Nature, Causes, and Prevention of Labyrinth Air Seal Failures

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This paper is devoted to the problems of labyrinth air seals where radially extending teeth are carried by rotating components. In view of the possibility of resonance, determination of the natural nodal frequencies of seal components and supporting structure by analysis, tests, or both is of vital importance in the design and development procedure. Construction of Campbell frequency diagrams is a basic requirement to assure adequate frequency mistuning margin against vibration and fatigue failures caused by rubs. Aerodynamic disturbing forces have also excited flexural vibration and fatigue failure of seals. A w/R stability criterion has proven effective in evaluating the hazard of aeroelastic vibration due to pressure drop across multi-tooth seals. Acoustic oscillations in the annular chamber into which the seal discharges have coupled with the corresponding natural nodal flexural vibration of seal components, which increases nodal flexural vibration of seal components, and increases the hazard of vibration and fatigue failure. Transient thermal instability may occur if the heat generated during rub causes the inner component to grow at a faster rate than outer. Seals must have compatible thermal expansions to maintain close clearances even when engine operating transients are rapid. The design and application of severed damper sleeves and ring to rotor components are effective in protecting air seals and the cylindrical walls of annular chambers aft of seals against flexural vibration and fatigue failures excited by aerodynamic disturbing forces. Since acoustically coupled oscillations persist over relatively long distances and thus involve additional components, severed damper sleeves have proven effective in protecting these thin-walled components excited by aerodynamic disturbing forces.

Introduction

LABYRINTH air seals are used extensively in turbomachinery. This paper describes structural problems and solutions of radial air seals where the teeth are carried by rotating components, a configuration much used on aircraft engines. The history of labyrinth air seal failures shows that most seal failures would have been prevented if the seal rotor and stator components had sufficient rigidity, i.e., the seal components would not distort. History further shows that the flexibility of some structures has been a primary factor in the problem. That is, if it were always possible to design seal components that did not distort, most of the known causes of labyrinth seal failures would be eliminated.

Known Causes

Causes of failure may be: a) Rub-induced flexural nodal vibration at resonance; b) Thermal instability during rub; c) Acoustically coupled nodal vibration and related aeroelastic self-excited vibration; d) Low cycle thermal fatigue; e) Incompatibility of rubbing materials resulting in material transfer between rotor and shaft components. The most common and serious hazards to seals will be discussed in turn.

Rub-Induced Flexural Nodal Vibration at Resonance

Forced resonance of labyrinth seals occurs when the angular velocity of propagation of a flexural wave in seal components is equal to the relative speed of rotation between rotor and stator components. In a simple case, resonance is present when there exists a nodal vibration pattern where the natural frequency f for a mode shape having n waves around the circumference are related to the actual speed of rotation N by $f/n = N$. The investigations of Campbell on axial nodal vibration of turbine disk wheels demonstrated the hazard of backward-traveling waves, which were amplified at resonance when the backward-traveling waves were stationary in space. This occurs when the backward speed of these waves in the disk coincides exactly with the forward angular velocity of the rotating disk so that the waves become stationary in space. Then a rub, stationary in space, or a constant circumferential variation in pressure, can feed energy into the vibrating seal component.

An example of resonance between the flexural nodal vibration of a thin-walled cylinder and speed of rotation is the stator component of the labyrinth seal at compressor discharge. The Campbell diagram constructed from laboratory measurements of stator natural nodal frequency is shown on Fig. 1. Resonance occurs at maximum engine

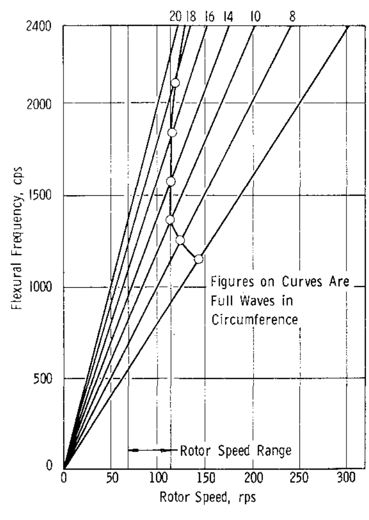

Fig. 1 Campbell frequency diagram of stator components of labyrinth air seal at compressor discharge.

