to apply a higher load. At an operating speed of 910 rpm the load was gradually increased up to 26 lbs. over the test period of 60 minutes. The test components were examined at the end of this run and the results were satisfactory. There was mild wear which was difficult to measure and some detached oxide film in random patches, mostly from the inside diameter of the contact zone. The seal ring surfaces showed a slightly polished coating and some K162B transfer films mixed or embedded with MoS₂ coating. There was definite thermal distortion in the radial direction which produced localized contact wear (line contact).

The MoS₂ coating on the seal ring was reburnished with fine emery paper prior to conducting the test at a higher speed of 1810 rpm. At room temperature a face load of 26 lbs was applied. Duration of this test was only 5 minutes due to undesirable rub noise. The seal runner and seal ring were examined and photomicrographs of wear tracks were prepared. Figures 5-4(a) and 5-4(b) show photos of the seal components after test with 26 lbs at room temperature and a rotor speed of 1810 rpm. The photomicrographs of the worn area of the seal ring and runner are shown in Figures 5-5 and 5-6. The dry film coating on this run was smeared on the wear track of the runner (Figure 5-5) and severe wear was observed. Evidently under high contact pressure the MoS₂ coating along with other wear debris acted as an abrasive material and had damaged both surfaces. Deep scratches were detected at the ID side of the wear track, and talysurf traces of the seal runner after the test, together with flatness measurements, are shown in Figure 5-7(a) and 5-7(b), respectively. The flatness measurements were conducted with a Pneumo-Centric[®] 5500 Roundness Gage, and four traces were taken within the wear tracks. The talysurf traces were from the ID towards the OD (i.e. perpendicular to the direction of motion) at four places 90° apart.

Evidence of severe plastic flow and surface delamination could be observed under a high magnification microscope. Interestingly, reformation of the oxide film in either surface was apparently absent. Although the runner prior to this test was oxidized, as can be seen from Fig. 5-5, there are sections of flaked off

Fig. 5-4 (a) Photo of K162B Seal Runner after Test (N = 1810, F = 26 lb at RT)

Fig. 5-4 (b) Photo of INC718 with MoS₂ Coating Seal Ring after Test (N = 1810 rpm, F = 26 lb at RT)

(X30)

Fig. 5-5 Photo of K162B Runner After Test (W = 26 lbs,
N = 1810 rpm @ RT)

(X30)

Fig. 5-6 Photo of INC 718 Seal Ring Coated with MoS₂ After Test (W = 26 lb, N = 1810 rpm @ RT)

Vertical Mag: each small division = 5 μ in.
 Horizontal Mag: each small division = 2000 μ in.

Fig. 5-7 (a) Talysurf Traces (taken four places in radial direction) of K162B, SN/1 Seal Runner After Test with INC 718 Coated with MoS₂, RS-CM

Fig. 5-7 (b) Flatness Measurement of Seal Runner (K162B SN/1)
After Test, Average Flatness = 14 μ in.

oxide films (white color patches) indicating the removal of the oxide film under abrasive wear.

From the results of this run, it can be concluded that the MoS₂ coating might be a good candidate for low load sliding contact but apparently it is not a good dry lubricant against Kentanium K162B under high contact pressure.

Test No. 1

Significant rework was done on the seal components. The runner and ring (SN/2-K162B pair) were reground and lapped. The seal ring and runner were laid over a piece of flat Al₂O₃ ceramic sheet inside an oven. The oven temperature was maintained at about 1500°F for 5 hours in order to oxidize the surface of the runner and ring.

For the first part of this test a 10 mil compliant support assembly was utilized (Test No. 1, K162B/K162B SN/2). The test was conducted at room temperature with rotor speeds of 910 rpm and 1810 rpm and face loads of 15 lbs, 20 lbs, and 26 lbs for the test period of 146 minutes. At the end of this test period the runner and ring were examined and they were observed to be in excellent condition.

There was no evidence of wear that could be measured. The test was continued with a 5 mil compliant mount support at room temperature with a constant rotor speed of 1810 rpm for another 130 minutes and again a maximum load of 26 lbs was applied. Detailed run conditions and the computed wear factor for this two-part test is tabulated and is shown in Table 5-2. The wear factor was about 3.5×10^{-11} which is an indication of low wear rate.

The mating surfaces were examined and surfaces which were originally perfectly parallel assumed a wavy pattern after this brief running period. The presence of a double wave along the rubbing trajectory could be clearly seen on the seal ring and is also visible on the photograph of Fig. 5-8. In the valleys of the wave a trace of fine mustard-color debris was present.

TEST RESULT FEB. 3 & 4 '87 TAMB. = 75 F
 S/N2 .. K162B/K162B W/ 10 MIL FOR TEST # 1 - 4
 5 MIL " " # 5 - 7

O WEAR DEPTH = 0.101 MIL

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	6.00	910.00	30.00	3.059973E+06	
2	6.00	910.00	60.00	6.119947E+06	
3	10.00	910.00	30.00	5.099960E+06	
4	6.00	1810.00	26.00	5.274815E+06	
5	5.00	1810.00	30.00	5.071938E+06	
6	8.00	1815.00	30.00	8.137522E+06	
7	10.00	1815.00	70.00	2.373443E+07	
			-----	-----	
			276.00	5.649858E+07	Totals

ϕ = 3.524795E-11 CM3/CM-KG

TABLE 5-2

RESULTS OF TEST NO. 1

Fig. 5-8 Wear Patterns on the TiC Specimens

Test No. 2

After Test No. 1 the seal ring was lapped and a surface finish of about $6\text{ }\mu\text{ in.}$ with a flatness between $20\text{--}50\text{ }\mu\text{ in.}$ was obtained. The specimens were preoxidized and assembled with a 5 mil thick compliant mount.

At ambient temperature, rotor speed and face load varied up to 26 lbs at 3600 rpm; the total run time was 339 minutes. The temperature at the seal ring tip, T_0 , was monitored continuously. The temperature rise was more sensitive to the increase in face load than rotor speed.

Wear debris was collected as it was ejected from the seal's outer periphery. Special care was taken to separately collect those wear particles that were outside the zone and those which remained and filled the valleys and clearance spaces between the interacting surfaces. Those particles that were found outside the seal were black and shiny. Fig. 5-9 shows an SEM micrograph of some of the evidence. The particles collected from the inside seal area (between the seal ID and over the journal surfaces) had a dull dark brown color. Figure 5-11 is an SEM micrograph, X1000, of the wear particles that stayed in the interspace during operation and upon disassembly were found present on the surfaces (Type I). The larger chunks of wear material (Type II) that, ejected from the interface, were found outside the wear track are shown in Figures 5-9 and 5-10.

The small wear particles were, of course, found mostly in the two valleys generated on the wear specimen. A three-dimensional microscopic view of the wear debris shows the particles to be of nearly spherical shape. In order not to alter the chemical nature of the wear debris, the SEM conducted a rapid scan and a semi-quantitative analysis was performed on the Type I and Type II wear debris. The results of the analysis are shown on Figure 5-12 and 5-13 respectively. In this analysis, those elements which were analyzed have been marked, and a brief summary is given on the right hand side of Figures 5-12 and 5-13. The most interesting finding is that the fine particles (Type I), showed variation in size from $0.1\text{ }\mu\text{m}$ to $2.0\text{ }\mu\text{m}$, while some of the ejected particles could be measured up to $80\text{ }\mu\text{m}$.

The chemical elements of the Type I particles were mostly made up of Ti, Ni, and oxygen with no evidence of C, and the Type II debris was heavy and the only major

Fig. 5-9 SEM Micrograph (X475) of Wear Debris Found Outside Wear Track, Type II

Fig. 5-10 SEM Micrograph (X475) of Wear Particles Found Outside Wear Track, Type II, Ejected to ID Side

Fig. 5-11 SEM Micrograph (X1000) of Wear Particles Found Inside Wear Track, Type I

ELECTRON MICROPROBE ANALYSIS

Test Debris Particulate - Type I (Inside Interface)

PERIODIC TABLE OF THE ELEMENTS

CODE (Crystal Structure)

Cubic, Face Centered ☐ Rhombohedral ☐

Cubic, Body Centered ☐ Orthorhombic ☐

Cubic ☐ Monoclinic ☐

Diamond ☐ Tetragonal ☐

Semi Quantitative Results

Ti - heavy

Ni - heavy

O - moderate

B - light - moderate

Nb - light

Fe - light - trace

Co - light - trace, possibly contamination

Mo - light

Specimen: Type I debris particulate -
(brownish-yellow powder)

Operating Conditions: 4x10⁻⁸A
25Kv (12.5Kv - oxygen, boron, carbon)
1000 x, 2cm box, rapid scan

- ◊ - indicates peak above background but in matrix (mount pedestal)
- △ - indicates peak above background levels
- ∩ - indicates element was analyzed for but not found - no peak above background levels

Figure 5-12

ELECTRON MICROPROBE ANALYSIS

Test Debris Particulate - Type II (Outside Interface)

PERIODIC TABLE OF THE ELEMENTS

CODE (Crystal Structure)
 Cubic, Face Centered ☐ Rhombohedral ☐
 Cubic, Body Centered ☐ Monoclinic ☐
 Cubic ☐ Tetragonal ☐
 Diamond ☐

Semi Quantitative Results

C - heavy - only major element
 Ti - hotspots on particle surface
 Ni - not part of bulk
 Al - hotspots on particle surface
 - not part of bulk
 Fe - very light
 Nb - very light
 Mo - very light

Specimen: Type II debris particulate
 (shiny black particles)

Operating Conditions: $4 \times 10^{-8} \text{A}$
 25Kv (12.5Kv for O, B, C, N)
 1000x, 2cm box, rapid scan

- ☒ - indicates peak above background levels but in mount pedestal
- ☒ - indicates peak above background levels
- ☒ - indicates element was analyzed for but not found

Figure 5-13

element was found to be carbon and some hot spots on the particle surfaces were Ti and Ni.

It is perhaps the larger wear debris that helped manufacture the surface waves (surface tapers) which subsequently retained the small wear particles in the form of a powder lubricant that after the initial wear period facilitated the successful operation of the flat-surfaced seal specimen. Test results are shown on Table 5-3.

Test No. 3

Rub tests of the seal components were continued at a higher load and speed with an ambient temperature of 100°F using the same material and arrangement that was described in Test No. 2. Detailed test results are given in Table 5-4 and specific talysurf and flatness conditions are shown on Figures 5-14 through 5-20.

Figure 5-14 presents a schematic drawing of the seal runner and ring together with an exaggerated wear track on the runner and particular locations where measurements were taken, including depth measurements and surface profile traces. The depth measurements, together with wear track area, both were utilized throughout this program to compute wear factors for a given test result.

Talysurf traces at radial locations C, D and E (see Fig. 5-14) taking measurements in the circumferential direction are given in Fig. 5-15, indicating a mean average CLA of 23 μ in., where the surface profile measured in the radial direction indicated a mean average CLA of 19.5 μ in. (Fig. 5-16). However, surface profiles of the seal ring after test showed a mean average CLA of 17 μ in. and 20 μ in for the circumferential and radial directions, respectively (Figs. 5-17 and 5-18).

Both the runner and ring were checked for flatness at the wear track. Results are given on Fig. 5-19 and 5-20. Deviation from flatness for the runner and ring was 59 μ in. and 140 μ in. respectively with a peculiar saddle shape.

TEST RESULTS OF FEB. 5 '87 TAMB = 75 F
K162B / K162B W / 5 MIL C.M.

O WEAR DEPTH = 0.300 MIL (SEAL RING&RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	5.00	2400.00	165.00	3.698872E+07	
2	5.00	3000.00	33.00	9.247181E+06	
3	5.00	3600.00	30.00	1.008784E+07	
4	10.00	2400.00	111.00	4.976666E+07	
			339.00	1.060904E+08	Totals

$$\phi = 5.631408E-11 \text{ cm}^3/\text{cm-kg}$$

TABLE 5-3

RESULTS OF TEST NO. 2

TEST RESULTS OF FEB. 6 '87 TAMB = 100 F
K162B / K162B W / 5 MIL C.M.

0 WEAR DEPTH = 0.300 MIL (SEAL S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	5.00	2400.00	60.00	1.345045E+07	
2	10.00	2400.00	75.00	3.362611E+07	
3	10.00	3000.00	30.00	1.681306E+07	
4	10.00	3600.00	30.00	2.017566E+07	
5	10.00	2400.00	15.00	6.725224E+06	
6	15.00	2400.00	60.00	4.035133E+07	
-----				-----	
270.00				1.311418E+08	Totals

$$\phi = 4.555664E-11 \text{ cm}^3/\text{cm-kg}$$

TEST RESULTS OF FEB. 6 '87 TAMB = 100 F
K162B / K162B W / 5 MIL C.M.

0 WEAR DEPTH = 0.305 MIL (RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	5.00	2400.00	60.00	1.345045E+07	
2	10.00	2400.00	75.00	3.362611E+07	
3	10.00	3000.00	30.00	1.681306E+07	
4	10.00	3600.00	30.00	2.017566E+07	
5	10.00	2400.00	15.00	6.725224E+06	
6	15.00	2400.00	60.00	4.035133E+07	
-----				-----	
270.00				1.311418E+08	Totals

$$\phi = 4.607963E-11 \text{ cm}^3/\text{cm-kg}$$

TABLE 5-4

RESULTS OF TEST NO. 3

K162B/K162B

Fig. 5-14 Schematic Drawing of Seal Components
Wear Depth Data After Test No. 3

At Location C, CLA = 22 μ in.

At Location D, CLA = 23 μ in.

At Location E, CLA = 24 μ in.

Vertical Mag. 5000, 20 μ in./dev, Horizontal Mag = 20)

Fig. 5-15 Circumferential Talysurf Traces of Seal Runner
K162B, SN/2 After Test No. 3

Fig. 5-16 Radial Talysurf Traces of Seal Runner K162B, SN/2 After Test No. 3

At Location A, CLA = 18 μ in.

At Location B, CLA = 16 μ in.

At Location C, CLA = 18 μ in.

Fig. 5-17 Circumferential Talysurf Traces of Seal Ring K162B,
SN/2 After Test No. 3

CLA = 20 μ in.

CLA = 20 μ in

CLA = 21 μ in.

