Current Seal Designs and Future Requirements for Turbine Engine Seals and Bearings

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Turbine engine mainshaft seals are a significant factor in the overall engine performance. Each engine application presents a different set of requirements often dictating solutions that employ one or more of the basic seal types; ring seal, face seal, or labyrinth seal. The aircraft turbine engines of the 1980's with co-rotational rotors will require seals that can operate at pressure differentials within the 345 N/cm^2 (500 psi) to 413 N/cm^2 (600 psi) range, gas temperatures within the 922°K (1200°F) to 1032°K (1400°F) range and surface velocities within the 152 m/sec (500 fps) to 183 m/sec (600 fps) range. Currently developed seal systems demonstrate limited capability for meeting advanced requirements without imposing severe performance and system penalties. Mainshaft bearings for these advanced engine applications will require operation in the 2.5 to 3.0 × 10^6 DN level range. Of the two basic bearing types employed, ball thrust and cylindrical roller, the latter presents the most challenge. Additional bearing and mainshaft seal development is required to meet the requirements of tomorrow's aircraft engines.

Introduction

TURBINE engine mainshaft seals are a significant factor in the overall engine performance. Each engine application presents a set of requirements often dictating solutions that employ one or more of the basic seal types: ring seal, face seal, or labyrinth seal. The selection of one type of seal system over another involves acceptance of certain advantages and penalties, the magnitude of which varies with the specifics of the particular engine design. The effects of seal selection on engine cycle performance, bearing/lubrication system, initial cost, weight, size, maintenance, reliability, and durability are reviewed toward providing some insight into the tradeoffs involved in the seal selection process. As gas turbine engines progress to higher rotor speeds and higher cycle pressures and temperatures, the rotor support system and sealing system problems increase accordingly. The required development of mainshaft bearings and seals to accommodate the increasingly severe environments to be found in advanced aircraft gas turbine engines will also be reviewed.

Engine Mainshaft Seal Design

The primary purpose of mainshaft bearing compartment seals is to protect the rotor support bearings and the bearing lubricant from the engine gas-path environment and to prevent lubricant leakage. Bearing compartment locations are selected on the basis of engine size, number of rotors, rotor critical speed characteristics, and rotor-stator concentricity and blade tip clearance control requirements. Gas pressure and temperature external to each bearing compartment vary with the compartment location, engine cycle characteristics, and secondary flow system design. Current production engine bearing–compartment seal systems consist of either one of three basic seal types or a combination of these seal types. The three basic seal types are ring seals, face seals, and labyrinth seals.

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Ring Seal

The ring seal is essentially a piston ring that may be either of the expanding or contracting design. The expanding design is the simpler of the two and is the one illustrated in Fig. 1. The seal rings are carbon and they seal radially against the inside diameter of the stationary cylindrical surface as well as axially against the faces of the adjacent metal seal seats. The metal seal seats are fixed to, and rotate with the shaft. The sealing closure force is provided by a combination of spring forces and gas pressures. Ring seals are employed where there is a large relative axial movement due to thermals between the shaft and the stationary structure. For high pressure sealing these seals can be used in multiples. Ring seals are limited to operation at air pressure drops and sliding speeds that are considerably lower than those allowed for face seals. However, they can be used to gas temperature levels in the same range as for positive-contact face seals which is approximately 756K (900°F). Generally, a minimum pressure differential of 1.4 N/cm^2 (2 psi) must be maintained to prevent oil leakage from the bearing compartment.

Face Seal

The positive-contact face seal was developed to provide better sealing at higher pressure differentials. A typical installation as shown in Fig. 2 utilizes a carbon nosepiece to seal in the axial direction, and separate secondary seals are used to seal radially. The carbon nosepiece is usually assembled with a shrink fit into a metal housing called a carbon carrier. The nosepiece assembly is loaded both pneumatically, and by springs, to ensure contact between

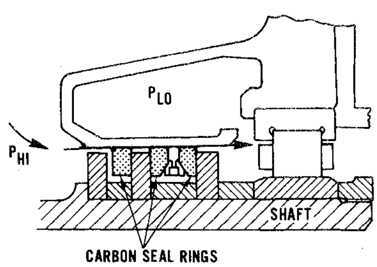

Fig. 1 Typical expanding type ring seal.

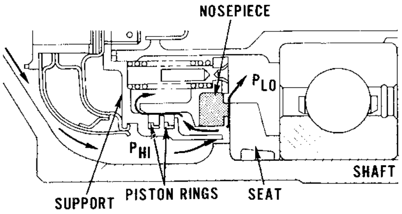

Fig. 2 Typical positive contact face seal.

the carbon nosepiece and the seal seat. Torque pins are used to prevent rotation of the nosepiece assembly. The surface of the rotating seal seat which mates against the stationary carbon nosepiece is usually chrome plated or hard face coated. The mechanical arrangements used in current face seal applications have evolved from the early designs used in the J57 engine; this development is outlined in Ref. 1. The face seal can operate at higher speeds, pressures and temperatures than ring seals since the interfacial pressures or rubbing loads can be made to approach zero by pressure balancing, and this seal design can be oil-cooled more readily.

