The J-Seal: Novel Concept for a Resilient Noncontacting Face Seal

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Turbine engine compressor seals must survive a severe environment of pressure and temperature while minimizing leakage. In addition, these seals must be tolerant to uncertainty and fluctuation in relative location of stationary and rotating members. Resilient seals overcome this uncertainty by making the stationary member flexible. Any interference is temporarily accommodated by elastic deformation, and, when the rotating member subsequently moves away, the stationary member springs back to follow it. The problem with resilient seals is to combine flexibility with strength. The J-Seal is a concept which supports the steady pressure load by membrane action, and minimizes the bending stresses which have to be supported when accommodating relative motion between stator and rotor. A design study has been performed and the resultant configuration is now at the fabrication stage. This paper describes the concept and the tradeoffs involved in its design, and presents the performance to be anticipated.

Nomenclature

a = distance measured parallel to the axis between part of attachment of seal ring to outer membrane and outer support, meter
b = radial extent of outer membrane (referred to as outer support leg), meter
c = radial distance between point of radial tangency of toroid and the inner support (referred to as inner support leg), meter
C_{xf} = translational elastic compliance of seal ring and membrane in response to an axial force applied uniformly around a circumferential ring located at the fluid film center of pressure, meter/Newton
$C_{\theta f}$ = derivative of seal ring slope relative to a plane penpendicular to the axis, with respect to an axial force applied uniformly around a circumferential ring located at the fluid film center of pressure, m/Newton
F_f = fluid film force, Newton
G_z = viscosity coefficient to account for turbulence effects in the fluid film. Values used were developed by Ng and Elrod.[1] Under laminar conditions $G_z = 1/12$
h = film thickness, meter
h_o = running heel gap, meter
h_{oo} = hot, unloaded heel gap, meter
N = viscosity, Newton sec/meter^2
p = pressure, Newton/meter^2
r = radial coordinate in fluid film equations, meter
R = toroidal radius, meter
r_i = inner radius of seal ring, meter
r_o = outer radius of seal ring, meter
t = membrane thickness, meter; also time in the fluid film equation, sec
Δh_o = running film taper, meter
Δh_{oo} = hot, unloaded film taper, meter
Δp = pressure difference across seal, Newton/meter^2
Δr = radial width of seal face, meter
ΔX_p = shift of seal inner radius towards runner in response to a pressure load equal to Δp, meter
$\Delta_{\theta p}$ = change in seal ring slope in response to a pressure load equal to Δp, radians

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I. Introduction

TURBINE engine seals have been identified by Mahler[2] as critical elements in the further development of advanced gas turbine engines. Seal leakage currently has a significant impact on the overall engine efficiency. The essential requirement for a seal is to maintain close clearances between rotating and stationary components. However, the seal must contend with gross dimensional changes, much larger than operating clearances, due to thermal and centrifugal effects and transient loading. The seal must also contend with dynamic runout which is of the same order as the operating clearance. New sealing concepts are required and yet, as the drive for higher efficiencies and higher specific thrust leads to increased temperature, pressure, and speed in advanced engines, any new concept must be able to survive an increasingly severe environment.

The commonly used labyrinth seal represents the state-of-the-art in gas turbine engine seals. As normally designed, the labyrinth seal resists axial flow and its critical clearance is radial; it is clearly affected by thermal and centrifugal growth of the rotor, but is unaffected by gross axial shifts of the rotor. To minimize the harmful effects of radial growth, the labyrinth seal employs abradable materials. However, no mechanism is provided to accommodate dynamic runout.

This paper presents a new, resilient seal concept entitled the "J-Seal." This is a high-pressure, high-temperature seal, which responds to dynamic runout by elastic deformation. The critical J-Seal clearance is axial; it is unaffected by thermal and centrifugal growth and can accommodate dynamic changes in axial dimensions due to distortion, runout, or axial shift, of up to ¼ mm, by flexure of the membrane. Gross shifts of the rotor such as occur during engine transients are accommodated by a slower moving seal carrier.

The target application for the J-Seal is as the high-pressure compressor end stage seal for advanced (1980s) turbine engines with the severe operating conditions as listed in Table 1. The design and analysis of the J-Seal was thus structured around these conditions.

The J-Seal also has applicability to near-term engines with less severe operating conditions and is being consid-

Table 1 Design conditions for the J-Seal

Compressor discharge pressure ~4,136,854 N/M² (600 psi)
Compressor discharge temperature ~816°C (1500°F)
Corrected core air flow ~45.5 kg/sec (100 lb/sec)
Rotor speed (high rotor) ~1,257 rad/sec (12,000 rpm)
Inner diameter 0.33 m (13 in.)