Vertical Magnification 2000, 50 μ in., Horizontal Magnification 20)

Fig. 5-18 Radial Talysurf Traces (120° Apart) of Seal Ring K162B, SN/2 After Test No. 3

Fig. 5-19 Deviation from Flatness of Seal Runner
K162B, SN/2 After Test No. 3

Fig. 5-20 Deviation From Flatness of Seal Ring K162B, SN/2
After Test No. 3

Test No. 4

During this test a higher load of 38.5 lbs at room temperature was applied to the seal components (K162, SN/2) at various rotor speeds ($N = 2400$ to 3600 rpm). The ambient temperature varied from 75°F up to 100°F . The inlet air temperature to the hydraulic loader was kept constant at 75°F and the temperature at the outer periphery of the seal ring was monitored (a maximum temperature of 340°F was recorded).

Table 5-5 summarizes the detailed test conditions and test results. The estimated wear factor for this run showed no change relative to the results of Test No. 3. Examination of the surface profile (wear tracks) of the runner showed no significant changes (mean avg. CLA = $19\text{ }\mu\text{in.}$) and talysurf traces of the seal ring after tests are shown in Fig. 5-21 (mean avg. CLA = 20 and $16\text{ }\mu\text{in.}$). The seal ring flatness measurements are shown in Fig. 5-22, indicating flatness of the order of $280\text{ }\mu\text{in.}$. While its deviation from flatness relative to run No. 3 has doubled, the saddle shape remained almost unchanged.

Typical data illustrating a characteristic pattern of friction and temperature versus speed was obtained. This pattern is shown in Fig. 5-23. The seal interface temperature is a function of contact pressure and relative speed, and Fig. 5-23 shows a drop in coefficient of friction η with speed and temperature. If one views the increase in temperature and speed as being primarily responsible for an increase in wear, then the drop in η can be associated with the lubricating function of the debris particles (titanium oxide powder which was analyzed after Test No. 2).

Tests No. 5 and 6

Testing of the K162B runner versus the compliant mount K162B seal ring (SN/2) was continued at an elevated ambient temperature (Test No. 5 at a maximum temperature of 500°F). After 60 minutes the seal ring outer periphery temperature (T_{out}) indicated a continuous rise over the oven temperature and after 160 minutes of testing T_{out} reached 590°F . Test results are given in Table 5-6. Wear factors showed an increase of 28% relative to the previous run (compared with Test No. 4). The surface profile and talysurf traces showed no significant changes after test No. 5. There was, however, a significant amount of debris

TEST RESULTS OF FEB.13 '87 TAMB = 100 F , TOUT = 280. F
K162B / K162B W / 5 MIL C.M.

0 WEAR DEPTH = 0.300 MIL (SEAL S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	15.00	2400.00	27.50	1.849435E+07	
2	15.00	2700.00	10.00	7.565874E+06	
3	15.00	3000.00	20.00	1.681306E+07	
4	15.00	3300.00	30.00	2.774154E+07	
5	15.00	3600.00	60.00	6.052702E+07	
			147.50	1.311418E+08	Totals

ϕ = 4.555664E-11 cm3/cm-kg

TEST RESULTS OF FEB.13 '87 TAMB = 100 F , TOUT = 280. F

K162B / K162B W / 5 MIL C.M.

0 WEAR DEPTH = 0.310 MIL (RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	15.00	2400.00	27.50	1.849435E+07	
2	15.00	2700.00	10.00	7.565874E+06	
3	15.00	3000.00	20.00	1.681306E+07	
4	15.00	3300.00	30.00	2.774154E+07	
5	15.00	3600.00	60.00	6.052702E+07	
			147.50	1.311418E+08	Totals

ϕ = 4.707519E-11 cm3/cm-kg

TABLE 5-5

RESULTS OF TEST NO. 4

CLA = 26 μ in.
(measured in radial direction)

CLA = 16 μ in.
(measured in circumferential direction)

Vertical Magnification 5000, 20 μ in./dev., Horizontal Mag 20

Fig. 5-21 Talysurf Traces of Seal Ring K162B, SN/2 After Test No. 4

Fig. 5-22 Deviation From Flatness of Seal Ring K162B,
SN/2 After Test No. 4 (Unit μ in.)

Fig. 5-23 Friction Coefficient as Function of Speed
for Material K162B against K162B

TEST RESULTS OF FEB. 19 '87 TAMB = 500 F.
K162B / K162B; W / 5 MIL C.M.

O WEAR DEPTH = 0.333 MIL (SEAL NOSE INSERT S/N2.)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	15.00	2400.00	75.00	5.043917E+07	
2	15.00	2700.00	25.00	1.891467E+07	
3	15.00	3000.00	20.00	1.681306E+07	
4	15.00	3300.00	20.00	1.849435E+07	
5	15.00	3600.00	20.00	2.017566E+07	
			160.00	1.248369E+08	Totals

$\phi = 5.312181E-11$ cm³/cm-kg

TEST RESULTS OF FEB. 19 '87 TAMB = 500 F.
K162B / K162B; W / 5 MIL C.M.

O WEAR DEPTH = 0.403 MIL (RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	15.00	2400.00	75.00	5.043917E+07	
2	15.00	2700.00	25.00	1.891467E+07	
3	15.00	3000.00	20.00	1.681306E+07	
4	15.00	3300.00	20.00	1.849435E+07	
5	15.00	3600.00	20.00	2.017566E+07	
			160.00	1.248369E+08	Totals

$\phi = 6.428853E-11$ cm³/cm-kg

TABLE 5-6

RESULTS OF TEST NO. 5

gathered outside and beneath the seal ring and over the compliant support. Again the powder which was found inside the wear track was very fine mustard-colored oxide debris; and the debris outside the contact area had shiny black chunky particles with a considerable large size particulate among them (some measured up to 4 μ in. in length as shown in Figure 5-24).

Next, the same test arrangement was prepared (Test No. 6) and tested at a maximum temperature of 750°F for 287 min. The face loading was kept constant at 41 lbs. at a constant rotor speed of 3600 rpm. Test results are shown in Table 5-7. Wear factors for the runner and ring were computed after this run and found to be $3. \times 10^{-11}$ and 5×10^{-11} respectively, which indicates a drop in the wear factor relative to the previous run by 31%.

Talysurf traces of the seal ring and runner after tests are shown in Fig. 5-25(a) and (b). The surface profiles of wear tracks were virtually mirror images of each other (Fig. 5-25). The polishing of the outer rim of the wear track in the high temperature tests was much more pronounced than in the low temperature tests.

Tests No. 7 and 8

The significant changes for these two tests were: a) the compliant mount was chosen to be 10 mils for high temperature strength and, b) the inlet air temperature to the hydraulic loading chamber was pre-heated during the tests. The rotor speed was kept constant during the tests at 3600 rpm and the load was varied from 51 lbs to 90 lbs. These tests were conducted at an ambient temperature of 1200°F. Test results were extremely good and the wear rate showed a decrease as the ambient temperature was increased. Detailed test conditions together with test results are given in Tables 5-8 and 5-9.

Talysurf traces of the seal runner and ring after Test No. 7 are on Figure 5-26.

The surface roughness appeared to be constant (CLA = 20 μ in.) after these tests at the specified high temperatures (up to 1200°F). The temperature near the outer periphery of the seal recorded the maximum temperature of 1220°F. The seal ring thickness after test was measured and tabulated as shown in Fig. 5-27.

X125

Fig. 5-24 SN/2 Type II Exterior Particulates
After Test No. 6

TEST RESULTS OF FEB. 25 '87 TAMB = 750 F
K162B / K162B W / 5 MIL C.M.

O WEAR DEPTH = 0.775 MIL (SEAL S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	16.00	3600.00	50.00	5.380174E+07	
2	16.00	3600.00	50.00	5.380174E+07	
3	16.00	3600.00	50.00	5.380174E+07	
4	16.00	3600.00	50.00	5.380174E+07	
5	16.00	3600.00	10.00	1.076035E+07	
6	16.00	3600.00	47.00	5.057363E+07	
7	16.00	3600.00	10.00	1.076035E+07	
8	16.00	3600.00	20.00	2.152069E+07	

287.00				3.088218E+08	Totals

$$\phi = 4.997645E-11 \text{ cm}^3/\text{cm-kg}$$

K162B / K162B W / 5 MIL C.M.

O WEAR DEPTH = 0.485 MIL (RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	16.00	3600.00	50.00	5.380174E+07	
2	16.00	3600.00	50.00	5.380174E+07	
3	16.00	3600.00	50.00	5.380174E+07	
4	16.00	3600.00	50.00	5.380174E+07	
5	16.00	3600.00	10.00	1.076035E+07	
6	16.00	3600.00	47.00	5.057363E+07	
7	16.00	3600.00	10.00	1.076035E+07	
8	16.00	3600.00	20.00	2.152069E+07	

287.00				3.088218E+08	Totals

$$\phi = 3.127558E-11 \text{ cm}^3/\text{cm-kg}$$

TABLE 5-7

RESULTS OF TEST NO. 6

a) Seal Ring

b) Seal Runner

Fig. 5-25 Radial Talysurf Traces of Seal Component After Test No. 6
(K162B vs. K162B, SN/2 at $T_{amb} = 500^{\circ}\text{F}$).

TEST RESULTS OF MAR. 9 '87 TAMB = 1200 F
K162B / K162B W / 10 MIL C.M.

0 WEAR DEPTH = 0.800 MIL (SEAL S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	20.00	3600.00	330.00	4.438643E+08	
			330.00	4.438643E+08	Totals

ϕ = 3.589314E-11 cm3/cm-kg

0 Wear Depth = 0.800 mil

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	20.00	3600.00	50.00	6.725219E+07	
2	20.00	3600.00	50.00	6.725219E+07	
3	20.00	3600.00	40.00	5.380176E+07	
4	20.00	3600.00	20.00	2.690088E+07	
5	20.00	3600.00	90.00	1.210540E+08	
6	20.00	3600.00	30.00	4.035131E+07	
7	20.00	3600.00	20.00	2.690088E+07	
8	20.00	3600.00	10.00	1.345044E+07	
9	20.00	3600.00	20.00	2.690088E+07	
			330.00	4.438638E+08	Totals

ϕ = 3.589318E-11 cm3/cm-kg

K162B / K162B W / 10 MIL C.M.
TEST RESULTS OF MAR. 9 '87 TAMB = 1200 F

0 WEAR DEPTH = 0.850 MIL (RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	20.00	3600.00	330.00	4.438643E+08	
			330.00	4.438643E+08	Totals

ϕ = 3.813647E-11 cm3/cm-kg

TABLE 5-8

RESULTS OF TEST NO. 7

TEST RESULTS OF MAR. 11 '87 TAMB = 1200 F
K162B / K162B W / 10 MIL C.M.

0 WEAR DEPTH = 1.240 MIL (SEAL S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	35.00	3600.00	280.00	6.590717E+08	
			280.00	6.590717E+08	Totals

$$\phi = 3.746801E-11 \text{ cm}^3/\text{cm-kg}$$

0 Wear Depth = 1.240 mil

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	35.00	3600.00	20.00	4.707656E+07	
2	35.00	3600.00	20.00	4.707656E+07	
3	35.00	3600.00	30.00	7.061483E+07	
4	35.00	3600.00	30.00	7.061483E+07	
5	35.00	3600.00	20.00	4.707656E+07	
6	35.00	3600.00	60.00	1.412297E+08	
7	35.00	3600.00	30.00	7.061483E+07	
8	35.00	3600.00	20.00	4.707656E+07	
9	35.00	3600.00	10.00	2.353827E+07	
10	35.00	3600.00	20.00	4.707656E+07	
11	35.00	3600.00	20.00	4.707656E+07	
			280.00	6.590717E+08	Totals

$$\phi = 3.746801E-11 \text{ cm}^3/\text{cm-kg}$$

TEST RESULTS OF MAR. 11 '87 TAMB = 1200 F
K162B / K162B W / 10 MIL C.M.

0 WEAR DEPTH = 0.900 MIL (RUNNER S/N2)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	35.00	3600.00	280.00	6.590717E+08	
			280.00	6.590717E+08	Totals

$$\phi = 2.719452E-11 \text{ cm}^3/\text{cm-kg}$$

TABLE 5-9

RESULTS OF TEST NO. 8

SEAL RUNNER

SEAL RING

a) After Test No. 7, and before Test No. 8

SEAL RING

b) After Test No. 8

Fig. 5-26 Radial Talysurf Traces of Seal Components K162B/K162B with 10 Mil CM

Pos	0	90	180	270
A	.1027	.1031	.1012	.1007
B	.1027	.10315	.1017	.1010
C	.1027	.1028	.1015	.1002

a) After Test No. 7, Before Test No. 8

Pos	0°	90°	180	270
A	.0998	.1003	.1008	.1011
B	.1002	.1010	.1012	.1012
C	.100	.1008	.1010	.1012

b) After Test No. 8

Fig. 5-27 Variation of Thickness, Radially and Circumferentially
Before and After Test (K162B/K162B with 10 Mil CM, SN/2)

Several repetitious measurements indicated a radially crowned wear pattern with an average crown height of 400 μ in. and an average waviness of 1.2 mil.

The collected wear debris at the end of each test was virtually unchanged. The results were obtained with the K162B/K162B combination and indicate that oxidation of the substrate, or of the oxide coated surfaces or complex oxide formation, is also influencing the performance of the seal. A characteristic pattern of friction and wear rate versus temperature was obtained. Typical data illustrating this pattern are shown in Figures 5-28 and 5-29. The results show that the lowest coefficients of friction were obtained at 1200°F. The wear rate also followed the same trend of η . When the ambient was cooled to an intermediate temperature or to room ambient there was a rise in friction to values of 0.6. Each time the test temperature was cycled from cold to hot or vice versa this frictional pattern was repeated. Results gathered from Test No. 2 through Test No. 8 were correlated against the multiple of load, F, and the travel distance, L, are presented in Figure 5-29. From this figure it can be concluded that wear rate and the coefficient of friction of K162B vs K162B is inversely proportional to ambient temperature.

Repeatability Test with K162B; SN/1

Tests of rigid surface-compliant mounted seals made of K162B that were preoxidized followed. This set of seal components (SN/1) with compliant mount INC X750, 5 or 10 mils thick was chosen to run at various temperatures, speeds and load to determine repeatability of the good performance of this particular seal concept (RS-CM). Prior to assembly the seal runner and ring were checked for their surface integrity, flatness, and roughness levels. It should be mentioned here that this set of seal rubbing components called SN/1 has been tested before under the conditions of rigid surface and rigid mounted seal concept. They were then re-ground, lapped and pre-oxidized. Via lapping a 14 and 46 μ in. flatness was achieved for the seal runner and ring respectively. Talysurf traces indicated a CLA of 4 μ in. and 6 μ in. finish for the runner and ring surfaces. The following tests (No. 9 through No. 12) were conducted for their determination of tribological characteristics at room temperature; therefore the seal chamber was omitted from the assembly.