Current practice has been to generally limit use of positive-contact face seals to conditions not exceeding a pressure differential of 69 N/cm^2 (100 psi), a gas temperature of 756K (900°F), and a sliding speed of 107 m/sec (350 fps) although engine operation at sliding speeds to 138 m/sec (450 fps) has been achieved with this type of seal at the expense of pressure differential capability. In engines with higher pressure and temperature requirements, oil wetting of the carbon nosepiece/seal seat interface has allowed reliable operation at pressure differentials to 104 N/cm^2 (150 psi), gas temperatures to 894K (1150°F), at sliding speeds in excess of 122 m/sec (400 fps). Generally a minimum pressure differential of 2.0 N/cm^2 (3 psi) must be maintained for both types of face seals to prevent oil leakage from the bearing compartment.

Labyrinth Seal

The labyrinth seal is utilized frequently in a multiple seal configuration with pressurization and/or venting between seal stages. Early gas turbine engines employed simple nonbuffered and nonvented systems because cycle gas pressures and temperatures were low with little corresponding bearing compartment fire hazard. The more advanced engines in current use operate at high pressures and temperatures and a simple labyrinth seal system would permit excessive leakage of high temperature air which could ignite the engine lubricant. Consequently, the labyrinth seal system has evolved into a stepped, three stage system as shown in Fig. 3. Three stages and an extensive plumbing system are required to decrease the temperature of the air leakage into the bearing compartment to satisfactory levels. The high pressure, high temperature air must not be permitted to enter the compartment directly so it is vented overboard between the first and second stages. Relatively low temperature pressurized air must be used between the second and third stages to provide a positive pressure drop across the third stage so that the leakage flow into the bearing compartment is relatively cool air, thereby reducing the possibility of a bearing compartment fire. Minimization of seal clearance is of prime importance for obtaining maximum labyrinth seal effectiveness since leakage is approximately proportional to clearance. Adequate clearance to guarantee structural integrity of the seal must be established, taking into consideration dynamic growth, anticipated thermal expansion, shaft motion, tolerance buildup, and runouts. The behavior of labyrinth seals is well documented and the leakages can generally be predicted within 10%. The general leakage prediction system is described in Ref. 2.

Simple, single stage labyrinth seal systems have been used at pressure differentials up to 34 N/cm^2 (50 psi) and gas temperatures to 589K (600°F). Multistage labyrinth seal systems have been used at pressure differentials up to 280 N/cm^2 (400 psi) and gas temperatures of approximately 922K (1200°F). Rotational speed is not a limitation for this type of seal design. Generally a minimum pressure differential of 2.0 N/cm^2 (3 psi) must be maintained to prevent oil leakage from the bearing compartment.

Buffered Face Seal

In some instances labyrinth seals are used as "buffer" seals in series with a face seal. Used in conjunction with a pressurizing and/or venting system the labyrinth seals serve to maintain acceptable pressure and temperature levels at a face type seal. One of the possible combinations would be to replace the third stage of multistage labyrinth seal shown in Fig. 3 with a face type seal.

Considerations In Seal Type Selection

Ring seals, face seals, and labyrinth seals are all widely used for gas turbine engine mainshaft sealing applications since no specific type is the optimum choice for all designs. The seal design selected must, of course, be capable of controlling air leakage to acceptable levels in the expected engine environment for the length of time it is required to operate between inspection or overhaul. However, the seal design selected affects the engine cycle performance, bearing/lubrication system, initial cost, weight, size, maintenance, reliability, and durability. The selection of one type of seal over another involves the acceptance of certain advantages and penalties. The possibility also exists that the penalties associated with the alternative seal designs or the remaining seal design when there are no alternative designs may be high enough to force either the development of a new type of seal or a new engine design configuration.

Engine Cycle Performance

The advantage of ring and face seals over labyrinth seals for air leakage control is well documented. Labyrinth seals generally have leakage rates approximately 10 times that of ring and face seals because of the relatively large clearances that must be maintained between the sealing surfaces. The total system air leakage of multistage labyrinth with a buffer air supply and overboard vent can be 100 times greater than a face seal capable of sealing the same conditions.

High pressure air that leaks past the seals into the bearing compartment or is vented overboard penalizes the engine cycle performance. The magnitude of the performance penalty varies with the particular cycle and engine configuration. A comparison of three alternate bearing compartment seal systems for application to the high rotor rear bearing compartment in a two-spool four-bear-

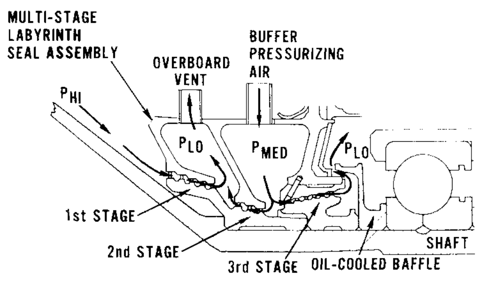

Fig. 3 Typical multi-stage labyrinth seal.

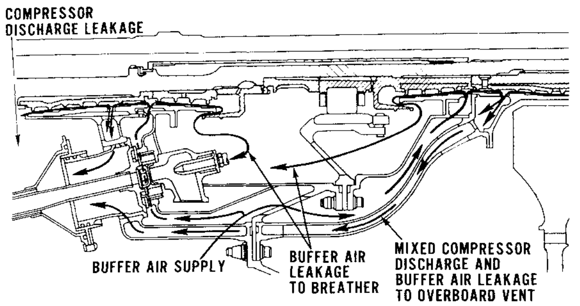

Fig. 4 Transport engine bearing compartment with multistage labyrinth seals.

ing transport engine illustrates the effect that seal air leakage can have on engine cycle performance.

