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INVESTIGATION OF VAPOR-PHASE LUBRICATION IN A GAS TURBINE ENGINE

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ABSTRACT

The liquid oil lubrication system of current aircraft jet engines accounts for approximately 10-15% of the total weight of the engine. It has long been a goal of the aircraft gas turbine industry to reduce this weight. Vapor-Phase Lubrication (VPL) is a promising technology to eliminate liquid oil lubrication. The current investigation resulted in the first gas turbine to operate in the absence of conventional liquid lubrication. A phosphate ester, commercially known as DURAD 620B, was chosen for the test. Extensive research at Wright Laboratory demonstrated that this lubricant could reliably lubricate rolling element bearings in the gas turbine engine environment. The Allison T63 engine was selected as the test vehicle because of its small size and bearing configuration. Specifically, VPL was evaluated in the number eight bearing because it is located in a relatively hot environment, in line with the combustor discharge, and it can be isolated from the other bearings and the liquid lubrication system. The bearing was fully instrumented and its performance with standard oil lubrication was documented. Results of this baseline study were used to develop a thermodynamic model to predict the bearing temperature with VPL. The engine was then operated at a ground idle condition with VPL with the lubricant misted into the #8 bearing at 13 ml/hr. The bearing temperature stabilized at 283°C within 10 minutes. Engine operation was continued successfully for a total of one hour. No abnormal wear of the rolling contact surfaces was found when the bearing was later examined. Bearing temperatures after engine shutdown indicated the bearing had reached thermodynamic equilibrium with its surroundings during the test. After shutdown bearing temperatures steadily decreased without the soakback effect seen after shutdown in standard lubricated bearings. In contrast, the oil lubricated bearing ran at a considerably lower operating temperature (83°C) and was significantly heated by its surroundings after engine shutdown. In the baseline tests, the final bearing temperatures never reached that of the operating VPL system.

NOMENCLATURE

C_p	specific heat (J/kgK)
h_{load}	heat load coefficient (W/deg C)
$\dot{m}$	fluid mass flow rate (kg/sec)
MDN	shaft diameter (mm) X shaft RPM/10^6
$\dot{Q}_{oil}$	oil heating rate (W)
$\dot{Q}_{air}$	air heating rate (W)
$\dot{Q}_{vap}$	Durad heating rate (W)
$T_{bearing}$	#8 bearing temperature (deg C)
T_{exit}	#8 vapor outlet temperature (deg C)
T_{inlet}	#8 vapor inlet temperature (deg C)
$T_{surrounding}$	#8 bearing air cavity temperature (deg C)
T_{T3}	compressor bleed temperature (deg C)
T_{T5}	intra-turbine temperature (deg C)