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speed for nodal patterns having $n = 12$, 13, and 14 full waves in the circumference. Resonance was further confirmed by 13 equispaced flats worn on the labyrinth rotor teeth tips. The radial depth of the flats was about 1/16 in. under the original cylindrical surface. This wear pattern demonstrates that the forward-traveling flexural wave in stator was exactly equal to the rotor speed. Flexural fatigue failures in the thin walls of conical shell supporting stator caused by resonance, were eliminated by the application of slip damping discussed later.

Thermal Instability

When a stable rub occurs, the dimensional changes resulting from heat generated reduce the rub interference, and thus tend to clear the rub. Changes in seal diameter are proportional to the mass-weighted average temperature. Radial teeth of adequate radial height on the rotating seal provide thermal insulation and isolation from the high temperatures generated at the tooth tip by the rub. The design objective is to have the mass-weighted temperature of rotor component little affected by the hot spots on the tips of rotor teeth. During thermal transients on start-up, the stator should expand in diameter as rapidly as rotor.

A portion of the heat generated by the rub is conducted away from the hot tip of rotor teeth. One design objective is to have nearly all of this heat from tip transferred from sides of tooth to surrounding air by forced convection. If this is well done, the walls of cylindrical tooth support will have less change in diameter than stator. Analysis of the heat transfer processes governing insulating effectiveness of tooth shows that the radial depth of tooth and the tooth shape and thickness are both primary parameters in seal thermal stability during a rub.

Design features to minimize heat generation and distortion resulting from rub are largely based upon the principle that the amount of heat generated during a rub is a function of the amount of metal that must be rubbed or melted away. Among the means used to improve seal thermal stability during rub are:

1) Rotors of critical seals should include an integral disk support to minimize radial growth.
2) During thermal transients, the stator should expand in diameter at least as rapidly as the rotor.
3) The selection of stator rubbing material against which the labyrinth teeth rub is naturally a primary consideration. For low to intermediate temperatures, successful rub materials have included: epoxy and similar nonmetallic materials, silver braze, aluminum braze, nickel graphite for applications of moderate pressure drop.
4) For high-temperature seals, honeycomb open-cell facing on stator has proven successful.
5) Experience has shown that thermal distortion of seals is apt to be most troublesome during rubs at low to intermediate speeds. The heat generated during rubs can be substantially reduced by coating the labyrinth rotor teeth surface that rubs an abrasive cutting material such as aluminum oxide. The abrasive coat adherence to labyrinth teeth tips is improved by first coating teeth with a compatible base bonding material.

For high temperatures, honeycomb open cell stators have long been successfully used for rub surface. The success of brazed honeycomb facing on stators has made them one of the preferred constructions for labyrinth air seals operating at high temperatures. For example, honeycomb interstage seals supported from inner circular wall of turbine diaphragm have had an excellent record of mechanical reliability. The thin walls provide high thermal resistance between the exposed inner edge of honeycomb where the rub first occurs, and the cylindrical wall of the stator to which the base of honeycomb is brazed. A rub causes an almost instantaneous generation of heat, and the local temperature of the honeycomb wall at the point of contact causes a local softening and yielding or melting of the thin wall. This is a primary factor in clearing the rub.

The cellular open surface of honeycomb assures that the leakage flow is not completely cut off during a rub. As a result, a residual airflow is available to provide some cooling to the sides of the radial teeth of rotor during rub. Most experience with successful labyrinth seals has had tooth tip thickness about 0.010–0.020 in. Wall thickness of stator honeycomb material has always been substantially less than tip width of rotor teeth. This helps to assure that wear is on the stator rather than the rotor. As mentioned previously, an abrasive coating on seal teeth reduces generation of heat and resulting thermal distortion during a rub.