The bearing compartment flow system using multistage labyrinth seals is shown in Fig. 4. This system uses a separate buffer air supply to reduce the temperature of the air leakage into the bearing compartment. A carbon face seal arrangement using labyrinth seals as buffers to reduce the pressure differentials across the face seal is illustrated in Fig. 5. Use of a carbon face seal arrangement exposed directly to the environmental conditions of compressor discharge pressure and temperature is shown in Fig. 6.

Although this bearing compartment location imposes the severe operating environment of compressor discharge pressure and temperature on the bearing compartment seal system, this bearing location is needed for adequate control of rotor dynamics and rotor-to-stator alignment. In a large transport engine with a 25/1 pressure ratio these sealing conditions include approximate maximum levels of seal pressure differential to 280 N/cm^2 (400 psi), gas temperatures to 922K (1200°F), and sliding speeds to 116 m/sec (380 fps). Table 1 summarizes losses in terms of total engine flow and the effect on fuel economy for the three bearing compartment seal systems. The projected labyrinth seal system leakage of 0.88% of total engine flow is charged to engine cycle performance. Recovery of this air for turbine cooling or other cycle uses may not be possible because of the low pressure at which this air leaves the bearing compartment. Breather flow and overboard vent flow totaling 0.88% of the total engine flow result in an engine thrust specific fuel consumption (TSFC) penalty.

As compared with the all labyrinth seal systems, the buffered face seal configuration bleeds less leakage air flow from the compressor and discharges this air from the buffering compartment at a sufficiently high pressure for recovery in the low-pressure turbine. The resulting TSFC gain is 0.6%. The operating parameters for the face seal are a pressure drop of approximately 69 N/cm^2 (100 psi), a gas temperature of 866K (1100°F), and a sliding speed of 116 m/sec (380 fps). As can be seen in Table 1, the total leakage across a face seal would be only about 0.002% of the total engine flow, resulting in a TSFC gain of 0.9%. This face seal would be required to operate at a maximum pressure drop of approximately 280 N/cm^2 (400 psi), a gas temperature of 922K (1200°F), and a sliding speed of 116 m/sec (380 fps).

The preceding example illustrates the beneficial effects on engine cycle performance, as measured by TSFC, through improved bearing compartment sealing for a large transport engine. It must be recognized, however, that the effect of seal system air leakage on engine cycle performance is different for each engine cycle.

Bearing/Lubrication System

The performance of the bearing/lubrication system is a factor to be considered in the selection of mainshaft bearing compartment seals. Increased seal air leakage breather flow reduces the deoiler efficiency and this causes an increase in engine oil consumption. Breather flow in the typical medium size transport engine which utilizes labyrinth seals extensively is 82×10^{-3} scms (175 scfm) at sea level take-off as compared to 14×10^{-3} scms (30 scfm) for a comparable carbon-sealed engine. Oil consumption for labyrinth-sealed engines is approximately 0.38 to 0.76 liters/hr (0.10–0.20 gal/hr) as compared to 0.13 liters/hr (0.035 gal/hr) for a typical carbon-sealed engine.

Increased seal air leakage also carries more contaminant into the bearing compartment which is ultimately entrained by the lubricant. Dirt damage to bearings and lubrication system components is a much more predominant problem with high seal air leakage labyrinth-sealed engines compared to low seal air leakage carbon-sealed engines. Oil contamination levels from 2 to 10 mgrams/100 mliters are common for labyrinth-sealed engines, while the typical carbon-sealed engine has a contamination level of approximately 1.5 mgrams/100 mliters. Filtered seal pressurization air and/or finer oil filtration[3] have been incorporated in labyrinth-sealed engines to reduce the effects of the air entrained contaminant on the bearings and lubrication system components in these engines. The heat generation rates and oil flow requirements for ring seals, face seals and labyrinth seals are about equal. In the labyrinth seal no heat is generated by the seal itself but the hot air leakage past the seals and the adjacent walls must be cooled to prevent lubricant coking in and around the bearing compartment. With ring seals and face seals, hot-air leakage is not a significant problem, but cooling is required to remove the heat generated at the rubbing carbon nosepiece/seal seat faces.

Labyrinth seals do not normally come in direct contact with the lubricant, and, therefore lubricant compatibility with the labyrinth seal materials is not a problem. This is not the case with carbon ring and face seals; therefore, new lubricants must be tested to ensure that they do not interact adversely with seal materials before they can be used in gas turbine engines. This is done in tests of individual components and finally in designated experimental engines. Only after successful completion of these tests are new lubricants introduced for service use.

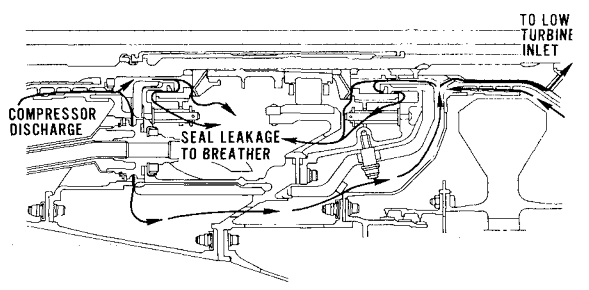

Fig. 5 Transport engine bearing compartment with buffered face seals.