Fig. 5-28 Relation Between η and Rate of Wear

Fig. 5-29 Relation Between Temp, η , and Rate of Wear as a Function of ΣFL

Test No. 9

The seal rubbing components were assembled with a 5 mil thick compliant mount for the room temperature test. The starting rotor speed was set at 2400 rpm with an increment of 600 rpm. Face loadings of 26 lbs and 38.5 lbs over the test period of 150 minutes were applied. Temperature and torque levels were monitored and test results were compared against SN/2 under the same test operating conditions. Detailed test conditions and results are given in Table 5-10. The wear factor resulting from this test (3.5×10^{-11}) $\text{cm}^3/\text{cm-kg}$ was compared against Test No. 1 and the repeatability level was excellent. The seal ring had waviness of 500 $\mu\text{in.}$ to 600 $\mu\text{in.}$ with a roughness level of 20 $\mu\text{in.}$. The seal runner wear track surface was also examined and an average CLA of 19 $\mu\text{in.}$ was recorded.

Test No. 10

The seal components SN/1 was assembled with a 10 mil thick compliant mount support to conduct a high temperature test.

Temperature and friction under various loads for a constant speed of 3600 rpm over a test period of 450 minutes is given in Fig. 5-30. The results show that the lowest coefficients of friction (0.12 - 0.15) were obtained at 800°F. When the test was cooled to an intermediate temperature or to near room ambient, there was a marked rise in friction to values of 0.27 to 0.28. Each time the test temperature was cycled from hot to cold this frictional pattern was repeated.

Table 5-11 illustrates test conditions and test results. The average wear rate was found to be 5.6×10^{-11} $\text{cm}^3/\text{cm-kg}$ which is comparable with the results of Test No. 5 and 6. At the end of this test period a slight amount of face load (8 lbs.) was applied and the rotor was rapidly decelerated to a complete stop to capture the intermittent, powdery debris which is believed to act as a lubricant. A thin layer of light brown-yellowish fine powder could be seen in two distinct sectors (valleys) of the contact faces. The remaining contact areas which were not covered with the powder was dark with some shiny spots. Examination of the surface profile has indicated a maximum peak-to-peak height for

TEST RESULTS FEB. 18 '87 TAMB = 72 F
(S/N1) K162B / K162B W/ 5 MIL C.M.

O WEAR DEPTH = 0.197 MIL (RUNNER)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)
1	10.00	2400.00	30.00	1.345045E+07
2	15.00	2400.00	30.00	2.017566E+07
3	15.00	3000.00	30.00	2.521958E+07
4	15.00	3300.00	30.00	2.774154E+07
5	15.00	3600.00	30.00	3.026350E+07

150.00 1.168507E+08 Totals

$\phi = 3.357425E-11 \text{ cm}^3/\text{cm-kg}$

TEST RESULTS FEB. 18 '87 TAMB = 72 F
(S/N1) K162B / K162B W/ 5 MIL C.M.

O WEAR DEPTH = 0.197 MIL (SEAL NOSE INSERT)

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)
1	10.00	2400.00	30.00	1.345045E+07
2	15.00	2400.00	30.00	2.017566E+07
3	15.00	3000.00	30.00	2.521958E+07
4	15.00	3300.00	30.00	2.774154E+07
5	15.00	3600.00	30.00	3.026350E+07

150.00 1.168507E+08 Totals

$\phi = 3.350610E-11 \text{ cm}^3/\text{cm-kg}$

TABLE 5-10

RESULTS OF TEST NO. 9

Fig. 5-30 Friction and Temperature vs Time, Test No. 10

TEST RESULTS OF MAR. 5 '87 TAMB = 850. F
K162B / K162B W / 10 MIL C.M.

O WEAR DEPTH = 1.300 MIL (SEAL S/N1)**

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	16.00	3600.00	70.00	7.532245E+07	
2	16.00	3600.00	70.00	7.532245E+07	
3	16.00	3600.00	47.00	5.057363E+07	
4	20.00	3600.00	173.00	2.326926E+08	
5	20.00	3600.00	5.00	6.725220E+06	
6	20.00	3600.00	10.00	1.345044E+07	
7	20.00	3600.00	10.00	1.345044E+07	
8	3.00	3600.00	10.00	2.017567E+06	
9	3.00	3600.00	10.00	2.017567E+06	
			405.00	4.715717E+08	Totals

$\phi = 5.489938E-11$ cm³/cm-kg

TEST RESULTS OF MAR. 5 '87 TAMB = 850. F
K162B / K162B W / 10 MIL C.M.

O WEAR DEPTH = 1.380 MIL (RUNNER S/N1)**

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	16.00	3600.00	70.00	7.532245E+07	
2	16.00	3600.00	70.00	7.532245E+07	
3	16.00	3600.00	47.00	5.057363E+07	
4	20.00	3600.00	173.00	2.326926E+08	
5	20.00	3600.00	5.00	6.725220E+06	
6	20.00	3600.00	10.00	1.345044E+07	
7	20.00	3600.00	10.00	1.345044E+07	
8	3.00	3600.00	10.00	2.017567E+06	
9	3.00	3600.00	10.00	2.017567E+06	
			405.00	4.715717E+08	Totals

$\phi = 5.827783E-11$ CM³/CM-KG

TABLE 5-11

RESULTS OF TEST NO. 10

waviness of about 1.8 mils. Talysurf traces of the wear track of the seal ring and runner after tests are shown in Figure 5-31.

Test No. 11 and 12

Test No. 11 was conducted with K162B seal ring and runner using 10 mil compliant support as well as preheating the air inlet to the hydraulic loading chamber of the seal ring arrangement. The goal of this test was to achieve the highest ambient temperature with the existing heater arrangement. At a constant rotor speed of 3600 rpm the ambient temperature reached 1200°F over a 140 minute test period with a face load of 51 lbs. A maximum ambient temperature of 1275°F was measured while T_0 was indicating 1315°F at a face loading of 64 lbs. Detailed test conditions are given in Table 5-12.

After 310 minutes of testing, the parts were examined, a wear rate of 2.8×10^{-11} to 3.0×10^{-11} cm³/cm/kg was the result of this run which is the lowest wear rate ever found with K162B pairs. A three-dimensional inspection of the seal ring and runner revealed a mean average waviness of 1.8 mil and 250 μin. respectively. Talysurf traces of wear track surfaces of the ring and runner indicated roughness levels of 20 to 24 μin..

The results of Test No. 11 were encouraging and an average wear factor of 2.84×10^{-11} was an indicator of the good wear resistance characteristics of K162B at a higher ambient temperature.

The next test (No. 12) consisted of the same arrangement as Test No. 11 except the ambient temperature was set for about 800°F with a maximum face loading of 90 lbs. (see Table 5-13). The test duration was 540 minutes and the average wear factor was found to be 3.4×10^{-11} cm³/cm/kg. Talysurf traces of worn (contact) surfaces in the rotational direction is given in Figs. 5-32 and 5-33. As can be seen from Figure 5-33, the existence of convergent and divergent profiles (which was referred to previously as a waviness) is evident. A three-dimensional inspection was conducted on runner and ring and a mean average waviness was 0.25 mil and 1.85 mil and radial distortion (dishing) levels of 0.2 mil and 0.5 mil respectively.

Since Test No. 12 was the final run for this series of evaluations of the K162B/K162B seal components, a thorough surface analysis and metallurgical evaluation was conducted. The summary of the investigation and the findings together with photomicrographs are presented in Appendix B.

a) Seal Ring

b) Seal Runner

Fig. 5-31 Radial Talysurf Traces of Seal Components K162B/K162B
After Test No. 10

TEST RESULTS OF MAR. 10 '87 TAMB = 1275. F
K162B / K162B W / 10 MIL C.M.

O WEAR DEPTH = 0.615 MIL (SEAL S/N1)**

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	20.00	3600.00	60.00	8.070264E+07	
2	20.00	3600.00	50.00	6.725219E+07	
3	20.00	3600.00	50.00	6.725219E+07	
4	20.00	3600.00	50.00	6.725219E+07	
5	25.00	3600.00	30.00	5.043914E+07	
6	20.00	3600.00	20.00	2.690088E+07	
7	25.00	3600.00	40.00	6.725219E+07	
8	25.00	3600.00	10.00	1.681304E+07	
			310.00	4.438636E+08	Totals

$$\phi = 2.759289E-11 \text{ cm}^3/\text{cm-kg}$$

TEST RESULTS OF MAR. 10 '87 TAMB = 1275. F
K162B / K162B W / 10 MIL C.M.

O WEAR DEPTH = 0.670 MIL (RUNNER SN/1)**

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	20.00	3600.00	60.00	8.070264E+07	
2	20.00	3600.00	50.00	6.725219E+07	
3	20.00	3600.00	50.00	6.725219E+07	
4	20.00	3600.00	50.00	6.725219E+07	
5	25.00	3600.00	30.00	5.043914E+07	
6	20.00	3600.00	20.00	2.690088E+07	
7	25.00	3600.00	40.00	6.725219E+07	
8	25.00	3600.00	10.00	1.681304E+07	
			310.00	4.438636E+08	Totals

$$\phi = 3.024183E-11 \text{ CM}^3/\text{CM-KG}$$

TABLE 5-12

RESULTS OF TEST NO. 11

TEST RESULTS OF MAR. 13 '87 TAMB = 775- 750 F.
K162B / K162B W / 10 MIL C.M.

O WEAR DEPTH = 2.185 MIL (SEAL S/N1)**

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	35.00	3600.00	20.00	4.707656E+07	
2	35.00	3600.00	30.00	7.061483E+07	
3	35.00	3600.00	70.00	1.647680E+08	
4	35.00	3600.00	60.00	1.412297E+08	
5	35.00	3600.00	60.00	1.412297E+08	
6	35.00	3600.00	100.00	2.353828E+08	
7	35.00	3600.00	60.00	1.412297E+08	
8	35.00	3600.00	80.00	1.883062E+08	
9	20.00	3600.00	20.00	2.690088E+07	
10	20.00	3600.00	10.00	1.345044E+07	
11	20.00	3600.00	10.00	1.345044E+07	
12	20.00	3600.00	10.00	1.345044E+07	
13	20.00	3600.00	10.00	1.345044E+07	
			540.00	1.210539E+09	Totals

ϕ = 3.594548E-11 cm³/cm-kg

TEST RESULTS OF MAR. 13 '87 TAMB = 775- 750 F.
K162B / K162B W / 10 MIL C.M.

O WEAR DEPTH = 2.000 MIL (RUNNER SN/1)**

INDX#	Load (psi)	RPM	Time (min)	LF (cm-kg)	
1	35.00	3600.00	20.00	4.707656E+07	
2	35.00	3600.00	30.00	7.061483E+07	
3	35.00	3600.00	70.00	1.647680E+08	
4	35.00	3600.00	60.00	1.412297E+08	
5	35.00	3600.00	60.00	1.412297E+08	
6	35.00	3600.00	100.00	2.353828E+08	
7	35.00	3600.00	60.00	1.412297E+08	
8	35.00	3600.00	80.00	1.883062E+08	
9	20.00	3600.00	20.00	2.690088E+07	
10	20.00	3600.00	10.00	1.345044E+07	
11	20.00	3600.00	10.00	1.345044E+07	
12	20.00	3600.00	10.00	1.345044E+07	
13	20.00	3600.00	10.00	1.345044E+07	
			540.00	1.210539E+09	Totals

ϕ = 3.286554E-11 cm³/cm-kg

TABLE 5-13

RESULTS OF TEST NO. 12

Fig. 5-32 Circumferential Talysurf Traces of Seal Runner After Test No. 12

Fig. 5-33 Circumferential Talysurf Traces (Flatness Traces) Showing Wear Patterns in K162B Cermet Seal Ring Run Against K162B Cermet

6.0 CHEMICAL COMPATIBILITY TESTING

The effects of high temperature gases containing sulfur compounds on the selected materials, was described as a potential problem area by material vendors and personnel of the Oak Ridge National Laboratory. Some of the potential candidate materials with good tribological properties were selected for static tests (thermal and chemical compatibility tests).

The objective of this experiment was to determine the effect of hydrogen sulfide gas on ceramic, metal bonded carbide, and nickel base alloy samples at elevated temperatures of 1500°F and 1850°F. It was intended as a limited simulation of a coal gasification reactor environment.

The results of a short time exposure test of the following materials to 2% H₂S in methane gas at elevated temperatures will be discussed in this section of the report.

6.1 Material Specifications

The following sample materials were tested - nominal compositions are as follows:

1. **Sialon**; SiN, Al₂O₃
2. **K162B**; TiC, Ni-Mo binder
3. **KT150**; TiC, TiN (60%), WC, MoC (20%), Co, Ni binder (10%)
4. **Carboloy**; TiC, WC, Co, Ni
5. **In 718**; Ni-62%, Cr-19%, Fe-18.5%, Mo-3%, Cb+Ta-5%, Ti-.9%, Al-.5%

6.2 Apparatus and Procedure

The following experimental apparatus and procedure were devised and utilized to expose the selected samples to a free flowing gaseous environment of 2% hydrogen

sulfide in methane (Airco) at elevated temperatures, (1500°F, 1850°F) and at near-atmospheric pressure.

6.2.1 Apparatus

A gas tight ceramic tube (Omegatite 450, 99.8% Al_2O_3 , 16" long x 3/4" I.D. x 1" O.D.) was placed in the cylindrical cavity of a hinged high temperature furnace (Lindberg Hevi-Duty, Mod. No. 55035-A, max. temp. 1850°F) as shown in Figures 6-1 through 6-4. Gas inflow and outflow lines (1/4" tubing, 316SS), were connected to opposite ends of the ceramic tube via fittings (316SS) secured to the tube by a moldable ceramic compound, (Cotronics Corp. 360-M), see Figure 6-2. Temperature monitoring was accomplished by two (2) type K thermocouples (Omega, accuracy $\pm 2.2^\circ\text{C}$, sheathed with 304SS) positioned as shown in Figure 6-4. Thermocouple readout was by an Omega digital thermometer with 1°C resolution. One thermocouple was at the center of the furnace, the other approximately 2 inches from the center, both being adjacent to the tube exterior. Initially a tube interior/exterior temperature correlation was made with one thermocouple inserted to the midpoint inside the tube. The corresponding interior temperature was found to be 20°F lower at test point temperatures. Exiting hot gas was cooled by passing the outflow tube through a constant flowing cold water bath. The entire apparatus, excluding the gas supply bottle, was located in an active exhaust fume hood.