Table 2 Test conditions

Compressor discharge pressure ~2,068,427 N/M² (300 psi)
Compressor discharge temperature ~649°C (1200°F)
Corrected core air flow ~45.5 kg/sec (100 lb/sec)
Rotor speed (high rotor) ~1257 rad/sec (12,000 rpm)

ered for such an application. The specific objectives in studying the J-Seal are to analyze the concept, to determine which parameters influence its performance and how, on the basis of this analysis, to design the J-Seal for the design conditions of Table 1. A rigid support for the J-Seal membrane has initially been considered in this study, so that the J-Seal performance can be studied independently of the carrier mechanism.

The next step of this program is to test the design, not only to prove its feasibility but also to verify the advantages claimed relative to present sealing methods. For the first series of tests, conditions more representative of a current high-performance engine have been selected as shown in Table 2.

The basic objectives of this paper are to present the J-Seal concept, to show that it is feasible from analytical considerations, and to document the present design configuration. In so doing, the design analysis is summarized and some typical results showing the influence of design parameters are presented. The leakage performance of the J-Seal is compared to that of a comparable labyrinth seal.

II. The J-Seal Concept

As illustrated in the cross-section of Fig. 1, the J-Seal consists of a sealing ring supported on a thin membrane and separated by a fluid film from the runner. Over its toroidal section, the J-Seal membrane is subjected to an unbalanced pressure load (constant for a constant pressure differential) and, in the region of the seal face, there is an unbalanced pressure load which increases as the fluid-film force decreases. Between the seal ring and the outer support the pressure is balanced.

The J-Seal is well-shaped for supporting the internal pressurization in its toroidal section, since predominantly in-plane stresses are so generated. However, as the unbalanced axial force across the seal ring becomes significant, the membrane deflects towards the runner and bending stresses are induced at the supports. Since the fluid-film force decreases as the separation of runner and seal ring increases, the seal ring is inclined to follow the runner when it moves away and, conversely, to deflect out of the way as the runner moves towards it. Since the fluid-film thickness range necessary to provide the required force variation is consistent with good resistance to leakage, the J-Seal can achieve good sealing performance under considerable variation in relative location of stator and runner.

The limitations to the behavior described are imposed, at one end of the scale, by the membrane's limited ability to deflect without yielding and, at the other end of the scale, by the fluid film's inability to generate a force greater than that corresponding to the product of pressure differential and seal area. Any higher axial force would result in direct contact between stator and runner.

In designing the J-Seal the following data are needed: the dependence of membrane stiffness and stresses on membrane geometry; the dependence of fluid-film force and flow on face width and the fluid film profile; and the way in which the membrane and fluid film forces interact to determine equilibrium of the sealing ring. Additionally, the distorting influence of the high temperature environment must be known and accounted for. Correspondingly, design analysis needs in four areas are identifiable. Flexibility and stress analysis, fluid-film analysis, interaction analysis, and distortion analysis. This paper is mainly concerned with the first three of these.

III. Design Parameters

The geometrical parameters of the J-Seal which significantly affect performance, as shown in Fig. 1, are toroidal radius R, membrane thickness t, outer support leg b, inner support leg c, outer support offset a, and face width Δr. These parameters are directly controllable by the designer.

In addition, the fluid-film thickness profile under running conditions is important to the seal's operation. However, because of the J-Seal's flexibility and the variable force generated in the fluid film, this running-film thickness profile cannot be directly controlled. It is necessary, by analysis, to relate the running profile back to the unloaded profile, which will be markedly different but is more directly controllable by the designer.

To provide a uniform nomenclature for discussion of the fluid film profile, the following terms are defined. "Heel gap": Separation of runner and stator at the inner radius of the sealing annulus. "Film taper": Difference in separation of runner and stator at inner and outer radius of sealing annulus (referred to in some places simply as "taper"). "Running": Qualifies "heel gap" and "taper" to denote values at hot, loaded, running condition. "Hot unloaded": Qualifies "heel gap" and "taper" to denote values at running temperature but under uniform pressure conditions. "Cold unloaded": Qualifies "heel gap" and "taper" to denote values at room temperature under uniform pressure conditions. The design analysis in Sec. IV has, as its purpose, the relation of seal leakage, membrane stresses, and running heel gap to the geometrical parame-

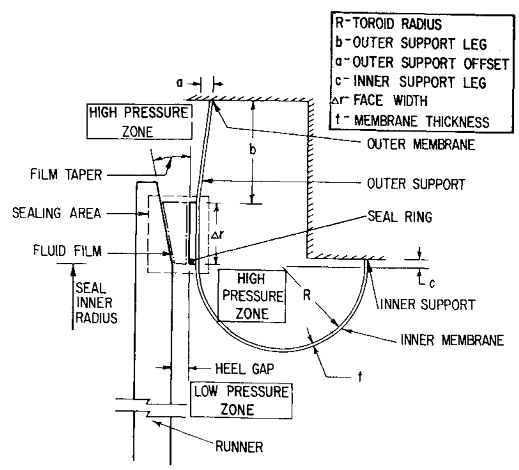

Fig. 1 **J-Seal in cross-section. Note: film taper is expressed throughout in terms of the difference in gap at *OD* and *ID* (m, or mils).**

