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Abstract

Experimental evaluation under simulated engine conditions revealed that conventional mainshaft seals have disadvantages of high gas leakage rates and wear. An advanced seal concept, the self-acting face seal, has a much lower gas leakage rate and greater pressure and speed capability. In endurance tests (150 hr) to 43 200 rpm the self-acting seal wear was not measurable, indicating noncontact sealing operation was maintained even at this high rotative speed.

A review of published data revealed that the leakage through gas path seals has a significant effect on TSFC, stall margin and engine maintenance. Reducing leakages by reducing seal clearances results in rubbing contact, and then the seal thermal response and wear determines the final seal clearances. The control of clearances requires a material with the proper combination of rub tolerance (abradability) and erosion resistance. Increased rub tolerance is usually gained at the expense of reduced erosion resistance and vice versa.

I. Introduction

A multiplicity of seals are used in gas turbines to restrict gas leakage, provide thrust balancing, maintain thermal gradients, meter cooling gas flow, and protect bearing sumps. Since a gas turbine has many sealing locations (blade tips, interstage, mainshaft, etc.), the accumulative effect of deficient sealing practice can be appreciable. In fact, operating experience reveals that sealing practice has a significant impact in three areas: engine performance, stall margin and general engine maintainability.

The effect of sealing on performance has received some documentation. Reference 1 calculates a 21 percent power loss attributable to gas seal leakages (mainshaft, gas path, flanges, etc.) in a small gas turbine engine. Reference 2 gives a detailed review of the effect of seal clearance on performance of transport and fighter engines. This study indicated that a $2\frac{1}{2}$ percent change in TSFC can be expected for a nominal change in seal clearances.

There is considerable evidence that stall margin can be affected by incorporating various geometries (e.g., grooves) into the blade tip shrouds;(3) also, the tip clearance magnitude affects stall margin.

A study on reliability and maintainability of seven small military engines now in use, shows that one of the main causes of unscheduled engine removal is oil leakage through carbon mainshaft seals.(4) In fact, it ranks as the third most common cause (foreign object damage and improper maintenance being first and second) out of a list of 28 failure modes; and in reference 4 it is stated that "many in the engine design, management, and procurement communities tend to minimize the seal problem and correspondingly downplay the value of improvement in this area."

Dirt ingestion through high leakage seals at the bearing sumps degrades bearing life. Other problems which can occur as a result of high airflow into the bearing lubrication system are increased gearbox pressure (tends to lower seal life) and increased bearing cavity pressure which may limit oil flow into the bearing compartment and thereby affect bearing life.

This paper will consider the state-of-the-art of two general types of seals in a gas turbine: the mainshaft seals and the gas path seals (e.g., compressor blade tip). A comparison of the wear and gas leakage rates obtained for three conventional mainshaft seals (ring, circumferential and face seals for small engines) and for an advanced seal, the self-acting seal, is made. The effect of leakage through gas path seals is reviewed and effect of rub interactions discussed. Experimental data (wear and erosion) are presented for abradable turbine shroud materials.

II. Apparatus and Test Seal Description

Mainshaft Seal Rig

The rig bearing compartment is typical of small high-speed gas turbine sumps. Sealing positions were located forward and aft of the bearing, which enabled simultaneous testing of two seal assemblies.

The bearing compartment drains by gravity into a static air-oil separator. Desired air pressure is introduced into the cavities adjacent to the test seals, and the air that leaks past the two test seals is passed through a flowmeter downstream of the air-oil separator to obtain a measure of seal leakage.

Conventional Mainshaft Seals

Mainshaft seals are used in gas turbine engines to restrict gas leakage into the bearing compartments (sumps). The air leaking through the seal prevents oil leakage out. In some engines the gas pressure and temperature are relatively low, so that a single labyrinth seal to restrict gas leakage into the sump is adequate (Fig. 1). At low pressures, the efficiency loss due to seal leakage is not significant. However, a major disadvantage of the labyrinth seal, as compared to the rubbing contact seals (circumferential and face), is that the higher leakage requires corresponding larger ducting requirements which results in easier passage of airborne water and dirt into the sump. It should be emphasized that dirt is very detrimental to bearing life. In addition, for labyrinth seals,

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reverse pressure drops must be avoided to preclude high oil loss. Further, high temperature compressor air leaking into oil sumps can lead to sump fires.

As engine pressures increase, the high leakage rates associated with labyrinth seals start to become significant from a performance penalty and sump design standpoint. In small engines, the available space for the bearing sumps tends to be very limited, hence the amount of gas leakage that can be handled in the sump is limited.

In large engines with high gas pressure differential (i.e., 241 N/cm^2 (350 psi)), multiple labyrinth seals (see schematic in Fig. 2) are used to protect the bearing from the high temperature, high pressure, turbine cooling gas. These multiple labyrinth seal systems take up considerable space and are generally too costly and difficult to accommodate in small engines. Thus, multiple labyrinth seal systems have significant disadvantages when applied to small high pressure engines.

The floating ring seal can be considered to be a type of labyrinth seal, which, because the ring is not restrained from moving in a radial direction, can be set up to operate with less leakage gap clearance than the conventional labyrinth seals.(5) The floating ring assembly (Fig. 3(a)), which is free to rotate in the seal case, is composed of a carbon ring shrunk into a steel retaining band. The retaining band is used to control the expansion rate of the composite ring and to reinforce it against compressive and rotational stress. The ring assembly is designed to have a coefficient of thermal expansion similar to that of the seal runner. Thus, the clearance between the runner outside diameter and the carbon inside diameter will be constant (independent of temperature). The ring assembly is not restrained from rotating and in operation may rotate (at a speed less than shaft speed) because of local and/or intermittent contact with the shaft.(5)

Circumferential seals operate with very low gas leakages and are therefore attractive, but the pressure differential capability for these pressure unbalanced seals is low (less than 41 N/cm^2 (60 psi)) because of rubbing contact. The circumferential segmented seal (Fig. 3(b)) is a carbon ring consisting of three 120° segments held together by a garter spring on the outside diameter. When the ring is installed on the runner, the gaps between the adjacent ends of the segments is a source of air leakage into the bearing cavity. During operation, when the carbon wears, the garter spring forces the segments radially inward. When about 0.127 mm (0.005 in.) of radial wear has occurred, the adjacent segment ends butt up and the seal then operates as a close-clearance labyrinth.(5) Other circumferential seal designs (not evaluated in this program) incorporate multiple rings or overlapping joints to eliminate the leakage at the gaps between the segments. In addition pressure balancing of the segments can result in a moderate increase in the pressure and speed capability.(6)

The conventional face seal (Fig. 3(c)) consists of a rotating seat that is attached to the shaft and a nonrotating carbon sealing nose that is free to move in an axial direction and thus accommodate engine thermal expansion. The secondary seal (piston ring) is subjected only to the axial motion (no rotation) of the nose; a wave spring provides axial mechanical force to maintain contact. The sealing nose is pressure balanced with an area ratio of 0.645 (for pressure balance definition, see Ref. 7). Pressure balancing is also applied to the secondary carbon seal ring, both axially and radially.