6.3 Procedure

Initially all sample weights were measured on a Sartorius analytical balance. Also all samples were cleaned in acetone, rinsed with water and blown dry with compressed freon. For each of four test runs, three selected samples were placed in a ceramic boat (Norton Alundum, Al_2O_3 , 3" long, 1/2" wide, 1/2" height). The samples were small enough, approximately 1/2" to 3/4" long, such that 3 could easily fit into the boat. The boat with samples was inserted into the ceramic tube such that the boat mid-point matched the tube mid-point. The 1/4" tube fixtures were then secured/sealed with ceramic paste to the ceramic tube and allowed to dry. Shortly after the furnace was turned on while temperatures were 300°F or less, air was purged from the tube system and gas flow was started. The gas flow rate was set and periodically monitored at 3-5 bubbles per second. The bubble rate was determined by placing the flexible end of the

Fig. 6-1 Furnace Open, Ceramic Tube Placed in Furnace Cavity, 1/4-in. Tubing Connected to Ceramic Tube Ends

Fig. 6-2 Furnace Closed, Note Location of Gas Inflow and Outflow (Arrows)

Fig. 6-3 Ceramic Tube Dimensions Closely Match Furnace Cavity

Fig. 6-4 Thermocouple Locations Adjacent to Ceramic Tube Exterior

outflow tube just beneath the surface of water in a beaker. Test runs were terminated by shutting off gas flow and turning off the furnace simultaneously. For each test run a new boat was used, also a new ceramic tube was used for each run when necessary, i.e., if it could not be completely cleaned out. Test runs were conducted as shown in Table 6-1.

TABLE 6-1

DATE	RUN	TEMPERATURE	TIME AT TEMPERATURE
2/23/87	#1	1500° ± 50°F	7 hrs.
2/27/87	#2	1850° ± 50°F	6 hrs.
3/10/87	#3	1500° ± 50°F	6 hrs.
3/12/87	#4	1850° ± 50°F	5.5 hrs.

The sample selection schedule, including temperature, initial and final weights is shown in Table 6-2:

TABLE 6-2

SAMPLE NO.	SAMPLE TYPE	TEST RUN	TEMP °F	INITIAL WT.	FINAL WT.
				(gms)	(gms)
1	Sialon	#1	1500°	1.6461	----
2	K162B	#1	1500°	3.0569	3.0722
3	KT150	#1	1500°	5.2968	5.2954
4	Sialon	#2	1850°	1.6348	----
5	K162B	#2	1850°	3.0980	3.0966
6	KT150	#2	1850°	5.2678	5.2743
7	KT150	#3	1500°	3.1265	3.1649
8	Carboloy	#3	1500°	3.2212	----
9	K162B	#4	1850°	3.0440	3.0086
10	Carboloy	#4	1850°	3.2263	----
11	IN718	#3	1500°	2.8268	---- *
12	IN718	#4	1850°	3.1423	---- *

*Final weight could not be measured because of the nature of the corrosion product.

6.4 Results

The as tested condition of all samples was documented. Figures 6-5 through 6-8 show the respective positions of the samples in the boats as tested. Close-up views showing the surface condition of individual samples after removal from the boat are presented in Figures 6-9 through 6-22.

The Sialon (Fig. 6-9, 6-10) and Carboloy (Fig. 6-11, 6-12) samples show little or negligible surface reaction at both 1500°F and 1850°F.

The K162B samples run at 1500°F (Fig. 6-13, 6-14) show some evidence of surface reaction. At 1850°F the K162B samples (Fig. 6-15, 6-16) were more highly reactive, especially sample 5 (Fig. 6-16); it formed a dual natured reaction product consisting of a soft outer layer and hard "scale" inner layer. This reaction product was easily removed from the base material and is shown in Figure 6-17. Although sample 9 (1850°F) showed similar evidence of reaction as in samples 2 and 7 (1500°F), interestingly it did not form a gross build-up of reaction product as did sample 5.

The KT150 samples reacted similarly to the K162B. At 1500°F the KT150 sample (Fig. 6-18) showed evidence of some reaction while at 1850°F another KT150 sample (Fig. 6-19) was highly reactive forming an easily removable, dual natured, reaction product (Fig. 6-20) similar to that evident on the K162B sample 5.

The Inconel 718 samples (Fig. 6-21, 6-22) appeared to be the most highly reactive of all the samples, forming relatively thick reaction product layers at both 1500°F and 1850°F. At 1850°F the reaction product formed consisted of a gross, dual natured build-up containing hard and soft layers; at 1500° only hard scale was evident.

To observe a microscopic profile of the affected surface layers, micro-structural cross-sections of all samples were prepared. Resulting micro-structures of the prepared cross-sections (excluding the Sialon and Carboloy samples which showed no effect in profile), are presented in the light microscope (LM) photographs, Figures 6-23 through 6-38. The microstructural profiles of the K162B, KT150, and IN718 samples all showed layering of reaction product on the surface

L8523

Fig. 6-5 Run No. 1, as Tested Samples No. 1, 2, and 3, 150C° F

L8522

Fig. 6-6 Run No. 2, as Tested Samples No. 4, 5, and 6, 1850° F

L8586

Fig. 6-7 Run No. 3, as Tested Samples No. 7, 8, and 11, 1500°F

L8588

Fig. 6-8 Run No. 4, as Tested Samples No. 9, 19, and 12, 1850°F

L8530

Fig. 6-9 Sialon, Sample 1, 1500°F, as Tested Condition; No Reaction

L8536

Fig. 6-10 Sialon, Sample 4, 1850°F, as Tested Condition; No Reaction (Material on Surface is Displaced Reaction Product from Sample 4)

L8592

Fig. 6-11 Carboloy, Sample 8, 1500°F, as Tested Surface Condition; Negligible Reaction

L8598

Fig. 6-12 Carboloy, Sample 10, 1850°F, as Tested Surface Condition; Negligible reaction (Material on Surface is Displaced Reaction Product from Sample 12)

L8528

Fig. 6-13 K162B, Sample 2, 1500°F, as Tested Surface Condition Shows Evidence of Surface Reaction

L8532

Fig. 6-14 K162B, Sample 7, 1500°F, as Tested Surface Condition Shows Evidence of Surface Reaction

Fig. 6-15 K162B, Sample 9, 1850°F, as Tested Surface Condition; Evidence of Reaction

Fig. 6-16 K162B, Sample 5, 1850°F, as Tested Surface Condition; Note Predominance of Surface Layer Reaction Product, Soft Outside, Hard Inner "Scale"

L8534

Fig. 6-17 K162B, Sample 5, 1850°F as Tested Condition after Removal of "Scale"

L8526

Fig. 6-18 KT150, Sample 3, 1500°F, as Tested Surface Condition; Some Reaction

L8538

Fig. 6-19 KT150, Sample 6, 1850°F, as Tested Surface Condition; Note Predominance of Reaction Product

L8542

Fig. 6-20 KT150, Sample 6, 1850°F as Tested Condition after Partial Removal of "Soft" Outer Reaction Product and "Hard Scale"

L8594

Fig. 6-21 IN718, Sample 11, 1500°F, as Tested Surface Condition; Note Layered Reaction Product "Scale"

L8600

Fig. 6-22 IN718, Sample 12, 1850°F, as Tested Surface Condition; Note Predominance of Reaction Product

and effected subsurface layers of varying depths at both test temperatures. All reaction product layers and their respective thicknesses are indicated in the figures. At 1500°F the K162B and KT150 samples (Fig. 6-23, 6-24, 6-29, 6-30) formed thin, two-phase surface layers and relatively thicker subsurface layers. The subsurface layers became evident during sample polishing; they appeared to lack the normal material cohesiveness. The K162B surface and subsurface layers were thicker than their counterparts in the KT150. At 1850°F one K162B sample (Fig. 6-26, 6-28) and the KT150 sample (Fig. 6-31, 6-32) both formed very thick, single phase, outer surface layers. The other K162B sample at 1850°F (Fig. 6-25) had a thin two-phase surface layer. Interestingly the K162B sample with the thick outer layer formed a subsurface layer thinner than that evident in K162B samples exposed at lower temperatures. This was also true at the KT150 sample, which has no substantial subsurface layer. The K162B sample without the thick layer, however, formed a sub-surface layer much thicker than those at lower temperature exposure. There were some areas in the K162B sample 5 (1850°F) on which a relatively thin multi-phase layer was evident (Fig. 6-28) at the inner scale surface. No multi-phase layer was seen in the KT150 sample at 1850°F.

Microstructural profiles through the IN718 sample surfaces are shown in Figures 6-33 - 6-38. Even though multi-layered, multi-phase reaction product layers are evident in both 1500°F and 1850°F samples the arrangement and type of layers appear quite different. The multi-phase layers in the higher temperature sample is quite thick; the even thicker outer layer (not present at 1500°) consists of an unusual network of globules.

Selected areas of the IN718, K162B and KT150 reaction product layers formed at high temperature were chemically analyzed by electron microprobe analysis (EMPA). The area and phases analyzed in the IN718 sample are shown in Figure 6-36, those in the K162B sample - Figure 6-28, and the KT150 sample - Figure 6-32. Results are listed in Table 6-3. All phases analyzed contained various concentrations of sulfur. Only the thick, outer, single phase layers contained carbon.

Fig. 6-23 K162B, Sample 2, 1500°F LM Photograph of x-Sectional Microstructure through Sample Surface; Note Presence and Thickness of Two-Phase Surface Layer (Reaction Product) and Affected Subsurface Layer

Fig. 6-24 K162B, Sample 7, 1500°F LM Photograph of x-Sectional Microstructure through Sample Surface; Note Presence and Thickness of Two-Phase Surface Layers

Fig. 6-25 K162B, Sample 9, 1850°F LM Photograph of x-Sectional Microstructure through Sample Surface. Note Presence and Thickness of Surface and Subsurface Layers

Fig. 6-26 K162B, Sample 5, 1850°F LM Photograph of x-Sectional Microstructure through Sample Surface (Reaction Product Layers removed); Note Remnant of Surface Layer Area Thickness of Affected Subsurface Area

L8629

85X

Fig. 6-27 K162B, Sample 5, 1850°F LM Photograph at x-
Sectional Microstructure of Detached Surface
Reaction Product; Note Presence and Thickness of
Single-Phase Outer Layer and Multiphase Inner
Layer

L8631

400X

Fig. 6-28 Higher Magnification of K162B Single and Multi-
phase Reaction Product Layers Shown Above; Maximum
Thickness is 0.0025 in. (Note Five Distinct Phas-
es)

Fig. 6-29 KT150B, Sample 3, 1500°F LM Photograph of x-Sectional Microstructure through Sample Surface; Note Presence and Thickness of Surface and Subsurface Layers

Fig. 6-30 High Magnification of KT150 Surface Layer Shown Above; Note Two or Possibly Three Phase Present in the Thin Layer

Fig. 6-31 KT150, Sample 6, 1850° LM Photograph of x-Sectional Microstructure through Sample Surface (Reaction Product Layer(s) Removed); Note Absence of Any Appreciable Affected Subsurface Layer

.011+
(Much of Softer Material has been Removed)

Fig. 6-32 KT150, Sample 6, 1850°F LM Photograph of x-Sectional Microstructure of Detached Surface Reaction Layer; Note Area Identified as "1" - Chemical Analysis Location

L8626

85X

Fig. 6-33 IN718, Sample 11, 1500°F, LM Photograph of x-Sectional Microstructure through Sample Surface; Note Multiphase and Multilayer Reaction Product/Scale

L8628

200X

Fig. 6-34 Higher Magnification of IN718 Outer Surface Layer Shown Above; Note Presence of Three Distinct Phases

L8619

85X

Fig. 6-35 IN718, Sample 12, 1850°F, LM Photograph of x-Sectional Microstructure than Sample Surface; Note Presence and Thickness of Reaction Product Layers

L8620

400X

Fig. 6-36 Higher Magnification of IN718 Single Outer Phase and Multiphase Reaction Product Layers Shown Above; Note Presence of Four Distinct Phases

L8623

85X

Fig. 6-37 IN718, Sample 12, 1850°F, LM Photograph of Cross-Section through Reaction Product Layers (Emphasis on Microstructure of Outer Most Layer); Note Loosely Connected Formation of Globules (Arrows)

L8622

200X

Fig. 6-38 Higher Magnification of IN718 Reaction Product Globule from Area Similar to that Shown Above; Note Phase at Center of Globule.

TABLE 6-3

**EMPA CHEMICAL ANALYSIS RESULTS OF IN718, K162B and KT150
REACTION PRODUCT FORMED AT 1850°F**

IN718

Phase	Fe	Ni	Cr	S	Ti	C	COLOR
1	Neg.	V. Low	Neg.	Low	Low	V. high	Dk. Gray
2	Low	High	V. Low	High	V. Low	Neg.	Tan
3	Med	Low	High	V. High	Low	Neg.	Lt. Gray
4	High	V. high	V. Low	V. low	Low	Neg.	White

K162B

Phase	Ni	Ti	Mo	S	C	COLOR
1	Low	V. low	Neg.	Low	V. high	Dk. Gray
2	Low	V high	Neg.	Low	Neg.	Blue gray
3	High	Med.	V. low	Med.	Neg.	Tan
4	Med.	High	Low	High	Neg.	Lt. Gray
5	V. high	Low	Low	V. low	Neg.	White

KT150

Phase	TI	W	Mo	Lo	Ni	S	C	COLOR
1	Neg.	Neg.	Neg.	Neg.	Neg.	Low	V. high	Lt. Gray

6.5 Conclusions/Recommendations

This preliminary study of reactions between 2% H₂S in methane and the ceramic, metal bonded carbide, and nickel base alloy samples, show that all materials reacted, some more some less, in the short duration (5 to 7 hrs.) tests, forming reaction products containing sulfur. The relative reactivity is shown in Table 6-4. The exact nature of each of the reaction products was not analyzed because of limited resources.

The brief examination of the reaction products revealed that the reactions are complex. The reaction products vary from a "cotton candy" type network to multi-phase dense corrosion product. It appears on some samples that either the surface or the initial reaction products may be catalyzing deposits of reaction products, such as the "cotton candy" material. The variety and type of reaction products may indicate that some of the reaction products could have useful properties for a bearing system in a coal gasification system. The reaction products may provide solid lubrication and the base material could be incorporated into the bearing structure as a supply.

The extent to which this experiment simulates the coal gasification reactor needs to be established. With other gaseous compounds in the environment, more and more complex reactions can be expected. The thermodynamics of the material interactions should be addressed to optimize material selections prior to any continuing experiments.