Protecting Labyrinth Air Seals from Self-Excited Vibration

Seals which have developed high-cycle fatigue due to vibration during engine operation have been investigated in static test rig. These seals have revealed acoustically coupled vibration phenomena, and also aeroelastic self-excited vibration. A typical example is the investigation disclosed by Armstrong in his interesting discussion in the published paper.[1] In this static rig test, aeroelastic flexural vibration of seal stator component was generated when the seal was pressurized with air, whose air temperature could be controlled. The frequency of resonant oscillation within the annular chamber into which the labyrinth seal discharged is proportional to the velocity of sound, which in turn is proportional to the square root of the absolute temperature of seal leakage air. When the natural flexural nodal frequency of the seal stator supporting the conical shell was resonant with the frequency of resonant oscillation within the annular chamber into which the labyrinth air seal discharged, a substantial increase in amplitude of the vibration occurred. Self-excited oscillations have also been observed on other seals which developed fatigue failures when the seal component was pressurized on a static rig. These investigations have clearly demonstrated that with seal components which can deflect, a mutual influence between pressure fluctuations within seal teeth, or the annular chamber into which seal discharges, can lead to instability and self-excited oscillations. Acoustic coupling and resonance with natural flexural nodal frequency of the structure much intensifies vibration.

Further investigations of seals which failed in service revealed the primary importance of location of seal support. Both experience and seal dynamic analysis show the characteristics of seal components supported on the high pressure side are distinctly different from those which are supported on the low pressure side. Air seals should be supported on discharge or low pressure side whenever possible. Examples and further discussion of the vital importance of seal support is given in Ref. 2. Figures 2 and 3 illustrate the relation of seal support to successful seals and also seals which have failed.

The annular chambers into which seals discharge usually have relatively large cross-sectional flow area, which normally equalizes any tendency for circumferential variation of static pressure. However, a very important exception is when nodal oscillatory flows are generated in the annular chamber into which the seal discharges. For example, this acoustic resonance was demonstrated by Armstrong in his generous published discussion of Ref. 1. Acoustic waves propagating in the circumferential direction may also be generated within the annular chambers

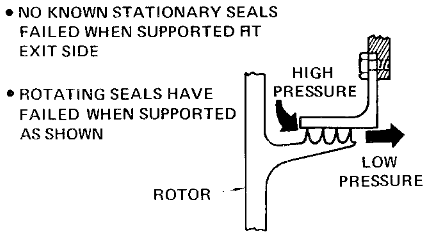

Fig. 2 Rotor seal supported at entrance side; stator supported at discharge side. Note: no known stationary seals failed; rotating seals have failed.

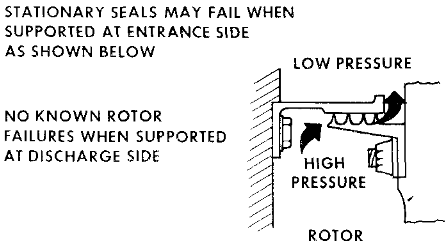

Fig. 3 Stator supported at entrance side; rotor supported at discharge side. Note: stationary seals have failed. No known rotor failures when supported as shown.

between adjacent seal teeth. Therefore, to avoid acoustic-mechanical resonance, the seal component natural flexural nodal frequencies should exceed by a margin of not less than 20% if possible the acoustic frequencies of resonant oscillation in the annular chambers between teeth and also the annular chamber into which the seal discharges. That is, to avoid the hazard of acoustically coupled vibration problems and also the related aeroelastic instability problems, the structural stiffness should be great enough that, if possible, all flexural nodal vibrations are above acoustic vibration frequencies. The air normally has some rotation, and this must be taken account of. Mechanical nodal vibration and acoustic oscillations should be of course referred to the same axis.

