JOURNAL OF PROPULSION AND POWER
Vol. 10, No. 4, July–Aug. 1994

Hysteresis and Bristle Stiffening Effects in Brush Seals

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Extensive testing of conventional brush seals has identified the phenomena of bristle "hysteresis" and "stiffening" with pressure as their two major drawbacks. Subsequent to any differential movement of the runner into the bristle pack due to its radial excursions or centrifugal/thermal growths, the displaced bristles do not recover against the frictional forces between them and the backing plate. As a result, a significant leakage increase is observed following any runner movement. Furthermore, the bristle pack exhibits a considerable stiffening effect with the application of pressure. This phenomenon may adversely affect the life of the seal and the runner due to a highly increased mechanical contact pressure at the sliding interface. In comparison with these conventional design seals, the characteristics of an improved design, known as the "low hysteresis" design, are presented here. This design shows a substantially lower degree of the detrimental effects mentioned above. This type of seal can maintain its reduced leakage characteristics throughout the running cycle with runner excursions and growths. The bristles also do not show any stiffening, up to a certain pressure threshold. Therefore, this seal also has a potential for a longer life than a brush seal of conventional design.

Nomenclature

D_b = back plate i.d., in.
D_i = brush seal i.d., in.
D_o = brush seal o.d., in.
D_r = retaining (side) plate i.d., in.
d = bristle diameter, in.
E = modulus of elasticity, psi
K_{bristle} = generalized bristle stiffness, psi/mil
L = bristle free length, in.
$\dot{m}$ = mass flow rate, lbm/s
N_{row} = number of bristle rows
P_r = pressure ratio, p_u/p_d
p_d = downstream pressure, psia
p_{mc} = mechanical contact pressure at bristle/runner interface, psi
p_u = upstream pressure, psia
R_s = bristle stiffness ratio, $K_{\text{bristle}}(\Delta p)/K_{\text{bristle}}(0)$
T = fluid temperature, °R
w = seal width, in.
Δp = pressure differential across the seal, $(p_u - p_d)$, psid
δ_r = radial interference between the bristles and runner, in.
θ = bristle angle, deg
Φ = flow parameter, $(\dot{m}\sqrt{T})/(p_u D_i)$, lbm·in·°R$^{1/2}$/lbf·s

Introduction

OVER the last decade, brush seals have emerged to be a very promising technology for gas-path sealing in gas turbine engines. Tests conducted by various investigators[1-5] indicate that a substantial reduction in secondary flow leakage can be achieved using brush seals over the state-of-the-art labyrinth seals. Studies[4,5] have shown that improvements to internal flow system to reduce leakage can yield an increase in thrust by as much as 17%, and decrease in specific fuel consumption by over 7%. The superior leakage performance of brush seals mainly results from the fact that these seals accommodate transient radial motions of the engine rotors without permanently enlarging the seal leakage area. Brush seals, therefore, can retain their performance even after large excursions, whereas labyrinth seals suffer degradation due to increased clearance. Consequently, brush seal development is a specific technology that the U.S. Air Force has identified to pursue as an "Integrated High Performance Turbine Engine Technology (IHPTET)" initiative because of its innovative design features and high-risk, high payoff potential in improving overall engine efficiency and thrust-to-weight ratio.

As part of a "Brush Seal Development Program," funded by the U.S. Air Force, extensive testing of conventional brush seals was conducted at EG&G Fluid Components Technology Group (FCTG) R&D Laboratory. This study has identified pressure induced bristle hysteresis and stiffening phenomena which are deleterious to the performance of conventional brush seals. Subsequent to any transient differential movement of the runner into the bristle pack, the displaced bristles do not recover due to the hysteresis effect. As a result, the leakage increases considerably over the initial level with concentric condition and interference. This phenomenon tends to undermine the very characteristic that discriminates a brush seal from a labyrinth seal. The bristles, however, do ultimately recover when the pressure differential is reduced to almost 0. In addition, the bristle stiffening with pressure considerably increases the contact pressure at the sliding interface, thereby accelerating the bristle/runner wear and diminishing seal life.

In this article, the pressure induced bristle hysteresis and stiffening effects of the conventional brush seals are discussed, and the experimental results are presented. In addition, the characteristics of an improved design, which is referred to as the low hysteresis design, are presented. The low hysteresis design minimizes the detrimental effects mentioned above.

Received May 7, 1993; presented as Paper 93-1996 at the AIAA/SAE/ASME/ASEE Joint Propulsion Conference, Monterey, CA, June 28–30, 1993; revision received Dec. 7, 1993; accepted for publication Dec. 8, 1993.

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This design has the ability to maintain low leakage even after rotor excursions and runouts, and also has the potential of an extended life due to a diminished contact load at the sliding interface.

