Rotating/Stationary Interface

The papers in this section are from a session with the same title chaired by William J. Anderson.

Design and Development of Low-Cost, Self-Contained Bearing Lubrication Systems for Turbine Engines

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This paper discusses methods to simplify bearing lubrication systems for gas turbine engines. The approaches presented are based on experience in the development of a small (660 lb thrust—41,000 rpm) turbojet engine for a missile application. During this development, the ability to operate under a very hostile environment was demonstrated; the environment included accelerating from standstill to full speed in the order of 12 sec after cold soaking in a −60°F environment, accelerating to full speed in less than half that time after hot soaking, running at temperatures experienced at speeds in excess of Mach 1, and operating at very high thrust loads (20 mm bearing at an axial load of 1200 lb). The final thrust bearing configuration involved a self-contained pot lube system, but a grease-packed thrust bearing system was also successfully developed for a slightly less stringent set of operating conditions. The development of the companion grease-packed roller bearing, which operates in the hot (over 600°F) turbine end environment, is also discussed. The paper discusses the design and development aspects of the problem, including such considerations as heat rejection, volume constraints, lubricant storage, and feed to meet the demanding environmental extremes, and maintaining design simplicity in the interest of cost.

Introduction

THE growing importance of jet engine propulsion for unmanned aircraft, missiles, and target systems places great emphasis on low cost, economically viable systems. The complex lubrication systems employed in conventional aircraft gas turbines are prime cost reduction candidates. These systems generally employ multiple oil sumps with multiple element oil pumps, associated control valves, filters, deaerators, coolers, oil tanks, and plumbing. One approach to elimination of this complexity, and reduction of system cost, is to replace the conventional systems with grease-packed or pot-lubed¶ bearing/lubrication systems.

This paper discusses the design and development of grease packed and pot lubed self-contained mainshaft bearing/lubrication systems for the J402-CA-400 turbojet engine[1,2] for the AGM-84A HARPOON Weapons System. The J402 is a small engine, but the basic approach is adaptable to larger engines. The design and development considerations are discussed, beginning with the grease-packed mainshaft ball (thrust) and roller bearing systems, and continuing through the final pot-lubed system for the ball bearing.

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¶A wet sump system, without external oil supply.

In spite of the short life expendable nature of the engine, the application of the J402 turbojet engine to a tactical missile results in a stringent set of requirements; specifically in terms of the small volume available for the lubrication system, environmental temperature, long untended storage life, rapid starts, and shaft thrust loading.

Design Requirements

The bearing/lubrication system development program was conducted in two phases, with the second phase requirements being more severe. The design requirements for both phases of bearing testing are summarized in

Table 1 Bearing design requirements

	First phase	Second phase
Speed, rpm	40,800	41,200
Life, min	24	38
Bearing load		
Thrust, maximum continuous, lb	500	1000
Thrust, maximum for 2 min, lb	500	1200
Radial, continuous, lb	100	100
Cold start		
Soak temperature, °F	−40	−60
Start time,[a] sec	11	11
Hot start		
Soak temperature, °F	160	160
Start time,[a] sec	5	5
Bearing cooling air temperature		
maximum continuous, °F	260	193
maximum for 2 min, °F	260	238

[a] Start time varies with altitude and flight speed; times quoted are the most severe conditions, i.e., the least allowable time to accelerate from 0 to 98% of maximum speed (40,800 rpm for both the first phase hot and cold temperature starting conditions and 41,200 rpm and 38,500 rpm for the second phase hot and cold temperature starting conditions, respectively).

Table 1. With the exception of the high shaft thrust loading (1200 lb) and the longer life (38 min) requirements, the second phase requirements were anticipated at the onset of the program.

In addition to the basic design requirements, there were a number of design considerations arising from the missile application of the engine. The HARPOON missile has a cylindrical propulsion section which dictated a maximum engine outer diameter of 12.5 in. and length of 29.25 in. The engine develops a sea-level static thrust of 660 lb and weighs less than 100 lb, including all required accessories and equipment. The major portion of the available space outside the engine's casing was required for the fuel control electronics package, the alternator power conditioning unit, the two pyrotechnic starter cartridges and various lines and connections. For all practical purposes, this space limitation dictated that the bearing, lubrication, and heat rejection systems be totally contained inside the engine nose cone and tail cone. The space and cost constraints resulted in an inline, shaft speed, alternator, and fuel pump, eliminating all gearing. The remaining useable volume available for the ball bearing and lubricant systems was 6.5 in.^{3}, and 3.4 in.^{3} for the roller bearing. The front-mounted inline alternator heated the thrust bearing cooling air, resulting in the 260°F maximum cooling air temperature. Elimination of the alternator in the second phase of the program made additional space available for the ball bearing lubrication system and lowered the maximum cooling air temperature to 238°F.