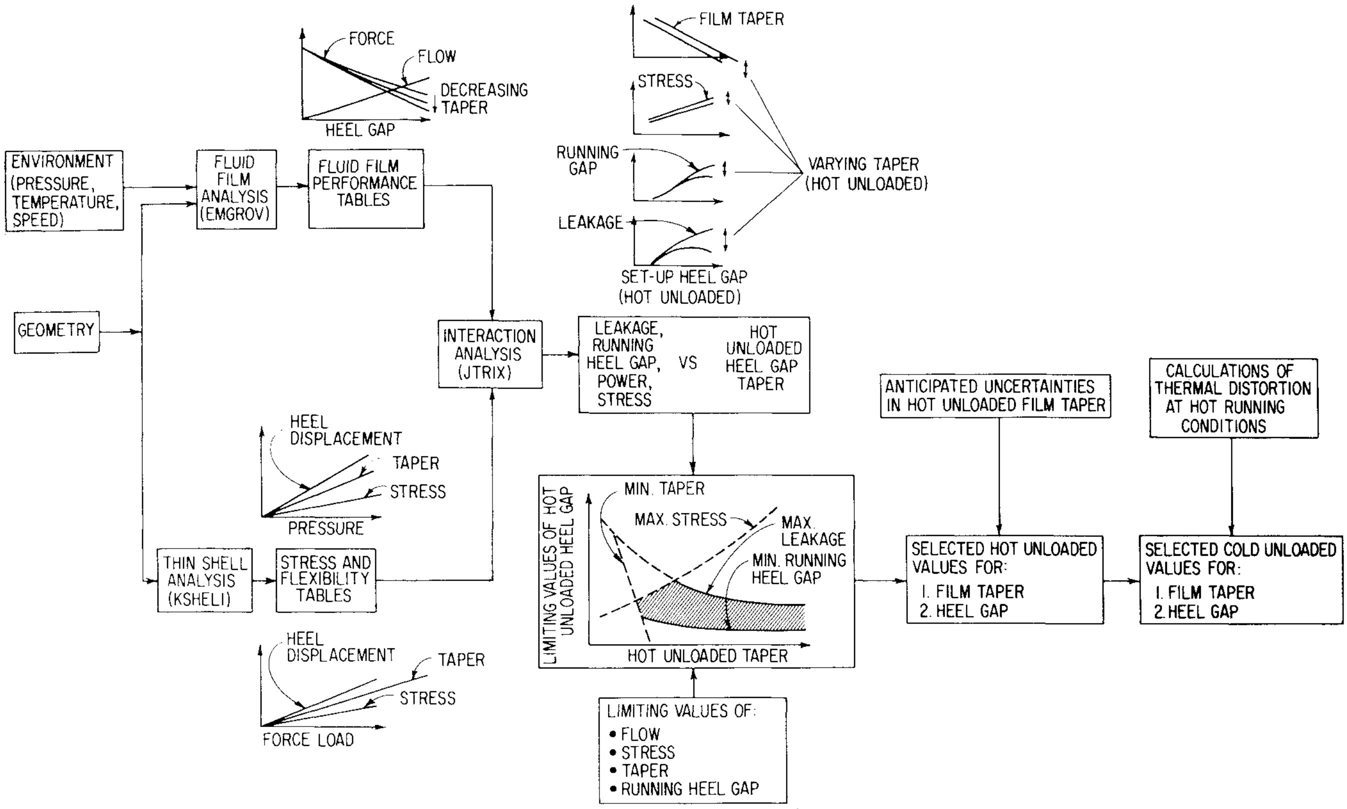

Fig. 2 Information flow for J-Seal analysis.

ters and to the unloaded values of heel gap and running taper.

IV. Analysis of J-Seal Operation

Analytical Summary

The need for flexibility and stress analysis, fluid-film analysis, interaction analysis and distortion analysis have been identified. The corresponding flow of information in the overall design process and the nature of the data involved are illustrated in Fig. 2. The twin paths of fluid-film analysis and flexibility-stress analysis are followed to produce tables of data for input to the interaction analysis. The interaction analysis, in turn, produces overall performance as a function of hot, unloaded taper, and heel gap, and provides the bounds on safe operation relative to imposed limits. On the basis of this information, the optimum value of hot, unloaded film taper from ideal considerations can be selected.

This paper is concerned primarily with the selection of J-Seal geometry and of hot unloaded values for heel gap and taper. This design analysis can be performed independently of the structure on which the J-Seal is mounted, and is, therefore, generally applicable. However, in any specific application, particularly where temperatures are high, thermal distortions of the particular structure must be accounted for in relating required hot values of heel gap and taper to corresponding cold values.

The following analytical approach was employed in implementing this design process: treat the membrane and fluid film as axisymmetric; include fluid inertia effects to the extent that they limit leakage flow predictions; account for turbulence effects by the method of Ng and Elrod[1]; interface fluid-film and flexibility analysis by means of a variable unbalanced, force acting on the seal face; and represent the fluid-film profile as varying linearly across the seal face.