Bristle Hysteresis Effect

Figure 1 shows the schematic diagram of a conventional brush seal. When in operation, Δp acts over the annular area between D_r and D_i. This unbalanced axial force causes the bristle pack to load against the backing plate with a high mechanical contact pressure over the annular surface between D_r and D_b. Figure 2 presents the free body diagram of the bristle pack. The resultant force due to the contact pressure as well as the frictional force are indicated on the figure. Contact forces are also generated at the bristle-to-bristle interfaces within the pack by varying degrees. These frictional forces give rise to bristle hysteresis with any radial movement. For example, during a rotor excursion, the bristle pack is forced radially outward. Subsequently, when the rotor withdraws, the bristles do not drop back down on the shaft, causing bristle "hang up," unless the applied pressure is decreased to a low value. In fact, experiments indicate that even a few psig pressure differential across the seal prevents free radial movement of the bristle pack. This bristle hysteresis leaves a large gap, and hence, causes an appreciable increase in leakage after any transient differential movement of the runner into the bristle pack. This has been experimentally demonstrated many times at EG&G FCTG R&D Laboratory. The leakage increase is also likely to be cumulative as the shaft excursions take place in different radial directions by different amounts, and along the entire circumference during rotor thermal or centrifugal growth.

Bristle Stiffening Effect

As mentioned earlier, another detrimental effect of the contact pressure between the bristle pack and the backing plate is to increase the bristle stiffness. This phenomenon is caused by the fact that the bristles are prevented from moving freely in the radial direction by the frictional force, thereby reducing their effective free length.

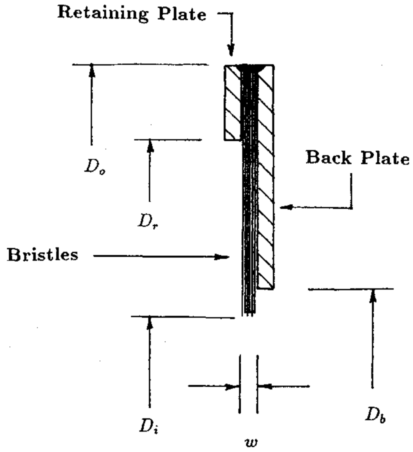

Fig. 1 Schematic diagram of a conventional brush seal.

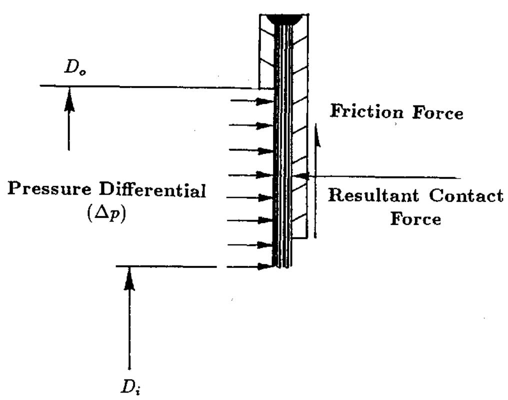

Fig. 2 Free body diagram of a conventional brush seal.

K_{bristle} is a fundamental parameter to characterize a brush seal. This is defined as the pressure required at the bristle tips to displace them radially by a unit magnitude. The following functional relationship can be written:

$$K_{\text{bristle}} = f(d, L, \theta, w, E, \Delta p) \tag{1}$$

For a given design, however, it is a function of Δp only, and is denoted as $K_{\text{bristle}}(\Delta p)$. Although $K_{\text{bristle}}(0)$ can be estimated theoretically at the 0 pressure differential condition, its value for a certain Δp is generally determined experimentally which is covered in a later section.

For a given δ_r, the p_{mc} at the bristle/runner interface can be estimated using the following relationship:

$$p_{\text{mc}} = K_{\text{bristle}}(\Delta p)\delta_r \tag{2}$$

The p_{mc} at the interface has a direct bearing on the wear rate of the bristles, the life of the runner surface, and the amount of heat generated at the interface. A lower p_{mc} would yield a lower wear rate and hence, a longer seal life.

Experiments done at EG&G FCTG R&D Laboratory demonstrated that the K_{bristle} of a conventional brush seal appreciably increases with pressure. Typically, an order of magnitude increase in bristle stiffness was measured with a differential pressure rise of 60 psig. Because of this bristle stiffening effect, any differential movement of the runner, e.g., radial excursion or growth into the bristle pack, will introduce very high p_{mc} at the sliding interface, causing an accelerated wear of the bristles and degradation of the runner surface. Subsequently, when the runner withdraws to its normal position, it will leave a clearance resulting in an increased leakage as well.

Experimental Procedure

In the following two sections, the experimental arrangements to measure the bristle stiffness and to introduce radial excursions during dynamic testing are described.