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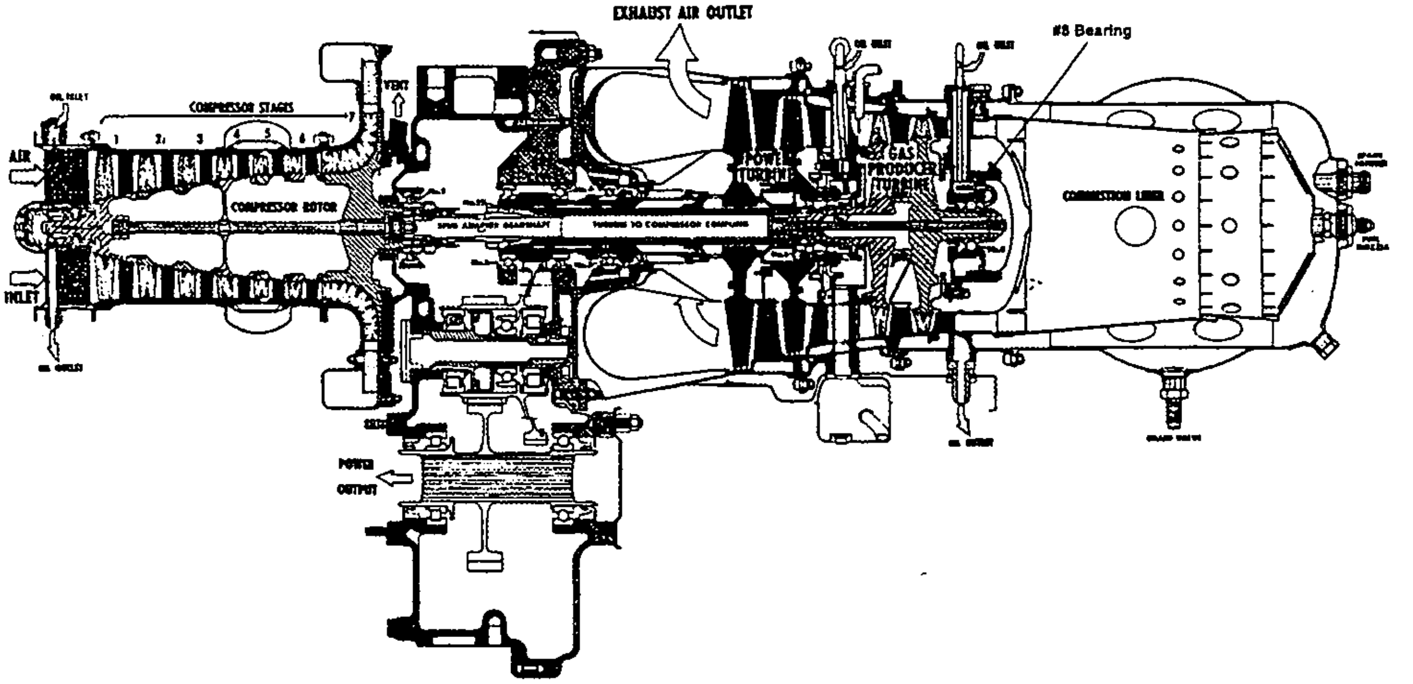

Figure 1 Allison T63 Turboshaft Engine (Allison, 1981)

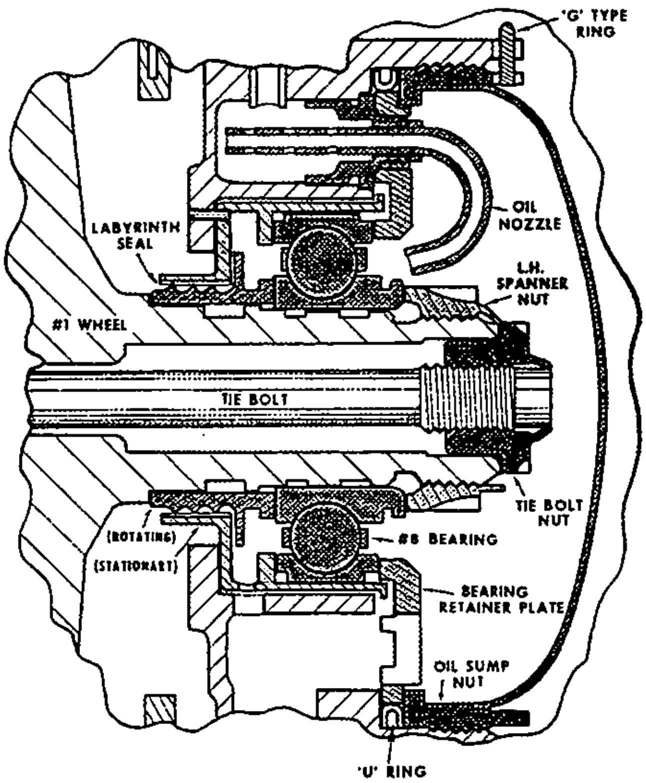

Figure 2 Sideview # 8 Bearing Housing (Allison, 1981)

INTRODUCTION

The development of gas turbines operating without a conventional liquid lubrication system has been a research objective for several decades. Potential benefits of eliminating the liquid lubrication system include reductions in cost, weight, engine cross-sectional area, and maintenance. Additional benefits are possible if the lubrication method also increases the operating temperature of the main shaft bearings reducing thermal gradients and therefore, thermal stresses in the rotating components of the engine. Currently, bearing temperatures in gas turbines are limited to approximately 204°C due to thermal limitations of the liquid lubricant. To maintain a 204°C operating temperature, the bearing compartment is cooled with compressor air, heat shielding is added to critical locations, and the lubricant is cooled via a fuel/oil heat exchanger. In advanced engines the use of these thermal management tools becomes increasingly more difficult.