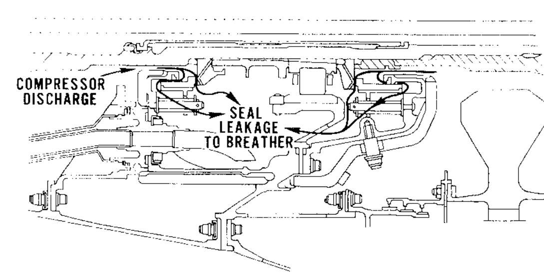

Fig. 6 Transport engine bearing compartment with face seals.

Table 1 Comparison of leakage losses for high-pressure rotor rear-bearing compartment seals in transport engine

	Type of seal Labyrinth	Buffered face seal	Face seal
Running clearance at sea-level takeoff μm (in.)	254–381 (0.010–0.015)	12.7 (<0.0005)	12.7 (<0.0005)
Total compressor air leakage at sea-level takeoff (% of total engine flow)	0.88	0.600	0.002
Leakage flow distribution (% of total engine flow)	breather—0.11 overboard vent—0.77	breather—0.002 buffer loss—0.598	breather—0.002
Projected net thrust specific fuel consumption improvement considering labyrinth seal system as base (%)	base	0.6	0.9

Initial Cost, Weight, and Size

At low pressures and temperatures the labyrinth seal is less costly and comparable in size and weight to carbon ring and face seal installations. However, once certain limitations on pressure and temperature are reached, the simple lightweight labyrinth seal designs give way to the multi-stage labyrinth seal design requiring pressure bleed-off and low temperature air pressurization with air filtration hardware. The multi-stage labyrinth seal design is more costly, heavier and requires a larger envelope than a comparable face seal installation.

A comparison of the seals used for a carbon-sealed engine and a labyrinth sealed engine, illustrates the initial cost and weight tradeoff between the use of carbon ring and face seals vs labyrinth seals. The engines are two-spool multi-bearing rotor support designs. Table 2 and Table 3 illustrate the relative cost and weight of the seals for these engines with the carbon-sealed engine No. 1 seal assigned the base value of 1.0. The carbon-sealed engine No. 1 seal is a carbon ring seal, the No. 2, No. 3, and No. 4 seals are carbon face seals. The labyrinth-sealed engine No. 1 and No. 2 seals are simple two stage labyrinth seals with pressurization between stages and pressure bleed off between stages respectively, while the No. 3 and No. 4 seals are three stage labyrinth seals with pressure bleed off, buffer air pressurization, and air filtration plumbing. The No. 4½, and No. 6 seals are ring seals in both engines; the No. 5 seal is a face seal in both engines. Hence, the cost and weight for the No. 4½, No. 5, and No. 6 seals are approximately the same for both engines.

Referring to Table 2, the labyrinth-sealed engine No. 1 and No. 2 seals cost less than the carbon-sealed engine No. 1 and No. 2 seals because of the simplicity of their design. The labyrinth-sealed engine No. 3 and No. 4 seals cost more than the carbon-sealed engine No. 3 and No. 4 seals because of the complexity of their design. The hardware associated with pressurizing and venting the labyrinth-sealed engine No. 3 and No. 4 seals plus air filtering equipment further increases the cost. The overall labyrinth seal system initial cost is greater than twice that of the carbon seal system. Referring to Table 3, the labyrinth seals weigh more than the carbon seals at each comparable location; the weight differential is especially high at the No. 3 and No. 4 seal positions. The overall labyrinth seal system weight is greater than twice that of the carbon seal system.

Maintenance

Carbon ring seals are particularly sensitive to breakage during handling. Carbon face seals are installed as a complete metal backed assembly; thus, carbon breakage with face seals is less of a problem. With labyrinth seals, the sharp knife edges can be easily damaged during installation or removal from an engine. Carbon ring seals are replaced if worn beyond overhaul limits. Carbon face seal assemblies can be repaired at overhaul, if required, through replacement of the carbon nosepiece and resurfacing and repair of the worn areas (i.e., piston ring bore, torque pin slots). Repair procedures are also available for worn or damaged labyrinth seal knife edges and lands if they are within overhaul limits.

Reliability and Durability

The primary advantage of the labyrinth seal is that it has an infinite life if neither major interferences nor erosion occur. Carbon ring and face seals necessarily rub and wear. The carbon seal ring and nosepiece wear rate, which limits the service life of these seals, is a complex function of unit contact force, surface speed, material properties, and thermal environment.[4] The analytical model used for these seals is quite empirical due to the difficulty in determining the precise relationship between the factors affecting carbon ring and nosepiece wear. Service experiences indicate that long life can be achieved with ring seals and face seals. Ring seals and face seals, where used at moderate conditions, in current commercial engines such as the JT3D, JT4, and JT8D currently meet time between overhaul (TBO) periods in excess of 10,000 hr. Face seals for military engines which operate at more severe conditions meet TBO periods up to 1,000 hr or greater. Carbon nosepiece wear rates for both ring and face seal

Table 2 Relative cost of carbon seal system vs labyrinth seal system

	carbon seal system	labyrinth seal system
No. 1 seal	1.0	0.5
No. 2 seal	1.3	1.1
No. 3 seal	1.2	2.7
No. 4 seal	1.9	3.0
hardware required for labyrinth seal system pressurization, venting, air filtration	...	3.8
Total	5.4	11.1

Table 3 Relative weight of carbon seal system vs labyrinth seal system

	carbon seal system	labyrinth seal system
No. 1 seal	1.0	1.3
No. 2 seal	0.4	0.7
No. 3 seal	0.3	1.2
No. 4 seal	0.8	2.2
hardware required for labyrinth seal system pressurization, venting, air filtration	...	1.3
Total	2.5	6.7