Stiffness Measurement—Static Test

A schematic of the test apparatus is shown in Fig. 3. In this setup, two identical seals with the same nominal K_{bristle} are mounted on a runner with a face-to-face arrangement. The cavity between the seals is pressurized to create a desired pressure differential across the seals. A certain amount of radial interference, typically 0.010 in., is maintained between the seals and the runner. Although, for the sake of clarity, the runner diameter is shown to be smaller than the seal i.d., it is actually slightly larger to provide for the necessary interference. The seals are secured in a seal holder which is moved up and down vertically with respect to the fixed, static runner, by turning the adjustment nut at the top. The force required to do so is measured with a load cell. The seal holder is initially made concentric with the runner so that there is no net force coming from the bristle/runner interface, and the load cell registers only the known dead weight of seal holder and seal assembly. As the holder is made eccentric with respect to the runner, the force on the load cell increases. The radial eccentricity Δ_r is measured with a Bently proximity probe, targeting on a flat-faced pin attached to the bottom of the seal holder. The forces at different radial eccentricities are measured during the ascending and descending travel of the seal holder. The initial slope of the ascending curve gives twice (since there are two identical seals) the seal stiffness K_{seal} which can be related to the K_{bristle} as follows:

$$K_{\text{seal}} = CK_{\text{bristle}} \tag{3}$$

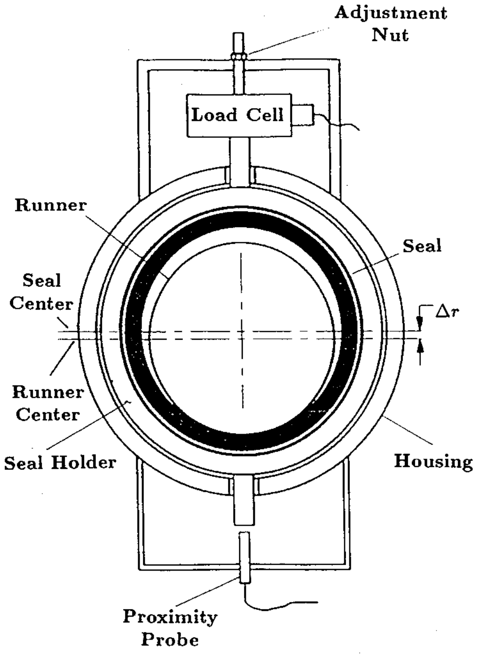

Fig. 3 Schematic diagram of the stiffness measurement setup.

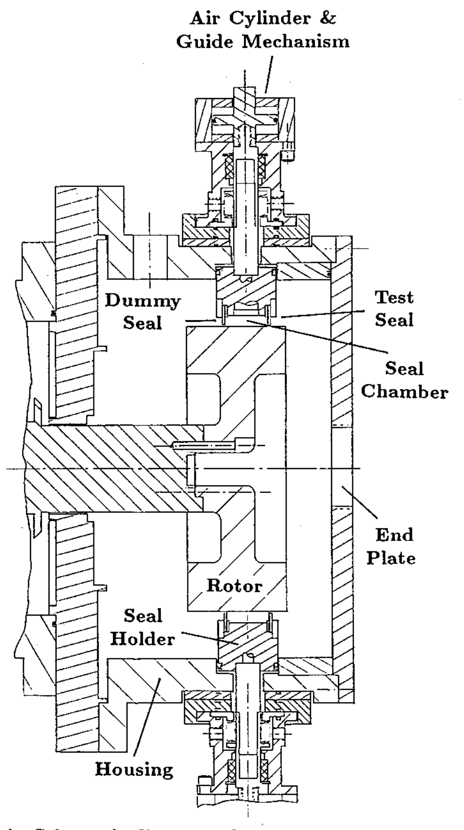

Fig. 4 Schematic diagram of the radial excursion setup.

where C is the constant of proportionality which depends on the seal geometry only.

The cavity between the seals is pressurized to create a certain pressure differential across the seals, as desired. The above test is repeated for various pressure differentials. The values of K_{seal} at different Δp are determined from the initial slopes of the corresponding load-eccentricity curves. The K_{bristle} values are next calculated using Eq. (3).

Radial Excursion—Dynamic Test

Figure 4 shows a sketch of the arrangement to impose a radial excursion of the seal into the runner during dynamic testing. In order to eliminate or minimize any axial load due to pressure on the test rig spindle, two brush seals are mounted face-to-face in a movable seal holder. One seal is the actual "test seal" under evaluation, and the other seal is the "dummy seal" required to pressurize the chamber. The movable holder is allowed to move along a vertical axis up to 0.032 in. away from the center. Vertical movement is guided by two shafts, attached to the seal holder, one at the 12 o'clock and the other at the 6 o'clock positions. Each shaft extends through the pod to an air cylinder and guide mechanism, which is attached to the test rig housing. The force to move the holder is supplied by these two air cylinders and the magnitude of the force can be regulated by adjusting the air pressure to the cylinders. Travel of the seal holder is controlled by an adjustable stop on each air cylinder.

Pressure is introduced into the seal chamber via flexible pipings which extend through the test pod and into the movable holder. Leakage through the test seal is measured with a flow meter in series with a pipe attached to a port on the end plate.