Stress and Flexibility Analysis

Stress and flexibility data define the elastic behavior of the flexible membrane and the seal ring it supports under the loads imposed by the environment. For the purposes of analysis, two types of loading have been considered. These are "pressure loading," which describes a uniform unbalanced pressure acting on the toroidal area between the inner support and the inner edge of the seal face; and "force loading," which describes an unbalanced force applied around a circumferential line on the back of the seal face. "Force loading" represents the resultant of pressure differences between back side and fluid-film side of the seal face. A computer program (KSHEL1) written by A. Kalnins of Lehigh University[3] was used for the flexibility analysis. It is capable of representing all the geometries, boundary conditions, and loads required for this application.

A set of flexibility data consists of axial displacement at the inner radius of the seal face; change in face slope; and maximum stress values; for given values of pressure and force loading. By linear superposition and displacement and stress values can then be obtained for any combination of pressure and force loading.

Fluid-Film Analysis

The equation for pressure in a narrow axisymmetric radial annulus (Reynolds equation), corrected for turbulence effects is

$$\frac{1}{r}\partial/\partial r \frac{rh^3G_z}{\mu}p(\partial p/\partial r) - (\partial/\partial t)(ph) = 0 \tag{1}$$

Subject to the compressor and seal discharge pressures as boundary conditions at OD and ID, respectively.

Equation (1) has been solved under steady conditions ($\partial(ph)/\partial t = 0$) by finite difference methods to determine the radial pressure distribution in film. The pressure distribution was integrated to yield the axial fluid film force. Additional calculations provided the frictional dissipation in the film and the leakage flow.

These calculations were performed for a meaningful range of heel gaps and film taper values, to yield a set of data such as is plotted in Fig. 3. The fluid-film force in Fig. 3 is referred to discharge pressure, that is, a zero force would correspond to a uniform pressure equal to the discharge value. The solid lines for leakage correspond to viscous flow calculations, corrected for turbulence. The broken line corresponds to choked flow, based on the upstream pressure, which is independent of viscous effects. The flow values used for a given taper and heel gap are the lower of the choked or viscous flow values. The power loss data shows a pronounced increase as the heel gap becomes small, and at large heel gaps the power loss tends towards a nonzero minimum value. This nonzero minimum value results from turbulence in the film. The time-dependent form of Eq. (1) was also solved to determine the frequency-dependence of the fluid film stiffness and damping.

Fluid-film performance data is influenced by compressor and seal discharge pressure values, by rotational speed, by radial face width, and by inner radius. Thus, a new set of data is required for each combination of interest for these parameters.

Interaction Analysis

To determine overall performance of the J-Seal, "interaction analysis" must be performed, in which the interaction of fluid-film forces, elastic deflection and fluid-film leakage are considered. The interaction analysis solves the following nonlinear equations relating running values of heel gap and taper (subscript$_o$) to hot, unloaded values of heel gap and taper (subscript$_{oo}$).

$$h_o = h_{oo} - \Delta X_p - C_{xf}\,[\pi\,(r_o^2 - r_i^2)\,\Delta p - F_f]$$

$$\Delta h_o = \Delta h_{oo} - \Delta r \Delta\theta_p - \Delta r C_{\theta f}\,[\pi\,(r_o^2 - r_i^2)\,\Delta p - F_f]$$

The nonlinearities arise because the fluid-film force, F_f, is a function of running heel gap and film taper, as determined by the fluid-film analysis.

A computer program was written to perform interaction analysis. Input to the program consists of tables of fluid-film and flexibility data for a given geometry and environmental condition. One set of output from the program is a table of stress, leakage, running film taper, and running heel gap, as a function of hot unloaded values of heel gap and film taper. In addition, for specified limiting values of stress, leakage, running film taper, and running heel gap, the program computes, over a range of hot unloaded film taper values, the values of hot, unloaded heel gap which will induce operation at each limit. In this way the upper and lower bounds for the hot unloaded heel gap are determined, and a safe operating region may be determined. This feature is illustrated in Sec. V on design selection. Output from the interaction analysis allows values of heel gap and film taper to be selected which will optimize performance of the seal.