Damping is of course a primary factor in vibration phenomenon. The three types of damping usually of concern are: internal material damping, sometimes called hysteretic damping; relative micro slippage between mating parts, such as bolted joints and between surfaces pressed together; and damping done against external environment, such as enveloping air mass.

Materials suitable for the elevated temperatures at which many components of the new aircraft engines operate unfortunately have relatively little internal hysteretic damping. Furthermore, the trend in new engine designs is to use less bolted joints. For example, new welding techniques permit a rotor to be constructed with adjacent portion inertia welded together, with large reduction in number of bolted joints. This reduces rotor weight, but unfortunately also reduces the available damping. A convenient measure of the absence is the resonance magnification factor denoted by the symbol Q where

$$Q = \frac{2\pi(\text{maximum vibration energy stored in system})}{\text{Energy dissipated per cycle}} \tag{1}$$

For example, the magnification factor at resonance has been measured on a one-piece thin-walled cylindrical welded shaft, and the damping found so low that the resulting magnification factor Q is in the range of one to five thousand. Therefore, at resonance the thin walls can be excited to substantial vibration amplitudes by relatively small magnitude disturbing forces. A parallel trend in aircraft engine design is high cycle pressure, especially in high bypass turbofan engines. This means increasing amplitudes of aerodynamic disturbing forces acting upon some of labyrinth air seals and the annular chambers into which the seals discharge.

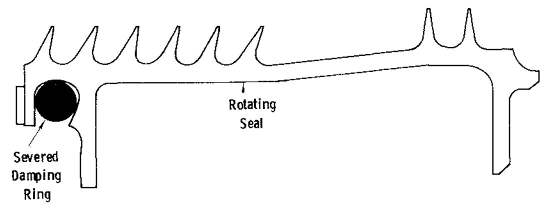

Fig. 4 Severed damping ring to protect labyrinth seals from flexural vibration and fatigue failures of the walls.

Protection of the labyrinth air seal rotor from self-excited vibration and fatigue can best be assured by providing additional damping where necessary. Severed damper rings, and particularly severed damper sleeves, have proven especially effective in protecting seal rotors, rotating thin-walled air ducts, and even thin-walled shafts from fatigue failures excited by aerodynamic disturbing forces. Figure 4 shows what is believed to be the first successful application of a severed damper ring to seals to resolve fatigue failures due to flexural nodal vibration of the thin walls of the seal rotor. In the initial design, fatigue failures originated at the tip of outer-most tooth adjacent to the free end of the cantilever seal. The fatigue failures then propagated in an axial direction due to flexural nodal vibration of the thin walls of the rotor. Analysis revealed acoustic-mechanical coupling and resonance at $n = 4$ nodal diameter mode. The severed damper ring has proven entirely successful over many years of operation in hundreds of engines. The severed ring is positively retained so it cannot fall out. The radial inward extending ring near the free end was necessary to increase stiffness to resolve a transient out-of-round distortion caused by heat generated by rub over about a 90° arc. Rubs at low speeds appeared to give the largest out-of-round seal distortion.

The annular chambers into which labyrinth seals discharge may have a rotating thin-walled tube or shaft as one of the boundaries. Rotating thin-walled air tubes and shafts must also be protected from flexural nodal vibration and fatigue failures. One possible solution would be to provide adequate frequency margin between the many natural flexural frequencies of the thin-walled cylinders and all of the sources of excitation. All the possible aerodynamic disturbing forces may be unknown during the design process. However, a condition of design is to avoid Campbell-diagram traveling wave resonances as discussed in the first section with the critical vibration patterns, especially the low-ordered modes, having $n = 1$ to $n = 6$ nodal diameters.

Severed damping sleeves and rods have proved effective in protecting labyrinth air seals on thin-walled rotating ducts and shafts from fatigue failure. Additional substantiated criteria are now needed to tell during the design phase when additional dampers are needed for rotating thin-walled ducts and shafts.

