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ANALYSIS OF EXPERIMENTAL SHAFT SEAL DATA FOR HIGH-PERFORMANCE

TURBOMACHINES - AS FOR SPACE SHUTTLE MAIN ENGINES

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SUMMARY

High-pressure, high-temperature seal flow (leakage) data for nonrotating and rotating Rayleigh-step and convergent-tapered-bore seals have been characterized in terms of a normalized flow coefficient. The results for normalized Rayleigh-step and nonrotating tapered-bore seals were in reasonable agreement with theory, but the data for rotating tapered-bore seals were not. The tapered-bore-seal operational clearances estimated from the flow data were significantly larger than calculated. Although clearances are influenced by wear from conical to cylindrical geometry and by errors in clearance corrections, the problem was isolated to the shaft temperature - rotational speed clearance correction. The geometric changes support the use of some conical convergence in any seal. Under these conditions rotation reduced the normalized flow coefficient by nearly 10 percent.

INTRODUCTION

Few data are available in the literature to guide seal designing for high-performance turbomachines. Although reference 1 discusses several examples of instabilities, along with some design recommendations, it gives few data for turbomachines characteristic of the space shuttle main engine (SSME). A seals study program was undertaken (ref. 2) to determine the leakage and wear performance of a Rayleigh-step and a tapered-bore seal. The Rayleigh-step seal's close tolerances and calculated low restoring forces (as designed) made it undesirable for this application. The tapered-bore seal, designed according to reference 3, appeared to have sufficient stiffness, acceptable flow leakages, and tolerable wear characteristics. Tests were conducted at various pressures, mostly above 24 MPa (3500 psi). Nominal initial temperatures were 500 K (440 °F), but some room-temperature checks were made for calibration and comparison. The nominal rotational speed was 483 Hz (29 000 rpm), but some checks were made without rotation. The working fluid was gaseous nitrogen.

The tabulated results of reference 2 could be analyzed by using the normalized flow coefficient method presented in reference 4. The working fluids of reference 4 were ambient-temperature gases. The seal configurations tested were nonrotating, of similar diameter to those of reference 2 but with nominal design clearances nearly three times larger. Using the method of reference 4 could therefore present a scaling problem. However, the Reynolds numbers for the data of both studies were estimated to be of the same order.[1]

The tapered-bore seal data of reference 2 were reanalyzed by using the real-gas theory of reference 4 to determine the applicability of such techniques and to assess the wear performance. First, the real-gas theory was shown to give results in reasonable agreement with those of other theories for flow rates in a Rayleigh-step seal. The theory was then compared with the tapered-bore-seal data of reference 2. For no rotation, with and without clearance corrections, there was reasonable agreement. However, for rotation with clearance corrections the experimental flow ratio differed significantly from the theoretical (to 1.6). These differences had to arise either from wear or from rotation or temperature clearance corrections. The wear rates are shown to be in reasonably good agreement with theory, on the basis of flow data. Therefore the problem must rest in the rotation or temperature clearance corrections. A 10 to 20 percent change in these values is shown to be sufficient to account for much of the deviation.

SYMBOLS

A	area
D	diameter
D_h	hydraulic diameter, $4A/p$
C_d	flow loss coefficient (e.g., entrance)
C_f	flow coefficient
E	Young's modulus
e	clearance tolerance correction

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[1] $Re = \dot{w}D_h/A$; for a shaft seal, $D_h = 2h$ and $D_h/A = 2/\pi\ (D + h) \sim 1/\pi R$. Reynolds similarity criteria requires $Re_1 = Re_2$ or $(\dot{w}_1/\dot{w}_2)(\eta_2/\eta_1)(D_2 + h_2)/(D_1 + h_1) = Y$. Here $D_1 = D_2 >> h_1 > h_2$ and $\eta_2 = \eta_1 \sqrt{T_2/T_1}$. With typical values of $\dot{w}_1/\dot{w}_2 = 1/2$ and $T_2/T_1 = 533/300$, $Y \rightarrow 2/3 = [0]1$.