Self-Acting Seal

A self-acting face seal (Fig. 4) is similar to a conventional face seal except for the added feature of a self-acting geometry, similar to that employed in a gas lubricated thrust bearing. When the shaft rotates, the sealing faces are separated a slight amount (in the range of 0.00025 to 0.00127 cm (0.0001 to 0.0005 in.)) by the force generated by this self-acting lift geometry. The positive separation results from the balance of seal forces and the gas film stiffness of the self-acting geometry. Analyses of the self-acting seal concept and experimental feasibility studies for large aircraft gas turbine engines have been detailed in several programs.(7-15)

The self-acting geometry can be any of the various types used in gas thrust bearings. In the seal shown in Fig. 4, the self-acting geometry consists of a series of shallow recesses (shrouded Rayleigh steps) arranged circumferentially inside the sealing dam (see Fig. 4, section A-A). An important point to note is that the lift pads are bounded at their inside diameter and outside diameter by the sealed pressure, P_1. (This is accomplished by feed slots that communicate with the annular groove directly inside the sealing dam.) Therefore, a pressure gradient due to gas leakage occurs only across the seal dam. Thus the effects of force changes due to seal face deformation are minimized.(10)

Gas Path Seal Rigs

The gas path seal materials were evaluated in wear (abradability) and erosion rigs. The wear rig consisted of a single rotating labyrinth knife edge into which a shroud specimen (60° arc) was moved at a predetermined rate in order to effect a rub penetration. The shroud specimen was heated to simulate engine operation.

In the erosion rig, a hot gas stream (Mach 0.35) containing Al_2O_3 particles was directed at shroud specimens. Typical erosion tests were 20 minutes in duration.

Materials

Four conventional type abradable materials and two experimental materials were evaluated in the wear and erosion rigs. The conventional type materials were:

(1) 1/16 inch size honeycomb of Hastelloy X material (material A)

(2) Porous cermet with metal honeycomb support (material B)

(3) Porous metal with diatomaceous earth and honeycomb support (material C)

(4) Sintered fiber metal of NiCrAlY (material D)

The experimental materials evaluated were:

(1) Sintered fiber metal of NiCrAlY with a plasma spray of nichrome/glass coating (material E),

(2) Sintered fiber metal of NiCrAlY with a plasma spray of nichrome/glass/CaF_2 coating (material F).

III. Results and Discussion

Comparison of Mainshaft Seal Performance

Three conventional mainshaft seals (floating ring, circumferential segmented ring, and face) and a self-acting face seal were evaluated in simulated gas turbine operation.

During high-speed testing (above 122 m/sec (400 ft/sec)) of the floating ring seal, substantial carbon wear occurred. (Wear was to be expected at the 213 m/sec (700 ft/sec) point since the calculated operating gap closes to 0.0025 mm (0.0001 in.).) The results of the testing revealed that the leakages were relatively high, and that it does no good to attempt to obtain lower leakages by reducing the initial clearances because the seal wears to a clearance value that is dictated by shaft runout and dynamic effects. In fact, seals starting with low initial radial clearances can end up with running clearances greater than the seal assembled with intermediate clearances; this is thought to be due to a thermal rub interaction (see section on gas path seals). The changes in diametral gap are shown in table I for two rings with different initial diametral gaps. The table shows the radial gaps after operating at sliding speeds of 91, 122, 152, 183, and 213 m/sec (300, 400, 500, 600, and 700 ft/sec). The ring with the smallest initial gap (forward seal) 0.0610 mm (0.0024 in.) wore, in the first test, to a diametral gap of 0.135 mm (0.00530 in.). The aft seal did not wear because the initial gap was large enough to accommodate dynamic shaft motion.

It was found during the test program that the circumferential segmented seals wore excessively at high speeds and pressures, and eventually operated as labyrinths when enough wear allowed the segments to butt together. At this point the amount of additional wear that takes place depends on running speed; the higher speeds cause more shaft radial motion and therefore more wear.

Conventional face seal evaluation covered a range of speeds and air presures at ambient temperatures. Typical conditions and resulting air flows, bearing cavity pressures, and seal temperatures are listed in table II. The seal operated successfully over a wide range of conditions, including 213 m/sec (700 ft/sec), with a pressure difference of 117 N/cm^2 (170 psi). The total face seal carbon nose wear for 20 hours was 0.0051 mm (0.0002 in.) on the forward seal and 0.0102 mm (0.0004 in.) on the aft seal. The fact that the temperature did not exceed 394 K (250° F) at sliding speeds of 152 m/sec (500 ft/sec) suggests the seals were operating on an air film during much of the running time.

Table III contains the gas leakage data for two self-acting seals operating over a pressure differential range from 23 to 111 N/cm^2 (34 to 161 psi) and a sliding speed range from 91 to 183 m/sec (300 to 600 ft/sec). This is a rotative speed range of 27 300 to 54 600 rpm. Neither the forward nor the aft carbon nose or seal seat showed any wear during this evaluation. Thus, the sealing surfaces were separated by a gas film over the entire matrix of operating variables. This suggests that the gas bearing film stiffness was sufficient to prevent rubbing contact under the high inertia forces that are associated with high rotative speeds.[14]

Note in table III that the seal leakage increases as the sliding speed increases (for any given pressure differential). This leakage increase is due to a slight increase of the sealing gap because the increased lift force produced by the lift pads (dynamic effect). As would be expected, the leakage increases as the pressure increases.

To further explore the operating limits of the self-acting seals, 150 hours of endurance operation at ambient temperature was conducted as follows:

Speed, m/sec	Speed, ft/sec	Air pressure differential, N/cm^2	Air pressure differential, psi	Time, hr
102	334	103	149.7	28
122	400	103	149.7	22
137	450	103	149.7	65
145	475	103	149.7	20
145	475	124	179.7	15

Air temperature varied throughout the test but was generally from 372 to 408 K (200° to 275° F). Inspection of the seals after the 150 hours revealed that the wear to each seal was insignificant (less than 0.0012 mm (0.00005 in.)).

A comparison of the gas leakage rates of the various seal configurations is shown in Fig. 5. In general, the plot shows that self-acting face seal has the potential of significantly reducing leakage as compared to the conventional seals.

Of the conventional configurations, face seals allowed the least air flow at high pressure differentials. Circumferential segmented seals are as tight as face seals at moderate operating conditions; however, experience and the subject test program results have shown that at pressure differentials above 41.4 N/cm^2 (60 psi) and speeds above 107 m/sec (350 ft/sec), these (unbalanced) circumferential segmented seals rapidly wear out and finally operate as labyrinths. In that case there is little to choose between circumferential, rotating ring, and labyrinth seals in terms of air flows.