The engine had to be capable of being stored for periods of up to five years, on land, at sea, and in the air. Consequently, the lubricants and the materials used in the bearing system had to be compatible with the environments that the engine would encounter in the various storage locations, including moisture, salt content, temperature, and vibration. The missiles (engines) have to be in a state of constant readiness for immediate deployment; no lubricant check or replenishment can be performed before firing.

Bearing System Design and Development

The major factors in the design of the bearing/lubrication system were the size and cost constraints, the operating speed and thrust load, and the fast starts, especially at the −60°F condition. The engine is started by two pyrotechnic cartridges that generate 620 hp and impinge on the radial compressor rotor. There is no motoring prior to start and no idle or hold in accelerating to full operating speed, so the 11 sec cold start condition is one of the most severe requirements of the bearing system.

A number of bearing systems were initially considered for the J402 engine. Those considered, in ascending order of complexity (primarily numbers of parts) and cost were: 1) air foil bearings; 2) dry bearings; 3) grease-packed bearings; 4) pot-lubed bearings; 5) wick systems (variant of pot-lube system with a wick serving as a lubricant conduit); 6) mist lube systems (lubricant sprayed on the bearing in an air mist); and 7) conventional aircraft turbine lubrication systems.

The air foil bearings offered the potential of being a low-cost system, but their load-carrying capability and state of development at the time excluded them from serious consideration for the J402 engine. The mist and conventional systems were undesirable from the standpoint of cost, weight, and space. Early testing of the dry bearing indicated a high development risk for the J402 application, so further testing was not pursued. From the remaining candidates, the grease-packed system was selected, primarily based on its cost advantage. Pot-lubed and wick systems were designed as backup systems.

Ball Bearing System Design And Development

The ball (thrust) bearing was designed and developed initially as a grease-packed system. The engine successfully completed its initial qualification and flight testing with the grease-packed bearing systems. Because the requirements were more severe, the grease-packed bearing was replaced with a pot-lubed system for the second phase of development, as discussed later.

Functionally, the ball bearing supports the front of the mainshaft and reacts the shaft thrust load. In the interest of cost and space, the engine incorporates only one mainshaft thrust bearing in place of the two that would be used for a longer life engine, and the engine pressure seals were located at a smaller diameter to favor aerodynamic performance rather than at a larger diameter to favor shaft thrust balance. This resulted in a high shaft thrust load and a reduced bearing life (B_{10} life of 7 hours at maximum conditions for the final thrust bearing). The bearing size was selected on the basis of shaft critical speed requirements, grease-packed bearing DN state-of-the-art, thrust load, and life. The smallest size (DN) and most inexpensive conventional design bearing that would result in a satisfactory shaft third critical speed and have a reasonable chance of satisfying the thrust load carrying requirement was a basic 204 bearing (20 mm) with a DN

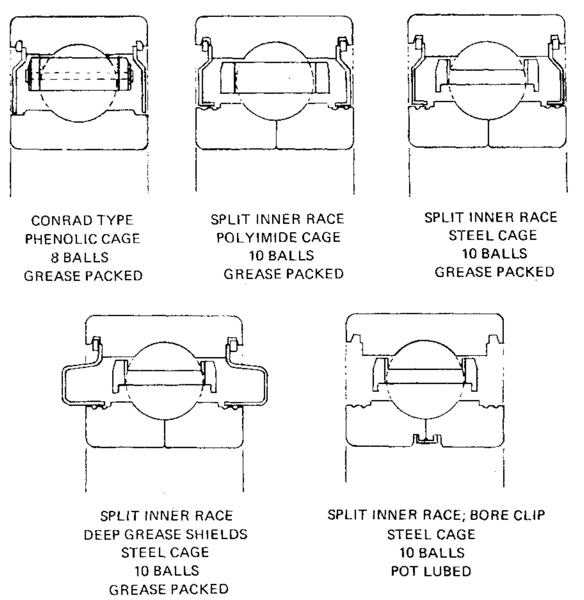

Fig. 1 Ball bearing evolution.