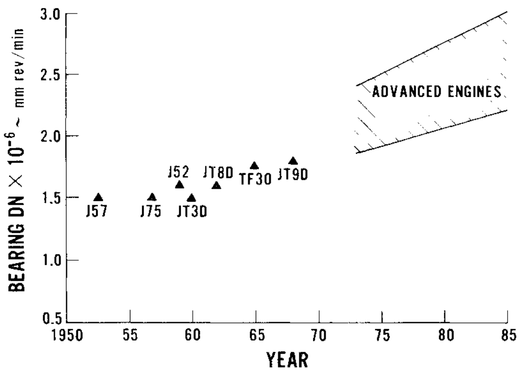

Fig. 7 Trend toward increased *DN* values for mainshaft bearings.

applications must generally be below 25 μ/100 hr (1 mil/100 hr) for military applications and 25 μ/1000 hr (1 mil/1000 hr) for commercial applications. However, because carbon seals are rubbing contact devices, their performance can be influenced by factors such as accumulation of coked oil deposits, anti-rotation pin and slot wear, and piston ring secondary seal fretting. Most premature engine removals attributed to ring and face seals are caused by these secondary factors which may cause the seal to leak excessive amounts of air or have excessive nosepiece wear.

Seal System Selection

In any engine design, it must be realized that one type of sealing system may not be appropriate for all or any bearing compartment locations. Many design analysis iterations are performed to select the most suitable sealing system for each bearing compartment environment allowing the least compromise to the engine design objectives.

The primary advantage of the carbon ring and face seals over labyrinth seals is that of leakage control. The labyrinth seal, however, provides the potential of extremely long life with maintenance free operation unlike carbon ring and face seals. In engines with low pressure differentials the labyrinth seal is simpler than carbon face and ring seals and provides adequate sealing. However, once certain limitations on pressure differential and temperature are reached, three-stage labyrinth seal designs are required. The seal system becomes costly, heavy, and bulky with undesirably high air leakage. The high air leakage results in engine cycle performance penalties and decreased bearing/lubrication system performance. Face seals in particular are effective in regions of high pressure differential although currently available designs cannot seal against maximum cycle pressures of more recent engine designs without the use of a labyrinth buffer seal. Thus, each application must be considered on its own merits taking into consideration the current state-of-the-art.

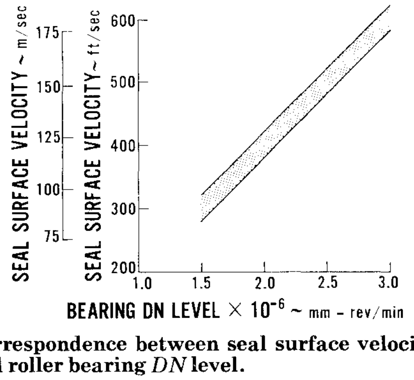

Fig. 8 Correspondence between seal surface velocity and ball and roller bearing *DN* level.

Current production models of a transport engine use the multi-stage labyrinth seal system for the high rotor rear bearing compartment. Tests during early phases of this engine development on carbon face seals indicated that the face seal configuration, similar to Fig. 6, was beyond the state-of-the-art and that additional development was required for the buffered-face seal configuration, similar to Fig. 5, to meet the durability requirements. Hence, the labyrinth seal system, similar to Fig. 4, despite its inherent engine performance and system penalty, was selected. Development testing on the buffered face seal configuration was continued. Improvements were made in the design configuration, carbon and piston ring materials, and cooling methods. The buffered face seal configuration is now the prime design for advanced models of this engine. The revised seal design is expected to meet a 10,000 hr TBO life. The conditions encountered with the unbuffered configuration (compressor discharge pressure and temperature) are still outside the range of currently developed face seal technology for a 10,000 hr life seal.

Bearing and Seal Requirements for Future Engines

Future bearing and seal requirements for gas turbines are expected to be more severe than for current applications. Advanced engines for both commercial and military applications are expected to continue the trends toward lower fuel consumption and higher thrust-to-weight ratios. The trend toward lower fuel consumption implies higher pressure ratios, higher turbine inlet temperatures, and higher bypass ratios.[5] The higher thrust-to-weight ratios are expected to place greater demands on engine structural systems. It will be difficult to develop structural systems for large transport engines which will allow the flow area needed for higher thrust without a corresponding increase in weight. "Piggyback" bearing systems† for the support of the turbine end of the rotor, while attractive for possible thrust-to-weight gains, increase certain bearing and seal requirements.

The achievement of higher levels of rotor system performance in advanced engines will result in higher rotor speeds. Increased rotor speeds, greater rotor torque capacity, and critical speed margin demands will result in higher mainshaft bearing *DN*‡ levels and seal surface velocity levels. The trend toward increased bearing *DN* values is shown in Fig. 7. The bearing *DN* levels of 2.5 to 3.0 × 10^6 *DN* and consequently seal surface velocity levels of 152 m/sec (500 fps) to 183 m/sec (600 fps), (Fig. 8), correspond to engines that will be in operation in the 1980's or are under various stages of development. The relationship between seal surface velocity and bearing *DN* level is linear as is to be expected since the seal diameter is proportional to the slightly smaller adjacent bearing bore size. Advanced engine designs with counter-rotating rotors may require intershaft bearings with *DN* values of 3.0 to 4.0 × 10^6 *DN* and intershaft seals with surface velocities between 183 m/sec (600 fps) and 305 m/sec (1000 fps). The future bearing and seal requirements discussed in this paper will be limited to those required for co-rotational rotor gas turbine engine designs and the nonintershaft bearings and seals of counter-rotating rotor designs.