Results and Discussion

In the following sections, the characteristics of conventional seals are compared with low hysteresis seals. Two seals of different designs in each of the above two categories have been chosen. These seals are denoted as "conventional seal 1, 2" and "low hysteresis seal 1, 2," respectively. D_o and D_i of all the above seals are 6.414 and 5.375 in., respectively.

Bristle Stiffness

Figure 5 presents the R_s band of two conventional seals of different bristle stiffness characteristics [i.e., different $K_{\text{bristle}}(0)$]. The lower and upper lines correspond to the conventional seal 1 and 2, respectively. The K_{bristle} at a given Δp is nondimensionalized with respect to the corresponding 0 pressure value, e.g., $K_{\text{bristle}}(0)$. As seen in Fig. 5, the bristle stiffness of the conventional seals increases by more than an order of magnitude over the range of pressure differential considered (e.g., 80 psig). As mentioned earlier, this phenomenon would cause a correspondingly higher mechanical contact pressure at the sliding interface, resulting in higher wear and shorter life.

The bottom curve in the figure shows the stiffness ratio curve of the low hysteresis seal 1. It is evident that the bristle stiffness does not increase over $K_{\text{bristle}}(0)$ until a pressure threshold of 30 psig is reached. However, this threshold value beyond which the stiffness begins to increase depends on a particular design and can be extended by changing various geometrical design parameters. Obviously, this design would cause a lower p_{mc} at the runner interface than a corresponding conventional seal, and hence, has a potential of longer life.

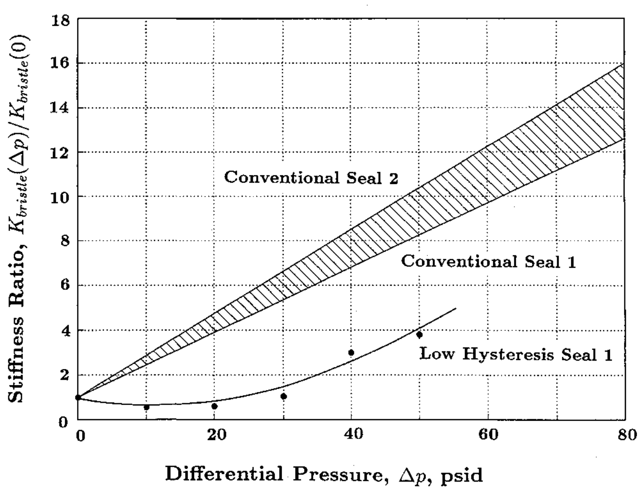

Fig. 5 Stiffness ratio vs pressure differential.

Leakage Hysteresis

The leakage hysteresis characteristics of the above seals have been demonstrated and evaluated with the following three types of dynamic tests: 1) dynamic test with concentric rotor, 2) dynamic test with eccentric rotor, and 3) dynamic test with radial excursion.

There are basically three different ways a runner can have differential movements with respect to the bristle pack. Firstly, the runner's center moves in an orbit eccentric with respect to the brush seal center. This is known as "rotor whirl." In this case, the amount of eccentricity is usually relatively small, e.g., of the order of a few mils. This is a steady-state phenomenon. For the low hysteresis design, it causes continuous wear of the bristle until the bristle i.d. becomes line-to-line with the orbiting rotor o.d. However, for the conventional design, the bristles are pushed apart by the runner during the initial rotations, and are not continuously subject to wear.

Secondly, there can be differential thermal and centrifugal growths between the runner and the brush seal. Thirdly, the brush seal could move into the runner due to the aircraft turning maneuvers, landing and takeoffs. These latter two types of differential movement are transient ones and could be of the order of 0.020–0.040 in., depending on the seal location in the engine. The third type of differential movement of the runner is simulated in the dynamic test with radial excursion. In this test, controlled unidirectional excursions of large magnitude were imposed on the seal. From the leakage data of this test, an attempt has been made to estimate the leakage when there is a uniform differential movement of the same magnitude all around the circumference.

The above tests and the corresponding results are described in the following sections. The leakage data is presented in terms of Φ which is defined as follows:

$$\Phi = (\dot{m}\sqrt{T}/p_u D_i)$$

It may be noted that the flow parameter definition, adopted here, is slightly different from the ones used by other investigators.[3-5]

Also, on most of the plots, P_r is used as an independent parameter instead of Δp, because under choked flow condition, which is quite usual, Φ becomes independent of P_r, but not of Δp. Other investigators[3-5] have also used P_r or its function. However, if the reader is interested in the value of Δp, it can be approximately calculated from P_r, since p_d is always maintained near the atmospheric value for the results presented here.