TABLE 6-4
RELATIVE REACTIVITY

BASE MATERIAL	TEMPERATURE	RELATIVE REACTIVITY WITH 2% H ₂ S IN METHANE
Carboloy	1500°F	Low
Sialon		
KT150		
KB162		
Inconel 718		High
Carbaloy	1850°F	Low
Sialon		
KB162		
KT150		
Inconel 718		High

7.0 CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

The following tests have been completed in this program.

1. A rigid surface-rigid mount seal combination with a K162B runner and K162B seal ring was run for several minutes and showed considerable distress. The parts were not coated nor preoxidized. There was a clear indication of thermal distortion and misaligned contact marks, which may have contributed to the failure, but the results clearly indicate that a rigid surface-rigid mount configuration with hard and high modulus of elasticity materials (such as cermets and ceramics) is a poor performer.
2. A rigid surface-compliant mount seal, with a moly-disulfide coating was tested next. After several minutes of testing under various loads and speed at room temperature, the coating was not able to survive for a higher load. It also wore through.
3. The most promising seal concept was the rigid surface-compliant mount seals made of K162B that were preoxidized. This configuration withstood speeds up to 3600 rpm (1900 ft/min), in an ambient temperature up to 1275°F and a load of 91 lbs. The test results were very good. Results are indicated in Table 7-1. Two sets of seals were run to determine repeatability performance; SN/1 and SN/2. Varying loads and temperatures up to 1275°F were run. Some 35 hours were run on the first seal and 23.4 hours on the second seal. During these tests, frequent metallographic and surface inspections were made. Wear rate ϕ of 3.88×10^{-11} on seal SN/1 and 3.79×10^{-11} on seal SN/2 are very good indeed. Friction levels indicated by the variable η are also very good. Several interesting interfacial phenomenon occurred which are subsequently explained.
4. An additional test was run using the NASA PS-200 coating to determine its suitability for the application. The runner was preoxidized K162B and the seal ring was a rigid surface-compliant mount configuration

TABLE 7-1

TRIBOLOGICAL PERFORMANCE OF K162B (TiC CERMET) FOR RING-DISK COMBINATION
USED FOR ADVANCED ROTATING SHAFT SEAL

Seal Ring - ID = 4.62 cm, OD = 5.6 cm, A = 7.84 cm²

Test No. Serial No.	Test Date Day/Mc.	v_{avg} cm/sec (RPM)	F kg (psi)	ΣFL cm kg (x10 ⁶)	T_i °C	$T_{o,max}$ °C	T_a °C	ζ_R um (MIL)	ζ_S um (MIL)	Wear Test $\phi S/\phi R, \bar{\phi}$ (cm ² /kg)x10 ⁻¹¹	Test Period (min)	η_{avg} (-)	Remarks
1-2	3&4/C2	364 (1360)	8.81 (16)	564.986	21 (7C)	110 (230)	24 (75)	2.5 (0.1)	2.6 (0.101)	3.53 = 3.5	276	0.32	Seal ring compliant mount with 5mil thick INC.x 750
2-2	5/02	668 (2500)	8.73 (16)	106.091	21 (7C)	110 (230)	24 (76)	7.6 (0.3)	7.6 (0.3)	5.63 = 5.6	339	0.32	c.m. 5 mil
3-2	6/02	668 (2500)	11.65 (25.7)	131.142	22 (72)	154 (310)	22 (72)	7.8 (0.305)	7.6 (0.3)	4.56 = 4.58 4.61	270	0.29	
4-2	13/02	882 (3300)	17.48 (31.7)	131.142	22 (72)	171 (340)	22 (72)	7.9 (0.31)	7.6 (0.3)	4.56 = 4.64 4.71	147.5	0.28	
5-2	19/02	722 (2700)	17.48 (31.7)	124.837	24 (72)	302 (575)	260 (500)	10.2 (.403)	8.46 (0.333)	5.31 = 5.96 6.61	160	0.26	
6-2	25/02	962 (3600)	18.64 (33.8)	308.822	22 (72)	399 (750)	416 (780)	12.4 (.487)	19.69 (.775)	5.0 = 4.1 3.13	287	0.17	
7-2	9/03	962 (3600)	23.30 (42.3)	443.864	Preheated	621 (1150)	649 (1200)	21.6 (0.85)	21.59 (0.85)	3.59 = 3.7 3.81	330	0.15	c.m. = 10 mil
8-2	11/03	962 (3600)	40.78 (73.97)	659.072	Preheated	660 (1220)	649 (1200)	22.9 (0.9)	31.50 (1.24)	3.75 = 3.235 2.72	280	0.15	c.m. = 10 mil - $\phi = 3.88 \times 10^{-11}$ ToT.EFL = 2.47×10^9
9-1	18/02	786 (2940)	16.3 (29.6)	116.851	21 (63)	160 (320)	21 (70)	5.0 (.197)	5.0 (0.197)	3.35 = 3.35 3.35	150	0.32	c.m. with 5 mil INC. x 750.
10-1	5/03	962 (3600)	21. (38)	471.572	13 (65)	427 (800)	454 (850)	35.1 (1.38)	33.0 (1.30)	5.49 = 5.66 5.43	405	0.18	c.m. with 10 mil INC. x 750
11-1	10/C3	962 (3600)	24.8 (45)	443.864	Preheated	713 (1315)	691 (1275)	17. (0.67)	15.6 (0.615)	2.76 = 2.89 3.02	310	0.15	
12-1	13/C3	962 (3600)	38.8 (70.5)	1210.539	Preheated	421 (790)	416 (780)	50.8 (2.0)	55.5 (2.185)	3.59 = 3.4 3.29	540	0.18	c.m. w/10 mil $\phi = 3.79 \times 10^{-11}$ ToT.EFL = 2.24×10^9

Nomenclature, Subscripts

i = inlet
o = outlet
a = ambient

s = seal ring
R = runner, disk
avg = average

v = average speed (disk)
 η = coef. of friction
T = temp

ζ = wear depth
 ϕ = wear rate
 $\bar{\phi}$ = mean avg. ϕ

F = normal force
L = Vt = sliding distance

with a substrate material made of XF818 coated with PS-200. The tests were only of short duration but long enough to obtain an indication of friction levels and wear rates at low and high temperatures. Subsequent surface inspection revealed that under moderate loading the coating wore rapidly. XF818 coated with PS-200 had a wear rate $\phi = 3.5 \times 10^{-9} \text{ cm}^3/\text{cm-kg}$ under 15 psi loading at room temperature, which is a high wear rate. An Inconel ring coated with PS-200 had a wear rate $\phi = 7.35 \times 10^{-10} \text{ cm}^3/\text{cm-kg}$ at an ambient temperature of 1250°F and a loading of 7 psi. It was concluded that the coating was not designed for high loading and does not appear to be a good face seal material.

General Conclusions

Sliding tests of rigid surface-compliant mounted seal concepts revealed several interesting facial phenomena, that bring dry sliding close to hydrodynamic behavior. Included are the behavior of friction, η , the formation of a wedge, and the role of wear debris as a lubricant.

One of the significant aspects of the frictional behavior of dry sliding is that it fits in more with hydrodynamic theory than the elastic behavior of dry contacts. One example indicated on Fig. 7-1 shows a drop in η with temperature and also with speed (shown in Fig. 5-28). Similar results, as shown by Semenov et al [34] are given in Fig. 7-2. They also show a drop in coefficient of friction with increasing temperature in vacuum for a graphite material and corundum ceramic (aluminum oxide). If one views the increase in temperature and speed as being primarily responsible for an increase in wear then the drop in η can be associated with the lubricating function of the wear debris particles. This is further supported by Fig. 5-28 where η is related directly to the wear rate in the specimen; when the rate of wear levels off so too does the friction coefficient. It is only on the basis of wear particles facilitating sliding that the drop of η with increasing load can be reconciled--because both a higher load as well as higher temperatures contribute to an increase in the amount of flash temperature and wear.

This frictional heating, and the thermal and thermochemical phenomena associated with it, can have a very important influence on the tribological behavior

Fig. 7-1 Relation Between η and Temperature for Cermet K162B

Effect of Temperature on Friction Coefficient in Vacuum Graphite Material AG-1500 on Corundum Ceramic

Fig. 7-2 Effect of Temperature on Friction Coefficient in Vacuum Graphite Material AG-1500 (1) and Corundum Ceramic (2)

of the sliding components as well as on the geometrical stability of the seal components.

Based on experimental observations and their interpretation with numerical computations contact surface temperatures were estimated. Computed values indicated interacted surface temperatures near the materials' melting point (flash temperature, see Appendix A). The numerical values of flash temperature for K162B seals under various loads are given in Appendix A.

Formation of Wedge Shaped Geometries

One of the more mystifying aspects of parallel plate interaction is that after a fairly short time the surfaces tend to generate their own tapers, as if they were aware of convergent wedges. After a brief running period the surfaces which were originally perfectly parallel (flatness was in order of light bands) assumed a wavy pattern. Fig. 7-3 and Fig. 5-33 give the produced waviness as a function of location and running time. The average wave height over the wave half width yields 1 to 2 mils of taper per inch of pad length--the very ratios used in hydrodynamic bearings. The presence of this double wave along the rubbing trajectory could be clearly seen on the specimen and is also visible on the photograph in Figure 5-8.

Presence of Wear Particles in the Interspace

Under the influence of interacting asperities at the surfaces, in sliding wear, various processes can operate to generate wear particles. In recent years, much attention has been focused on formation of lamellar wear particles due to the introduction of the delamination theory of wear by Suh [35] and hot spot theory by Quinn [36]. Thin plate-shaped wear particles are commonly found in the wear debris recovered from pin on disk test apparatus. The lateral dimension of these particles range from a fraction of a micron to several microns, with a thickness of the lateral dimensions. According to the delamination theory of wear these lamellar wear particles result from plastic deformation and subsequent fracture. As shown in figures given in Appendix B this wear forms fairly small-sized plateaus which produce very small-sized particles (see also Fig. 5-11), with half of them below the 1 μm (40 $\mu\text{in.}$) range (particle size range - 0.1 μm to 2.0 μm). These particles would thus be perfectly capable of filling

Fig. 7-3 Formation of a Geometric Wave as Function of Time, (Σ FL)

the valleys and clearance spaces between interacting surfaces and remaining there for prolonged periods. This is indeed what was found on the specimens, and Fig. 5-11 shows a photograph of some of the evidence. It is perhaps the tapers which subsequently retained the small wear particles in the form of a powder lubricant that after the initial wear period facilitated the successful operation of the flat land specimen. More direct proof of the quasi-hydrodynamic action of wear particles in a self-generated converging wedge in dry friction is given by Godet [37]. Therefore it can be concluded that wear debris or wear particles, which form rapidly, alter the nature of the contact which gradually changes from a two to a three-body contact as in hydrodynamic or in elastohydrodynamic action.

7.2 Recommendations for Future Research

Since the test results demonstrated the advantages of powdered lubrication in dry contact face seals, further efforts should be conducted to exploit this phenomenon. The powder in the experiments conducted in this program was self-generating, but it may also be possible to externally introduce the powdered lubricant. Thus it is recommended that:

1. Investigations be conducted to determine appropriate lubricant powders that could be introduced into the sealing interface. Characteristics of the powdered lubricant would include: temperature capability to 2000°F; lubricity at both cold and hot temperatures; particle sizes in the order of 1 μm ; chemical compatibility with surrounding gases and seal materials.
2. Means for external injection of the powdered lubricant should be investigated as this will be an important and integral part of any future seal system.
3. Comprehensive testing should be conducted using candidate seal materials and powdered lubricants to establish wear rate, friction coefficient, temperature speed and load capability, and estimated life.

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APPENDIX A

FRICTIONAL HEAT THERMAL ANALYSIS

FLASH INTERFACE TEMPERATURE CALCULATION

Frictional Heat - Thermal Analysis

When solids are in sliding contact, frictional heating causes a significant rise in temperature in all but the most lightly loaded, low speed systems. This temperature increase has an important influence on friction and wear. It may modify the mechanical and metallurgical properties of the sliding solids and control the oxidation of their surface.

It is not surprising therefore that there have been numerous attempts to determine the temperature of sliding surfaces. Temperature flashes arise due to frictional heating at the real contact areas. These are small: typically 10^{-2} mm, of variable but short duration, and their location within the apparent contact area is continually changing. Thus direct measurement of the flash temperature is virtually impossible.

The work of Blok [38], Jaeger [39], and Barber [40] would permit calculation of the real contact temperatures if the boundary conditions, i.e., size, relative motion of the contacts, and location of heat generation could be accurately specified. Unfortunately, these boundary conditions are particularly ill defined. However, the above authors show that the division of frictional heat between sliding solids is sensitive to the boundary conditions. Further, the division of heat, and the bulk surface temperature (i.e., that of the apparent or nominal contact area) are comparatively easy to measure. This suggests an alternative to either direct measurement or purely analytical determination of the flash temperature. Measuring the division of heat for a wide range of sliding conditions should allow the nature of the surface interactions to be deduced, permitting a more precise statement of the boundary conditions for any subsequent thermal analysis; for example estimates of flash temperature.

For a seal ring rubbing against a seal rotor, heat transferred at the remote boundary causes a temperature rise there which significantly affects the division of heat and material properties. Superimposing the heat flow caused by the nonzero temperature at infinity, T_1 and T_2 for solids 1 and 2, respectively, the heat flow into each solid resulting from total frictional heat input Q_T is:

$$Q_1 = \frac{Q_T r_2 + (T_2 - T_1)}{r_1 + r_2} \quad (A-1)$$

$$Q_2 = \frac{Q_T r_1 + (T_1 - T_2)}{r_1 + r_2} \quad (A-2)$$

for a heat source of intensity $Q(F/A)$ traveling over a half-space. For a source of length B (in the direction of sliding) moving over the half-space at velocity V , the time for the source to pass over any point is $t = B/V$. In friction studies the source strength Q is set as $Q = \eta FV/A$, where η is the friction coefficient and F/A is the interface pressure, P .

Setting Q and t the total frictional heat generation per unit time Q_T can be obtained and the maximum flash temperature, T_f , evaluated by Blok's equation.

$$T_f = nP \left[\frac{(B)^{1/2} (V_1 + V_2)}{(K_1 \rho_1 C_1 V_1 + K_2 \rho_2 C_2 V_2)^{1/2} (\frac{\pi}{4})^{1/2}} \right] \quad (A-3)$$

when

- B = source length in direction of travel
- V_1 = sliding speed, surface 1
- V_2 = sliding speed, surface 2
- K_1, K_2 = thermal conductivity
- C = specific heat
- P = Hertzian maximum pressure
- ρ = material density

A maximum allowable surface temperature is defined here because the material dictates the frictional heat not be easily conducted from the interface and it is assumed that a limiting condition (such as melting or oxidation) will occur

as load and speed are increased, beyond which frictional heating is actually the dominant factor. Also retention of simplicity in the analysis allows quantitative expressions for friction to emerge that are based only on heat flow.