An effective way of protecting labyrinth seal rotor structures from self-excited aeroelastic vibration and fatigue failure is by the use of severed damper rings and/or severed damper sleeves. These severed dampers are installed inside the rotating cylindrical structure, and centrifugal force presses the severed ring or sleeve against the inside surface of the rotating cylinder. Vibration energy is dissipated by relative micro-slippage between the severed damper and the cylindrical wall. Micro-slippage damping reduces the vibration amplitudes to relatively low levels. That is, aerodynamic disturbing forces are inca-

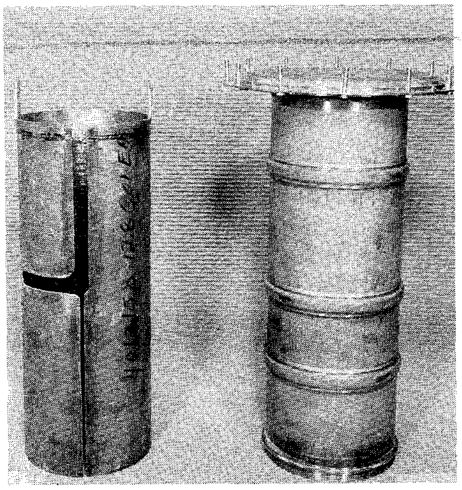

Fig. 5 View of severed damping sleeve removed from rotating air tube.

pable of producing vibratory stress levels in the thin walls of cylinders high enough to cause fatigue failure. Figure 5 shows the application of a severed sleeve damper inside of a rotating air duct.

A very desirable characteristic of the severed damper sleeve is that it provides effective damping over a wide range of frequencies. Since a thin-walled cylinder has many natural frequencies it may be difficult, if not impossible, to provide adequate frequency mistuning margin of the cylinder against high-frequency aerodynamic disturbing forces during the design process. The characteristic of providing effective slip-damping for all complex modes covering a wide range of natural frequencies is an outstanding virtue of the severed sleeve damper. In general, severed sleeve dampers are preferred over severed ring dampers for several reasons:

1) The nodal patterns of complex rotor systems are not known during the design process. Therefore, a severed damper ring may unfortunately be so located as to contact the cylinder wall at a point of negligible motion, and hence negligible damping results.

2) Although severed damper wires have shown negligible wear after many thousands of hours of use, it is still true that severed sleeves provide greater contact area.

3) Severed sleeves may be of perforated sheets, thus giving the designer more design options.

The (w/R) seal stability criterion is useful in identifying those critical seals where there may be a hazard of aerodynamic excitation and fatigue failure and the need for additional damping. This criterion is the ratio of radial deflection to the seal radius, and is defined as

$$\left(\frac{w}{R}\right) = \frac{\text{Potential Aerodynamic Disturbing Force}}{\text{Structural Stiffness of Seal Component to Resist Out-of-Round Vibrations Excited by Aerodynamic Disturbing Forces}} \tag{2}$$

For example, if the aerodynamic disturbing forces occur in length L between initial and final tooth of a labyrinth seal, the criterion is derived in Ref. 2 and is,

$$\left(\frac{w}{R}\right) = \frac{n^2}{n^2+1}\,\frac{\Delta P l g}{4\pi f^2 W_e} \tag{3}$$

To help protect against aerodynamic disturbing forces arising within the labyrinth seal itself, provide damper rings if the (w/R) exceeds about 0.4×10^{-3} in./in. of seal radius. Supporting stator seals at both ends permits the criterion (w/R) to rise to 2.0×10^{-3} in./in. before there is a hazard of aeroelastic self-excited vibration of the stator.