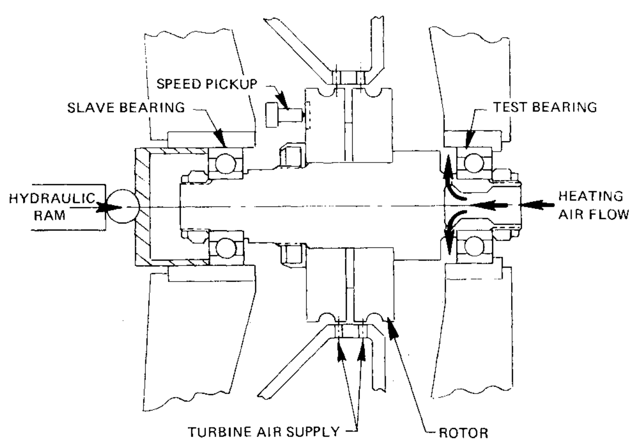

Fig. 2 Greased-packed ball bearing test rig.

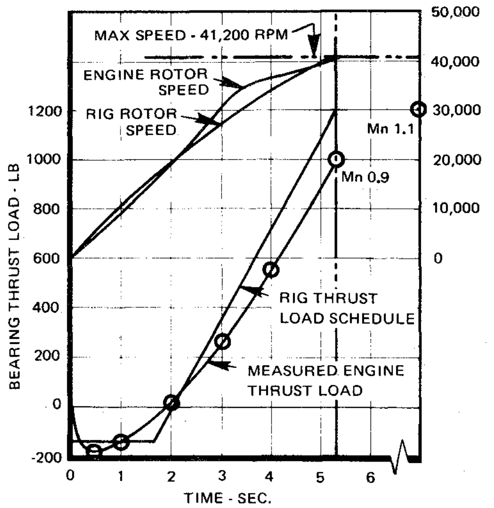

Fig. 3 Speed and ball bearing thrust as a function of time.

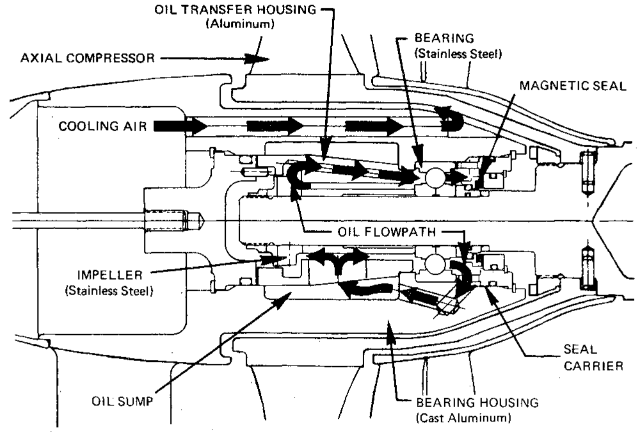

Fig. 4 Pot-lubed ball bearing system.

of 824,000. The bearing initially selected was a 440C stainless steel, Conrad type, bearing, with a riveted, aluminum reinforced, phenolic cage (Fig. 1). The basic approach was to test this "low-cost" bearing and make changes only as needed to meet the required life with a margin.

The grease-packed bearing test rig (Fig. 2) was a modified air turbine, with its rotor supported on the test bearing and a slave bearing. The thrust load was applied by a hydraulic ram, loading through the slave bearing outer race, and a radial load was induced by unbalancing the turbine rotor. Operating temperature conditions were simulated by directing a stream of hot air around the bearing for the hot condition, and by packing dry ice and discharging CO_2 around the bearing for the cold condition.

The primary considerations addressed in the first phase of ball bearing rig testing were lubricant selection, rolling element geometry, cage construction, and grease depletion. Testing conducted to provide data for grease selection included testing of Unitemp 500 and several Krytox grease formulations. The results showed that only the Krytox 260AC survived as long as 15 min; therefore, this grease was used for the remainder of the first phase of the development program.

Problems encountered affecting the rolling element geometry included excessive thermal inertia (loss of internal clearance) and insufficient thrust load capability. Bearing failure during rapid acceleration of the rig suggested the need for increased internal clearance to compensate for the loss in clearance resulting from the slower heating of the outer race and its supporting housing. An increase from 0.0022 in. to 0.0028 in. remedied the problem. Subsequent engine testing showed that the balls were running high on the inner race of the Conrad bearing, indicating the need for increased thrust loading capability. The increased thrust capability was achieved by switching to a larger internal clearance (0.0036 in.), higher shoulder, split-inner-race bearing and by increasing the ball complement from 8 to 10. The ball and race material was also changed to 440C modified stainless steel to obtain additional capacity at temperature from the higher hardness (Rc 60 vs Rc 56).