Dynamic Test with Concentric Rotor

In this test, the rotor assembly was balanced very carefully to minimize any dynamic runout. The seal chamber is pressurized to the desired level. The speed was increased from the static condition to 16,000 rpm, in steps of 5000 rpm, and then back to 0 rpm. At the 5000- and 10,000-rpm conditions, the seal was run for 15 min each, during both ascending and descending parts of the cycle. At 16,000 rpm, the seal was run for 30 min. The leakage was continuously monitored during the cycle. Steady-state leakage at each operating point is reported here.

Figure 6 presents the flow parameter vs speed curve during the entire cycle for the conventional seal 1. During this speed cycle, an attempt had been made to maintain the pressure differential around 30 psig. This means a P_r of about 3 having a near-atmospheric pressure at the seal downstream. At each data point, the corresponding pressure ratio achieved is indicated within parentheses.

As the speed was increased from static to 16,000 rpm, the leakage also increased. When the speed was subsequently decreased, the leakage continued to increase, indicating an appreciable amount of leakage hysteresis. The static leakage at the end of the speed cycle, denoted by point B, was about three times higher than that at the beginning of the cycle, point A. At point B, the seal chamber was completely depressurized and then pressurized back to the earlier level. The leakage dropped to the point C which is close to the static leakage level (A) at the beginning of the cycle. This is indicative of bristle recovery from their "hung up" positions against the back plate. Any small discrepancy between the points C and A may be attributed to minor changes in operating pressure, different runner orientation with respect to the seal, or bristle wear.

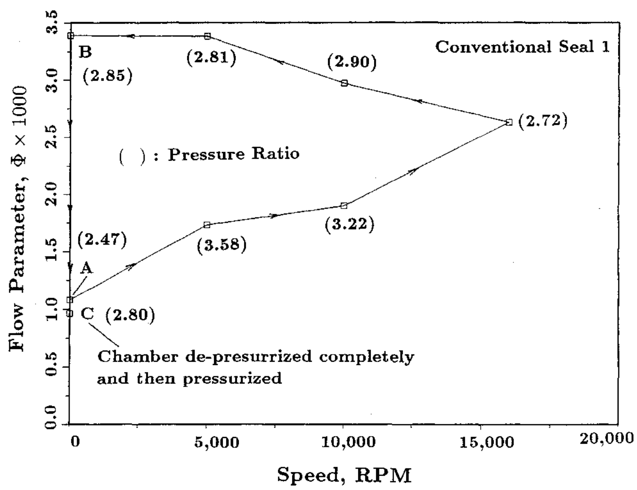

Fig. 6 Leakage hysteresis with speed—conventional seal 1.

The subject seal was almost new and had seen only about 1 h of testing. The interference between the bristle and the runner was about 10 mils under the static condition. Hence, the flow parameter at point A is seen to be about 1×10^{-3}, which is somewhat low. If the seal were to be run continuously over an extended period of time, the bristles would likely wear almost line-to-line with the runner, and the flow parameter at A could be about 2 to 3×10^{-3} under practical concentric, running condition with no bristle hang up.

The above hysteresis phenomenon can be attributed to differential growth of the runner with respect to the bristle i.d. During the ascending part of the speed cycle, the runner grew centrifugally, thereby increasing the effective interference, pushing the bristles radially outward. As a result, during the transients (not shown in the figure), the leakage first decreased and the interface heat generation increased. Due to the combined effects of higher heat generation and lower leakage flow (responsible for less cooling), the interface temperature started increasing. The high interface temperature caused further local growth of the runner, and hence, higher interference. Eventually, the heat diffused slowly into the body of the seal from the interface through the porous bristle pack, causing it to grow thermally as well. With this delayed thermal growth of the seal, the effective interference finally decreased somewhat. The interface heat generation dropped and the runner contracted thermally, leaving a certain amount of radial gap since the bristles, being held by the frictional forces, did not follow the runner. However, during this thermal transient, the centrifugal growth remained unchanged. The intensity of the above thermal transient, and hence, the corresponding clearance due to the thermal contraction, depend on the operating speed, initial interference, and the bristle stiffness at the applied pressure. During the descending part of the cycle, the runner contracted centrifugally, thereby further increasing the clearance, and therefore, the leakage, as seen in the figure. At the end of the speed cycle, denoted by the point B, there was a fairly large clearance between the runner and the bristle tips which was the sum total of the centrifugal and thermal growths, and some possible wear during the cycle. At point B, as the seal chamber was depressurized completely, the frictional forces were reduced to 0 level and the bristles recovered almost fully, closing the above gap. Subsequently, as the chamber was repressurized, the leakage went back to level C which was close to that which existed at the beginning of the cycle, i.e., A.

Figure 7 presents a similar curve for the low hysteresis seal 2. This seal had about the same level of interference as the conventional seal 1, described above. The initial static leakage was higher than the conventional seal. However, the leakage hysteresis associated with this seal was much smaller in magnitude.