G mass flux

G* mass flux normalizing parameter, $\sqrt{\rho_c P_c / Z_c}$ (G* = 6010 g/cm^2 s (85.5 lb/in^2 s) for nitrogen

G_r normalized mass flux, G/G*

h clearance

h* clearance based on profilometer data

L seal length

R radius

$\mathcal{R}$ gas constant

Re Reynolds number

P pressure

P_r reduced pressure, P/P_c

p "wetted" perimeter

T temperature

T_r reduced temperature, T/T_c

u_r radial displacement

w mass flow rate

X area ratio parameter

Y similarity measure parameter

Z compressibility, $P/\rho \mathcal{R} T$

η viscosity

υ Poisson's ratio

ρ density

χ flow parameter (eq. (17))

Ω angular velocity

Subscripts:

c thermodynamic critical point

calc calculated

cyl cylinder

des design

e exit

exp experimental

inst manufacturing, alignment, and other clearance tolerances

i inlet

op operational

ΔP pressure clearance tolerance

Δrpm rotational speed clearance tolerance

s seal; static component

sh shaft

ΔT temperature clearance tolerance

0 stagnation or reference

Superscript:

$-$ average

APPARATUS AND INSTRUMENTATION

The materials in this section are adapted from reference 2 and are presented here to clarify the nature of the experiment and the types of measurement available for analysis. For detailed information see reference 2. The turbine seal test assembly (fig. 1) could be driven to 3142 rad/s (30 000 rpm) by the apparatus motor. High-pressure nitrogen gas (to 25 MPa; 3600 psia) was heated in a counterflow heat exchanger to 533 K (500 °F) before it passed through the seal assembly. The seal inlet temperature decreased over the course of the run because of the high mass flow rates through the heat exchanger. The basic measurements were temperature, pressure, and flow rate. Typical instrumentation locations are illustrated in figure 2, with the flowmetering system not shown. A sketch of the tapered-bore seal is shown in figure 3 and the actual seal, in figure 4.

RESULTS AND ANALYSIS

The flow rate (leakage) analysis of reference 4 (without rotation) was applied to the Rayleigh-step and tapered-bore seals described in reference 2 (with and without rotation). The contribution of the friction terms was small relative to that of the inertia terms and for gas flows may be related by using the normalized flow coefficient

$$\frac{G_r \sqrt{T_{r,0}}}{P_{r,0}} = C_f = \frac{C_d}{5} \tag{1}$$

For the tests conducted in reference 4

$$C_f = 0.153 \quad \text{for concentric position (no rotation)}$$

$$= 0.158 \quad \text{for fully eccentric position (no rotation)} \tag{2}$$

which implies that $C_d = 0.8$ for those similar (geometric and kinematic) configurations.

In the following sections we demonstrate the application of equations (1) and (2). Theoretical comparisons were made for the Rayleigh-step seal, and several cases were investigated for the tapered-bore seal, with and without rotation and correlation of flow data, to isolate a clearance correction problem.

Rayleigh-Step Seal - Theoretical Comparisons

The flow rate (leakage) analysis of reference 4 was compared with the theoretical results of reference 2 (p. 24) for the design of the Rayleigh-step seal.[2] For the primary seal with a design pressure of 27.6 MPa (4000 psia) and a design temperature of 533 K (500 °F), the parameters used in equation (1) became[3] $P_{r,0} = 8.07$, $T_{r,0} = 4.22$, $A_e = 0.103$ cm^2, $G^* = 6010$ g/cm^2 s, $C_d = 0.6$, $G = 2833$ g/cm^2 s and the flow rate became $\dot{w} = GA_e = 292$ g/s (0.64 lb/s, or 495 ft^3/min). For the secondary seal with a design pressure of 0.059 MPa (8.5 psia) and a design temperature of 533 K (500 °F), the flow rate became $\dot{w} = 6.21$ g/s (0.014 lb/s, or 10.5 ft^3/min). Flow rates predicted by the methods of references 4 to 6 (table I) are in reasonable agreement.

Tapered-Bore Seal - Flow Rate Comparisons

Since the theoretical agreement between the results of references 4 to 6 is reasonably good, we now propose to apply the normalized flow coefficient method to the tapered-bore-seal data of reference 2. The seal was designed for operation at 483 Hz (29 000 rpm) with an inlet-to-exit clearance ratio of 1.8 for a nominal exit diametral clearance of 0.051 mm (0.002 in) at 26.2 MPa (3800 psia) and 533 K (500 °F). The average flow area for a conical tapered-bore seal is (ref. 4)

$$\bar{A} = \frac{A_i + 2A_e}{3} \tag{3a}$$

or in terms of the inlet and exit clearances (fig. 3)

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[2]The seal passage was slightly divergent.