Other significant factors influencing the change to a split-inner-race configuration were cage charring, cracking, and separation problems encountered in running the phenolic cages at temperatures near 450°F. A one-piece polyimide cage (DuPont SP-21) was initially used in the split-inner-race bearing. Most of the cage problems were alleviated with the polyimide cage, but side-rail cracking was experienced in some of the cages in engines that had accumulated nearly an hour of operation with significant time at the high inlet temperature conditions. To correct this, the cage material was changed to silverplated alloy steel (AMS 6415). The cage initially incorporated an undercut in the center portion of the i.d. of the cage to collect lubricant and feed it centrifugally through the ball pockets. Because an indication of skidding (light race frosting) was observed after rapid acceleration tests, the center section of the o.d. of the cage was also undercut to minimize the mass polar moment of inertia of the cage. This eliminated the skidding indications.

Engine testing at high temperatures and high shaft thrust load conditions indicated a tendency for the cage silverplating to be extruded at the edges of the ball pockets. This was attributed to the high contact pressure resulting from the area reduction associated with the cage o.d. and i.d. undercuts. The silver extruding was eliminated by reducing the plating thickness from 0.0015 in. to 0.00075 in.

Another problem which emerged from the engine test program was premature grease depletion. This problem was alleviated by replacing the standard shields with deep pocket grease shields which extended 0.120 in. beyond the edges of the races (Fig. 1), and by incorporating a windback seal** on the outer diameter of the bearing spacer. This spacer fits, with a small radial clearance, inside the inner diameter of the extended grease shield.

One series of engine tests, utilizing a flexible ring between the ball bearing and bearing housing to damp engine vibration, indicated the importance of heat conduction into the housing for cooling the bearing. With the normal installation of the grease-packed ball bearing directly into the housing, the bearing temperature stabilized between 420 and 470°F, depending on the engine operating conditions. However, with the flexible ring in place, the bearing temperature did not stabilize, and the test was terminated at a bearing temperature above 500°F. As a result, the ball bearing outer race o.d. and thrust face were designed to contact the housing directly, providing maximum heat transfer area without secondary interfaces. Inlet air is passed through the bearing housing for cooling.

A single grease-packed bearing of the final configuration was subjected to the rig testing sequence shown in Table 2. The bearing was cleaned and regreased between Parts A and B of the test; Part A was a continuous 1 hr and 10 min run.

Late in the first phase of the test program, engine testing was conducted to determine the shaft thrust loading for the operating conditions of the second phase of the

**A helical groove cut into the o.d. of a rotating part which runs with a small clearance inside a stationary diameter.

Table 2 Grease packed ball bearing demonstrated capability

Part A: Load/Endurance

Test time at indicated thrust load (min)	Thrust load (lb)
10	500
10	600
10	700
10	800
10	900
10	1000
10	1100

speed, 41,000 rpm; radial unbalance, 100 lb, cooling air temperature, 260°F

Part B: Accelerations

Soak temperature (°F)	Acceleration time 0–41,000 rpm (sec)	Endurance time at 41,000 rpm and 500 lb thrust load (min)
70	8	10
−40	8	10
−65	8	10

program. The results of these tests (Fig. 3) indicated a very high shaft thrust (1200 lb) at the maximum short time condition, a 1000 lb load at the maximum continuous condition, and a thrust reversal during engine acceleration, at approximately 19,000 rpm. Rig testing under these conditions indicated that the grease-packed ball bearing was marginal in terms of heat dissipation and maintaining the lubricant film necessary to support the 1200 lb thrust load. Consequently the grease-lubricated ball bearing development was terminated in favor of a pot-lubed system.

The pot-lubed bearing system (Fig. 4) is contained totally within the inner diameter of the axial compressor rotor. The system incorporates a wet sump and utilizes a centrifugal pump to assist the natural pumping action of the bearing, especially in the nose down attitude and during cold starts. The lubricant pumped by the impeller is collected in a groove near the forward end of the oil transfer housing and flows to the bearing through holes which discharge lubricant directly onto the front of the bearing. The oil transfer housing also separates the return flow, directs it into the impeller inlet and permits oil to flow directly into the bearing. The lubricant flow is returned to the sump through holes drilled in the seal carrier and bearing housing. The system is symmetrical about the engine centerline, making it insensitive to roll angle.

The pot-lubed bearing system utilized the final grease-packed ball bearing configuration, except the grease shields and the associated snap rings were eliminated and a bore clip was added to keep the inner race halves together during assembly. No functional changes were made to the bearing itself in the pot-lubed system development program.