Increases in gas turbine pressure ratios produce corresponding increases in compressor discharge pressures and

† "Piggyback" bearing system is a dual rotor support arrangement whereby one of the rotors is supported by the other through a bearing rather than from the engine case structure as in conventional dual rotor support arrangements.

‡*DN* values are a measure of the severity of a bearing application where: *D* is bearing bore in millimeters and, *N* is shaft speed in rpm.

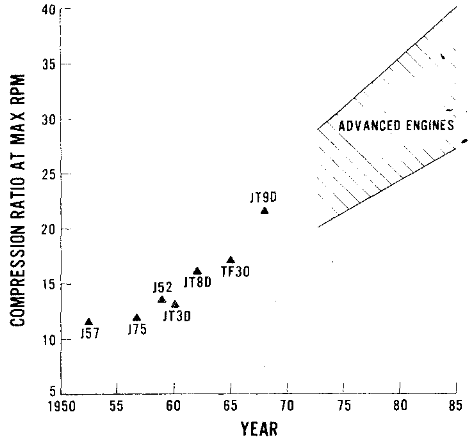

Fig. 9 Trend toward increased compression ratios for aircraft gas turbine engines.

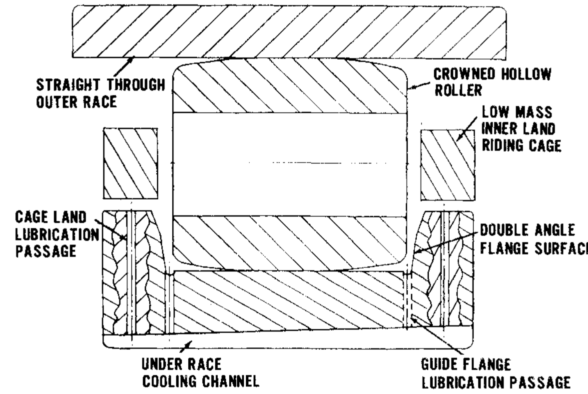

Fig. 10 Typical candidate design features for high speed roller bearings.

temperatures. The gas turbine pressure ratio trend is shown in Fig. 9. To avoid undue performance losses, particularly at the engine bearing compartment between the high compressor and turbine, low leakage seals must be developed to operate at pressures approaching high compressor discharge static pressures. Aircraft gas turbine engines in the 1980's will require seals capable of operation at pressure differentials within the 345 N/cm^2 (500 psi) to 413 N/cm^2 (600 psi) range, at gas temperatures within the 922K (1200°F) to 1032K (1400°F) range, in addition to the previously mentioned surface velocities within the 152 m/sec (500 fps) to 183 m/sec (600 fps) range.

Bearings for Future Engines

Extensive analysis and development testing is required to establish the rolling-contact bearings needed for operation at speeds in the 2.5×10^6 *DN* to 3.0×10^6 *DN* range expected in advanced engines for the desired B_{10} life.§ The selection of proper internal bearing geometry, removal of increased heat generated, design of cages with increased strength, reduced weight, and reduced cage friction, coping with the reduced load range, development of fatigue resistant materials, and proper lubrication of rolling and sliding contacts are all factors to be considered in the design of high-speed rolling contact bearings. Of the two basic types, ball thrust and cylindrical roller, the ball bearing has received the most attention. The development of high speed roller bearings has not kept pace with the development of ball bearings.

A parametric study on 120-mm bore angular-contact ball bearings has indicated that long term operation at 3.0×10^6 *DN* is possible. Testing performed under this program has indicated that operation at 3.0×10^6 *DN* can be achieved with a high degree of reliability using sophisticated but currently available state-of-the-art bearing designs, materials, lubricants, and lubrication techniques.[6] However, no comparable results for roller bearings are available.

Utilization of an approach for roller bearings similar to that used on the angular-contact ball bearings should not be ruled out. Experience with high speed roller bearing testing has revealed that most failures are related to design as opposed to fatigue limitations. Surface distress rather than subsurface fatigue is often the cause of roller bearing failure under high speed conditions. The design of a successful 3.0×10^6 *DN* roller bearing requires that many factors which influence bearing operation be carefully adjusted until all operating characteristics are suited to the application. The qualities, aside from rolling contact fatigue resistance, which provide a measure of the operating characteristics of a high speed roller bearing are; skid resistance, roller tracking control, wear resistance of sliding contacts, cage dynamics, operating temperatures and clearances, and heat generation.

A conventional design utilizing advanced features is shown in Fig. 10. Hollow rollers are incorporated as a means of reducing the outer race loading due to roller centrifugal loads. These rollers may also be used to preload the bearing. The inner-fiber stress level in a hollow roller however, must be kept low to prevent bending fatigue initiating at the bore. This design utilizes under race cooling and lubrication of roller guide flanges and cage guide surfaces and as Fig 13 indicates, the guide flange contour should be carefully selected. The roller end clearance must also be closely controlled. Cage pocket and guide surface clearances should be set for optimum lubrication and dynamic characteristics.