To gain some perspective of the magnitude of air flow under discussion, engine experience has shown that excessive air flow into a bearing package incorporating seals of the size used in the test program would be in the order of 0.012 kg/sec (0.029 lb/sec). Taking midpoint values of the range of pressure differentials in Fig. 5, the face seal could not meet this criterion at pressure differentials above approximately 85 N/cm^2 (123 psi); and the limiting pressure differential for circumferential segmented seals (which wear rapidly), rotating ring seals, and simple labyrinths would be approximately 40 N/cm^2 (58 psi). The self-acting seal, however, did not reach the limiting leakage rate and had a leakage of 0.0046 kg/sec (0.0102 lb/sec) at a pressure differential of 107.6 N/cm^2 (156.0 psi). In general the self-acting seal had about one third the leakage of the conventional face seal.

Gas Path Seals

Gas path sealing either in the primary or secondary gas flow paths, is a complex engineering problem since these seals operate in extreme and varying temperature environments under large centrifugal stress conditions, high peripheral speeds, and must withstand occasional rubbing contact. These rotating and stationary seal elements can move radially and axially with respect to one another due to engine thermals and transient conditions. However, in proposing improvements, it must be kept in mind that the foremost requirement is reliability; that is, the seal system must not enter a self-destruct rub mode when hard rubs occur and adverse blade damage must not be produced.

Considering all the seals, the primary gas path seals (Fig. 6) have the greatest impact on TSFC. Generally, the gas path sealing clearances change with each engine condition (idle, takeoff, climb, etc.); the support structure dimensional changes being large relative to the seal clearances. One portion of the design problem is that of compensating for these relatively large displacements through design features and judicious use of materials. The trend toward higher engine pressures and temperatures will tend to increase these seal displacements.

Typically, a secondary gas path system contains flow restrictions in parallel and series. The functions of the secondary seals include cooling, maintaining proper thermal balance, pressurizing, venting and thrust balancing. Because of the cavity interconnections across the seals, a problem at one seal location can affect other parts of the system (such as changing the thrust bearing load). Changes in the turbine cooling flow are of particular concern since the trend is to use materials at temperatures which result in the minimum acceptable low cycle thermal fatigue life. The total seal clearance, of course, is a function of differential thermal growth, wear, and erosion. The crux of the gas path seal (primary or secondary flow) problem is to maintain control of clearances that need to be made as small as practical.

Effect of Clearance

Reference 1 points out that gas leakage effects become more acute as engine size decreases because the ratio of engine circumference to engine flow area increases. For example, the leakage analysis reported in Ref. 1 indicates that the engine flange and labyrinth seal leakage is 2 percent of the mass flow for a 5 pound per second engine, and only 0.3 percent for a 120 pound per second engine. Reference 1 presents a detailed analysis of the clearance effects in a hypothetical engine with a 5 pound per second air flow. The engine was designed for zero leakage and performance calculated; performance was then determined with typical leakage values (based on seal tests) assigned to each seal position. The results are shown in Fig. 7. The most significant losses are in the labyrinth seals, and in particular the labyrinth seals at the discharge of the centrifugal compressor and at the turbine bearing locations. The calculated overall effect (which includes losses through flanges and vane pivots) is a 21 percent loss in power and a 10 percent increase in SFC.

In large axial compressors, overall efficiency losses of 2 to 6 percent have been indicated[2] and typical data are shown in Fig. 8 where compressor efficiency is plotted as a function of clearance to blade span ratio.

In turbines the blade tip clearances tend to be large because of significant thermal growths and thermal transients. Tip clearance to blade height ratios of 1 to 4 percent are common. The efficiency loss, however, depends on the degree of turbine reaction because the pressure difference across the stage has a direct effect on the amount of leakage through the tip clearance. Experimental efficiency effects for reaction and impulse turbines have been published;[16,17] these data are shown in Fig. 9. In general high reaction turbines show about a 3 percent loss in efficiency for an increase in tip clearance of 1 percent of blade height; for impulse turbines the loss is about 1.6 percent for an increase in clearance of 1 percent of blade height.

Effect of Shroud Geometry

Stall margin is affected by both clearance and seal shroud geometry (casing treatment). The term "casing treatment" has been used when holes, slots, grooves, etc., are added to the shroud surface in the region over the blade tips. Many geometries have been investigated.[3,18] In general, it has been found that stall margin can be improved by adding casing treatment if stall inception is at the outer wall. Reference 19 attributes the improvement to reduced downwash from the tip leakage flow.

Effect of Rub Interaction

As previously stated, clearance control is the crux of the seal problem and generally, the sealing clearances change with each engine condition. Relative displacements of the seal components can be grouped into four categories:[2]

(1) Symmetric differential growth - This effect is principally due to thermal growth differences, but pressure and body forces are usually also significant. It is vital to consider not only the total growth but also the thermal response rate. Thus, the shroud cooling and mounting of the seal to the mating parts represents an integrated design problem.

(2) Ovalization - An example is displacement due to localized introduction of hot gas that causes asymmetric displacements. Also mounting practice may allow case sag.

(3) Beam bending displacements - These occur for various reasons; some are: rotor thermal bow, mounting, aeroelastic vibration of seal parts, and maneuver loads. (Maneuver displacements can be large.)

(4) Machining tolerance buildup

In attempting to maintain close clearances, accumulation of adverse displacements causes rubs (usually localized). If no wear occurs to the blade (or knife edge) and all wear occurs in the shroud, then only the local clearance increase is generated. If wear occurs to the blades (knife), then the clearance has increased around its full annulus. Thus, an ideal situation would be zero wear to blades with all wear occurring in the shroud; a practical goal is 10 percent of the wear to the blade (knife) and 90 percent in the shroud; this can be termed a 10:1 rub tolerance.

Thermal Response

Intimately associated with rub tolerance is the thermal response of the seal system when a rub occurs. This thermal response can be divided into two parts - an overall response and a local response.

The overall response is dependent on the rub severity and the overall thermal expansion of the static and rotating parts. The differential expansions can lead to a self-destruct wear mode. That is, the rub itself generates enough heat to cause thermal growth to increase the severity of the rub. This type of rub interaction has been responsible for a series of engine failures. Thus, the first seal design criteria is avoidance of self-destruct rub interaction.

It has been found that the wear, in stable rub interactions, is dependent on the initial seal clearance. In some engines if the labyrinth seal is assembled with a small initial clearance, a rub interaction is severe and produces significant wear that results in large final clearance. But smaller final clearances can be obtained if the seals are assembled with intermediate size initial clearances, because then, the thermal response associated with the rub interaction is less severe and less wear occurs.

The change in stator and rotor diameters depends on the temperature change (due to the heat input) and coefficient of thermal expansion. The heat transfer problem is complex because of the complex geometry and temperature gradients. But, in general, it is desirable to maximize the heat input into the shroud and minimize the heat input into the rotor; this tends to minimize the differential thermal growth and the corresponding normal force between the rotor and shroud.