As far as thermal boundary conditions are concerned, three boundary conditions (Equations A-1, A-2 and A-3) are sufficient to model the seal components using the finite element model and the ANSYS Computer Program.

Thermal mappings were generated. In this study $p = 50$ psi was kept constant and the values for η were taken to be 0.2, 0.3 and 0.4 and ambient temperatures were considered to be 200°F, 500°F, 1000°F and 1500°F. Rotational speeds of interest were taken from 600 rpm up to 3600 rpm and appropriate thermal boundary layer thicknesses to those surfaces exposed to ambient as indicated in Figure A-1 were assigned.

To identify any problem areas associated with an increased ambient temperature (up to 1500°F); rotational speed (up to 3600 rpm) and values of friction coefficients, the thermal analysis in a parametrical grouping was conducted for the material K162B seal ring rubbing against K16B seal runner. The results of this analysis are presented in Figures A-2 through A-5, which serves to verify materials selection and surface coating, and upon which the seal design analysis, materials properties and seal surface coating selection were based.

The overall temperature distribution in the seal ring is a strong function of η and the ambient temperature. Although rotation speed has great influence on the seal temperature, due to the inverse relationship between velocity and η , the effect of speed will be discussed later.

As shown in Figure A-5, the bulk temperature of the seal ring at 3600 rpm varies from 1250°F up to 2250°F for the increase in T_{amb} of 1000°F. Therefore, the T_{amb} is of prime importance in the selection of the seal material to be used at higher temperatures. The other important parameter which must be considered is the coefficient of friction. By comparing the results of Figure A-2 with Figure A-4, when η was doubled, the seal bulk temperature was almost increased by a factor of 2 for the constant speed and seal loading. These analytical results suggest that selection of the material on the basis of reduced η and materials which are capable of high temperature operation.

Fig. A-1 Finite Element Model of Radial Cross-Section of Ring

3600 RPM MU=.3 TAMB=500 F

1278.	1285.	1286.	1281.	1272.	1259.						
1267.	1274.	1276.	1272.	1262.	1249.						
1247.	1257.	1260.	1256.	1245.	1231.						
1110.	1138.	1172.	1224.	1240.	1245.	1241.	1229.	1207.	1149.	1130.	1088.
1112.	1139.	1171.	1210.	1227.	1233.	1229.	1216.	1193.	1149.	1131.	1090.
1110.	1137.	1168.	1202.	1219.	1224.	1221.	1208.	1185.	1146.	1129.	1088.

3600 RPM MU=.3 TAMB=1000 F

1778.	1785.	1786.	1781.	1772.	1759.						
1767.	1774.	1776.	1772.	1762.	1749.						
1747.	1757.	1760.	1756.	1745.	1731.						
1610.	1638.	1672.	1724.	1740.	1745.	1741.	1729.	1707.	1649.	1630.	1588.
1612.	1639.	1671.	1710.	1727.	1733.	1729.	1716.	1693.	1649.	1631.	1590.
1610.	1637.	1668.	1702.	1719.	1724.	1721.	1708.	1685.	1646.	1629.	1588.

3600 RPM MU=.3 TAMB=1500 F

2278.	2285.	2286.	2281.	2272.	2259.						
2267.	2274.	2276.	2272.	2262.	2249.						
2247.	2257.	2260.	2256.	2245.	2231.						
2110.	2138.	2172.	2224.	2240.	2245.	2241.	2229.	2207.	2149.	2130.	2088.
2112.	2139.	2171.	2210.	2227.	2233.	2229.	2216.	2193.	2149.	2131.	2090.
2110.	2137.	2168.	2202.	2219.	2224.	2221.	2208.	2185.	2146.	2129.	2088.

Fig. A-2 Seal Ring Thermal Mapping for $n = 0.2$, $T_{amb} = 200^{\circ}\text{F}$,
 $N = 600\text{-}3600\text{ rpm}$

600 RPM MU=.4 TAMB=200 F

	373.	374.	375.	374.	371.	369.					
	370.	372.	372.	371.	369.	366.					
	366.	368.	369.	368.	366.	362.					
336.	342.	349.	361.	364.	365.	365.	362.	357.	344.	340.	331.
336.	342.	349.	358.	362.	363.	362.	359.	354.	344.	340.	331.
336.	342.	348.	356.	360.	361.	360.	357.	352.	343.	340.	331.

1200 RPM MU=.4 TAMB=200 F

	545.	549.	549.	547.	543.	537.					
	541.	544.	545.	543.	538.	533.					
	532.	536.	538.	536.	531.	525.					
471.	484.	499.	522.	529.	531.	529.	524.	514.	488.	480.	461.
472.	484.	498.	516.	523.	525.	524.	518.	508.	488.	480.	462.
471.	483.	497.	512.	519.	522.	520.	514.	504.	487.	480.	461.

2400 RPM MU=.4 TAMB=200 F

	891.	898.	899.	894.	886.	874.					
	881.	888.	890.	886.	877.	866.					
	864.	873.	876.	872.	863.	850.					
742.	707.	797.	843.	858.	862.	858.	848.	828.	777.	760.	723.
744.	768.	797.	831.	846.	851.	848.	836.	816.	776.	761.	724.
742.	767.	794.	824.	839.	844.	840.	829.	809.	774.	759.	723.

3600 RPM MU=.4 TAMB=200 F

	1237.	1246.	1248.	1242.	1229.	1211.					
	1222.	1232.	1234.	1229.	1215.	1199.					
	1196.	1209.	1213.	1208.	1194.	1174.					
1014.	1051.	1096.	1165.	1186.	1193.	1187.	1171.	1143.	1065.	1040.	984.
1015.	1052.	1095.	1147.	1169.	1177.	1172.	1154.	1124.	1065.	1041.	986.
1013.	1050.	1091.	1136.	1158.	1166.	1161.	1143.	1114.	1061.	1039.	984.

Fig. A-3 Seal Ring Thermal Mapping for $\eta = 0.3$,
 $T_{amb} = 200$, and $N = 600 - 3600$ rpm

600 RPM MU=.3 TAMB=200 F											
			330.	331.	331.	330.	329.	326.			
			328.	329.	329.	329.	327.	325.			
			325.	326.	327.	326.	324.	322.			
302.	306.	312.	321.	323.	324.	323.	321.	318.	308.	305.	298.
302.	307.	312.	318.	321.	322.	321.	319.	316.	308.	305.	298.
302.	306.	311.	317.	320.	321.	320.	318.	314.	308.	305.	298.

1200 RPM MU=.3 TAMB=200 F											
			459.	462.	462.	460.	457.	453.			
			456.	458.	459.	457.	454.	450.			
			449.	452.	453.	452.	448.	444.			
403.	413.	424.	441.	447.	448.	447.	443.	436.	416.	410.	396.
404.	413.	424.	437.	442.	444.	443.	439.	431.	416.	410.	397.
403.	412.	423.	434.	440.	441.	440.	436.	428.	415.	410.	396.

2400 RPM MU=.3 TAMB=200 F											
			718.	723.	724.	721.	714.	706.			
			711.	716.	717.	714.	708.	699.			
			698.	705.	707.	704.	697.	687.			
607.	625.	648.	683.	693.	696.	694.	686.	671.	633.	620.	592.
608.	626.	647.	674.	685.	688.	686.	677.	662.	632.	621.	593.
607.	625.	645.	668.	679.	683.	680.	672.	657.	631.	619.	592.

3600 RPM MU=.3 TAMB=200 F											
			978.	985.	986.	981.	972.	959.			
			967.	974.	976.	972.	962.	949.			
			947.	957.	960.	956.	945.	931.			
810.	838.	872.	924.	940.	945.	941.	929.	907.	849.	830.	788.
812.	839.	871.	910.	927.	933.	929.	916.	893.	849.	831.	790.
810.	837.	868.	902.	919.	924.	921.	908.	885.	846.	829.	788.

Fig. A-4 Seal Ring Thermal Mapping for $\eta = 0.4$
 $T_{amb} = 200^\circ\text{F}$ and $N = 600 - 3600 \text{ rpm}$

600 RPM MU=.2 TAMB=200 F											
			286.	287.	287.	287.	286.	284.			
			285.	286.	286.	286.	285.	283.			
			283.	284.	284.	284.	283.	281.			
268.	271.	275.	280.	282.	283.	282.	281.	279.	272.	270.	265.
268.	271.	275.	279.	281.	281.	281.	280.	277.	272.	270.	266.
268.	271.	274.	278.	280.	280.	280.	279.	276.	272.	270.	265.

1200 RPM MU=.2 TAMB=200 F											
			373.	374.	375.	374.	371.	369.			
			370.	372.	372.	371.	369.	366.			
			366.	368.	369.	368.	366.	362.			
336.	342.	349.	361.	364.	365.	365.	362.	357.	344.	340.	331.
336.	342.	349.	358.	362.	363.	362.	359.	354.	344.	340.	331.
336.	342.	348.	356.	360.	361.	360.	357.	352.	343.	340.	331.

2400 RPM MU=.2 TAMB=200 F											
			545.	549.	549.	547.	543.	537.			
			541.	544.	545.	543.	538.	533.			
			532.	536.	538.	536.	531.	525.			
471.	484.	499.	522.	529.	531.	529.	524.	514.	488.	480.	461.
472.	484.	498.	516.	523.	525.	524.	518.	508.	488.	480.	462.
471.	483.	497.	512.	519.	522.	520.	514.	504.	487.	480.	461.

3600 RPM MU=.2 TAMB=200 F											
			718.	723.	724.	721.	714.	706.			
			711.	716.	717.	714.	708.	699.			
			698.	706.	707.	704.	697.	687.			
607.	625.	648.	683.	693.	696.	694.	686.	671.	633.	620.	592.
608.	626.	647.	674.	685.	688.	686.	677.	662.	632.	621.	593.
607.	625.	645.	668.	679.	683.	680.	672.	657.	631.	619.	592.

Fig. A-5 Seal Ring Thermal Mapping for N = 3600 rpm,
 $\eta = 0.3$ and $T_{amb} = 500, 1000, \text{ and } 1500^\circ\text{F}$

Flash Interface Temperature Calculation

The flash (instantaneous) temperature rise at the interface depends on the mechanical, thermal, topographical, and the tribological properties of the mating surfaces and sliding conditions. From Blok [38] the temperature rise for a slider rubbing on a semi-infinite solid in low speed sliding pertinent to the seal problem has been modified and has been given as Equation (A-3)

$$T_f = \eta P \left[\frac{(B)^{1/2} (V_1 + V_2)}{(K_1 \rho_1 C_1 V_1 + K_2 \rho_2 C_2 V_2)^{1/2} (\frac{\pi}{4})^{1/2}} \right] \quad (A-3)$$

For soft material as a coating, Equation (A-3) takes the following form:

$$\bar{\theta} = \frac{fV}{k_1 + k_2} [0.44 H \bar{d}_{\max} + 2\bar{\sigma}] \quad (A-4)$$

$$\text{If } L \text{ (Peclet Numer)} = \frac{V \bar{d}_{\max}}{K} < 1$$

where the surface temperature is equal to T_f plus the material's bulk temperature.

$$T_s = T_b + T_f \quad (A-5)$$

where,

- $\bar{\theta}$ = average flash temperature, °C(°F)
- f = coefficient of friction
- V = sliding speed, mm/s (in/s)
- k_1, k_2 = thermal conductivity, W/(m.K) (lb/(sec°F)) of sliding members
- K = thermal diffusivity, mm²/s (in.²/sec)
- H = bulk hardness of the softer material, Pa (psi)
- $\bar{\sigma}$ = mean contact stress, Pa (psi)
- $\bar{d}_{\max}$ = maximum value of junction diameter during the life of an asperity contact, mm(in)
- l = half length of square slider, mm(in)

Relevant thermal, mechanical, and topographical properties (for K162B) needed are:

$$\eta = 0.1 \text{ \& } 0.2$$

$$C = 77.8 \text{ ft.lb/lb.}^\circ\text{F}$$

$$\gamma = 363 \text{ lb/ft}^3$$

$$K = 9.2 \text{ lb.ft./sec.ft.}^\circ\text{F (extrapolated value for } T_{\text{amb}} 1500^\circ\text{F)}$$

$$U = 3.16 \text{ ft/sec}$$

$$B = 0.58 \text{ ft}$$

$$P = \text{Hertzian maximum pressure (lb/ft}^2\text{)}$$

At higher speeds and loads it is seen that the interface temperatures are very high and as a result there may be formation of a protective oxide layer at the interface which may reduce the friction considerably. For demonstration purposes, it is considered that $0.1 \leq \eta \leq 0.2$.

Substituting these values in Equations (A-3) and (A-4) produces the results which are tabulated for various loads and for two values of η in Table A-1.

TABLE A-1

**CALCULATED AVERAGE FLASH TEMPERATURE FOR SEAL RING
AND RUNNER (SN/1)**

W (lb)	11.25	22.2	45	91	Coeff. of Friction
PH (psi)	12,193	15,362	19,355	24,476	
T_f (°F)	310	390	492	622	$\eta = 0.1$
	610	780	984	1,244	$\eta = 0.2$
T_s (°F)	2,010	2,186	2,384	2,644	$\eta = 0.2$

Relevant thermal, mechanical, and topographical properties (for K162B) needed are:

$$\eta = 0.1 \text{ \& } 0.2$$

$$C = 77.8 \text{ ft.lb/lb.}^\circ\text{F}$$

$$\gamma = 363 \text{ lb/ft}^3$$

$$K = 9.2 \text{ lb.ft./sec.ft.}^\circ\text{F} \text{ (extrapolated value for } T_{\text{amb}} 1500^\circ\text{F)}$$

$$U = 3.16 \text{ ft/sec}$$

$$B = 0.58 \text{ ft}$$

$$P = \text{Hertzian maximum pressure (lb/ft}^2\text{)}$$

APPENDIX B

SURFACE ANALYSIS OF K162B/K162B,

RS-CM SEAL RUNNER AND RING

Surface Analysis

The seal runner and ring made of Kentanium K162B (SN/1) after a successful run (Tests No. 9 through 12), accumulated a total of 1405 minutes, at a rotor speed of 3600 rpm (surface linear velocity of 1900 ft/min) with a sum of $FL = 2.24 \times 10^9$ cm-kg at low and high temperatures up to 1275°F. These specimens were sent to the RPI lab for a thorough examination. It should be mentioned that the seal arrangement was Rigid Surface-Compliant Mount throughout the tests.

