Advances in Labyrinth Seal Aeroelastic Instability Prediction & Prevention

D. R. Abbott

INTRODUCTION

Labyrinth air seals are widely used in turbo-machinery. Their purpose is to provide air seals between adjacent annular spaces where these spaces are enclosed partially by static structure and partially by rotating structure. A labyrinth seal therefore comprises a rotating part and a static part. The rotating part of the seal has several fin-shaped seal teeth typically supported by cylindrical structure. The static part of the seal usually has a structure supporting a surface which is tolerant of rubbing by the tips of the rotating seal teeth. This rubbing surface is sometimes made of abrasible material; but, for high temperature seals, is usually made of very thin metal honeycomb.

Labyrinth air seals are not perfect seals. The clearance between each seal tooth and the static rubbing surface acts as an annular orifice and the seal as a whole acts as a series of orifices. The seal leakage flow depends on the number of seal teeth, the seal diameter, clearance, and the pressures in the annular spaces being sealed. A small diameter aircraft engine seal with characteristics which allow operation at tight clearances may have a leakage flux as little as 0.1% of the main engine flow. A larger diameter seal where the final operating clearances are set by rubs during transient operation will typically have a leakage flow of 0.5% of the main engine flow. The available energy in the leakage flow is considerable and has often caused aerodynamically excited seal vibration with subsequent fatigue failure.

The purpose of this paper is to describe a particular aerodynamically excited vibration problem and how it was solved, and to contribute a new understanding of the aeroelastic vibration phenomenon and how it may be prevented.

VIBRATION PROBLEMS IN LABYRINTH SEALS

Labyrinth seals have often been susceptible to vibration. Several different mechanisms have been identified as the cause of vibration. Some of these mechanisms have been described in references (1) - (3). Each of these references describes at least one case where the vibration was caused by aerodynamic instability excited by the seal leakage flow.

References (2) and (5) describe how aerodynamic instability may be prevented by adjusting the seal clearances. Although this does result in some increase in seal leakage flow, the effect of this increased leakage in reducing engine performance was not considered to be significant.

References (1), (3), and (4) describe how the aerelastic instability of rotating seals may be prevented by mechanical damping devices and also records the fact that there are no known cases of vibration where a rotating seal has been supported on the low pressure side.

References (1), (2), (3), and (4) also define design parameters which can be used to determine if aerelastic instability would occur and, hence, if clearance adjustments or mechanical dampers are needed for its prevention.

The vibration problem which will be described led to a new analytical model which predicts some of the empirically observed phenomena noted in the above references.
Mechanical damping devices which provide friction damping have generally been very successful. Reference (3) describes the solution to a stationary seal vibration problem and also describes the use of damper rings to protect rotating seals from vibration. Reference (4) describes further applications of damper rings to rotating seals and also describes the use of a damper sleeve to protect a rotating tube from vibration.

A typical application of a damper ring on a rotating seal is shown in Figure 1. Two seals are shown in this figure. The outer (upper) seal is supported on its low pressure side and is not provided with a damper ring. As previously noted in references (1), (3), and (4), there are no known cases of vibration problems with rotating seals supported in this way. The inner (lower) seal is supported on its high pressure side and is provided with a damper ring.

The criterion used in determining damper ring effectiveness is to select a location on the seal where the flexural circumferential strain of the seal is a maximum. An analysis of several low frequency vibration modes was carried out and a damper ring position selected which would be effective in all of these vibration modes.

Based on all of the design experience available at the time, both seals were considered to be completely protected from aeroelastic instability. This proved to be the case for the outer seal which has always operated without problems. The inner seal, however, did not operate as successfully as all the earlier designs which were provided with damper rings.

SOME UNEXPECTED PROBLEMS

Although the inner seal was considered to be completely protected from aeroelastic instability, during factory testing of engines a total of four seals suffered from fatigue cracks. Two cases occurred on an earlier version of the inner seal shown in Figure 1. This earlier seal had its damper ring located at the extreme left-hand end of the seal. Since the damper ring was obviously not providing enough damping, a careful analysis of the possible vibration modes was carried out and it was determined that the damper ring position shown in Figure 1 would be considerably more effective. The seal was redesigned as shown but again two seals cracked. A typical fatigue crack is shown in Figure 2. The fatigue cracks never progressed beyond the extent shown, and were not detrimental to engine operation. In fact, none of the cracks were discovered until routine teardown inspections were carried out.

As a result of these problems, a program was set up to design and evaluate an effective damper and also to understand the vibratory mechanism which was causing the cracks. That part of the program which led to the understanding of the vibratory mechanism will be described first.