In addition to the overall thermal response, the local thermal response has a significant influence on the rub interaction. For example, experimental data show that when a labyrinth knife edge rubs against a shroud segment, the rubbing can take place over just a small segment ($\sim 5^{\circ}$ arc) of the 360° of knife edge (see Fig. 10). Thus, the heat input is highly localized and a local thermal bump is generated that expands, rubs harder, and finally wears away. This is then followed by rubbing over a second small segment which grows and then wears, etc. Evidence of localized rubbing is indicated by the heat discoloration of the knife edge (Fig. 11) that was rubbed against a shroud specimen. This type of local rub interaction has been investigated from a fundamental analytical standpoint, but application of the tip seal problem has not been attempted.[20-22]

Rub Tolerance

In addition to the thermal considerations, the rub tolerance of tip seals is of prime importance, and research effort is currently being directed toward development of rub tolerant tip seals that can operate in turbine environments for long periods of time without degradation.[23] As an example of this work, Fig. 12 shows the torque produced by rubbing of a simulated labyrinth knife edge against a turbine shroud specimen. The data are for a rubbing interaction at 183 m/sec (600 ft/sec) with the labyrinth knife edge penetrating the shroud at a rate of 0.254 mm (0.010 in.) per second. The four conventional type materials showed a wide variation in rubbing torque with the magnitude of the rubbing torque being an indication of the wear that occurred to labyrinth knife edge. The experimental materials evaluated have nickel-chromium-glass flame sprayed coatings on a fiber metal base. One experimental material contained calcium fluoride, as a high temperature lubricant, in the coating. The effect of this high temperature lubricant can be seen (Fig. 12) to have reduced the reaction torque by a factor of two. Figure 13 shows the clean cut (groove) produced by the knife edge in this experimental material.

In addition to reaction torque, erosion rate is a measure of the material potential in engine applications. Figure 14 shows erosion rates for the four conventional type materials and the two experimental materials. Note that conventional material type D, a sintered fiber metal that had low reaction torque and hence caused low knife edge wear, had a high erosion rate. On the other hand, materials B and C had low erosion rates but high torque which suggests low abradability (high blade wear). The experimental materials (E and F) with the flame sprayed coatings showed the lowest erosion rates. Generally materials with good abradability had poor erosion characteristics and vice versa. The experimental materials showed a potential for achieving both good erosion and good abradability. The experimental materials were the first attempts to use the surface film concepts. Optimization of both film materials and supporting deformable structure may give further improvements in tribiological and erosion properties.

Summary of Results

Four types of shaft seals were evaluated under simulated gas turbine operating conditions which included pressures to 148 N/cm^2 (215 psi). The results of this experimental evaluation revealed the following:

1. The self-acting face seal operated without rubbing contact. This was evidenced by lack of wear. Of particular interest was the successful operation at 54 600 rpm (183 m/sec (600 ft/sec)); this was taken as evidence that the gas film stiffness was high enough to prevent rubbing contact under high inertia force conditions.

2. Self-acting face seal leakage was significantly lower than that of the three conventional seal types.

3. Conventional contact seals may not be satisfactory in future advanced engines because of wear life and excessive air leakage flow.

a. Of the conventional seals tested, the face seal configuration was the most successful at limiting air leakage flow; however, at air-to-oil pressure differentials above approximately 85 N/cm^2 (123 psi), air flow was considered excessive.

b. The circumferential segmented seal (unbalanced type) configuration operated well at moderate conditions, but at air-to-oil pressure differentials of 41.4 N/cm^2 (60 psi) and speeds above approximately 107 m/sec (350 ft/sec), it wore very rapidly and eventually operated as a labyrinth.

c. Ring seal clearances, which are determined by shaft dynamics and thermal rub response, results in air leakage rates that are comparable to labyrinth seals.

Published data on gas path seals was reviewed; wear and erosion evaluations were made on commercial turbine shroud materials and on some experimental materials using simulated labyrinth knife edge rubbing into a shroud specimen. This study revealed:

1. Gas path seals have significant effects on engine efficiency and compressor stall margin.

2. The thermal response and rub tolerance of the gas path seal are significant factors in the determination of the final operating clearances.

3. In four conventional materials there was a wide variation in their wear, torque reaction, and erosion resistance. Generally, materials with good abradability had poor erosion and vice versa. Shroud materials with flame spray coatings on a deformable substrate show promise of providing more optimal tribiological as well as erosion properties.

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Table I Floating ring seal post test diametral measurements

		Test speed, m/sec (ft/sec)					
		New	91 (300)	122 (400)	152 (500)	183 (600)	213 (700)
Forward seal	Diametral gap, mm (in.)	0.06096 (0.00240)	0.12446 (0.00530)	0.12700 (0.00500)	0.12700 (0.00500)	0.12700 (0.00500)	0.21590 (0.00850)
Aft seal	Diametral gap, mm (in.)	0.13462 (0.00520)	0.13462 (0.00530)	0.13716 (0.00540)	0.13462 (0.00530)	0.14478 (0.00570)	0.16510 (0.00650)

Table II Typical face seal test data

Rpm	Speed		Air pressure differential		Air flow (two seals)		Seal temperature	
	m/sec	ft/sec	N/cm^2	psi	kg/sec	lb/sec	K	^oF
27 300	91	300	61.4	89.0	0.007	0.016	350	170
36 400	122	400	61.0	88.5	.007	.015	363	195
45 500	152	500	62.1	90.0	.006	.013	394	250
36 400	122	400	111.7	162.0	.025	.055	---	---
45 500	152	500	113.1	164.0	.023	.050	---	---
54 600	183	600	113.8	165.0	.021	.045	---	---
63 700	213	700	117.2	170.0	.016	.035	---	---

Table III Self-acting face seal evaluation

Rpm	Speed		Air pressure differential		Air flow (two seals)		Seal temperature	
	m/sec	ft/sec	N/cm^2	psi	kg/sec	lb/sec	K	^oF
27 300	91	300	23.4	34.0	<0.0006	<0.0013	333	140
36 400	122	400	23.1	33.5	<.0006	<.0013	352	174
45 500	152	500	23.1	33.5	<.0006	<.0013	371	210
54 500	183	600	22.1	32.0	.0011	.0024	392	246
27 300	91	300	111.4	161.5	.0023	.0050	364	196
36 400	122	400	110.7	160.5	.0032	.0070	373	212
45 500	152	500	109.6	159.0	.0036	.0079	386	236
54 600	183	600	107.6	156.0	.0046	.0102	402	263

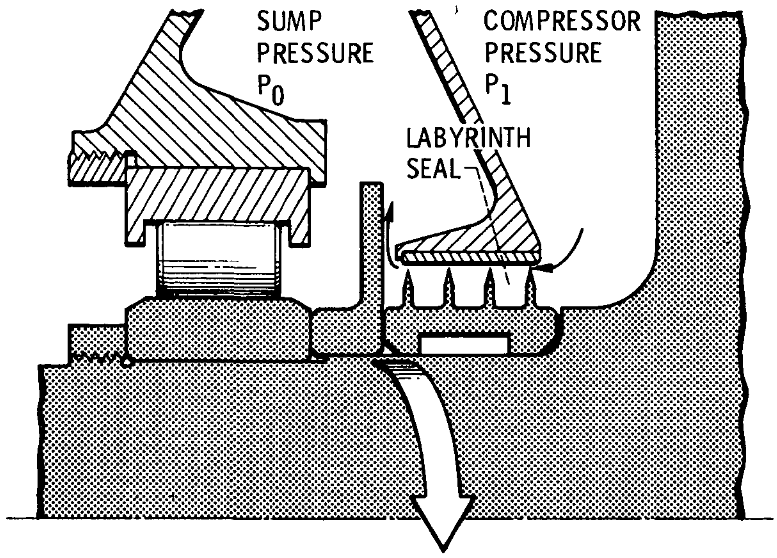


Fig. 1. - Single stage labyrinth seal.