V. Design Selection

Section IV described the analysis and data sets which have been used in design of the J-Seal. Three areas of analysis have been considered which are recast as design loops, together with the geometrical variables whose selection they most strongly influence, as follows:

1) fluid film: Δr
2) flexibility-stress: b, t, R, c, and a
3) interaction: unloaded values of heel gap and film taper

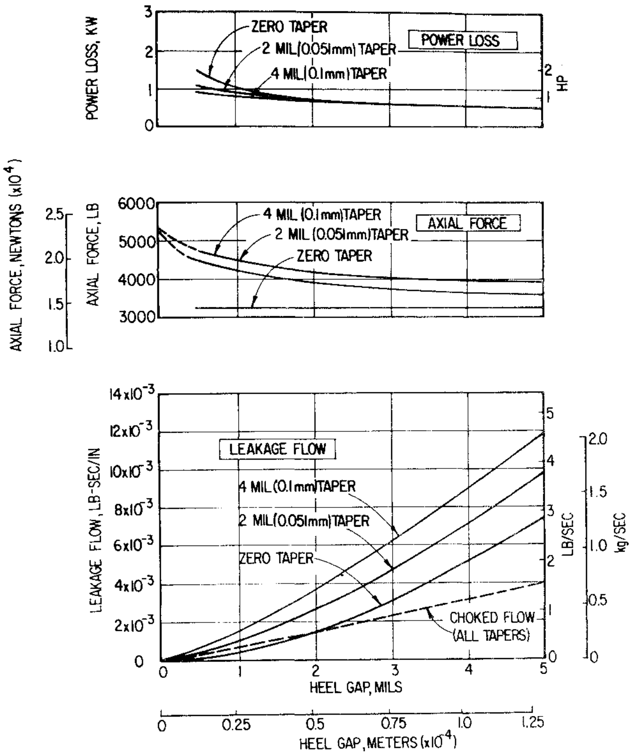

Fig. 3 J-Seal fluid film performance—¼ in. face width; P_{HI} = 600, P_{LO} = 100, T = 1500°F, i.d. = 13 in., N = 12,000 rpm (design conditions).

The design loops have been followed under the design conditions defined in Sec. I. The inputs to the fluid-film loop are typified by the data of Fig. 3. The flexibility-stress loop is based on a study in which the geometrical quantities Δr, b, t, R, c, a, were varied in turn about a reference combination. The reference combination of these quantities is defined as follows:

Membrane thickness (t), 1.143×10^{-3} m (0.045 in.)
Face width (Δr), 12.7×10^{-3} m (0.500 in.)
Outer support leg and face width ($b + \Delta r$), 0.0508 m (2.0 in.)
Inner support leg (c), 1.700×10^{-3} m (0.067 in.)
Outer support offset (a), 0.000 m
Toroid radius (R), 0.0254 m (1.0 in.)

On the basis of the fluid-film and flexibility loops, two geometries were selected: a design geometry intended for the design conditions and a test geometry intended for the less severe test conditions. For each geometry the interac-

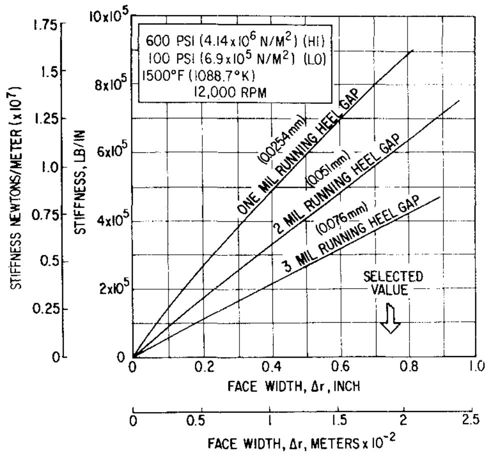

Fig. 4 Fluid film stiffness as a function of face width—design conditions (viscous flow assumption).

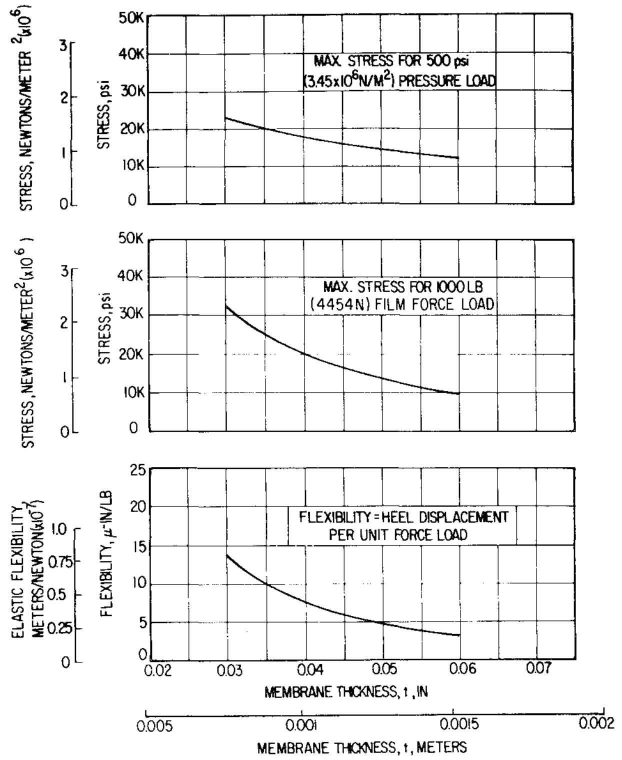

Fig. 5 J-Seal—influence of membrane thickness.