The pot-lubed system rig testing more closely simulated actual engine operation from the standpoint of rig hardware, soak and operating temperatures, attitude, radial unbalance, and speed and thrust load transients, including the thrust reversal. The rig is shown in Fig. 5. Typical rig transients and corresponding measured engine transients are shown in Fig. 3. Two problems were identified in the pot-lubed system rig test program: 1) temperature instability at the high temperature (238°F) inlet condition and 2) cold start (−60°F) and 30 deg nose down attitude start failures.

Rig testing of a bearing at the 1200 lb thrust load and 238°F condition resulted in a gradual, but consistent, bearing tempeature rise. Analysis of the oil after testing indicated significant carbon particle contamination. The carbon particle size approached the oil film thickness at the high inlet temperature, high thrust load, test conditions, resulting in increased heat generation and the unstable bearing operating temperature. The contamination was attributed to blistering and flaking of the carbon seal due to heat generation at the interface of the wiper and carbon. By changing to a higher grade carbon, the amount of contamination was reduced to an acceptable level.

The cold start problem was the more persistent of the two. It was attacked in steps and finally eliminated by a combination of changes. Fast accelerations after cold soaking (bearing and oil temperature stabilized at −60°F) resulted in bearing failures after 30 sec of operation. An acceleration test was conducted under the same conditions to determine whether the failures were due to skidding or lack of lubrication. The rig was stopped immediately upon reaching maximum speed and load (9 sec). The bearing showed no evidence of skidding, indicating that lack of lubrication was the cause of failure. Tests were then conducted to determine if increased oil level would improve bearing lubrication during cold start. Preliminary tests were run to select the highest acceptable oil level, based on oil loss and bearing temperature after a 15 min run. Initially, the sump was filled to a level 0.65 in. below the shaft centerline (to the center of the lowest ball), with the engine in the horizontal attitude. Based on the oil level test results, a level 0.3 in. below the shaft centerline was selected. The higher oil level improved the horizontal cold start performance of the bearing, but subsequent cold start and 30 deg nose-down room temperature start failures indicated the need for improved lubricant flow into the bearing.

A redesigned, simplified, oil pump impeller and modified oil transfer housing (Fig. 6) were introduced to improve the oil feed to the bearing. To reduce cost, the impeller was designed as a simple turning with 3 holes drilled radially through to form the vanes. The rig was modified to measure the lubricant flow and pressure in the oil flow channel by blocking off one of the three flow channels and inserting a tube to direct the flow out of the

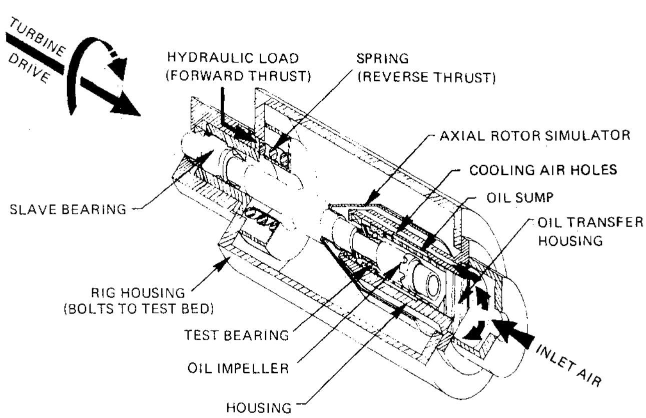

Fig. 5 Pot-lubed ball bearing development test rig, cut away section.

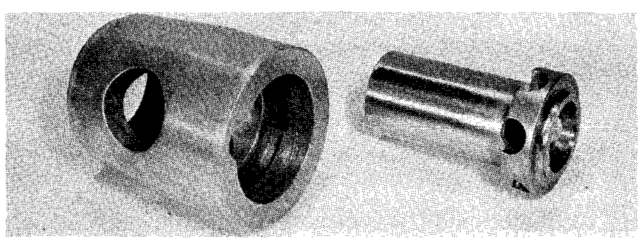

Fig. 6 Oil transfer housing and pump impeller.

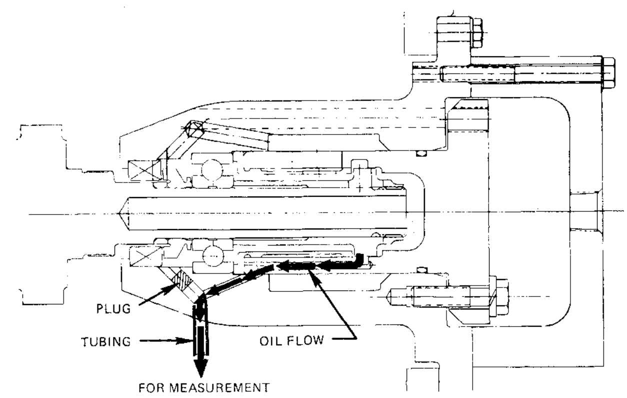

Fig. 7 Oil flow and pressure measurement.