E-8096

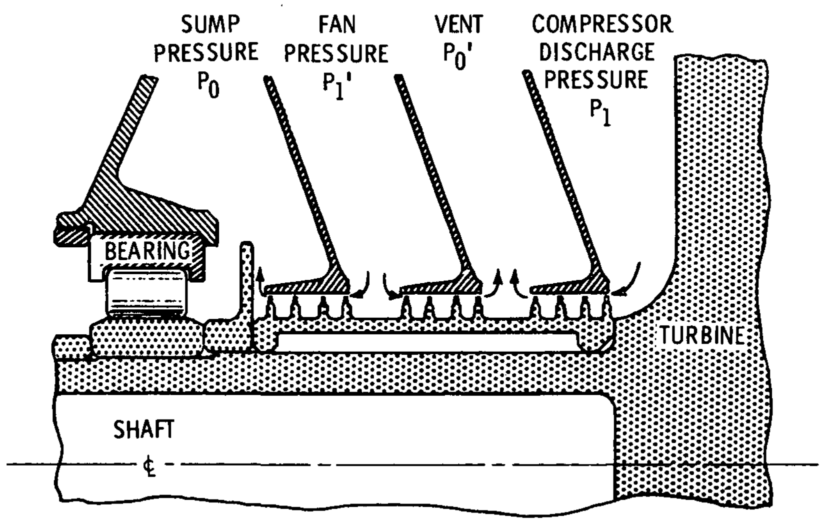
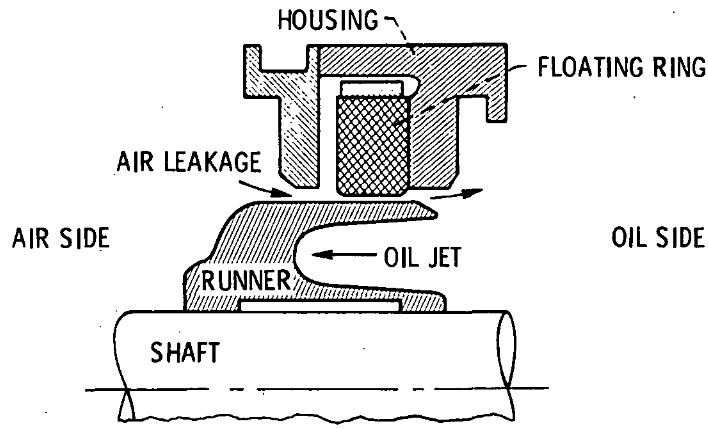
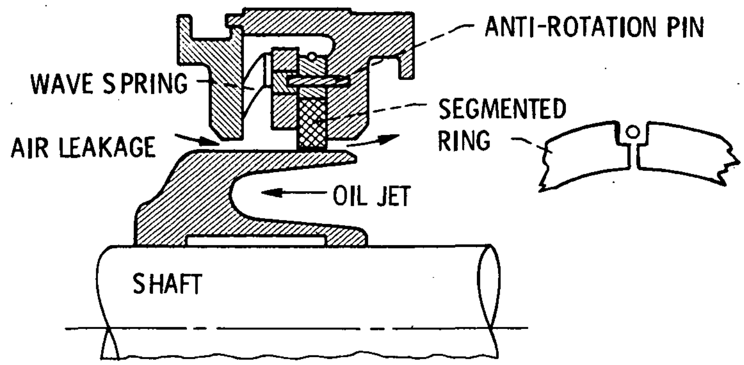


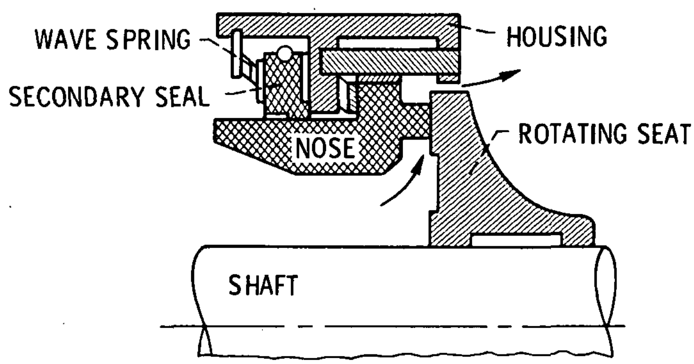
Fig. 2. - Multiple stage labyrinth seal system.



(a) FLOATING RING SEAL.

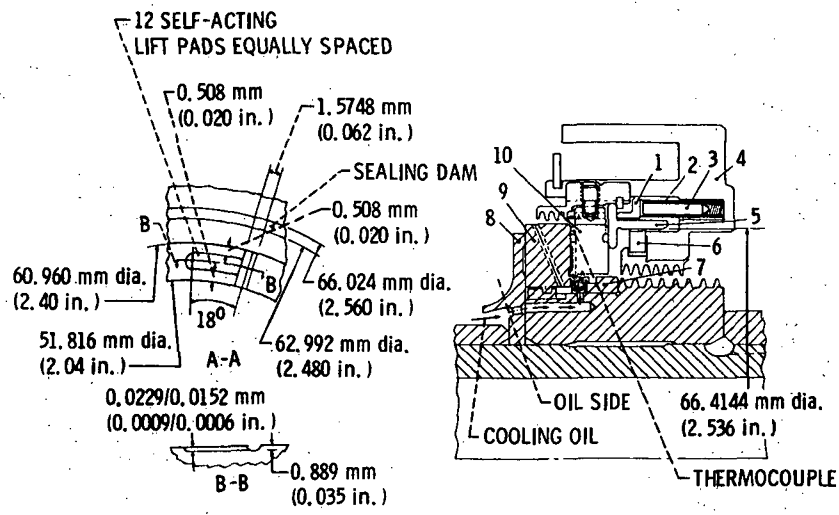


(b) UNBALANCED CIRCUMFERENTIAL SEGMENTED-RING SEAL.



(c) FACE SEAL.

Figure 3. - Conventional shaft seals (ref. 5).



1. SPRING PLATE
2. COMPRESSION SPRING
3. SPRING PIN
4. HOUSING
5. CARRIER
6. PISTON RING (SECONDARY SEAL)
7. BELLOWS SPACER
8. OIL DAM AND HEAT SHIELD
9. ROTATING SEAT
10. NOSEPIECE

Figure 4. - Self-acting face seal design (ref. 5).