Pending development of a successful roller bearing design, the introduction of new concepts such as; the dual diameter roller bearing, series hybrid bearing, and tapered roller bearing may be required to provide a durable means of radial support for high speed applications. The dual diameter roller concept, shown in Fig. 11, was evaluated during limited tests at speeds to 3.5×10^6 *DN*.[7] This configuration reduces the cage operating speed. The series hybrid bearing is a relatively new concept for improving the fatigue life of a rolling contact bearing. This concept couples a rolling contact bearing in series with a fluid-film bearing so as to reduce the rotational speed of the rolling contact bearing. Application of this design concept to roller bearings would result in a design similar to Fig. 12. Both the roller and fluid film bearing carry the full radial load; however, the inner race of the roller bearing will operate at some intermediate speed with respect to the shaft resulting in a lower effective *DN* operating range for the roller bearing. Tapered roller bearings are being developed for speeds to 3.5×10^6 *DN*.[8] In development testing of a

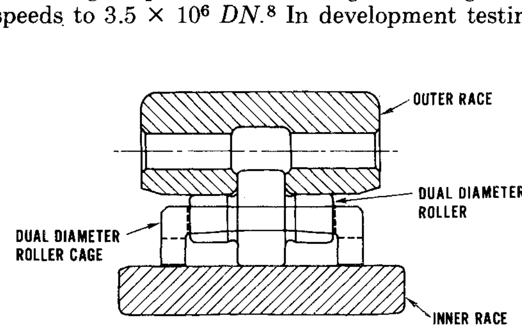

Fig. 11 Dual diameter roller bearing configuration.

§B_{10} life-number of hours by which 10% of a group of similar bearings will have failed.

88.9 mm bore and 125.8 mm bore bearing to 1.4×10^6 *DN* and 1.8×10^6 *DN*, respectively, tapered roller bearings did not exhibit any detectable slip. The prime problem encountered was that of maintaining a lubricant supply to the cone thrust rib. Provisions for a second source of lubricant supply, Fig. 13, to the cone thrust rib alleviated the problem.[9] However, investigation of each of the alternative rotor support concepts will require development time and financing to reach current roller bearing operating levels.

The lightweight rolling element is one approach to higher bearing speeds by reducing the mass of the rolling elements. This reduces the centrifugal load which these elements apply to the bearing outer race and may be expected to lead to increased bearing fatigue life at high speeds. Experimental investigation of hollow balls[10] and drilled balls[11] has demonstrated operational capability at 3.0×10^6 *DN;* however, they have not demonstrated the hoped for improvement over conventional solid balls. The high tensile stresses developed on the inside wall and consistent "tracking" of the hollow and drilled balls is believed to have shortened their fatigue life. Utilization of solid materials such as silicon nitride for bearings appears to be more promising.[12] Silicon nitride has a density which is 60% lower than that of steel, and this is a greater weight reduction than hollow steel elements can provide.

With the development of the piggyback rotor support arrangement, intershaft roller bearings are being used for the rotor radial support function. Use of *DN* level, calculated using the difference in rotor speeds for the speed factor, does not give a valid representation of the complete bearing operating level. The *DN* value so calculated indicates the rolling element *DN* level; however, the rolling element centrifugal loads on the outer race, the bearing cage centrifugal loads on the guide surfaces, and cage speed are that of a considerably higher *DN* bearing. Intershaft bearings with cage speeds typical of conventional bearings operating at 2.5×10^6 *DN* are currently being used with anticipated requirements of 3.0 to 3.5×10^6 *DN*. Additional development and analysis of intershaft bearings is required.

Seals for Future Engines

Considerable effort is required to develop the seals with the capability to operate at pressure differentials within the 345 N/cm² (500 psi) to 413 N/cm² (600 psi) range, gas temperatures within the 922°K (1200°F) to 1032°K (1400°F) range, and surface velocities with the 152 m/sec (500 fps) to 183 m/sec (600 fps) range for the desired life. Current technology seals will impose severe engine design restraints by their continued usage. The pressure/temperature/speed conditions for advanced engines appear to be outside the range of current technology ring and face seal designs. However, exclusive use of labyrinth seals will result in increased penalties on engine cycle performance, weight, cost, and size. Further increase in labyrinth seal capabilities is likely to be incremental. These changes are likely to come from the use of advanced materials and the application of design and fabrication techniques which enhance seal temperature capabilities and minimize dimensional changes and deflections under operating conditions.

Development testing of currently used positive contact face seals has also demonstrated a limited growth potential.[13] The operating range and service life of positive-contact face seals will continue to be incrementally extended through seal nosepiece and mating ring materials development as well as through the use of improved cooling techniques. However, future use of rubbing contact face seals will be restricted to seal locations which do not depart significantly from current operating limits.

The "oil-wetted" face seal has a wider operating range than the conventional positive contact seal. However, at extremely high sliding speeds, the oil film heat generation leads to distortion of the seal. Also, high air pressure differentials across the seal tend to disturb oil distribution in the seal interface and alter the axial force balance of the seal. Additional experimentation and analysis are needed to develop a design that is distortion controlled to the extent of permitting long life and low leakage operation at the speeds and pressure differentials expected in advanced engines.