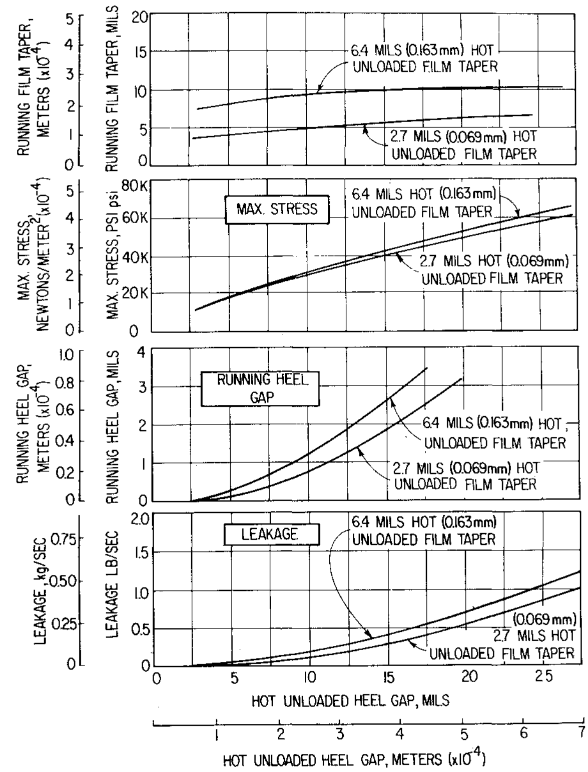

Fig. 7 J-Seal performance—test conditions (test geometry) 300 psi *(HI)*, 50 psi *(LO)*, 12,000 rpm, 1200°F.

tion loop then allowed the selection of optimum values for hot unloaded heel gap and film taper.

The details of the design analysis are not reproduced here. Typical working curves are shown in Figs. 4–8, which show, respectively, the influence of: face width on fluid-film stiffness; membrane thickness on flexibility and stress; inner support leg on flexibility and stress; hot unloaded heel gap on running taper, and heel gap, maximum stress and leakage; and hot unloaded taper on limiting heel gap values.

The selected geometries for both test and design conditions are summarized in Table 3. As discussed, only hot values for unloaded heel gap and taper are given. Cold values must be determined for each specific support

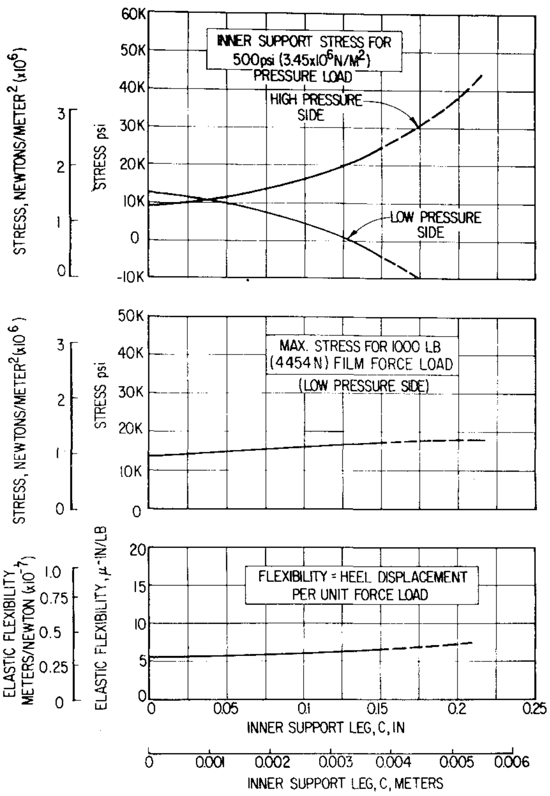

Fig. 6 J-Seal—influence of inner support leg.

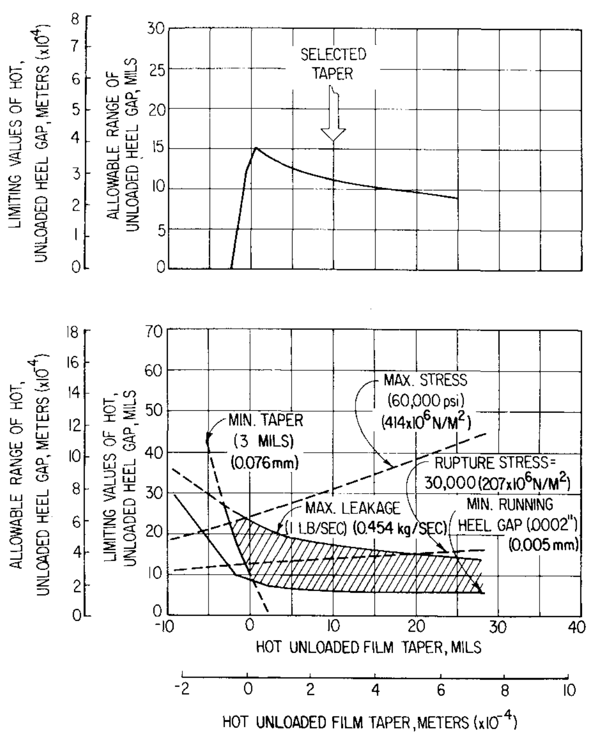

Fig. 8 J-Seal performance limits—design conditions (design geometry) 600 psi *(HI)*, 100 psi *(LO)*, 1500°F, 12,000 rpm.