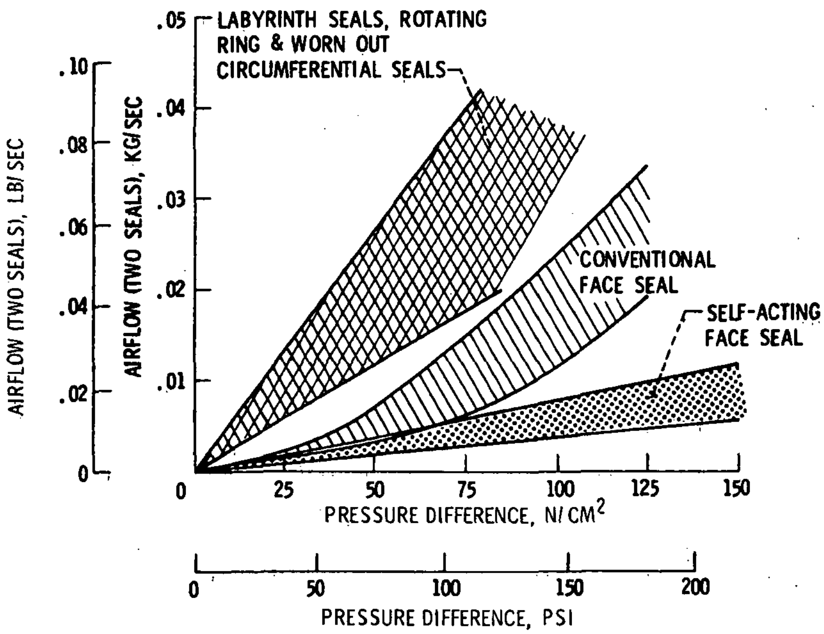


Fig. 5. - Comparison of seal configurations.

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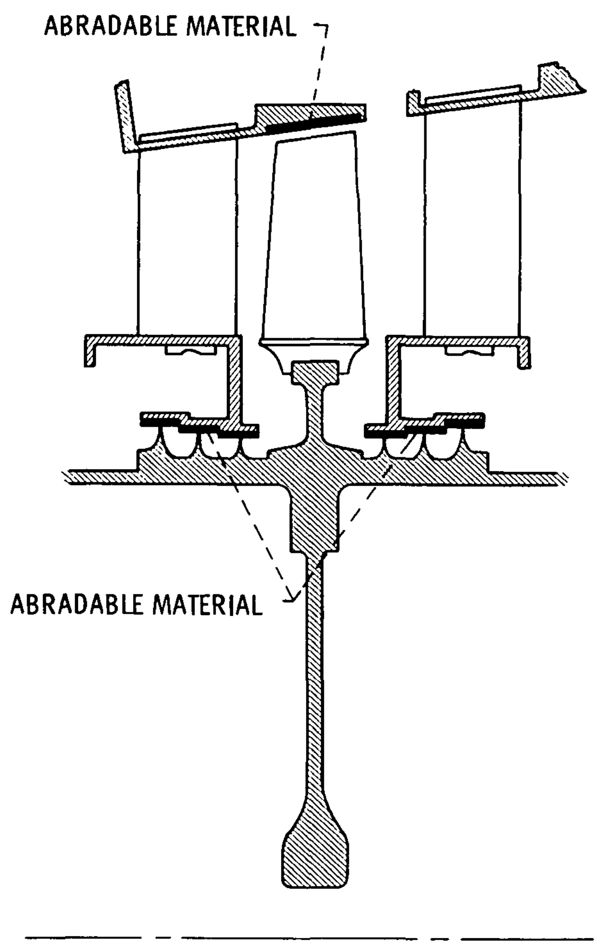

Figure 6. - Primary gas path seals (ref. 2).

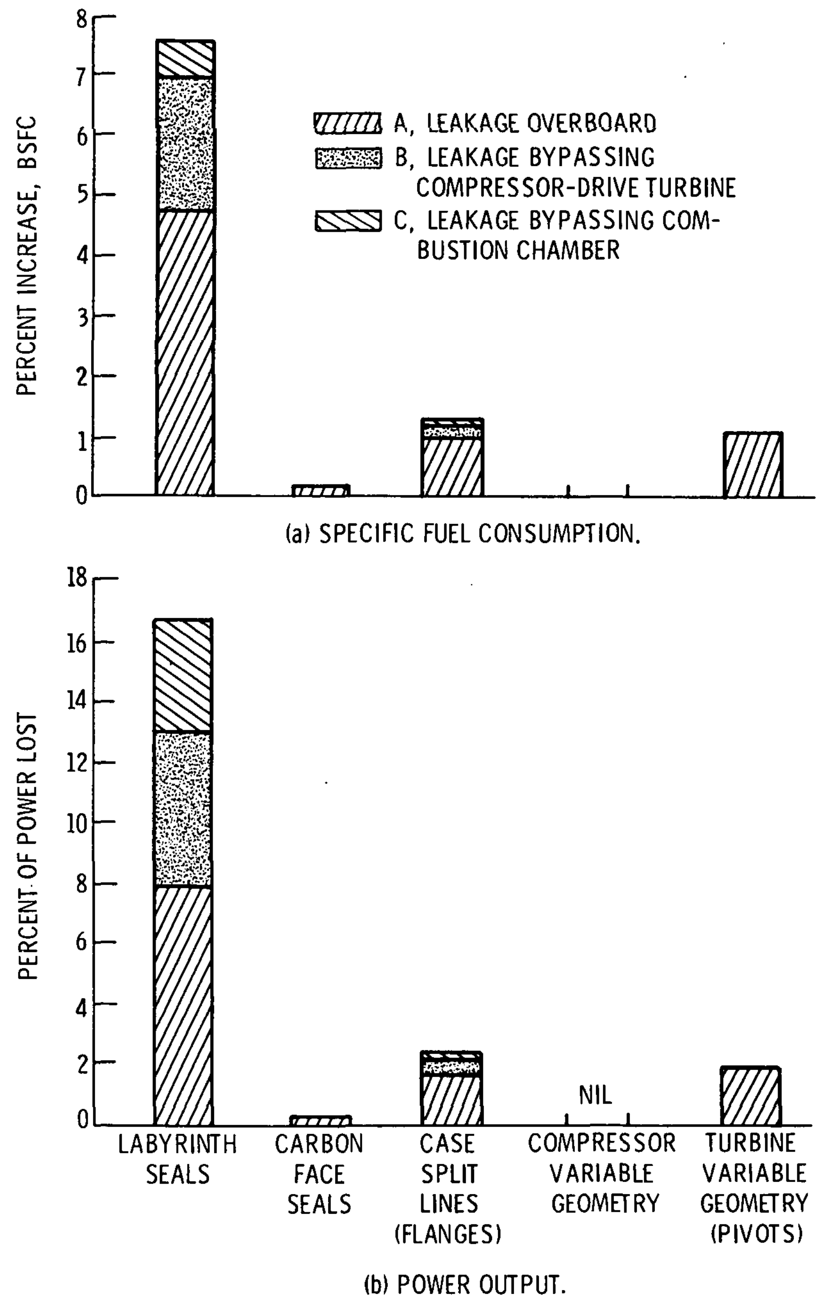

Fig. 7. - Effects of engine leakage (Ref. 1).