STABILITY TESTING

A static rig was designed to test the seal at the leakage flow conditions which would occur in an engine. A photograph of the rig is shown in Figure 3. The space between the inner and outer seals was pressurized over a range of pressures which would simulate engine operation. Three displacement probes were positioned as shown in Figure 3. These displacement probes measured vibratory displacements of the inner seal. Three dynamic pressure probes were placed in the honeycomb cells of the static seal (see Figure 1) to measure any pressure oscillations between the seal teeth.

The results of this test were that the inner seal would commence to vibrate over a wide range of leakage flows. Typical displacement and pressure traces are shown in Figure 4. The vibration mode was identified as a three nodal diameter mode which is the lowest frequency vibration mode for this seal. It may be noted from Figure 4 that the pressure oscillations lead the displacement in phase. This is a classic case of aeroelastic instability where the pressure forces lead the motion and provide enough work input to overcome any natural damping in the system.

Another objective of this test was to determine the effect of the natural frequency of circumferential acoustic waves on the seal stability. Since the unstable mode had been identified as a three nodal diameter mode, it was
decided to add six weights to the inner seal as shown in Figure 3. By successively adding increasingly heavy weights, it was possible to decrease the natural frequency of the inner seal until it was equal to or less than the natural frequency of a three nodal diameter acoustic wave. The result of this phase of testing was that with the mechanical frequency equal to or less than the acoustic frequency, the seal vibration could no longer be excited by the leakage flow. This was a new result which could not be predicted by existing analytical models. Based on this newly observed phenomenon, the existing analytical models were extended to include the effect of acoustic natural frequency on seal stability.

A NEW ANALYTICAL MODEL

The model which had previously been used for the study of seal aeroelastic instability is described in reference (2). This model was extended to account for circumferential air flow and pressure variations within the seal teeth. The new model is shown in Figure 5. The equations which govern the pressure and flow in the control volume shown in the figure are as follows:

1. A continuity equation relating the net mass rate of flow into the control volume with the rate of increase of mass within the volume.
2. A circumferential momentum equation relating the time rate of change of momentum within the control volume with the net pressure force acting on the control volume.

3. A pressure-density relationship which was taken to be isentropic. This relationship introduces the velocity of sound in the circumferential direction.

Thus, for each seal cavity, there are three equations with four independent variables. These variables are the air density, pressure, circumferential velocity and the seal displacement which changes the magnitude of the control volume. These three equations can be reduced to a single differential equation relating pressure and displacement.

The method of solution employed was to assume a mechanical vibration of given amplitude, number of nodal diameters and frequency. This allows calculation of the corresponding amplitude and phase of the pressure fluctuations within the cavity. For a seal with N teeth and, hence, (N-1) cavities, the pressure amplitude and phase are unknown for each cavity, so that there are 2(N-1) simultaneous equations to be solved. A computer program
was written to allow input of any seal geometry and solve these simultaneous equations. The program then calculates the damping coefficient $C_p$ and stiffness coefficient $K_p$ in the following simplified differential equation for the seal motion.

$$M \frac{d^2 X}{dt^2} + (C_M + C_p) \frac{dX}{dt} + (K_M + K_p) X = 0$$  \hspace{1cm} (1)

Where: $M$ = seal effective mass 
$X$ = seal displacement 
$C_M$ = mechanical damping coefficient 
$C_p$ = aerodynamic damping coefficient 
$K_M$ = mechanical stiffness coefficient 
$K_p$ = aerodynamic stiffness coefficient

The final output of the program is the aerodynamic damping expressed as a fraction of the critical damping.

In the cavities between the seal teeth, the natural frequency of circumferential acoustic waves is given by:

$$\text{Acoustic Frequency} = \frac{nc}{2\pi R} \hspace{1cm} (2)$$

Where: $n$ = number of circumferential waves 
$c$ = velocity of sound 
$R$ = seal tooth cavity mean radius

For a given seal diameter, the acoustic frequency does not depend on the other mechanical details of the seal.

Typical outputs from the program are shown in Figure 6. The characteristic for a seal with low pressure side support is shown in Figure 6a. With a low mechanical frequency, the aerodynamic damping is negative. As the seal is stiffened and its frequency increased, there occurs a change to positive damping just below the acoustic natural frequency. A further increase in mechanical frequency produces a peak in positive damping which then decreases in magnitude but always remains positive even at very high mechanical frequencies. The characteristic for a seal supported on the high pressure side is shown in Figure 6b. For this type of support, the damping characteristic is reversed. The damping is positive for low mechanical frequencies and has a peak positive value just below the acoustic natural frequency. There occurs a change to negative damping just above the acoustic natural frequency and the damping remains negative even at very high mechanical frequencies.

The figures show clearly that a low pressure side supported seal can only be unstable if its mechanical natural frequency is less than the acoustic natural frequency and that a high pressure side supported seal can only be unstable if its mechanical natural frequency is greater than the acoustic natural frequency. This analytical prediction was certainly in accord with the stability testing which has been described for the engine seal. The engine seal had a high pressure side support and was unstable when tested. However, as was previously described, it became stable and could no longer be excited by its leakage flow when weights were added to reduce its natural frequency below the acoustic natural frequency.