[3]For fluid nitrogen, $P_c = 3.417$ MPa (495.5 psia); $T_c = 126.3$ K (-232.3 °F); $G^* = 6010$ g/cm^2 s (85.5 lb/in^2 s).

$$\frac{\bar{A}}{A_e} = \frac{h_i + 2h_e}{3h_e} = 1.27 \qquad (3b)$$

In turbomachinery, radial clearance h and tolerance corrections e due to temperature, pressure, manufacturing, alignment, and operations are significant and must enter into the analysis. The equation for h can be written as

$$h = h_{inst} + (e_{\Delta T} + e_{\Delta P})_s - (e_{\Delta T} + e_{\Delta rpm})_{sh} \qquad (4)$$

Flow rate comparison without rotation. - For the data given in table XII of reference 2 (build 10 - pretest 108,6) without rotation (0 rpm), we find for 20.1 MPa (2915 psia), 299 K (78 °F), and the geometry of figure 3 that $T_{r,0} = 2.36$ and $P_{r,0} = 5.88$. The nominal clearance can be estimated by using the diametral clearances presented in table II, assuming that $e_{\Delta T}$ and $e_{\Delta P}$ vary linearly with temperature and pressure, respectively, and that half of the shaft growth is due to temperature and is linearly dependent on both temperature and rotational speed. Since the test (build 10 - pretest 108,6) was run at ambient temperature (300 K; 78 °F) and 20 MPa (2900 psia) without rotation, the diametral clearance corrections were estimated (table III). Substituting these estimated clearance corrections into equation (4), the diametral clearance 2h becomes 0.1956 mm (0.0077 in). From equation (3) the effective flow area becomes 0.227 cm^2, and from equation (1) the estimated flow rate, $\dot{w} = 1044 \times C_d$ g/s ($2.30 \times C_d$ lb/s), is about 20 percent higher than the measured value of 700 g/s (1.54 lb/s) for $C_d = 0.8$.[4] For this run, $C_d = 0.7$ would bring satisfactory agreement.[5]

Flow rate comparison with rotation. - We now apply the same method to the rotational data (build 12 - test 120), where the rotational speed is 469 Hz (28 150 rpm); pressure, 24.8 MPa (3590 psia); and temperature, 510 K (458 °F). With the clearance corrections given in table III the average flow area is determined from equations (3) and (4) with $2h_i = 0.114$ mm (0.0045 in) and $2h_e = 0.064$ mm (0.0025 in) as $A = 0.082$ cm^2. From equation (1), the mass flow rate becomes $\dot{w} = 356 \times C_d$ g/s ($0.785 \times C_d$ lb/s), which is a factor of 1.6 lower than the experimental value of 584 g/s (1.29 lb/s) for $C_d \rightarrow 1$ and a factor of 2.3 lower for $C_d \rightarrow 0.7$. So how can one account for being nearly correct at no rotation and a factor of 2.3 low with rotation. It would appear that the clearance tolerances of tables II and III and equations (3) and (4) require further investigation to determine whether the disparity is solely the effect of rotation.

Tapered-Bore Seal Operational Flow Area and Rotation Effects

As an initial effort to determine the effects of operational flow area and rotation on flow rate, some of the data of reference 2 (p. 69, table XII) were recalculated in terms of the normalized flow coefficient. Rewriting equation (1) gives

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[4]Without clearance correction and without rotation, from equations.
[5]At high pressure the seal face is strained, rotates, and may change C_d.

$$C_f = \frac{\dot{w}_{exp}}{\dot{w}_{calc}} = \frac{\dot{w}_{exp}}{G^*\bar{A}P_{r,0}/\sqrt{T_{r,0}}} \tag{5}$$

and, with $\bar{A} = 1.27\ A_e$, the values are given as a function of $P_{r,0}$ in figure 5.