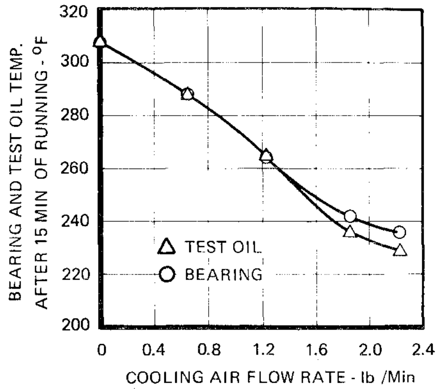

Fig. 9 Ball bearing and test oil temperature variation with cooling air flow rate.

rig for measurement (Fig. 7). These tests showed that the new impeller increased the average flow rate from 2.1 to 6.3 ml per min and the head rise from 0.17 to 0.20 psi, measured at room temperature. The modification to the oil transfer housing changed the oil pump feed from the rear side of the bearing to the front, thereby augmenting the natural pumping action of the bearing. Based on additional oil level testing (Fig. 8), an oil level at the shaft centerline (70 ml of oil) was selected for this system. Because of the temperature vs cooling airflow characteristics (Fig. 9) previously measured on the rig, and the fact that the engine cooling airflow had not been measured, the oil level tests were run with no cooling airflow.

The evolution of the ball bearing is pictured chronologically in Fig. 1, and the developed ball bearing is shown disassembled in Fig. 10. The pot-lubed ball bearing system successfully completed the rig testing indicated below. All endurance tests were run at maximum speed (41,200 rpm), thrust load (1200 lb), and radial unbalance load (100 lb). 1) Horizontal attitude, room temperature start, followed by 2 hr endurance (room temperature cooling air through the housing; start time, 4.5 sec). 2) 30° nose down attitude, bearing, and oil temperature −60°F, cold start, followed by 15 min endurance (no cooling air through the housing; start time, 10 sec). 3) Horizontal attitude, bearing, and oil temperature −60°F, cold start, followed by 90 min of endurance (no cooling air through the housing; start time, 10 sec). 4) Horizontal attitude, bearing, and oil temperature of 160°F, hot start, followed by 92 min of endurance (238–193°F) cooling air through the housing; start time 4.5 sec). 5) 30° nose up attitude, bearing, and oil temperature of −56°F, cold start, followed by 15 min of endurance (no cooling air through the housing; start time 11 sec).

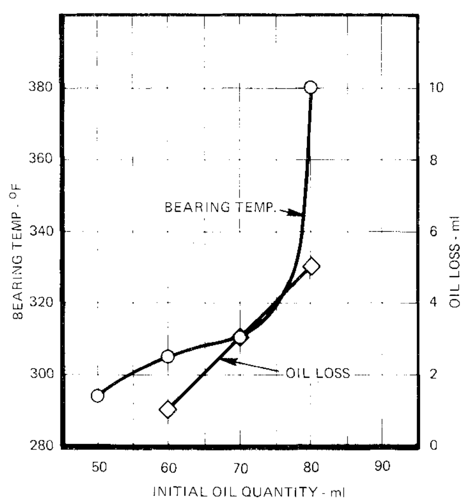

Fig. 8 Ball bearing temperature and oil loss after 15 min run.

With the final configuration pot-lubed system, a total of 4 hr and 41 min of rig testing was accumulated on one bearing. No engine development of this bearing system was required.

The pot-lubed bearing system development program was conducted using MIL-L-7808 as the lubricant. This oil was selected because of its relatively low viscosity at the −60°F temperature condition and its general suitability for turbojet engine applications. However, because of concern for its stability (water absorption and subsequent acid formation) during 5 years of storage, the lubricant was changed to a mixture of synthetic hydrocarbon, fuel and super refined heavy resin. The synthetic hydrocarbon provides a very stable base, the fuel serves as a viscosity depressant at the low temperature condition, and the heavy resin serves to enhance the lubricant film for the high temperature condition.