As a result of this clear analytical prediction on stability, the test results for other seals were obtained for correlation with the prediction.

The analysis confirmed the vibration problems encountered with the engine seal, there had been a series of tests on the stability of labyrinth seals in a rig similar to that used for the engine seal shown in Figure 3. Three seals had been specially made as part of a labyrinth seal technology program to investigate the stability of labyrinth seals. These three seals could be supported either on their high pressure or low pressure sides and had been tested in the same way that the engine seal was tested. The results of these stability tests were available for correlation with the new analytical predictions about stability.

The correlation of the results for these three seals together with the results for the engine seal are shown in Figure 7. In these figures, the acoustic frequency is plotted against the mechanical frequency for each vibration mode of each seal. For each seal, the lowest frequency mode on the acoustic frequency scale is a two nodal diameter mode. The higher frequencies shown for each seal are successively the three nodal diameter mode, four nodal diameter mode, etc. The filled symbols in each figure indicate the vibration modes in which self-excited vibration due to the leakage flow occurred. A line of equal mechanical and acoustic frequency has been plotted in each figure. The results for seals with low
pressure side support are shown in Figure 7a. The only vibration modes excited were those where the mechanical frequency was less than the acoustic frequency. The lowest frequency modes were most susceptible to vibration. The results for seals with high pressure side support are shown in Figure 7b. In this case, the only vibration modes excited were those where the mechanical frequency exceeded the acoustic frequency. The lowest frequency modes which exceeded the acoustic frequency were most susceptible to vibration.

The excellent correlation of all of these test results with the analytical prediction clearly confirms the validity of the analytical model.

![Fig. 7a Low Pressure Side Support](image)

![Fig. 7b High Pressure Side Support](image)

![Fig. 7 Correlation of Test Results](image)

THE PREVENTION OF FURTHER FAILURES

It may be seen from Figure 7b that the engine seal mechanical natural frequencies become close to the acoustic natural frequencies at engine operating conditions. Referring to Figure 6b, this condition corresponds to the peak of negative damping for seals with high pressure side support.

The close proximity of the acoustic and mechanical natural frequencies during engine operation now indicated that the aerodynamic destabilizing forces were greater than had been experienced with the many seals which had previously successfully operated with damper rings. More damping than could be provided with a damper ring was required for this particular seal. A damper sleeve was designed to fit the inner labyrinth seal. This design is shown in Figure 8.

To confirm that this design would provide more damping, a forced vibration test was carried out in a rig. It was considered essential to fully simulate centrifugal effects so the rig was operated to a speed of 18,000 rpm to cover the full operating range of the engine. The seal was excited by a pulsed air jet through a range of frequencies which included the three nodal diameter mode. The seal was tested with the damper ring in place and then again with the damper sleeve in place. The seal was strain gaged to measure the stresses during each test. The results of the tests showed that the damper sleeve reduced the vibratory stress level to 40% of that obtained with the damper ring. This was considered to be completely acceptable.

The engine was qualified with the damper sleeve which has now demonstrated its effectiveness for eight years of service and half a million hours of operation without incident.
CONCLUSIONS

1. Mechanical damping devices have proven to be completely effective in preventing labyrinth seal vibration.

2. The largest aerodynamic destabilizing forces occur when the acoustic and mechanical natural frequencies are almost equal. These conditions require more effective mechanical dampers such as sleeve dampers.

3. Based on the test results shown in Figure 7, some conclusions can be made on the type of seal support which gives inherent stability. However, it should be noted that the test results were obtained for stationary seals which were specifically designed to have either high or low pressure side support. For a rotating seal, the mechanical frequency is the frequency which would be measured by an observer moving at the same tangential velocity as the air in the seal tooth cavities. Thus for each vibration mode, both the forward and backward mechanical wave frequencies must be considered. Some seals do not clearly have either type of support. An example of such a seal is shown in Reference 5, where support is provided at both ends. The conclusions which apply to a well-defined high or low pressure support as illustrated in Figure 1 do not apply to seals supported at both ends. Despite these complications, it should be clear from Figure 6 and Figure 7 that it is possible to design seals with inherent stability. A seal supported on the high pressure side can be made stable if at all operating conditions and in all vibration modes its mechanical frequency is less than the acoustic frequency. Unfortunately there are other criteria - e.g., Campbell criteria - which make it undesirable for a seal to be designed with a low natural frequency. This approach is not recommended so that seals with high pressure support should be provided with mechanical dampers or other means to provide stability. However, a seal with low pressure side support can be made inherently stable if it is stiff enough to ensure that its mechanical frequency always exceeds the acoustic frequency. The outer seal shown in Figure 1 is an example of the successful application of this principle.

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REFERENCES