To make this comparison, the clearance tolerances of equation (4) were held fixed (i.e., constant clearance for expediency). As is clear from figure 5, there is considerable scatter for the pump-end primary seal. Similar results were found for the turbine-end primary seal. However, the trend with pressure was predictable since the clearance was held constant. This further indicates the necessity of properly incorporating equation (4) in the design analysis. It is possible that the experimental flow area for these tests differs from the design area by a factor of 3.[6]

Effect of rotation. - The effect of rotation (fig. 5) can be estimated as follows:

$$\frac{C_{f,rotation}}{C_{f,no\ rotation}} = \frac{0.137}{0.15} = 0.91 \tag{6}$$

This establishes, experimentally, about a 10-percent decrease in flow rate due to rotation independent of flow area.

Effect of wear on flow rate. - If the tapered-bore seal wore cylindrical, the flow area ratio would be

$$\frac{\bar{A}_{cyl}}{\bar{A}_{des}} = \frac{1.84\ A_e}{1.27\ A_e} = 1.45 \tag{7}$$

Because the seals did not exhibit such extensive wear and the area increase cannot account for the disparity in flow rates, the problem must reside in the shaft rotational speed clearance correction.

Operational flow area. - To resolve the flow area problem, it is first necessary to establish confidence in the flow data of reference 2. A statistical analysis of the 14 high-pressure, high-temperature data points with rotation for the pumpend and turbine-end primary seals is given in figure 6 and summarized in table IV. In general, although the scatter is apparent, it is much less than would be anticipated from the Monte Carlo sampling of a normally distributed sample space of 14 members. This gives confidence in the reproducibility of the flow data.

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[6]To place it in better perspective, a 10-percent error in design diametral clearance 2h is 5 μm. (Recall diametral clearance is 0.0508 mm (0.002 in).)

If the flow data with rotation were in agreement with the results of reference 4, as they are for no rotation with a $C_d = 0.7$ instead of 0.8, the ratio of the operational flow clearances would be defined in terms of the flow coefficients of table IV and equation (2).

$$\frac{\bar{A}_{op}}{\bar{A}_{e,des}} = \frac{C_{f,op}}{C_{f,des}} = \left(\frac{0.8}{0.7}\right)\left(\frac{0.41}{0.153}\right) = 3.1 \tag{8}$$

From equations (3) and (8) the clearance becomes

$$3.1 = \left\{D_i^2 - (D_s + 2h_0)^2 + 2\left[D_e^2 - (D_s + 2h_0)^2\right]\right\}\frac{\pi}{12} \tag{9}$$

Solving gives $2h_0 = 0.117$ mm (0.0046 in) and

$$h_e = 0.083 \text{ mm } (0.00325 \text{ in}) \tag{10}$$

or about three times the design value.[7] This suggests an error in rotation or temperature design clearance (table II).

For a cylinder the radial displacement u_r can be approximated as

$$u_r = \frac{\left(1 - \nu^2\right)\rho_{sh}D_{sh}^3\Omega^2}{64E} \tag{11}$$

and solving gives $u_r = 0.018$ mm (0.0007 in), which is significantly less than cited in table III. For a 50-percent decrease in both the rotation and temperature shaft clearance corrections, the predicted and experimental flow rates would agree, but such an error is difficult to verify and remains only bracketed. We now turn our attention to the problem of wear.

WEAR ANALYSIS

Wear - from Geometry

Combining the definition of the effective flow area $\bar{A}_1$ for a tapered-bore seal (eq. (3)), the optimum taper recommended (ref. 3), and the geometry (fig. 3) gives a pretest, or initial, clearance of

$$h_{i,1} = 1.8\, h_{e,1} \tag{3c}$$

and a flow area of

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[7]It must also be recognized that during the blowdown test the full effect of the temperature may not be felt. Furthermore, some immediate rub-in is imminent.

$$\bar{A}_1 = 1.27\ A_{e,1} \approx 1.27\ \pi D h_{e,1} \tag{3d}$$

The post-test flow area $\bar{A}_2$ is not so easy to establish as it varies from conical to cylindrical; with the aid of figure 7, it can be expressed as

$$\bar{A}_2 = \left[\left(\frac{h_{i,2} + 2h_{e,2}}{3h_{e,2}}\right)(1 - X_0) + X_0\right] A_{e,2} \tag{12}$$

where $h_{i,1} = h_{i,2}$ and X_0 represents the fraction of the seal length that is cylindrical. The values of h_i, h_e, and X_0 are determined from profilometer data (e.g., fig. 8).