Another type of seal that holds much promise for the future is the gas film face seal. The gas film lubricated face seal has the low air leakage feature of the face seal and the non-contacting feature of the labyrinth seal. Seal heat generation is low and the gas film load capacity and stiffness may be matched to the seal operating conditions through design of the seal interface according to techniques developed for air bearing applications. Extensive analytical and experimental work has been done on gas-film seal development for mainshaft sealing applications.[14-15] An example of a gas film seal typical of those evaluated is illustrated in Fig. 14; the self acting geometry is illustrated in Fig. 15. Seal performance without rubbing contact has been demonstrated to a maximum pressure differential of 345 N/cm² (500 psi), to a maximum sliding speed of 183 m/sec (600 fps), and to a maximum gas temperature of 922K (1200°F).[16]

Considering its potential impact on overall engine performance and its demonstrated capabilities in rig testing, the gas-film seal is a very good candidate as a seal for advanced engines where harsh sealing conditions exist. Although special manufacturing techniques, tolerance control, exotic materials, and handling requirements will add to the cost, it is anticipated that these factors will be far outweighed by the gains in fuel savings due to improved engine performance. Application of gas-film seal technology to piston ring type and segmented contracting ring type seals also shows promise of extending the surface ve-

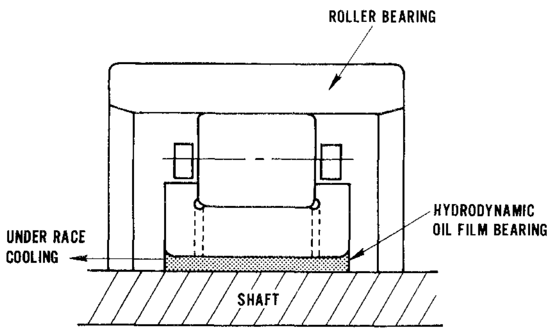

Fig. 12 Series hybrid bearing.

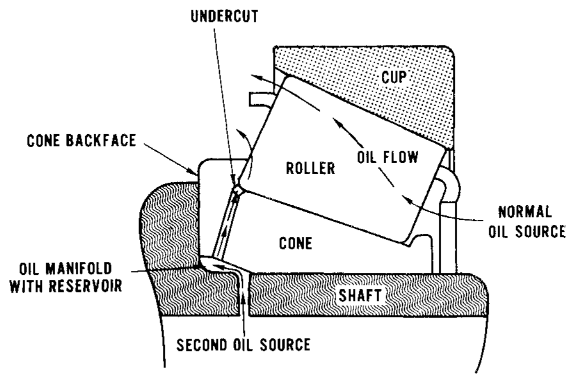

Fig. 13 High speed tapered roller bearing.

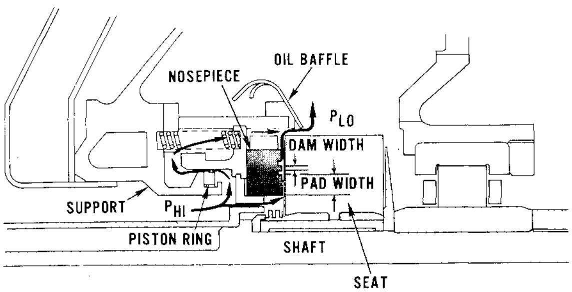

Fig. 14 Typical gas-film face seal with self-acting Rayleigh step pads.

locity range of ring seals. As rotor speeds and compartment pressure differentials increase and rotor support systems such as the "piggyback" arrangement are developed, new and improved ring seal designs will be required.

Summary

The development of bearings and mainshaft seals to accommodate the increasingly severe environments found in advanced aircraft gas turbine engines will require extensive effort. Each new engine design will have specific requirements for mainshaft bearings and seals, often dictating unique solutions with significant deviations from past experience. Providing a means of rotor support, particularly the high-speed rotor, for future aircraft gas turbine engines is becoming increasingly difficult. The development of bearings for rotor support systems capable of operation under the requirements of advanced engines with acceptable B_{10} lives will require extensive development. Failure to increase the capabilities of roller bearings will either limit future turbine speeds or cause the adoption of radically new rotor support systems. Considering the favorable impact of low leakage face and ring type seals on overall engine performance, their incorporation is generally preferred. However, the pressure drop, temperature, and surface velocity capabilities of these seal types have not kept pace with the growth of engine pressures, temperatures, and rotor speeds. Exclusive use of labyrinth type seal systems for future engines will impose severe performance and system penalties. Therefore, continued development of improved ring and face type seals is a prerequisite for acceptable bearing compartment sealing in future aircraft turbine engines.

Initial development of these bearings and seals will of necessity be exploratory toward the expected ranges listed in the various parameter areas of *DN*, fps, etc. The initial testing will be used to establish and advance the analytical model design systems. As more specific requirements are established, the bearings and seals which differ significantly from the development model will require further demonstrator engine or rig testing with additional refinements to the analytical models previously established. Development of bearings and seals must be done well in advance of the establishment of specific engine requirements to permit sufficient lead time to advance the state of the art and to permit integration of new bearing and seal designs into the final advanced engine configurations.

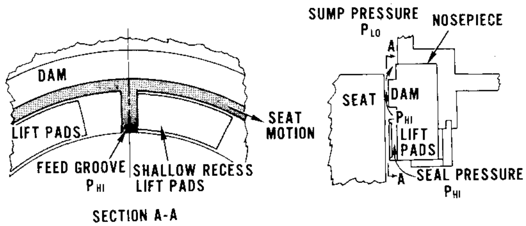

Fig. 15 Face geometry for gas film face seal with self-acting Rayleigh step pads.

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