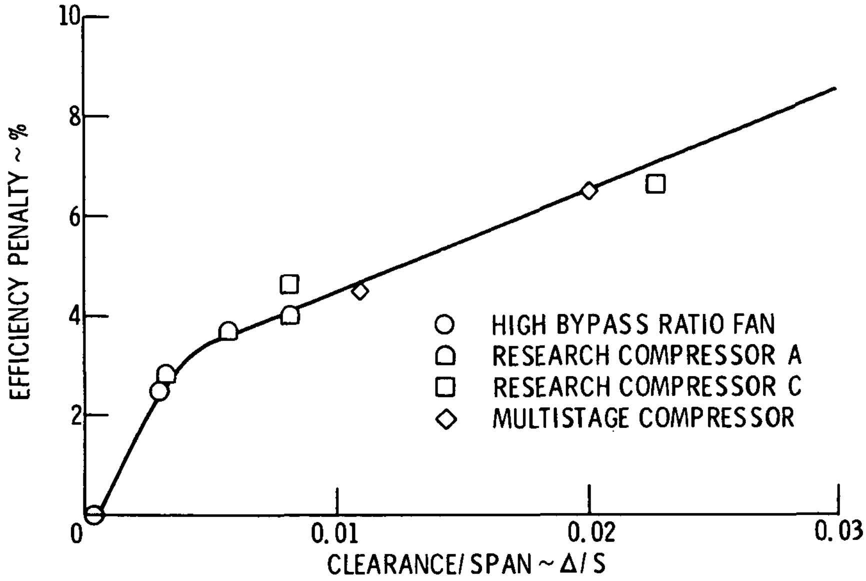

Fig. 8. - Typical compressor efficiency penalty as a function of blade clearance-to-span ratio (Ref. 2).

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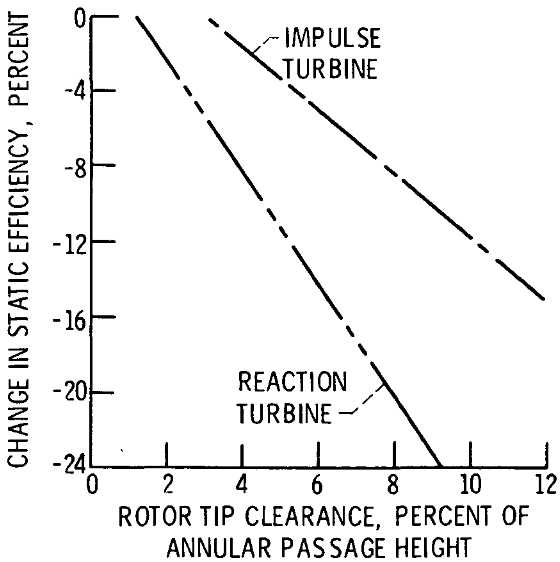

Fig. 9. - Effect of rotor tip clearance on performance for various turbines (Ref. 16).

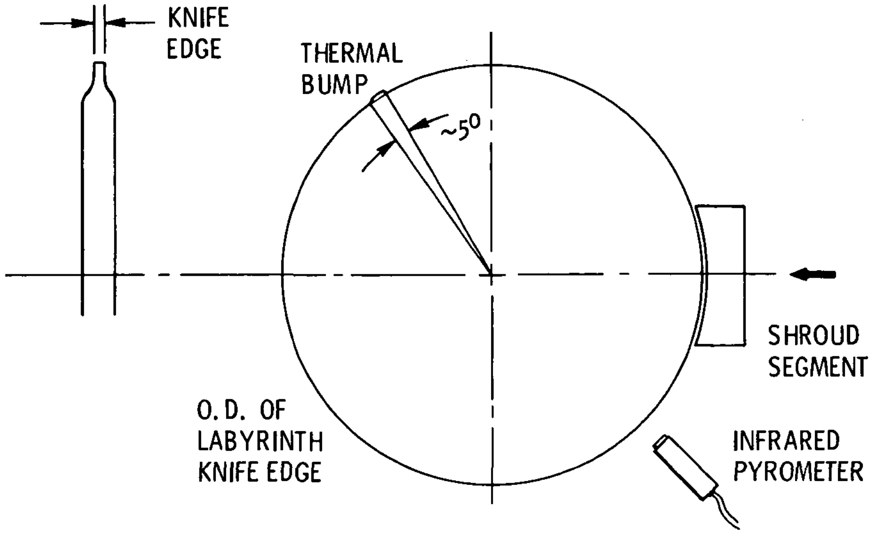

Figure 10. - Local thermal response of labyrinth knife edge rubbing against a shroud segment in bench tests.

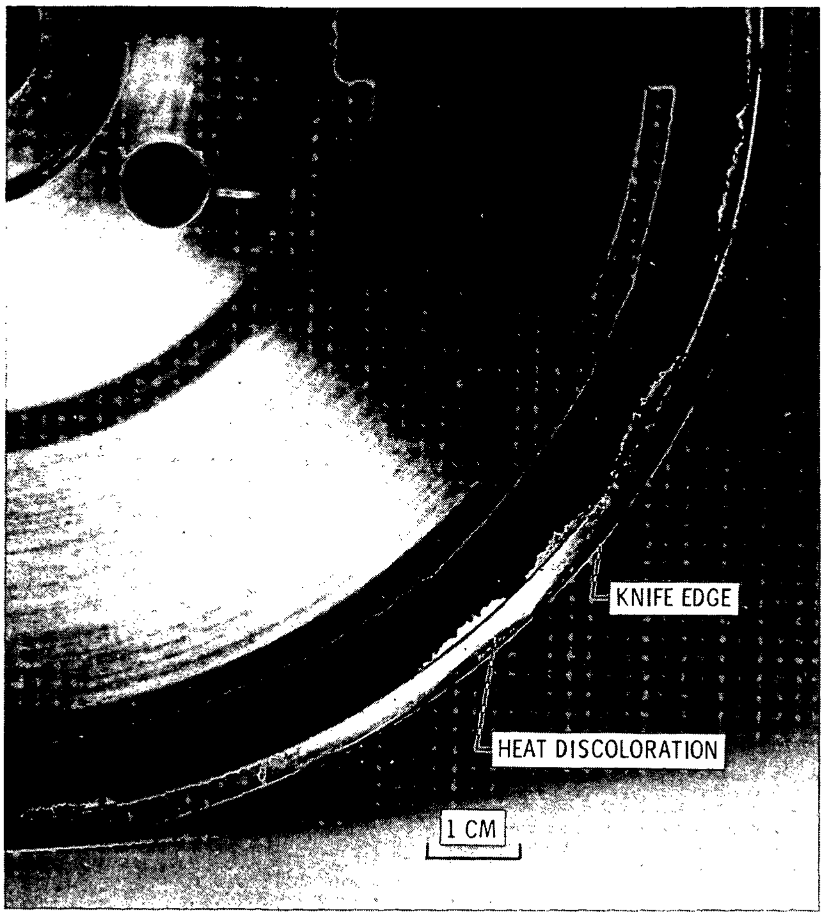

Figure 11. - Labyrinth disk and knife edge showing heat discoloration due to thermal bumps generated in rubbing contact against a shroud specimen. Rubbing speed, 183 m/s (600 ft/sec).