Previous efforts to develop a high temperature lubrication system focused primarily on solid lubricants delivered as powders [Macks et al. (1951), Anderson (1965), Wilson (1962), and Wallerstein (1965)] or as lubricant films transferred from the sliding surfaces of bearing cages [Devine et al. (1961), Dayton (1971), Boes (1978), and Gardos (1984)]. Both of these methods provide adequate lubrication for lightly loaded, low speed applications. However, both methods have demonstrated limited success at conditions required for a gas turbine engine i.e. , bearing speeds of 1.5 to 2.5 MDN (MDN = shaft diameter (mm) x shaft rpm / 10^6) and bearing stress loads of 1.0 to 2.0 GPa. Bearing wear, cage fracture, and seizure due to thermal growth, are the common modes of failure in high speed, solid lubricated bearings.

During the 1980's and early 1990's a new form of lubrication known as vapor phase lubrication began to show promise as an alternative high temperature lubrication concept [Gardos (1984), Graham and Klaus (1985), Gunsel (1986), Klaus et al. (1989), Klaus et al. (1990), Maki and Graham (1990), Maki and Graham (1991), Klaus and Duda (1991), Graham et al. (1993), Rao (1993), Morales et al. (1994) and Hanyaloglu and Graham (1994)]. To accomplish VPL, a small quantity of organophosphorus material is vaporized and transported to a metallic bearing surface where the vapors chemically react with the surface to form the lubricating film. Analyses of bearings, lubricated with a tertiary-butylphenyl phosphate, DURAD 620B, indicates that the lubricating film is primarily composed of condensed phosphates and graphite (Forster, 1996a). The phosphate serves as an antioxidant and binder for the lubricant. The extremely high flash point of the lubricant also allows a thin layer of liquid lubricant to exist in the bearing contact at extremely high temperatures.

During the 1990's Wright Laboratory initiated an extensive in-house research program to investigate vapor phase lubrication for use in gas turbine bearings. The primary emphasis of this research was to identify non-toxic vapor lubricants which can successfully lubricate rolling element bearings in an air environment (Forster, 1996a, 1996b). Over the past five years the technology has evolved to the point where gas turbine bearings can be reliably lubricated for periods of several hours. This period of operation is sufficient for consideration of the technology in expendable class engines. One of the key shifts in the vapor phase approach has been delivering the lubricants as an oil-mist rather than in the vapor-phase. The increased momentum of an oil-mist droplet allows better penetration of the pressure differential created by windage in high speed bearings. Upon entering the bearing, the bearing surface temperature provides the heat input required to complete vaporization and to initiate the chemical reactions. The oil-mist delivery approach also provides additional cooling of the bearing.

HIGH SPEED BEARING RIG TESTS

The Allison T63 turboshaft engine is an excellent platform to test VPL because of its small size and wide use in the commercial and military helicopter market (Figure 1). The #8 bearing in the T63 (Figure 2) was chosen to test VPL because its design places the #8 bearing immediately downstream of the combustor, providing a harsh temperature environment for the bearing cavity, and because the #8 bearing can be isolated from the other bearings in the liquid lubrication system. A shroud around the bearing housing is supplied with cooling air from the compressor. The shroud serves to shield the bearing housing from direct combustor flow, but at this location, high cooling rates for the bearing are still required to maintain normal operating temperatures. The T63 #8 bearing is a 20 mm bore split outer race bearing, with M50 steel balls and races, and a one piece cage of silver plated 4340 steel. It is typical of existing bearings currently used with conventional oil lubrication systems.