Wear - from Flow Data

Changes in the pretest and post-test flow rates provide a relative estimate of wear. From equation (1) we have

$$\frac{C_{f,1}}{C_{f,2}} = \frac{\left(\frac{G_r\sqrt{T_{r,0}}}{P_{r,0}}\right)_1}{\left(\frac{G_r\sqrt{T_{r,0}}}{P_{r,0}}\right)_2} = \left(\frac{\dot{w}_1}{\dot{w}_2}\right)\left(\frac{\bar{A}_2}{\bar{A}_1}\right)\left(\frac{P_{0,2}}{P_{0,1}}\right)\sqrt{\frac{T_{0,1}}{T_{0,2}}} = \chi\left(\frac{\bar{A}_2}{\bar{A}_1}\right) \tag{13}$$

Although χ is readily determined from the flow measurements, changes in C_f are difficult to assess.

Wear - from Profilometer

We can now make wear estimates from the profilometer data (fig. 8) and the flow data for pretest run 159 and post-test run 218 (fig. 108 and tables X and XII of ref. 2).

<u>Geometric wear</u>. - The average diametral wear is 0.038 mm (0.0015 in) at the seal outlet and 0.013 mm (0.0005 in) at the seal inlet (ref. 2, p. 179) in agreement with figure 8. Assume the operating clearances to be $2h_{e,1}$ = 0.165 mm (0.0065 in) (see eq. (10)) and $2h^*_{e,2}$ = 0.203 mm (0.008 in). As both profiles are tapered ($X_0 = 0$), $h^*_{i,1} \approx h^*_{i,2}$ 0.216 mm (0.0085 in).

$$\frac{\bar{A}_2}{\bar{A}_1} = 1.10 \tag{14}$$

<u>Wear by flow parameters</u>. - With $(L/2h) \rightarrow 40$, assume that the passage transition from conical to cylindrical lowers C_f by 5 percent. The operating conditions at 483 Hz (29 000 rpm) are run 159: P_1 = 22.8 MPa (3305 psia), T_1 = 464 K (375 °F), $\dot{w}_1$ = 608 g/s (1.340 lb/s) and $C_{d,1}$ = 0.8; and run 218: P_2 = 23.4 MPa (3390 psia), T_2 = 533 K (500 °F), $\dot{w}_2$ = 610 g/s (1.346 lb/s) and $C_{d,2}$ = 0.76.

The ratio A_2/A_1 becomes

$$\frac{\bar{A}_2}{\bar{A}_1} = \frac{C_{f,1}}{x C_{f,2}} \tag{15}$$

and from the flow parameters

$$\frac{\bar{A}_2}{\bar{A}_1} = \sqrt{\frac{533}{464}} \left(\frac{3305}{3390}\right) \left(\frac{1.346}{1.340}\right) \left(\frac{0.8}{0.76}\right) = 1.1 \tag{16}$$

The agreement between measured and calculated wear is reasonable. However, rub-in is common and, if the initial rub were 10 percent of the total wear, $h^*_{e,1}$ = 0.1689 mm (0.00665 in), calculated and measured wear would not agree.[8] Even so, for these results it is difficult to justify agreement to better than 10 to 15 percent.

STABILITY CONSIDERATIONS

As established in reference 4 and in experimental tests with similar low-L/D tapered- and straight-bore and high-L/D straight-bore seals (three seal configurations, ref. 7), the convergent-tapered-bore seal is at least as stable as the straight-bore seal and more tolerant to clearance problems. Thus a tapered-bore seal worn cylindrical gains dynamic stiffness from the increase in flow leakage.

The wear characteristics for a cylindrical passage could produce a divergent (or convergent) flow passage, leading to decreased (or enhanced) stability. If both divergence and convergence were produced, the leakage rate would increase because $C_f \rightarrow 1$, similar to a venturi with a conjectured loss of stability. Such geometric changes are the subject of another study.