Dynamic Leakage with Eccentric Rotor

For this test, a composite runner was used in which radial eccentricity in discrete steps up to 0.010 in. could be dialed in to simulate dynamic shaft vibration. The runner was subsequently balanced in the eccentric condition. Figure 8 presents the flow parameter vs speed for the conventional seal 1 with a radial eccentricity of 0.0025 in. which was half of the total indicated reading (TIR) on a dial indicator. As seen, the hysteresis loop is much larger (note: the scale is different here) than that with no runout as in Fig. 6. The bristles were pushed radially outwards by the initial sweep of the eccentric runner and stayed at that position, leaving a large uniform gap all around. During the descending part of the cycle with centrifugal contraction, the gap was so large and the leakage was high enough that the pressure in the seal chamber could not be maintained at the desired level due to the losses in the piping system. For example, the pressure ratio achieved at point B was only 1.65. Had the pressure ratio been maintained at the desired level (i.e., about 3 as in Figs. 6 and 7), the leakage over the descending part of the cycle would have been much higher, making the hysteresis loop even bigger. Hence, it is demonstrated that in a conventional design, the bristles are prone to hysteresis, the extent of which depends on the magnitude of bristle displacement, caused by the differential movement of the rotor. In Fig. 6, the excitation is smaller than that in Fig. 8, and consequently, the hysteresis loop is also smaller.

This eccentric rotor test was not performed on the low hysteresis design at the time of writing this article. However, as will be described in the following section, the low hysteresis design has a much greater ability to overcome the impeding friction and follow the shaft motion, and it is likely to exhibit much smaller hysteresis loop in response to direct displacement excitation of the bristles.

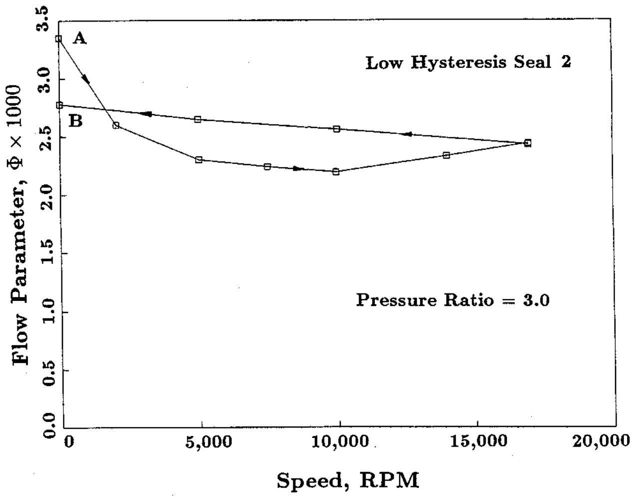

Fig. 7 Leakage hysteresis with speed—low hysteresis seal 2.

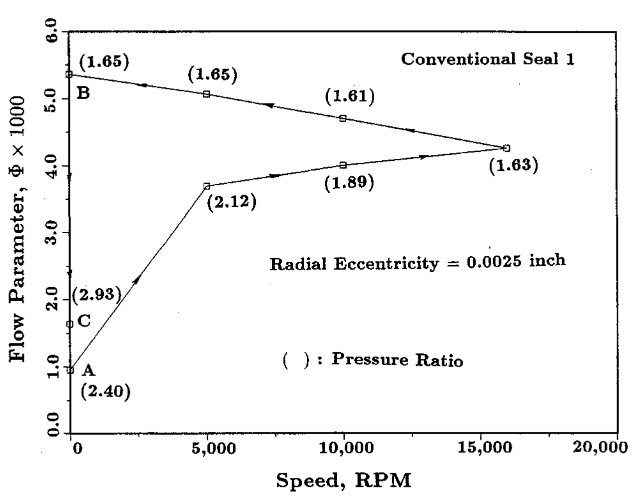

Fig. 8 Leakage hysteresis with speed and rotor radial eccentricity—conventional seal 1.

Dynamic Test with Radial Excursion

In this test, a certain magnitude of radial excursion was imposed for a brief period of time and then withdrawn. As described earlier, the seal holder was moved into the runner by the desired magnitude. In this case, the displacement excitation on the bristles was maximum along the line of action, diminishing to 0 at ±90-deg positions. In case of a whirling rotor, however, the radially eccentric runner would sweep a uniform gap all around, and hence, open up a larger leakage area. Also, in case of a differential thermal growth of the rotor during a thermal burst condition, the bristle displacement will be uniform around the circumference, as well.