Table 1 Component test data on damper sleeves

No. of nodal diameters	Vibration stress without damper, kpsi double amplitude	Vibration stress with severed sleeve damper, kpsi double amplitude
2	2.6	2.4
3	32	2.0
6	9	0

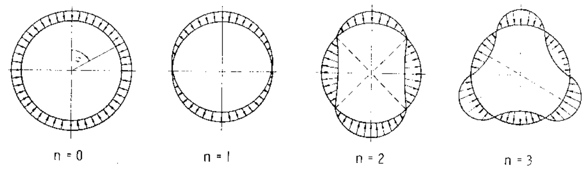

Fig. 6 Several modes of circumferential variation of static pressure differential across wall of cyclindrical shell. Note: the figure *n* denotes the number of full waves in circumference. *n* = 0 is axisymmetrical case.

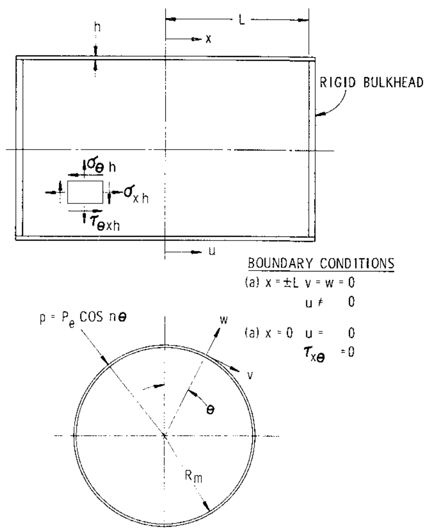

Fig. 7 Model of cylindrical membrane with end bulkheads used to analyze stresses, displacements, and slip damping.

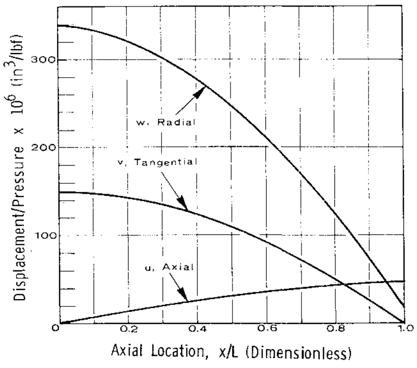

Fig. 8 Displacements of cylindrical membrane due to a static pressure differential having *n* 3 nodal diameters in circumference. Magnitude of membrane displacement for 3 nodal diameters, ½ axial wave vs axial distance for given pressure P_e = (lbf/in²); L = 6.124 in.; R_m = 6.59 in.; h = 0.80 in.; E_m = 25.93 × 10⁶ lbf in.²; α = 0.29; n = 3.

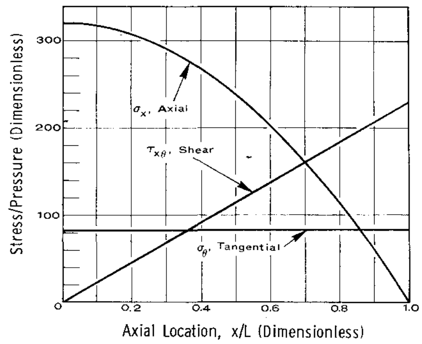

Fig. 9 Stresses of cylindrical membrane caused by a static pressure differential having n nodal diameters in circumference. Magnitude of membrane stress for 3 nodal diameters, ½ axial wave vs axial distance for given pressure P_e (lbf in.^{2}), L = 6.124 in.; R_m = 6.59 in.; h = 0.080 in.

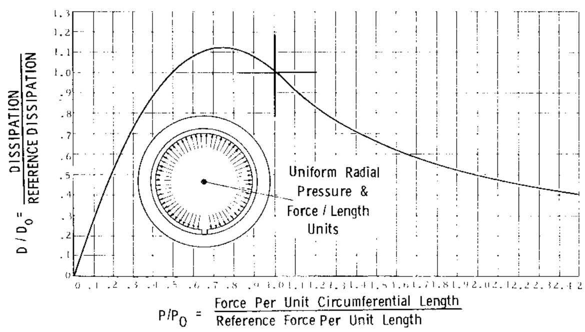

Fig. 10 Slip damping calculated for radial vibration of cylindrical shell and sleeve. Both outer shell and severed damper sleeve analyzed as extensible membrane. Radial motion for n = 0 where motion is in phase over entire circumference.