Table 3 J-Seal selected geometry, taper, and setup heel gap

	Test conditions	Design conditions
Outer support axial offset (a)	0.00 m	0.00 m
Outer leg length and face width ($b + \Delta r$)	0.0508 m	0.0508 m
Face width (Δr)	0.01905 m	0.01905 m
Membrane thickness (t)	7.02×10^{-4} m	11.43×10^{-4} m
Inner leg length (c)	2.54×10^{-3} m	2.54×10^{-3} m
Toroid radius (R)	0.0254 m	0.0254 m
Hot unloaded film taper	1.27×10^{-4} m	2.54×10^{-4} m
Hot unloaded heel gap	3.302×10^{-4} m	2.794×10^{-4} m
Inner diameter	0.3302 m	0.3302 m

structure accounting for thermal growth and distortion due to the operating temperature environment and heat generation over the seal face. The performance of the two designs under normal operating conditions is described in Sec. VI.

VI. Performance Calculations

Steps in the selection of design configurations for the "design" and "test" conditions were defined in Sec. V. The corresponding two geometries selected are defined in Table 3. This section presents the performance of these configurations under nominal conditions.

Static Performance

In Table 4, static performance at the test and design geometries is presented for operation under nominal conditions. Nominal stresses for either design are very similar, reaching about 1.79×10^8 N/m^2 (26,000 psi) at the inner support. Some differences can be seen in the values of *running* taper and heel gap and these can be related to similar differences in the design values of hot *unloaded* taper and heel gap. The leakage is, of course, higher for the design conditions because of the higher pressure. Slightly more power dissipation is to be expected from the design geometry because of its smaller running heel gap.

Dynamic Performance

Dynamic stiffness of the fluid film over the range of frequencies of interest was determined to be indistinguishable from static stiffness employed in the design analysis. On the basis of the fluid film and membrane acting in parallel on the seal ring the first natural frequency of the seal was calculated as higher than 1250 Hz, which is over six times the rotational frequency. Thus, the dynamic tracking ability of the J-Seal will not differ from the predictions of static analysis.

VII. Comparison of the J-Seal to a Labyrinth Seal

As has been previously noted, the J-Seal is intended to replace, in some applications, an abradable labyrinth seal. Because of the drastically different design geometries of the two seals and their differences in flow path orientation (radial for the J-Seal, axial for the labyrinth), it is difficult to make a direct performance comparison. However, as a measure of the sensitivity of each seal to runout or uncertainty in the clearance, the leakage has been calculated as a function of relative displacement of stator and runner, with zero flow (contact) as a common reference. The leakage calculations are based on the following assumptions: both seals are at the same diameter; the labyrinth seal has eleven teeth; and the clearance is circumferentially uniform.

Table 4 J-Seal performance summary

	Test geometry	Design geometry
High pressure	2.07×10^6N/m^2 (300 psi)	4.14×10^6N/m^2 (600 psi)
Low pressure	3.45×10^5N/m^2 (50 psi)	6.89×10^5N/m^2 (100 psi)
Temperature	649°C (1200°F)	816°C (1500°F)
Speed	12,000 rpm	12,000 rpm
Running heel gap	3.556×10^{-5}m (0.0014 in.)	2.54×10^{-5}m (0.001 in.)
Running taper	1.854×10^{-4}m (0.0073 in.)	3.226×10^{-4}m (0.0127 in.)
Leakage	0.09525 kg/sec (0.21 lb/sec)	0.1406 kg/sec (0.31 lb/sec)
Stress, outer support	1.338×10^8N/m^2 (19,400 psi)	1.448×10^8N/m^2 (21,000 psi)
Stress, outer face edge	1.283×10^8N/m^2 (18,600 psi)	1.241×10^8N/m^2 (18,000 psi)
Stress, inner support	1.773×10^8/N/m^2 (25,700 psi)	1.724×10^8N/m^2 (25,000 psi)
Power	1716 N m/sec (2.3 hp)	2238 N m/sec (3 hp)
Stress fluctuation for 4 mil runout (peak-to-peak)	6.69×10^7N/m^2 (9700 psi)	6.08×10^7N/m^2 (9400 psi)
Leakage per unit seal length	0.0918 kg/sec-m (5.14×10^{-3} lb/sec-in.)	0.136 kg/sec-m (7.58 lb/sec-in.)