Figure 9 shows the leakage characteristics of the conventional seal 1. The lower solid line curve Φ_L gives the leakage under concentric condition with a nominal radial interference of about 0.004 in. The pressure differential across the seal was about 45 psi with a near-atmospheric condition at the seal downstream. The upper solid line curve Φ_H illustrates the leakage after a radial excursion of about 0.018 in. of the seal into the runner. The pressure differential after the excursion dropped to about 25 psi, due to the higher losses in the piping system resulting from an increased leakage. A large gap was created about the direction of excursion because of the bristle hysteresis effect. The displaced bristles did not seem to overcome the frictional forces even if a reasonable time was allowed. The only means of bristle recovery was to reduce the applied pressure to almost 0 level. The flow parameters before and after excursions are called Φ_L and Φ_H, respectively. The abscissa of the plot indicate the excursion cycle number. After each excursion, Φ_H was recorded after reaching the steady state. Next, the pressure differential was decreased to 0 and then increased to the earlier level of 45 psi at which point Φ_L was again achieved. It is evident that

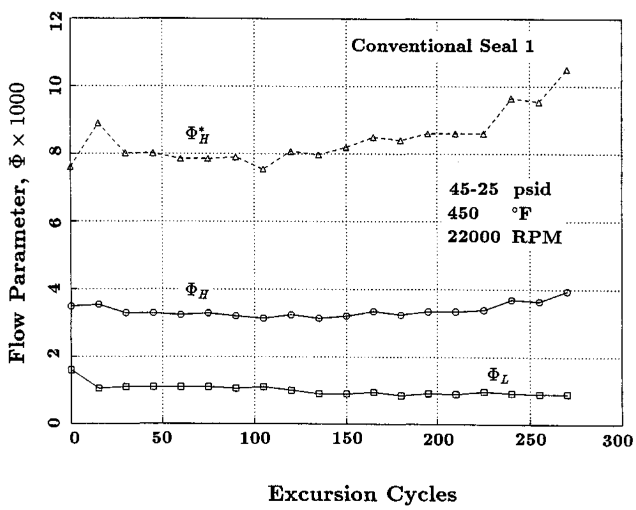

Fig. 9 Leakage hysteresis with radial excursion—conventional seal 1.

the bristle hysteresis causes an appreciable increase in leakage through the conventional seal. This observation has been made repeatedly with many seals of conventional design. Visual observation with a borescope confirmed the bristle hang up following an excursion. Variations of bristle parameters such as its angle, free length, etc., did not change this hysteresis effect by any appreciable amount.

As mentioned earlier, in this test, the stationary seal was moved into the runner and hence, the gap created was not uniform around the circumference, but maximum along the direction of excursion and diminished as a cosine function on its either side. In an actual flight environment, an eccentric whirling rotor may move into the seal, or the rotor may grow radially caused by thermal burst and centrifugal force during a full power condition. This type of differential movement would develop a uniform gap all around the circumference upon rotor withdrawal. The flow factor Φ_H^* under this circumstance for the same amplitude of excursion as the performed test, can be estimated as follows:

$$\Phi_H^* = \Phi_L + \pi(\Phi_H - \Phi_L)$$

The uppermost dashed curve in Fig. 9 shows the estimated value of Φ_H^*. It is evident that under this circumstance, the leakage increase would be considerable.

Figure 10 presents the similar characteristics of the low hysteresis seal 1. The solid curves in the figure show the Φ_H and Φ_L values with about 0.018 in. of radial excursion, as before. The pressure differentials before and after the excursion were about 30 and 25 psi, respectively. As seen, this particular design exhibits very little hysteresis. Visual observation with a borescope again confirmed the bristle recovery immediately following an excursion. With a uniform gap all around, the estimated level of Φ_H^* is much less than that of a conventional seal, as in Fig. 9.

As seen in Fig. 10, the flow parameter Φ_L, under concentric condition, for the low hysteresis seal 1 is appreciably higher than that of the conventional seal 1. The main reason for such difference is the level of interference. The low hysteresis seal 1 was running with a clearance of 0.002 in. (i.e., −0.002 interference), whereas the conventional seal 1 had an interference of 0.004 in., as mentioned earlier. The bottom-most dashed curve Φ_L^* shows the estimated leakage through the low hysteresis seal 1, had the proper interference been maintained. In that case, Φ_H^* would have been even lower. Based on theoretical consideration and experience, the flow factor Φ increases approximately by 1×10^{-3} for a clearance increase of 0.001 in., under inertia-dominated flow conditions.

Looking at Figs. 9 and 10, it may first be deceiving that the low hysteresis seal 1 seems to leak more than the conventional

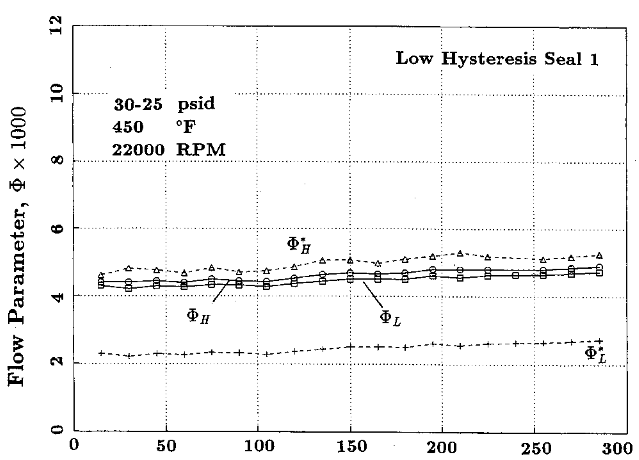

Fig. 10 Leakage hysteresis with radial excursion—low hysteresis seal 1.