The seal runner (ROTOR) and seal ring (STATOR) have been analyzed. Four separate measurements were made which are described in detail below.

1. Stylus Surface Profiles

A Form Talysurf was used to generate the output on Figures B-1 through B-17. On all of these outputs, the rectangular title area gives the following information:

- The part and the area on the part measured.
- The mode: either "unfiltered" or "roughness".

In the "roughness" mode, the data is filtered with a standard 0.010 in. cutoff.

There is also supplementary information showing the approximate area being profiled on the wear tracks of the specimen and explaining some of the data. For most of the measurements, three separate outputs are shown: a profile, statistics tables and a histogram of surface heights.

One further result of the surface topography measurements, shown on Figures B-13 through B-17 of the data is that there is a slight variation in overall depth of the wear area of the rotor face, indicating a misalignment of about 0.00025 inches. The diagram on Figure B-17 shows the actual total depth of the wear area measured in the eight profiles given on Figures B-13 through B-16.

Surface Profiles of Unworn Area

Data given in Figures B-18 through B-29 show the results of several surface profile measurements on the unworn surface of the rotor. The filtered average roughness values were consistently in the 4 to 5 μ in. range (Figs. B-23, B-25, B-27 and B-29). After abrading the unworn surfaces of the rotor, surface roughness data in the abraded area (Figs. B-18 through B-21), both along and across the abraded scratches was taken. These show that the silicon-carbide paper does put a number of parallel scratches into the surface (see bottom profile, Fig. B-18).

2. Optical Surface Profiles

A non-contacting Wyko-3-D Digital Optical Surface Profiler was used in an attempt to map the topography of the smoother worn areas on the stator face. The data are shown on two pages (Figures B-30 and B-31). The roughness levels of these surfaces are out of range of this instrument, and therefore the surface maps serve the purpose of an orientation relative to the direction of center face motion.

3. SEM Micrographs

An ISI Model 60-A scanning electron microscope was used to make the enclosed photomicrographs of the rotor and stator surfaces. Each photomicrograph has a short description and in all cases the photos were taken with the sliding direction running between the lower left and upper right corners (approximately). The sketch on Figure B-32 shows the general orientation of the parts in all of the micrographs.

4. SEM EDX X-Ray Element Analysis

A PGT System II Energy Dispersive X-Ray Microanalysis system was used to determine the elemental composition of selected areas on the surface.

The results of the analyses showed considerable wear of both the rotor and stator face areas. The duller areas on both faces had average roughnesses measuring 18, 23 and 29 microinches (CLA or R_a - see data Figs. B-3, B-5, and B-12).

The more reflective areas were considerably smoother with CLA roughnesses of typically around 2 to 3 microinches (see Figs. B-1, B-9, and B-10 of the data).

SEM photomicrographs revealed that the rougher areas resulted from both a general roughening of the cermet and from numerous raised plateaus on the surfaces. Fig. B-33, taken at the inside of the rotor contact area shows apparent delamination due to cracking of the cermet. The adjoining Fig. B-34 shows the resulting roughened surface. Figs. B-35 and B-35a show some of the smoothest areas near the outer edge of the rotor contact. The rest of the rotor surface consisted of raised plateaus and roughened spots as shown in Figs. B-36, B-37 and B-38. These raised plateaus could either be the result of material transfer between the two components, or they could be high spots left over as the majority of the surfaces flakes off as a result of delamination. Fig. B-39 shows features that may be plateaus forming at the edge of a dark, roughened area on the rotor surface, with an enlarged view of one area in Fig. B-40. It is still not conclusive, however, if these are building up as a result of transfer to the edge of this rough spot, or whether they are themselves delaminating flakes.

Three photomicrographs of the stator showed a very similar pattern of roughening plus raised plateaus (Figs. B-41, B-42 and B-43). In addition, Fig. B-41 shows the outside edge of the raised stator contact area where significant wear is taking place.

A total of 16 X-ray microanalyses were performed at various locations on the rotor and stator surfaces. Four analyses on non-contact areas of the rotor and stator were made as a standard, yielding significant amounts of Titanium, Nickel and Molybdenum, as expected for this material. The ratio of elements, which is only approximate with this system, was as follows:

<u>Rotor</u>	<u>Stator</u>
$\frac{\text{Ni}}{\text{Ti}} = .46 \text{ to } .56$	$\frac{\text{Ni}}{\text{Ti}} = .37 \text{ to } .38$
$\frac{\text{Mo}}{\text{Ti}} = .10 \text{ to } .11$	$\frac{\text{Mo}}{\text{Ti}} = .10 \text{ to } .11$

The presence of Niobium may make the Mo/Ti ratio somewhat higher than the true value.

Within the wear areas on both surfaces, the Mo/Ti ratio was consistently the same as the standard reading, but the Ni/Ti ratio was generally lower than the standard. The Ni/Ti ratio was typically between 0.25 and 0.28 in the wear area on both parts. Three analyses, taken directly within the rough, dark spots like those shown in Figs. B-37 and B-39, had even lower Ni/Ti ratios, averaging 0.20.

All of the photomicrographs were made using a back-scattered electron detector which can yield brightness variations due to atomic number differences on the surface being imaged. The dark areas on the surfaces may therefore appear dark because they have the lowest amount of Nickel within their boundaries. No chemical composition difference was noted for other dark areas on the surfaces (such as those on Fig. B-33) unless they also showed significant roughening. Also note that as compared with the standard measurements, all areas within the contact had a significantly lower Nickel content, whether roughened or not. This overall reduction in Nickel may be the result of an oxidative wear process that preferentially removes Nickel by forming Nickel-oxide wear debris.

An unworn area on the rotor face was abraded with silicon-carbide abrasive paper to remove any oxidation and scale. SEM photomicrograph (Fig. B-44a) shows the appearance of the surface before abrading and SEM photo (Fig. B-44b) shows the appearance of this abraded surface.

Four X-ray analyses were performed, corresponding to the four photomicrographs. The approximate ratios of elements (those that can be detected with this system) are:

<u>Abraded Area</u>	<u>Nonabraded Area</u>
$\frac{\text{Ni}}{\text{Ti}} = .15 \text{ to } .16$	$\frac{\text{Ni}}{\text{Ti}} = .48 \text{ to } .49$
$\frac{\text{Mo}}{\text{Ti}} = .09$	$\frac{\text{Mo}}{\text{Ti}} = .08 \text{ to } .10$

Note the significant reduction in the Nickel binder in the abraded areas.

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Rotor, Face(smooth spot) MTIRSHTHI

TIME: 14:13
DATE: 27-MAR-87

-1-

Taylor-Hobson

min =
• mils
• milli inches
• thousands of
an inch

uin = μm

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Rotor, Face(smooth spot) MTIRSHTHI				

Rt1 = 23.22 uin
Rpm = 5.05 uin
Ry = 31.47 uin
Rt1 = 31.47 uin
Rt2 = 19.98 uin
Rt3 = 30.36 uin
Rt4 = 22.08 uin
Rt5 = 12.19 uin

SLOPE = -.11 Deg
Lo = 49.272 in
Rp = 6.32 uin
Rv = 25.76 uin
Rt = 32.08 uin

Ra = 1.99 uin
Rq = 3.58 uin
Rsk = -3.6
Rku = 28.0
Delq = 2.16 Deg
Lamq = 594.84 uin
S = 336.33 uin
Sm = 763.98 uin

Ra = CLA
Rq = RMS
uin =
• microinches
• millionths of
an inch

TIME: 14:13
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Rotor, Face(smooth spot)				MTIRSMTH!

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 14:13
DATE: 27-MAR-87

-3-

Taylor-Hobson

HISTOGRAM

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Rotor, Face(rough spot)				MTIRRGH!

TIME: 14:16
DATE: 27-MAR-87

-1-

Taylor-Hobson

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Rotor, Face(rough spot)				MTIRCHI

STATISTICS

Rtm = 141.49 uin	SLOPE = .03 Deg	Ra = 28.94 uin
Rpm = 80.38 uin		Rq = 35.80 uin
Ry = 187.77 uin		Rsk = .0
Rt1 = 187.77 uin	Lo = 49.272 min	Rku = 2.9
Rt2 = 142.54 uin	Rp = 97.40 uin	Delq = 9.10 Deg
Rt3 = 133.47 uin	Rv = 99.50 uin	Lamq = 1.405 min
Rt4 = 144.55 uin	Rt = 196.90 uin	S = 504.50 uin
Rt5 = 99.12 uin		Sm = 2.056 min

TIME: 14:16
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Rotor, Face(rough spot)				MTIRCHI

HISTOGRAM

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 14:16
DATE: 27-MAR-87

-3-

Taylor-Hobson

- F1 - Analysis
- F2 - Reference.....STRAIGHT
- F3 - Compute
- F4 - Expand
- F5 - Datum
- F6 - Left Limit
- F7 - Right Limit *

MULTIPLE REFERENCE ANALYSIS

DATUM CO-ORDINATES
 Z-AXIS = 2.723 min
 X-AXIS = 12.099 min

ACTIVE CURSOR: C1

Taylor-Hobson

EXPANSION
 OF
 AREA
 BETWEEN
 CURSORS
 ↓

- F1 - Analysis
- F2 - Graph
- F3 - Dump
- F4 - Expand
- F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.09 in	STRAIGHT	0 %
MT1 Seal Stator Face 12:00		MTIS12

Peak To Valley = 410.56 uin

TIME: 14:36
 DATE: 27-MAR-87

-1-

Taylor-Hobson

UNFILTERED

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face 12:00			MTIS12	

STANDARD
ROUGHNESS
FILTERING

TIME: 14:37
DATE: 27-MAR-87

-1-

Taylor-Hobson

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face 12:00			MTIS12	

STATISTICS
(FILTERED)

Rt0 = 122.96 uin
Rpm = 55.55 uin
Ry = 261.40 uin
Rt1 = 133.25 uin
Rt2 = 87.01 uin
Rt3 = 89.02 uin
Rt4 = 145.73 uin
Rt5 = 261.40 uin
Rt6 = 94.89 uin
Rt7 = 49.44 uin

SLOPE = .18 Deg
dSLOPE = .00 Deg
Lo = 72.872 min
Rp = 98.65 uin
Rv = 198.56 uin
Rt = 297.21 uin

Ra = 17.95 uin
Rq = 30.50 uin
Rsk = -2.6
Rku = 17.8
Delq = 5.76 Deg
Lamq = 1.900 min
S = 709.29 uin
Sm = 1.533 min

TIME: 14:37
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face		12:00	MTIS12	

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 14:37
DATE: 27-MAR-87

-3-

Taylor-Hobson

F1 - Analysis
F2 - Reference.....STRAIGHT
F3 - Compute
F4 - Expand
F5 - Datum
F6 - Left Limit *
F7 - Right Limit

MULTIPLE REFERENCE ANALYSIS

DATUM CO-ORDINATES	
Z AXIS =	2.723 min
X AXIS =	12.099 min

ACTIVE CURSOR: C2

Taylor-Hobson

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face 12:00			MTIS12	

EXPANSION
OF AREA
BETWEEN
CURSORS
(FILTERED)

TIME: 14:40
DATE: 27-MAR-87

-1-

Taylor-Hobson

STATISTICS

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face 12:00			MTIS12	

Rtm = 30.00 uin
Rpm = 9.62 uin
Ry = 42.21 uin
Rt1 = 42.21 uin
Rt2 = 25.15 uin
Rt3 = 22.64 uin

SLOPE = -1.08 Deg
dSLOPE = .00 Deg
Lo = 37.563 min
Rp = 32.12 uin
Rv = 30.13 uin
Rt = 62.25 uin

Ra = 7.38 uin
Rq = 9.57 uin
Rsk = .8
Rku = 4.5
Delq = 1.37 Deg
Lamq = 2.516 min
S = 886.10 uin
Sm = 2.848 min

TIME: 14:40
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
HTI Seal Stator Face		12:00	MTIS12	

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 14:40
DATE: 27-MAR-87

-3-

Taylor-Hobson

HISTOGRAM

F1 - Analysis
F2 - Reference.....STRAIGHT
F3 - Compute
F4 - Expand
F5 - Datum
F6 - Left Limit *
F7 - Right Limit

MULTIPLE REFERENCE ANALYSIS

DATUM CO-ORDINATES	
Z_AXIS	= 2.723 min
X_AXIS	= 12.099 min

ACTIVE CURSOR: C2

Taylor-Hobson

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.003 in	PC	STRAIGHT	0 %
MTI Seal Stator Face		12:00	MTIS12	

STATISTICS
ON AREA
BETWEEN
CURSORS

Rtm = 11.74 uin	SLOPE = -.95 Deg	Ra = 2.43 uin
Rpm = 5.04 uin	dsLOPE = .00 Deg	Rq = 3.10 uin
Ry = 17.14 uin	Lo = 17.155 min	Rsk = -.2
Rt1 = 7.92 uin	Rp = 7.76 uin	Rku = 3.3
Rt2 = 12.42 uin	Rv = 9.38 uin	Delq = .94 Deg
Rt3 = 17.14 uin	Rt = 17.14 uin	Lamq = 1.183 min
Rt4 = 14.53 uin		S = 531.69 uin
Rt5 = 6.67 uin		Sm = 1.446 min

TIME: 14:42
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.003 in	PC	STRAIGHT	0 %
MTI Seal Stator Face		12:00	MTIS12	

HISTOGRAM

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 14:42
DATE: 27-MAR-87

-3-

Taylor-Hobson

PROFILE

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face(smooth spot)				MTISSMTH1

TIME: 14:46
DATE: 27-MAR-87

-1-

Taylor-Hobson

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face(smooth spot)				MTISSMTH1

Rtm = 18.71 uin	SLOPE = .05 Deg	Ra = 2.29 uin
Rpm = 6.23 uin		Rq = 3.56 uin
Ry = 26.19 uin	Lo = 49.272 in	Rsk = -1.9
Rt1 = 5.70 uin	Rp = 9.27 uin	Rku = 9.0
Rt2 = 21.43 uin	Rv = 18.08 uin	Delq = 1.63 Deg
Rt3 = 26.19 uin	Rt = 27.34 uin	Lamq = 788.58 uin
Rt4 = 25.21 uin		S = 325.12 uin
Rt5 = 15.03 uin		Sm = 939.75 uin

TIME: 14:46
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - Rk RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face(smooth spot)				MTISSHTH1

AMPLITUDE DISTRIBUTION

BEARING RATIO

Height	Depth
3.39 uin	5.88 uin

Ip%
10.0

TIME: 14:46
DATE: 27-MAR-87

-3-

Taylor-Hobson

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face(rough spot)				MTISRGH1

Peak To Valley = 155.88 uin

TIME: 14:48
DATE: 27-MAR-87

-1-

Taylor-Hobson

PROFILE
(STATOR)

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face(rough spot)				MTISRGH1

Rtm = 129.00 uin	SLOPE = -.06 Deg	Ra = 23.42 uin
Rpm = 78.37 uin		Rq = 38.19 uin
Ry = 155.08 uin	Le = 49.272 min	Rsk = .8
Rt1 = 108.44 uin	Rp = 94.51 uin	Rku = 3.3
Rt2 = 118.96 uin	Rv = 60.57 uin	Delq = 9.57 Deg
Rt3 = 137.26 uin	Rt = 155.08 uin	Lamq = 1.125 min
Rt4 = 125.27 uin		S = 434.05 uin
Rt5 = 155.08 uin		Sm = 1.300 min

TIME: 14:48
DATE: 27-MAR-87

-2-

Taylor-Hobson

F1 - Analysis
F2 - RR RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI Seal Stator Face(rough spot)				MTISRGH1

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 14:48
DATE: 27-MAR-87

-3-

Taylor-Hobson

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 12:00 MTIR12

TIME: 12:21
DATE: 25-MAR-87

-1-

Taylor-Hobson

PROFILES
RADIALLY
OUTWARD
ACROSS ROTOR
FACE. 12:00
POSITION
CORRESPONDS TO
BLACK MARK,
AND ALSO TO
PIT ON BACK
OF PART.
POSITIONS ARE
REFERENCE TO
VIEWING PART
FACE-ON.