SUMMARY OF RESULTS AND RECOMMENDATIONS

Primary and secondary seals for the pump and turbine ends of an advanced turbomachine have been experimentally studied in reference 2. The high-pressure, high-temperature nitrogen gas flow data of reference 2 have been characterized in terms of a normalized flow coefficient

$$\frac{G_t \sqrt{T_{r,0}}}{P_{r,0}} = C_f$$

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[8] Table X of reference 2 (p. 201) gives total wear values with an average of 0.028 mm (0.0011 in) for system post-test 278, but clearly, from figure 8, post-test 278 values give a wear of 0.057 mm (0.00225 in), a factor of 2 higher than the tabulated data. We feel that figure 8 is correct.

The data for normalized Rayleigh-step and nonrotating tapered-bore seals were in reasonable agreement with theory, but the data for rotating tapered-bore seals were not. The operational flow area had to be enlarged by a factor of 3 in order to match the data. The problem was isolated to the shaft temperature - rotational speed clearance correction. Rotation was found to reduce the normalized flow coefficient by nearly 10 percent independently of the flow area problems.

The flow area increase measured from the wear data and calculated from the flow data was 10 percent. However, rub-in is commonplace and agreement cannot be justified to better than 10 to 15 percent.

Even when normalized in terms of flow rate, the conical convergent seal appears to be more stable than the straight-bore seal. As it wears, it usually becomes more cylindrical; at the same time the flow rate (leakage) increases and the system stiffness can increase.

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TABLE I. - COMPARISON OF THEORETICAL FLOW RATES FOR A RAYLEIGH-STEP SEAL (DESIGN CLEARANCE, h = 0.051 mm (0.002 in)

Theory	Leakage				Percent deviation	
	Primary		Secondary		Primary	Secondary
	g/s	ft^3/min	g/s	ft^3/min		
Isentropic (ref. 5)	15.06	532	0.32	11.3	-7	-8
QUASC (ref. 6)	14.16	500	.303	10.7	-1	-2
Real gas (ref. 4)	14.00	495	.297	10.5	--	--

TABLE II. - DIAMETRAL CLEARANCES FOR TAPERED-BORE PRIMARY SEAL WITH MINIMUM CLEARANCE (ref. 2, p. 29)

[Nominal seal outlet diameter, 6.4668 cm (2.5460 in); nominal inlet-to-exit clearance ratio, 1.8; nominal seal inlet diameter, 6.4720 cm (2.5480 in); design pressure, 26.2 MPa (3800 psia); design temperature, 533 K (500 °F).]

	Diametral clearance,	
	mm	in
Design	0.508	0.002
Installed	.254	.0100
Seal:		
Temperature effect	.112	.0044
Pressure effect	-.0787	-.0031
Shaft - temperature and speed effect	.2286	.009

TABLE III. - ESTIMATED DIAMETRAL CLEARANCE CORRECTIONS FOR SELECTED DATA OF REFERENCE 2, WITH AND WITHOUT ROTATION

	Build 10 - pretest 108,6		Build 12 - test 120	
	mm	in	mm	in
Seal:				
Pressure correction, $2e_{\Delta P}$	-0.061	-0.0024	-0.074	-0.0029
Temperature correction, $2e_{\Delta T}$	.018	.0007	.10	.004
Shaft:				
Temperature correction, $2e_{\Delta T}$	.015	.0006	.10	.004
Rotational speed correction $2e_{\Delta rpm}$	0	0	.117	.0046

TABLE IV. - STATISTICAL VALUES[a] OF NORMALIZED FLOW COEFFICIENT FOR SELECTED DATA OF REFERENCE 2

Seal	Averaged normalized flow coefficient, C_f	Standard deviation	Correlation coefficient
Turbine end	0.414	0.0069	0.985
Pump end	.414	.0066	.977

[a]For a more definitive measure of the normalized flow coefficient, a much larger, well-controlled sample space will be required.