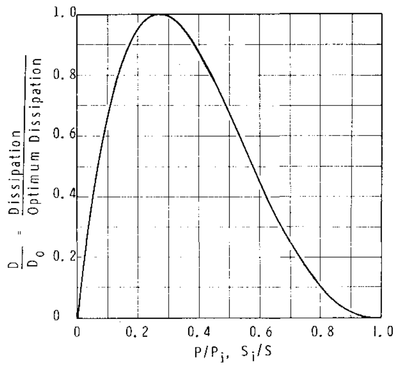

Fig. 11 Slip damping calculated for flexural nodal vibration of thin walls of cylindrical shell and sleeve. Both outer shell and damping sleeve analyzed as extensible membranes. P/P_i, S_i/S = radial force per unit circumferential length acting at interface of ring-shell combination.

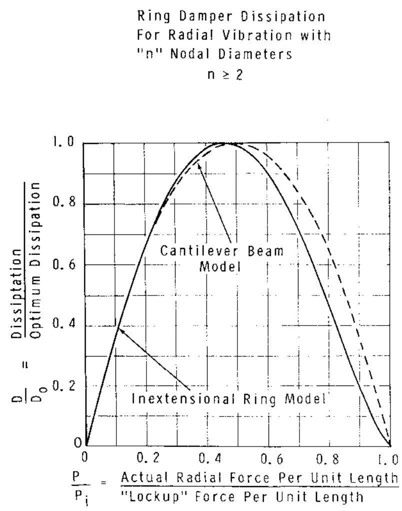

Fig. 12 Slip damping calculated for flexural nodal vibration of thin walls of cylindrical shell and sleeve. Outer cylinder assumed inextensible, and damping sleeve as extensible. Ring damper dissipation for radial vibration with n nodal diameters n = 2.

An example of the effectiveness of a severed damping sleeve installed inside a rotating thin-walled cylindrical air tube showed the amplitude of flexural nodal vibration patterns was reduced by a factor of 2 to 10 to 1. The high-frequency complex modes are particularly strongly attenuated in well-designed damper rings and damper sleeve applications.

The effectiveness of damper rings for high-frequency complex modes is illustrated in Table 1. The flexural vibrations of the walls of a cylindrical tube were excited by an interrupted siren impinging on the side of the rotating tube. The levels of excitation with severed sleeves remained the same with and without severed sleeve. The rpm was also maintained the same on component tests with and without the sleeve damper.

All higher and complex natural flexural frequencies of thin walls of tubes, say above 2000 Hz, had insignificant amplitudes when the damper sleeve was installed. These tests demonstrate that damper rings and sleeves are particularly effective for flexural vibration at the higher nodal patterns. Other experimental investigations of the effect of severed damper rings or sleeves inside a rotating thin-walled cylinder have shown that the amplitude of flexural nodal vibration patterns was reduced by a factor of 2 to 10 or greater. Analysis and test show that smaller attenuation with low-ordered flexural nodal vibration such as n = 2 to 3 modes, and much larger attenuation with higher ordered modes.

Typical circumferential variations of static pressure are shown in Fig. 6. A model of a cylindrical membrane with bulkheads is shown in Fig. 7. The displacements and membrane stresses are shown in Figs. 8 and 9, respectively. Most labyrinth air seal vibration problems involve flexural nodal vibrations which propagate in a circumferential direction. One interesting case involved flexural waves propagating in an axisymmetric mode along the slant thin wall of a conical shaft at the compressor discharge labyrinth seal. There was no wave propagation in the circumferential direction. This was proven because strain gages on the No. 2 stationary bearing support and strain gages on the rotating shaft gave the same frequency. That is, the vibration mode had zero nodal diameters. A severed damper ring, hollow in this case, was installed and was an effective solution to the problem. The results of the analysis of slip damping for waves propagating in an axial direction, but with zero nodal diameters, are shown on Fig. 10. The calculated slip damping provided by severed sleeves and rings for the more common modes having nodal flexural modes, and hence wave propagation in the circumferential direction is given in Figs. 11 and 12. Note that all cases of calculations of slip damping for flexural nodal vibration of Fig. 11 were made by assuming extensional deformation and membrane stresses and strains for both outer and inner damper sleeves. Calculation of slip damping for flexural nodal vibration of cylindrical shell and sleeves (Fig. 12) assume that the outer vibrating cylinder has inextensional deformation, while the damping ring has extensional deformation. Bending effects such as distance from the neutral axis are included.