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 1:30 MTIR130

TIME: 12:23
DATE: 25-MAR-87

-1-

Taylor-Hobson

F1 - Analysis
 F2 - Graph
 F3 - Dump
 F4 - Expand
 F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 3:00 MTIR3

TIME: 12:25
 DATE: 25-MAR-87

-1-

Taylor-Hobson

F1 - Analysis
 F2 - Graph
 F3 - Dump
 F4 - Expand
 F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 4:30 MTIR430

TIME: 12:26
 DATE: 25-MAR-87

-1-

Taylor-Hobson

F1 - Analysis
 F2 - Graph
 F3 - Dump
 F4 - Expand
 F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 6:00 MTIR6

TIME: 12:27
 DATE: 25-MAR-87

-1-

Taylor-Hobson

F1 - Analysis
 F2 - Graph
 F3 - Dump
 F4 - Expand
 F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 7:30 MTIR730

TIME: 12:28
 DATE: 25-MAR-87

-1-

Taylor-Hobson

F1 - Analysis
 F2 - Graph
 F3 - Dump
 F4 - Expand
 F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 9:00 MTIR9

TIME: 12:29
 DATE: 25-MAR-87

-1-

Taylor-Hobson

F1 - Analysis
 F2 - Graph
 F3 - Dump
 F4 - Expand
 F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.50 in	STRAIGHT	0 %
MTI Seal Rotor, Across face at 10:30 MTIR1030

TIME: 12:30
 DATE: 25-MAR-87

-1-

Taylor-Hobson

Total depth of wear track measured at eight locations around track.

Upper Number - Depth relative to average unwork surface level (mils)

Lower Number - Total Depth relative to highest point on shoulder of wear track (mils).

UNWORN AREA
ON ROTOR ABRASION
WITH S:C PAPER.
PROFILE ACROSS
ABRASIONS

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.10 in	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, ABR. RADIAL MTISTDI

TIME: 11:33
DATE: 18-APR-87

-1-

Taylor-Hobson

UNFILTERED
PROFILE.

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, ABR. RADIAL MTISTDI

TIME: 11:34
DATE: 18-APR-87

-1-

Taylor-Hobson

FILTERED
WITH .010 in.
CUT-OFF

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNIFORM AREA, ABR. RADIAL MTISTDI				

Rt1 = 36.58 uin
 Rq = 11.03 uin
 Ry = 49.17 uin
 Rt1 = 40.35 uin
 Rt2 = 35.18 uin
 Rt3 = 27.44 uin
 Rt4 = 39.30 uin
 Rt5 = 41.54 uin
 Rt6 = 35.57 uin
 Rt7 = 24.02 uin
 Rt8 = 49.17 uin

SLOPE = -.07 Deg
 Lo = 88.345 min
 Rp = 15.40 uin
 Rv = 33.77 uin
 Rt = 45.17 uin

Ra = 5.10 uin
 Rq = 7.09 uin
 Rsk = -1.6
 Rku = 6.5
 Delq = 3.42 Deg
 Lamq = 745.93 uin
 S = 326.42 uin
 Sm = 624.02 uin

STATISTICS
 Ra = CLA =
 5.10
 micro inches

TIME: 11:34
 DATE: 18-APR-87

-2-

Taylor-Hobson

F1 - Analysis
 F2 - PK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNIFORM AREA, ABR. RADIAL MTISTDI				

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 11:34
 DATE: 18-APR-87

-3-

Taylor-Hobson

HISTOGRAM

ABRADED
AREA,
ALONG
ABRASIONS

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.10 in	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, ABR. CIRCUMF, MT1STD2

Peak To Valley = 43.03 uin

TIME: 11:36
DATE: 18-APR-87

-1-

Taylor-Hobson

UNFILTERED

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, ABR. CIRCUMF, MT1STD2

Peak To Valley = 19.88 uin

TIME: 11:37
DATE: 18-APR-87

-1-

Taylor-Hobson

FILTERED

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, ABR. CIRCUMF, MTISTD2				

RtM = 11.64 uin
 Rpm = 4.18 uin
 Ry = 19.27 uin
 Rt1 = 10.31 uin
 Rt2 = 8.59 uin
 Rt3 = 9.23 uin
 Rt4 = 9.93 uin
 Rt5 = 17.62 uin
 Rt6 = 9.58 uin
 Rt7 = 8.60 uin
 Rt8 = 12.37 uin

SLOPE = .01 Deg
 Lo = 80.346 min
 Rp = 5.21 uin
 Rv = 14.68 uin
 Rt = 19.88 uin

Ra = 2.01 uin
 Rq = 2.62 uin
 Rsk = -1.2
 Rku = 5.6
 Delq = .83 Deg
 Lamq = 1.140 min
 S = 495.95 uin
 Sm = 1.277 min

FILTERED

TIME: 11:37
 DATE: 18-APR-87

-2-

Taylor-Hobson

F1 - Analysis
 F2 - RM RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, ABR. CIRCUMF, MTISTD2				

AMPLITUDE DISTRIBUTION

BEARING RATIO

FILTERED

TIME: 11:37
 DATE: 18-APR-87

-3-

Taylor-Hobson

UNWORN
AREA ON ROTOR
PROFILE RADIAL
OUTWARD.

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	10 in	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL, HTISTD3

TIME: 11:36
DATE: 18-APR-87

-1-

Taylor-Hobson

UNFILTERED

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL, HTISTD3

TIME: 11:40
DATE: 18-APR-87

-1-

Taylor-Hobson

FILTERED

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL,				MTIST23

Rpm = 30.42 uin
 Rpm = 13.16 uin
 Rv = 40.13 uin
 Rt1 = 27.51 uin
 Rt2 = 33.96 uin
 Rt3 = 29.96 uin
 Rt4 = 28.66 uin
 Rt5 = 27.53 uin
 Rt6 = 40.13 uin
 Rt7 = 28.46 uin
 Rt8 = 29.27 uin

SLOPE = -.88 Deg
 Lo = 80.346 min
 Rp = 16.59 uin
 Rv = 23.54 uin
 Rt = 40.13 uin

Ra = 4.26 uin
 Rq = 5.47 uin
 Rsk = -.5
 Rku = 3.5
 Delq = 3.37 Deg
 Lamq = 582.88 uin
 S = 303.79 uin
 Sm = 542.25 uin

TIME: 11:40
 DATE: 18-APR-87

-2-

Taylor-Hobson

FILTERED

F1 - Analysis
 F2 - R4 RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL,				MTIST23

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 11:40
 DATE: 18-APR-87

-3-

Taylor-Hobson

FILTERED

UNWORN
AREA,
CIRCUMFER-
ENTIAL

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	10 in	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF., MTISTD4

UNFILTERED

TIME: 11:41
DATE: 18-APR-87

-1-

Taylor-Hobson

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF., MTISTD4

FILTERED

TIME: 11:42
DATE: 18-APR-87

-1-

Taylor-Hobson

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF.				MT1STD4

Rtm = 31.50 uin	SLOPE = .01 Deg	Ra = 4.02 uin
Rpm = 11.58 uin		Rq = 5.34 uin
Ry = 41.76 uin	Lo = 80.346 min	Rsk = -1.1
Rt1 = 29.41 uin	Rp = 13.52 uin	Rku = 6.3
Rt2 = 24.38 uin	Rv = 29.33 uin	Delo = 3.02 Deg
Rt3 = 41.76 uin	Rt = 42.85 uin	Lamq = 635.10 uin
Rt4 = 33.82 uin		S = 301.78 uin
Rt5 = 36.19 uin		Sm = 554.32 uin
Rt6 = 25.06 uin		
Rt7 = 28.41 uin		
Rt8 = 32.95 uin		

TIME: 11:42
DATE: 18-APR-87

-2-

Taylor-Hobson

FILTERED

F1 - Analysis
F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF.				MT1STD4

AMPLITUDE DISTRIBUTION

DEARING RATIO

TIME: 11:42
DATE: 18-APR-87

-3-

Taylor-Hobson

FILTERED

UNWORN
AREA, RADIAL
OUTWARD

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.10 in	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL, MTISTDS

Peak To Valley = 88.66 uin

TIME: 11:44
DATE: 18-APR-87

-1-

Taylor-Hobson

UNFILTERED

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL, MTISTDS

Peak To Valley = 43.99 uin

TIME: 11:44
DATE: 18-APR-87

-1-

Taylor-Hobson

FILTERED

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL, MTISTD5				

Rm = 34.79 uin
 Rpm = 13.70 uin
 Ry = 38.55 uin
 Rt1 = 32.68 uin
 Rt2 = 37.40 uin
 Rt3 = 38.55 uin
 Rt4 = 35.13 uin
 Rt5 = 37.14 uin
 Rt6 = 36.17 uin
 Rt7 = 31.72 uin
 Rt8 = 39.58 uin

SLOPE = -.08 Deg
 Lo = 80.345 uin
 Rp = 20.19 uin
 Rv = 23.80 uin
 Rt = 43.99 uin

Ra = 4.66 uin
 Rq = 6.10 uin
 Rsk = -.9
 Rku = 4.3
 Delq = 3.42 Deg
 Lamq = 641.30 uin
 S = 301.51 uin
 Sm = 557.64 uin

TIME: 11:44
 DATE: 18-APR-87

-2-

Taylor-Hobson

FILTERED

F1 - Analysis
 F2 - RK RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, RADIAL, MTISTD5				

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 11:44
 DATE: 18-APR-87

-3-

Taylor-Hobson

FILTERED

UNWORN
AREA,
CIRCUMFERENTIAL

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Traverse Length	Reference	Ignore
UNFILTERED	.10 in	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF., MTISTD6

UNFILTERED

F1 - Analysis
F2 - Graph
F3 - Dump
F4 - Expand
F5 - Exclude

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF., MTISTD6

FILTERED

TIME: 11:47
DATE: 10-APR-87

-1-

Taylor-Hobson

FILTERED

F1 - Analysis

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF.,				MTISTD6

Rtm = 29.62 uin	SLOPE = .00 Deg	Ra = 4.11 uin
Rpm = 12.75 uin		Rq = 5.41 uin
Ry = 40.33 uin	Lo = 90.346 min	Rsk = -.6
Rt1 = 32.79 uin	Rp = 21.34 uin	Rku = 4.8
Rt2 = 22.20 uin	Rv = 26.96 uin	Deiq = 2.96 Deg
Rt3 = 36.60 uin	Rt = 48.30 uin	Lamq = 656.15 uin
Rt4 = 25.12 uin		S = 300.00 uin
Rt5 = 40.33 uin		Sm = 535.30 uin
Rt6 = 40.08 uin		
Rt7 = 23.33 uin		
Rt8 = 21.49 uin		

TIME: 11:47
DATE: 18-APR-87

-2-

Taylor-Hobson

F1 - Analysis
F1 - RA RESULTS

Mode	Cut Off	Filter	Reference	Ignore
ROUGHNESS	.010 in	PC	STRAIGHT	0 %
MTI ROTOR, UNWORN AREA, CIRCUMF.,				MTISTD6

AMPLITUDE DISTRIBUTION

BEARING RATIO

TIME: 11:47
DATE: 18-APR-87

-3-

Taylor-Hobson

FILTERED

STATOR 1 Time: 11:20 Date: 03/31/87

RMS: 63.1nm SURFACE P-V: 500nm

RA: 47.9nm

SLIDING
DIRECTION

Attempted WYKO 3-D Optical Profile
of Smooth Area on Wear Track of Stator
Face

STATOR 2 Time: 11:34 Date: 03/31/87

RMS: 42.0nm

SURFACE

P-V: 263nm

RA: 33.8nm

SLIDING
←————→
DIRECTION

Attempted WYKO 3-D Optical Profile
of Smooth Area on Wear Track of
Stator Face

No. 1

25X

SEM Photomicrograph of Rotor Wear Area

No. 2

200X

No. 3

200X

SEM Photomicrographs of Rotor Contact Area

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No. 4a

300X

No. 4b

1000X

200X

Photo No. 5

Raised Plateaus in Rough Area of Rotor Wear Track Surfaces

No. 6

300X

No. 7

1000X

SEM Photomicrographs of Typical Dark and Rough Spots
on Rotor Wear Track Surfaces

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No. 8

250X

No. 9

1000X

Rough Area on Wear Track of Rotor and Enlargement of
Edge of Rough Spot

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No. 10

200X

Outer Edge of Interfacial Zone (Wear Track) on
Stator Surface

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No. 11

(Smoother Area)

500X

No. 12

300X

SEM Photos of Stator Wear Area, Smoother Area, Raised Plateaus

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Abraded Area on Rotor

500X

Nonabraded Area on Rotor

500X

DOE/MC/22074-2533

DEVELOPMENT OF HIGH TEMPERATURE, HIGH PRESSURE ROTATING SHAFT SEALS

DOE