Roller Bearing System Design And Development

The roller bearing supports the rear of the mainshaft. Its housing extends inside the bore of the turbine wheel (Fig. 11), resulting in a hot operating environment for the bearing. Consequently, provisions were made to pass cooling air from the inner section of the shaft over the bearing and o.d. of the bearing housing, resulting in a bearing temperature in the range of 600 to 800°F, depending on engine operating conditions.

The approach to designing this bearing system was basically the same as for the ball bearing. The bearing size was selected from consideration of shaft critical speed, shaft stiffness and bearing DN. The development started with the most inexpensive bearing having the potential for meeting the engine requirements noted in Table 1.

The initial bearing design (Fig. 12) was a basic 203 size, 17 mm, with an extended inner race to accommodate the

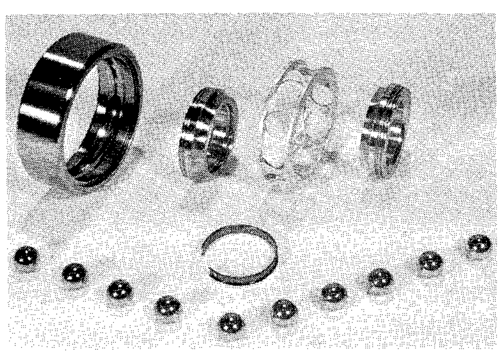

Fig. 10 Disassembled ball bearing.

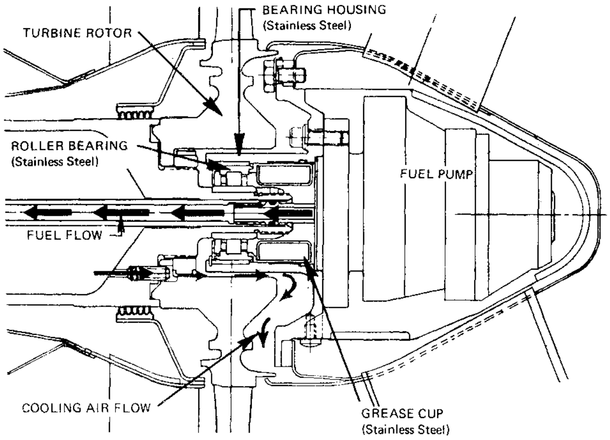

Fig. 11 Greased-packed roller bearing system.

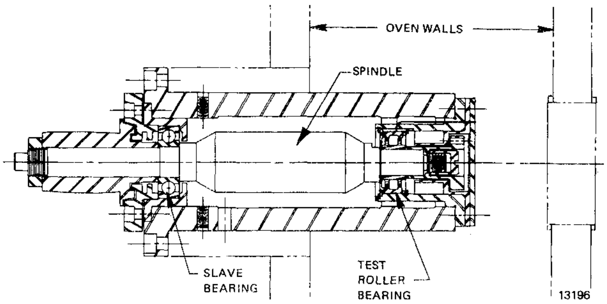

Fig. 13 Roller bearing test rig.

axial tolerance stackup and differential thermal expansion of the engine. This bearing operates at a DN value of 700,000. The races and rollers are made of 440C modified stainless steel. A small roller complement, 8 vs the standard 10, was selected to provide added cage separator thickness to enhance the cage strength. A one piece, outer land riding, polyimide cage was initially utilized because of its lubricity and resultant minimal heat generation.

The roller bearing rig testing was performed at the Marlin Rockwell Div. of TRW. The rig (Fig. 13) consisted of a belt driven spindle supported by the test roller bearing at one end and a grease-packed slave ball bearing at the other. The test bearing end of the support housing was enclosed in an oven with a regulated temperature range of up to 1000°F. The significant considerations addressed in testing were grease selection and grease depletion.

Several high temperature greases, including blends of polyphenyl ether and silicon fluids in various thickening or binding agents and various Krytox formulations, were evaluated. Of these, Krytox 280AC proved superior for high temperature endurance, and was utilized throughout the rig and early engine test programs. In the later phases of the engine test program, indications of high temperature bearing corrosion were observed. To reduce this corrosion to an acceptable level, the grease was changed from Krytox 280AC to 283AC, which contains a higher percentage of corrosion inhibiter (5% vs 1%).

Premature grease depletion was the only significant problem encountered during the rig test program. In some tests, grease depletion was evident after 20 minutes of testing. Increasing the grease capacity by increasing the bearing width failed to consistently provide the desired rig test life. As a result, a grease cup was designed to provide approximately 1 in.^{3} of additional grease storage capacity.