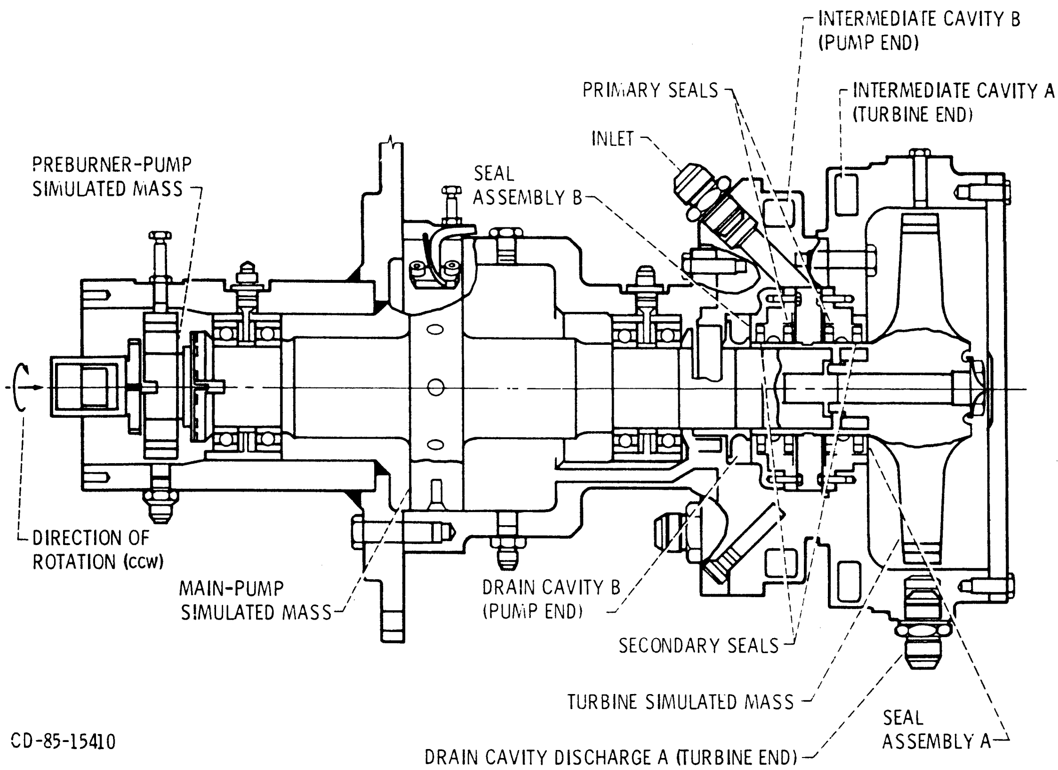

Figure 1. - Schematic of turbine seal test apparatus (ref. 2).