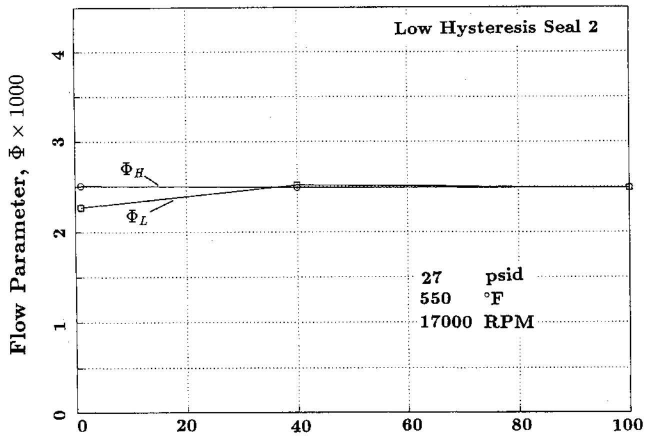

Fig. 11 Leakage hysteresis with radial excursion—low hysteresis seal 2.

seal 1 after the excursion. However, if the appropriate circumstances such as the rotor movement into the seal all around its circumference and the proper interference level with both seals, are taken into consideration, it becomes evident that the low hysteresis design seals would perform substantially better than the conventional seals.

Figure 11 presents the leakage characteristics of the low hysteresis seal 2. The radial excursion level corresponding to Φ_H was 0.013 in., and the pressure differential was about 27 psig. As seen, this seal exhibited very little hysteresis with excursion. Also, the average flow parameter was much less than the previous seal in Fig. 10, since proper interference was maintained with this seal. Again, the bristle recovery after an excursion was confirmed with a borescope.

Conclusions

The bristle hysteresis and the stiffening effects have been identified as two major drawbacks of conventional brush seals. Following a differential movement of the runner into the bristle pack, either due to a radial excursion or centrifugal/thermal growths, the displaced bristles do not recover, causing a large increase in leakage. This phenomenon of bristle hang up has been confirmed with a borescope. Furthermore, the bristle stiffness of the conventional seals is seen to increase considerably with the application of pressure. This phenomenon may adversely affect the longevity of the seal and runner, because of the increased mechanical contact pressure at the sliding interface, particularly during large excursions.

The characteristics of a low hysteresis design have been presented here which exhibits a substantially lower degree of the above hysteresis and stiffening effects. The bristles almost fully recover after any runner excursion or growth, and the seal maintains its low leakage characteristics throughout the running cycle. The bristle recovery has also been visually observed with a borescope. In addition, the bristle stiffness does not increase up to a pressure threshold which is about 30 psid for one of the low hysteresis seals, presented here. However, this threshold value could be extended by varying the pertinent design parameters. Because of the reduced bristle stiffening effect, this design has the potential for a longer life than a conventional seal, without sacrificing its low leakage characteristics throughout the operating cycle.

The "life" or "usefulness" of a brush seal may be considered to have expired, either permanently or temporarily, when its leakage becomes comparable to that of a labyrinth seal. The performance degradation is permanent when the bristles wear to the extent of the operating clearance of a corresponding labyrinth seal, whereas it is temporary when the bristles are pushed apart by the runner to the same extent. In the latter case, if the pressure differential across the seal is not reduced to a very low value during the operating cycle of the engine, the brush seal loses its usefulness during that cycle. As discussed earlier, the brush seals of conventional design tend to lose their usefulness to a varying degree whenever there is any differential radial movement of the runner. On the contrary, the brush seals of low hysteresis design retain their usefulness under the above circumstances. During an excursion or a rotor growth, the mechanical contact pressure at the sliding interface is much higher for a conventional seal than that for a low hysteresis seal due to the bristle stiffening effect. Hence, the amount of seal wear is likely to be more with the former design. However, once the bristles are in a hung up position in the above seal, the rotor differential movement causes no or limited wear. In case of the low hysteresis design, however, every rotor differential movement will cause some wear, since the bristles are always in close proximity to the rotor. Although the amount of such wear is likely to be much less than that for a conventional design in contact with the runner. It appears that long-term engine and rig testing is necessary to determine comparative wear characteristics of conventional and low hysteresis design brush seals. However, it is evident that a conventional brush seal is likely to lose its usefulness during a much longer period of engine operating time compared to a low hysteresis design.

Acknowledgments

The authors wish to acknowledge that the research conducted and reported herein was part of a project under the Integrated High Performance Turbine Engine Technology program and was funded by a U.S. Air Force Contract, F33615-90-C-2001. It was directed by Constance A. Dowler, USAF, the Government Project Engineer. The authors would also like to thank Robert Prior who was also involved in some of the testings, reported here.

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