The results are plotted in Fig. 9, and show the J-Seal, in the range from zero to 2.54×10^{-4} m (0.010 in.) of relative displacement, to have an average of 68% lower leakage. Of course, shifts larger than 10 mils in relative displacement are to be expected over the operating range of an engine. However the time span of these larger displacements, since they involve solid body shifts of the engine rather than dynamic runout, is such that they can be accommodated by a slower moving (more massive) seal carrier.

VIII. Discussion

The design analysis described in this paper demonstrates, analytically, the feasibility of the J-Seal concept, both under conditions representative of advanced (1980s) gas turbine engines and under conditions typical of more near-term and current engines. Test results are now need-

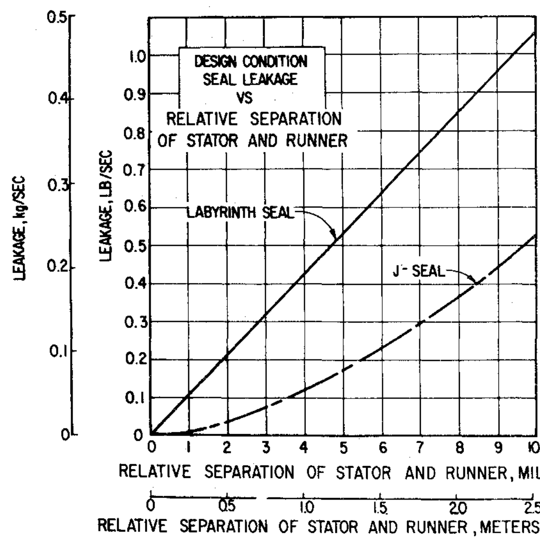

Fig. 9 Comparison of labyrinth seal and J-Seal.

ed to prove the validity of the concept in practice, and these results will form the substance of a future publication. Following rig tests, an application test in a currently designed engine is being scheduled.

When the concept has been proven by test, the analysis described in this paper should be advanced on several fronts. First, the combined effects of viscous and inertia forces on the pressure distribution should be investigated. The present analysis includes fluid inertia effects only to the extent that they limit the seal leakage. An additional analysis indicated that similar fluid-film forces would be obtained if the fluid equations of motion were solved preserving inertia terms and neglecting the viscous terms retained in the present analysis. However, a more thorough investigation of the fluid-film behavior, with viscous and inertia effects included both for static and dynamic conditions is called for. The recent work of Zuk and Smith[4] offers a means of performing this more complete analysis under static conditions. Secondly, the flexibility of the membrane under action of one- and two-per-rev force variation and the interaction of this flexibility with corresponding one- and two-per-rev variations in the fluid film thickness should be investigated. A brief study utilizing the "wave-number" capability of the KSHEL1 computer program indicated as much as 40% reduction in flexibility of the membrane under one- and two-per-rev excitations, but corresponding increases in fluid-film stiffness are to be expected as a result of hydrodynamic effects. The interaction deserves further study.

The dynamic behavior of the J-Seal during transient shifts in relative location of stator and rotor should be investigated analytically. This analysis should include interaction of the J-Seal with its slower moving carrier. Now that the components of the design analysis have been identified and investigated on an individual basis, a more complete systems analysis is required so that final performance prediction, such as presented in Table 2, can be optimized explicitly with respect to the geometrical design parameters.

At several points in this paper, the difference between test conditions and design conditions has been identified. The test conditions, while severe, do simplify the problems of verifying the concept and the analysis, while remaining close enough to the design conditions to allow valid extrapolation. Moreover they are representative of current and near-term engine applications. The design condition involves first a higher pressure (4.4×10^6 vs 2.07×10^6 N/m^2) (600 psi vs 300)—hence a stiffer fluid film, but also the potential for higher stresses in the material; and secondly, higher temperature (816°C vs 649°C) (1500°F vs 1200°F)—hence increased material flexibility, but reduced material strength and creep rupture limit. In developing the design geometry of this paper these effects have been accounted for, even though they limit the available design options. However, the selected configuration demonstrates, analytically, the feasibility of designing for these conditions. Success under the test conditions may be interpreted as indicating good likelihood of success under the more severe design conditions.

Finally, it should be pointed out that the paper has concentrated on the concept of the J-Seal and its analytical feasibility. At the same time there are many critical design questions concerned with fabrication of the J-Seal, joining of the seal ring to the membrane, and the design of the supporting structure (carrier). These will be discussed in more detail in a future publication.

IX. Conclusions

1) The J-Seal concept is analytically feasible. 2) The J-Seal configuration as designed will provide acceptable sealing performance while the relative axial location of stator and runner varies by over 2.54×10^{-4} m (10 mils). 3) The J-Seal shows substantial performance improvement with respect to a comparable labyrinth seal, in terms of tolerance to runout and uncertainty. 4) Test results are now required to prove the J-Seal's feasibility. 5) Extensions to the analysis presented are required in the consideration of inertia effects; in the treatment of nonaxisymmetric effects, and in the system optimization analysis.

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