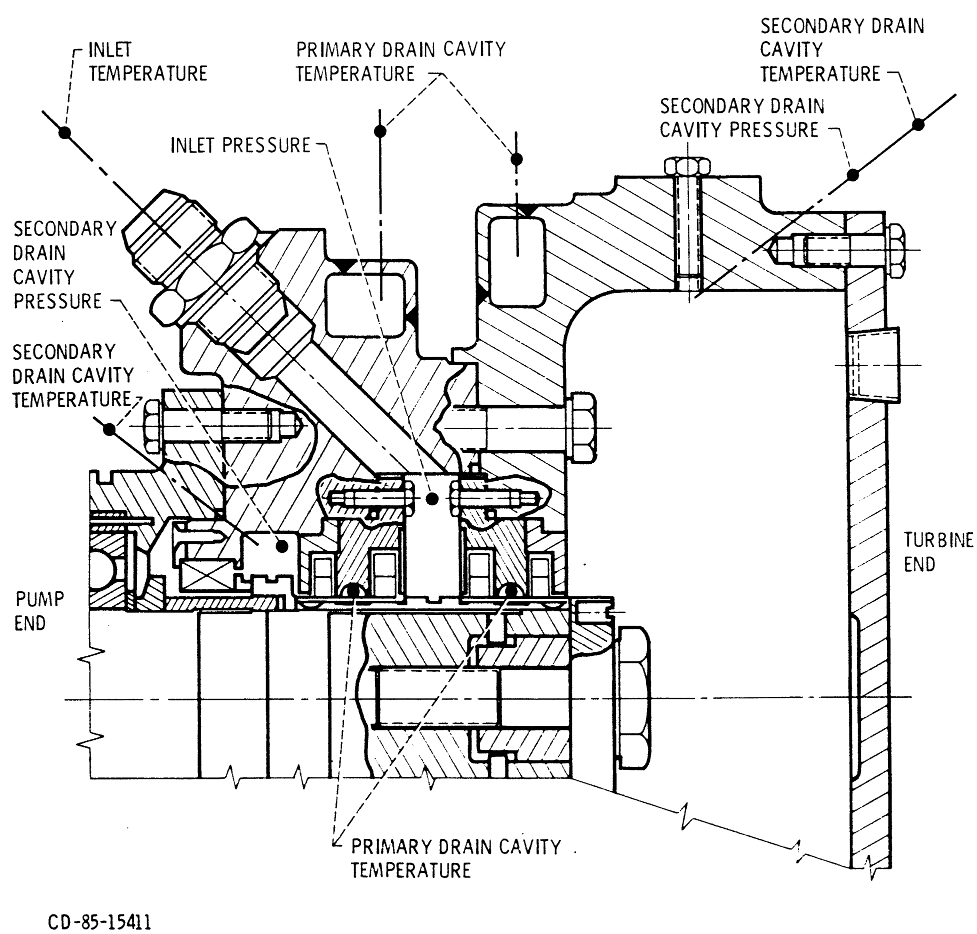

Figure 2. - Schematic of test instrumentation locations (ref. 2).

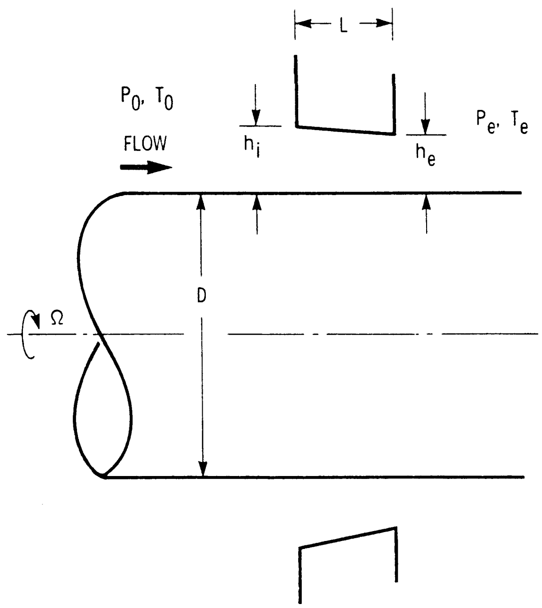

Figure 3. - Sketch of the tapered-bore seal configuration.

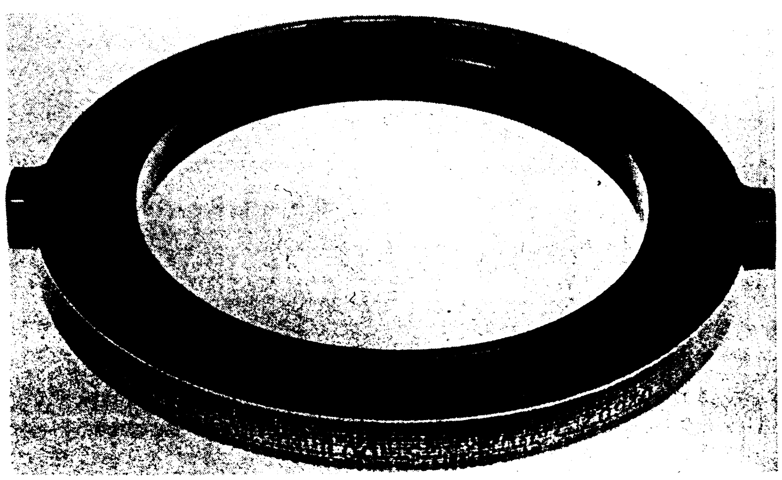

Figure 4. - Tapered-bore seal, (from ref. 2.).

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16. Abstract

High-pressure, high-temperature seal flow (leakage) data for nonrotating and rotating Rayleigh-step and convergent-tapered-bore seals have been characterized in terms of a normalized flow coefficient. The data for normalized Rayleigh-step and nonrotating tapered-bore seals were in reasonable agreement with theory, but data for the rotating tapered-bore seals were not. The tapered-bore-seal operational clearances estimated from the flow data were significantly larger than calculated. Although clearances are influenced by wear from conical to cylindrical geometry and errors in clearance corrections, the problem was isolated to the shaft temperature - rotational speed clearance correction. The geometric changes support the use of some conical convergence in any seal. Under these conditions rotation reduced the normalized flow coefficient by nearly 10 percent.

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