Investigation of S-N curves of typical metals shows that a relatively modest reduction in vibratory stress gives a large, often indefinitely large, increase in cycles to failure. This is the basic reason why the dry Coulomb damping provided by several sleeves and rings has proved so successful.

References

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Philosophy, Design, and Evaluation of Soft-Mounted Engine Rotor Systems

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Gas turbine engine rotors are conventionally supported by bearings mounted on relatively stiff supports. The resulting vibratory loads and deflections can be reduced significantly by judiciously soft-mounting the bearings through squirrel cages and/or squeeze films. In addition to minimizing loads and stresses in an engine, it is important that clearances during conditions of maneuvers, thermal bow, and rotor whirl due to unbalance (even under extraordinary conditions such as loss of blades) be controlled. For high-speed rotors, it becomes necessary to support the rotors on resiliently mounted bearings to achieve vibration-free, long-life, close-clearance engines. In this paper, the design philosophy, criteria, and methods of evaluation for soft-mounted turbine engine rotor systems used in General Electric aircraft engine design are described. A major constituent of this method is a computer program for system vibration and static analysis [VAST]. This program is capable of finding natural frequencies, normalized modes, and responses due to any distribution of exciting forces considering gyroscopic and shear-deflection effects. Aircraft mounting and excitations from the helicopter rotor are also included in the computer analysis. General Electric's T700 turboshaft engine, under development for the U.S. Army, serves to illustrate the squeeze film, soft-mounting concept of design. Results from tests of the T700 engine, Advanced Technology Axial Centrifugal Compressor (ATACC), T64 turboshaft, TF34 turbofan, and other engines are summarized verifying the advantages of soft-mounted rotor systems.

Introduction

THE ultimate objective of an aircraft gas turbine engineer is to design an ultra-lightweight, low cost rotor system free from excessive vibrations throughout the operating range, so as to achieve: 1) infinite life for supporting bearings and structural components; 2) a high performance engine with very little clearance-loss taken at critical locations of the engine due to engine vibrations or due to maneuver loads imposed on the engine; 3) a design that has "built-in" vibration fixes to solve vibration problems if encountered during the development of the engine; 4) a simple design with good growth potential and excellent reliability and maintainability features; 5) an engine tolerant to abusive situations where very high unbalances are introduced for a short time.

Since the objectives are contradictory, a difficult task of balancing various requirements lies in the hands of the engineer and hence, different approaches are used to design an optimum rotor compromised to the situation under consideration. Obviously, no single way of designing a rotor for all requirements and constraints can be formulated. However, an attempt is made by the author to highlight the merits of partially or fully soft-supporting the rotors which goes a long way towards achieving the forementioned objectives.[2-5]

The T700 is a unique engine whose gas generator rotor is solely mounted on two resilient squeeze film damped bearings and its power shaft damped by two squeeze film bearings. These design features were selected as a result of extensive analysis and rig and full-scale engine tests. The T700 has excellent vibration characteristics with less than 5 g's acceleration on casings under normal operating conditions. In this engine the designer's ultimate dynamics objective has virtually been achieved. The design concept, details, analyses, and development effort that went into the soft-mounted rotor system are enumerated in detail in this paper. Experience from other aircraft engines at General Electric is also cited verifying the benefits of soft mounting.

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Index category: Structural Dynamic Analysis.

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