The grease cup is basically an annular grease storage container with four internal radial dividers to make the system insensitive to roll angle. There are two holes in the front face of each compartment to permit grease flow, venting and filling of the grease cup. Rig tests of up to 1 hr and 30 min duration were achieved with the grease cup. Based on subsequent engine testing, the grease cup was modified slightly to minimize grease leakage by adding a lip to fit into the bearing outer race counterbore and an "O" ring seal at the outer diameter of the grease cup.

The only other significant problem in the roller bearing system development was that of cage failure. As early development engines accumulated time and rebuilds, cracking of the polyimide cages was observed. A silver plated steel cage was incorporated to prevent the cage cracking, which was attributed to assembly and disassembly abuse. The cage has a circumferential undercut at the o.d., and the roller pockets are formed by an intersecting axial broaching operation. This construction minimizes cage inertia, provides added grease capacity and exposes a greater portion of the ends of the rollers to the lubricant. The resultant bearings worked well through the remainder of the initial phase of the engine test program. Subsequent minor changes in the engine configuration in the area surrounding the roller bearing produced a change in temperature distribution within the bearing which resulted in a loss of internal clearance and precipitated two cage failures. This problem was eliminated by increasing the bearing internal clearance from 0.001 in. to 0.00175 in. The resultant roller bearing configuration (Fig. 14) performed successfully throughout the remainder of the engine development program.

Summary

Two basic bearing systems have been developed for the J402-CA-400 engine. These systems include a pot-lubed ball bearing system and a grease-packed roller bearing

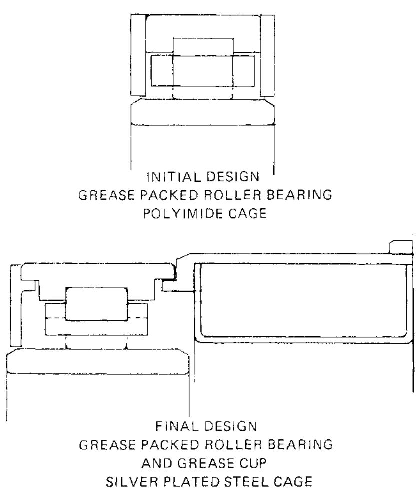

Fig. 12 Roller bearing evolution.

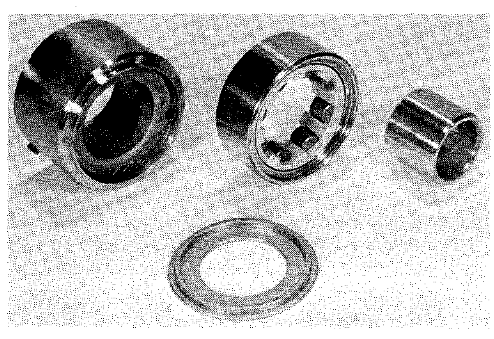

Fig. 14 Disassembled roller bearing.

system. A third system, a grease-packed ball bearing system, was developed to a lesser degree. These systems were developed for a relatively short life application with severe acceleration and environmental temperature requirements, and, in the case of the ball bearing, for an extremely high thrust load.

The use of a rig test program in conjunction with full-scale engine testing proved to be an expedient approach to developing a new bearing and lubrication system. Rig testing provided initial bearing development data and, with quantitative load data fed back from engine testing, provided a means of totally developing the pot-lubed ball bearing system.

Engine testing of the developed bearing systems has been extremely successful. Since the pot-lubed front main bearing system was incorporated 69 engine tests have been conducted, and a total of 73 hr and 45 min of engine test time has been accumulated. No failures were experienced in this testing except for one caused by an improper cool down procedure which was not representative of the engine application. A single ball bearing was run in an engine for 4 hr and 43 min.

The final grease-packed roller bearing has operated in the engine without incident during 32 engine tests, accumulating 20 hr and 34 min of test time. A single roller bearing ran in an engine for 1 hr and 28 min.

In addition, the bearing systems have performed without incident throughout the flight test program. This testing included engines with grease-packed and engines with pot-lubed ball bearings.

Conclusions

1) Pot-lubed and grease-packed bearing lubrication systems are practical for short-life gas turbine engines. 2) The use of grease-packed and pot-lubed bearing systems is a viable means of reducing engine complexity. 3) Pot-lubed and grease-packed bearing systems are capable of withstanding extremely fast engine starts for both hot (180°F) and cold (−60°F) soak temperatures. 4) Pot-lubed ball bearing systems are capable of operating at DN values of 824,000 with high thrust loads (1200 lb with a 20 mm bearing). 5) Grease-packed ball bearings are capable of operating at DN values of 824,000 with high thrust loads (1100 lb with a 20 mm bearing).

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