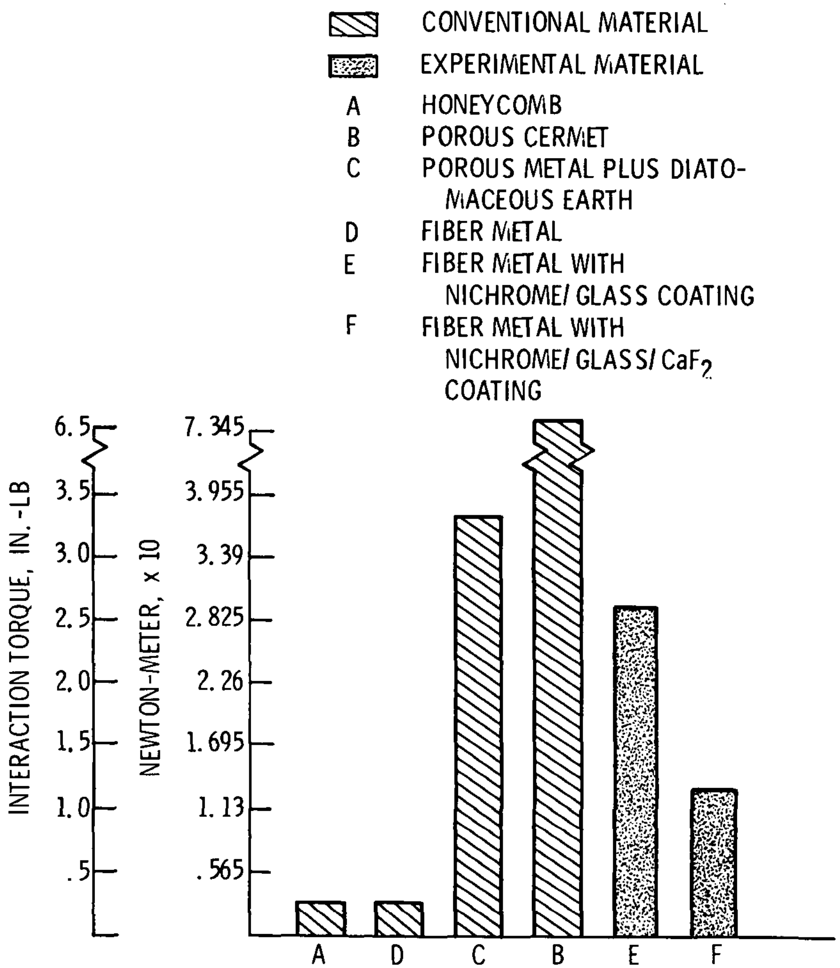

Fig. 12. - Torque produced when rubbing a knife edge against a shroud specimen; rubbing speed, 183 m/sec (600 ft/sec); penetration rate, 0.254 mm/sec (0.010 in./sec).

Figure 13. - Groove in shroud specimen produced by labyrinth knife edge. Material, sintered felt metal with plasma spray coating of nichrome/glass/CaF_2.

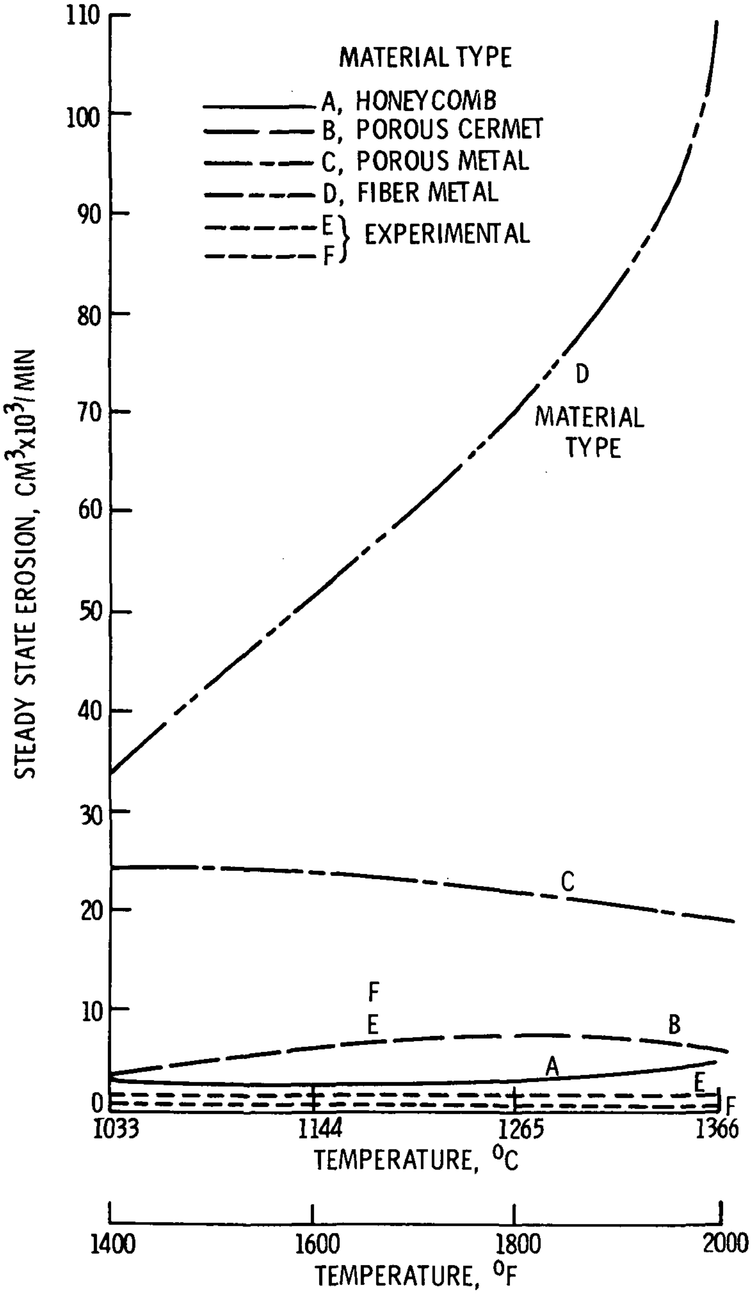

Fig. 14. - Erosion rates of shroud material specimens; gas velocity, 0.35 Mach number; Al_2O_3, particle rate, 2.72 kg/hr (6 lb/hr).