SEALING TECHNOLOGY FOR AIRCRAFT GAS TURBINE ENGINES

by L. P. Ludwig and R. L. Johnson
Lewis Research Center
Cleveland, Ohio

TECHNICAL PAPER proposed for presentation at
Tenth Propulsion Conference sponsored by
American Institute of Aeronautics and Astronautics
and Society of Automotive Engineers
San Diego, California, October 21-24, 1974
SEALING TECHNOLOGY FOR AIRCRAFT GAS TURBINE ENGINES

L. P. Ludwig * and R. L. Johnson **
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio

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 63,200 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 (ablability) 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 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,
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/cm² (350 psi)), multiple labyrinth seal designs (see 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. 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/cm² (60 psig)) 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. The 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 seal 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 0.00027 cm (0.0001 to 0.00005 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 self-acting seal (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, $P_s$. (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 Al2O3 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 nickel/chromium coating (material E),
2. Sintered fiber metal of NiCrAlY with a plasma spray of nickel/chromium/CaF₂ 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 seal carbon nose wear for 20 hours was 0.0051 mm (0.0001 in.).

During low-speed testing (152 m/sec (500 ft/sec)), the floating ring seal was generally from 372 to 408 K (200° to 275° F).

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:

<table>
<thead>
<tr>
<th>Speed (m/sec)</th>
<th>Air Pressure (N/cm²)</th>
<th>Time (hr)</th>
<th>Differential (psl)</th>
</tr>
</thead>
<tbody>
<tr>
<td>102</td>
<td>334</td>
<td>103</td>
<td>129.7</td>
</tr>
<tr>
<td>122</td>
<td>103</td>
<td>149.7</td>
<td></td>
</tr>
<tr>
<td>137</td>
<td>475</td>
<td>103</td>
<td>149.7</td>
</tr>
<tr>
<td>145</td>
<td>145</td>
<td>124</td>
<td>179.7</td>
</tr>
</tbody>
</table>

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/cm² (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 it is little to choose between circumferential, rotative 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/cm² (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/cm² (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/cm² 
(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 sec-
tary gas flow paths, is a complex engineering 
problem since these seals operate in extreme and 
varying temperature environments under large cen-
trifugal 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 condi-
tions, venting, and in some balancing. Because, 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 clear-
ances. One portion of the design problem is that 
of compensating for these relatively large dis-
placements through design features and judicious 
use of materials. The trend toward higher engine 
presures and temperatures will tend to increase 
these seal displacements.

Typically, a secondary gas path system con-
tains flow restrictions in parallel and series. 
The functions of the secondary seals include cool-
ing, maintaining proper thermal balance, presuriz-
ing, and venting. The design problem is that 
of compensating for these relatively large dis-
placements through design features and judicious 
use of materials. The trend toward higher engine 
presures and temperatures will tend to increase 
these seal displacements.

Effect of Clearance

Reference 1 points out that gas leakage ef-
fects 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 per-
cent 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 perform-
cance 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 lab-
yrinth 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 re-
action 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 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 in-
crease 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 im-
provement 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 
problem is grouped into four categories:\n
(1) Symmetric differential growth - This effect 
is principally due to thermal growth differences, 
but pressure and body forces are usually also sig-
nificant. 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
heat input is highly localized and a local thermal
response is induced by the heat discoloration of the knife
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 re-
response.

The overall response is dependent on the rub
severity and the overall thermal expansion of the
static and rotating parts. The differential expan-
sions 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 de-
pends 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 differen-
tial 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 in-
fluence on the rub interaction. For example, ex-
perimental data show that when a labyrinth knife
dge rubs against a shroud segment, the rubbing can
take place over just a small segment (~5° 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 rub-
ing over a second small segment which grows
and then wears, etc. Evidence of localized rubbing is
indicated by the heat discoloration of the knife
dge (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 rub-
ing interaction at 183 m/sec (600 ft/sec) with the
labyrinth knife edge penetrating the shroud at a
rate of 0.256 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 tem-
perature 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 exper-
imental material.

In addition to reaction torque, erosion rate
is a measure of the material potential in engine
applications. Figure 14 shows the erosion rates for
the four conventional type materials and the two ex-
perimental 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 experi-
mental materials showed a potential for achieving
both good erosion and good abradability. The ex-
perimental 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 tribological 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/cm² (215 psi). The re-
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 stiff-
ness was high enough to prevent rubbing contact
under high inertia force conditions.

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

3. Conventional contact seals may not be satis-
factory 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 tribological as well as erosion properties.

References


Table I Floating ring seal post test diametral measurements

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

Table II Typical face seal test data

<table>
<thead>
<tr>
<th>Rpm</th>
<th>Speed m/sec ft/sec</th>
<th>Air pressure differential N/cm² psi</th>
<th>Air flow (two seals) kg/sec lb/sec</th>
<th>Seal temperature K °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>27 300</td>
<td>91 300</td>
<td>61.4 89.0</td>
<td>0.007 0.016</td>
<td>350 170</td>
</tr>
<tr>
<td>36 400</td>
<td>122 400</td>
<td>61.0 88.5</td>
<td>0.007 0.015</td>
<td>363 195</td>
</tr>
<tr>
<td>45 500</td>
<td>152 500</td>
<td>62.1 90.0</td>
<td>0.006 0.013</td>
<td>394 250</td>
</tr>
<tr>
<td>36 400</td>
<td>122 400</td>
<td>111.7 162.0</td>
<td>0.025 0.055</td>
<td>--- ---</td>
</tr>
<tr>
<td>45 500</td>
<td>152 500</td>
<td>113.1 164.0</td>
<td>0.023 0.050</td>
<td>--- ---</td>
</tr>
<tr>
<td>54 600</td>
<td>183 600</td>
<td>133.8 165.0</td>
<td>0.031 0.065</td>
<td>--- ---</td>
</tr>
<tr>
<td>63 700</td>
<td>213 700</td>
<td>177.2 170.0</td>
<td>0.036 0.079</td>
<td>402 263</td>
</tr>
</tbody>
</table>

Table III Self-acting face seal evaluation

<table>
<thead>
<tr>
<th>Rpm</th>
<th>Speed m/sec ft/sec</th>
<th>Air pressure differential N/cm² psi</th>
<th>Air flow (two seals) kg/sec lb/sec</th>
<th>Seal temperature K °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>27 300</td>
<td>91 300</td>
<td>23.4 34.0</td>
<td>&lt;0.0006 &lt;0.0013</td>
<td>333 140</td>
</tr>
<tr>
<td>36 400</td>
<td>122 400</td>
<td>23.1 33.5</td>
<td>&lt;0.0006 &lt;0.0013</td>
<td>352 174</td>
</tr>
<tr>
<td>45 500</td>
<td>152 500</td>
<td>23.1 33.5</td>
<td>&lt;0.0006 &lt;0.0013</td>
<td>371 210</td>
</tr>
<tr>
<td>54 600</td>
<td>183 600</td>
<td>22.1 32.0</td>
<td>0.0011 0.0024</td>
<td>392 246</td>
</tr>
<tr>
<td>27 300</td>
<td>91 300</td>
<td>111.4 161.5</td>
<td>0.023 0.050</td>
<td>364 196</td>
</tr>
<tr>
<td>36 400</td>
<td>122 400</td>
<td>110.7 160.5</td>
<td>0.0032 0.0070</td>
<td>373 212</td>
</tr>
<tr>
<td>45 500</td>
<td>152 500</td>
<td>109.6 159.0</td>
<td>0.0036 0.0079</td>
<td>386 236</td>
</tr>
<tr>
<td>54 600</td>
<td>183 600</td>
<td>107.6 156.0</td>
<td>0.0046 0.0102</td>
<td>402 263</td>
</tr>
</tbody>
</table>
Fig. 1. - Single stage labyrinth seal.

Fig. 2. - Multiple stage labyrinth seal system.
Figure 3. Conventional shaft seals (ref. 5).
Figure 4. - Self-acting face seal design (ref. 5).

Figure 5. - Comparison of seal configurations.
Figure 6. - Primary gas path seals (ref. 2).

Figure 7. - Effects of engine leakage (Ref. 1).
Fig. 8. - Typical compressor efficiency penalty as a function of blade clearance-to-span ratio (Ref. 2).

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).
Fig. 14. - Erosion rates of shroud material specimens; gas velocity, 0.35 Mach number; Al₂O₃, particle rate, 2.72 kg/hr (6 lb/hr).