Prior to engine testing, single-bearing rig tests were performed to establish safe operating limits, thereby reducing risk to the engine. This was considered necessary because the bearing is designed to operate with standard liquid lubrication at outer race temperatures below 150°C, and temperatures with VPL were expected to be in excess of 260°C. The T63 #8 bearing, its support, and the vapor mist system were installed in the High Speed Bearing test rig. The rig is an air turbine driven test stand designed to test complete bearings at speeds up to 55,000 rpm. Operating conditions in the engine were simulated by blowing hot air over the bearing compartment and applying a constant thrust load of 267 Newtons to the bearing outer race. The rig could not simulate the rapid heating experienced in the engine, so the bearing was preheated to a steady state temperature for 30-45 minutes prior to each test. All tests began with a slow (10-15 minute) ramp up to the engine idle speed of 35,000 rpm, and then operation at idle to establish steady state conditions. From there, heater and speed controls were adjusted to establish other desired operating conditions. The precision to which any given test condition could be maintained was ±14°C in bearing temperature and ±1000 rpm in speed. Bearings were generally run until a sharp increase in friction was detected, as indicated by a rapid decrease in rig speed (~1000 rpm/sec) and a rapid increase in outer race temperature (~3°C/sec).

A total of eight bearings were rig tested. Results are summarized in Table 1. Setup and procedural errors which are believed to have caused premature suspension to bearing tests 1, 3, 4, and 6 are noted. Visual inspection of the bearing cages revealed wear at the cage-outer land and ball pocket surfaces in all cases, similar to the engine bearing. The bearing races and rolling elements varied in condition, but were all worn more severely than the engine bearing. This was attributed to the considerably longer run times and more severe test conditions experienced by the rig bearings.

Based on the series of bearing rig tests and the success demonstrated during bearing test eight, it was concluded that safe and continuous operation of the bearing could be achieved at outer race temperatures below 379°C at speeds from idle to full power (35,000-51,000 rpm). Continuous engine operation at temperatures above the limit was deemed inadvisable. The greater wear of the races and balls of the rig bearings indicates that improved materials for these components will be required for extended life at higher temperatures. Finally, although the engine races and balls showed virtually no wear, its cage was worn similarly to the rig bearings. This indicates that cage wear is a problem at mucb less severe conditions than race and ball wear. Therefore, a primary focus of future efforts should be on the improvement of cage lubrication.

T63 EXPERIMENTAL SETUP

With the success of the High Speed Bearing Rig tests, the next step was to test VPL under engine operating conditions. These experiments were conducted on an Allison T63-700 turboshaft engine. The engine test cell was equipped with a complete engine control system and with a SUPERFLOW SF-740 data acquisition system capable of 320 channels of data. The engine was fully instrumented and all normal engine health parameters monitored. The acquisition system also incorporated a dynamometer to measure engine torque during testing. In addition to the standard instrumentation, the # 8 bearing was fully instrumented to provide information on the operating conditions during both oil lubrication and VPL operations (Figure 3). The bearing housing contains five support struts. Two of these struts are ordinarily used to feed the lubricant to the bearing and remove the lubricant from the bearing.

Bearing	Speed (rpmx1000)	Bearing Temp (C)	Run Time (hr)	Comments
1	38	385-391	0.6*	extended preheat cycle without lube, unintended
2	38	338-354	12.0	
3	33	304-346	0.08*	bearing misalignment; alternate lube.
4	33	327-416	0.4*	bearing misalignment.
5	38	382-416	2.7*	
6	38	327-349	0.7	extended preheat cycle without lube, unintended
	40	338-343	0.1	
	43	343-360	0.01*	
			0.8 Total	
7	38	399-416	3.3*	
8	35-39	332-377	4.7	
	40-50	316-343	3.5	
	50-55	299-354	8.9	
	55	349-379	3.1	
	55	379-418	0.6	
	55	418-443	0.05*	
			20.9 Total	
				*Test terminated due to sudden friction increase

Table 1: High Speed Test Rig Results

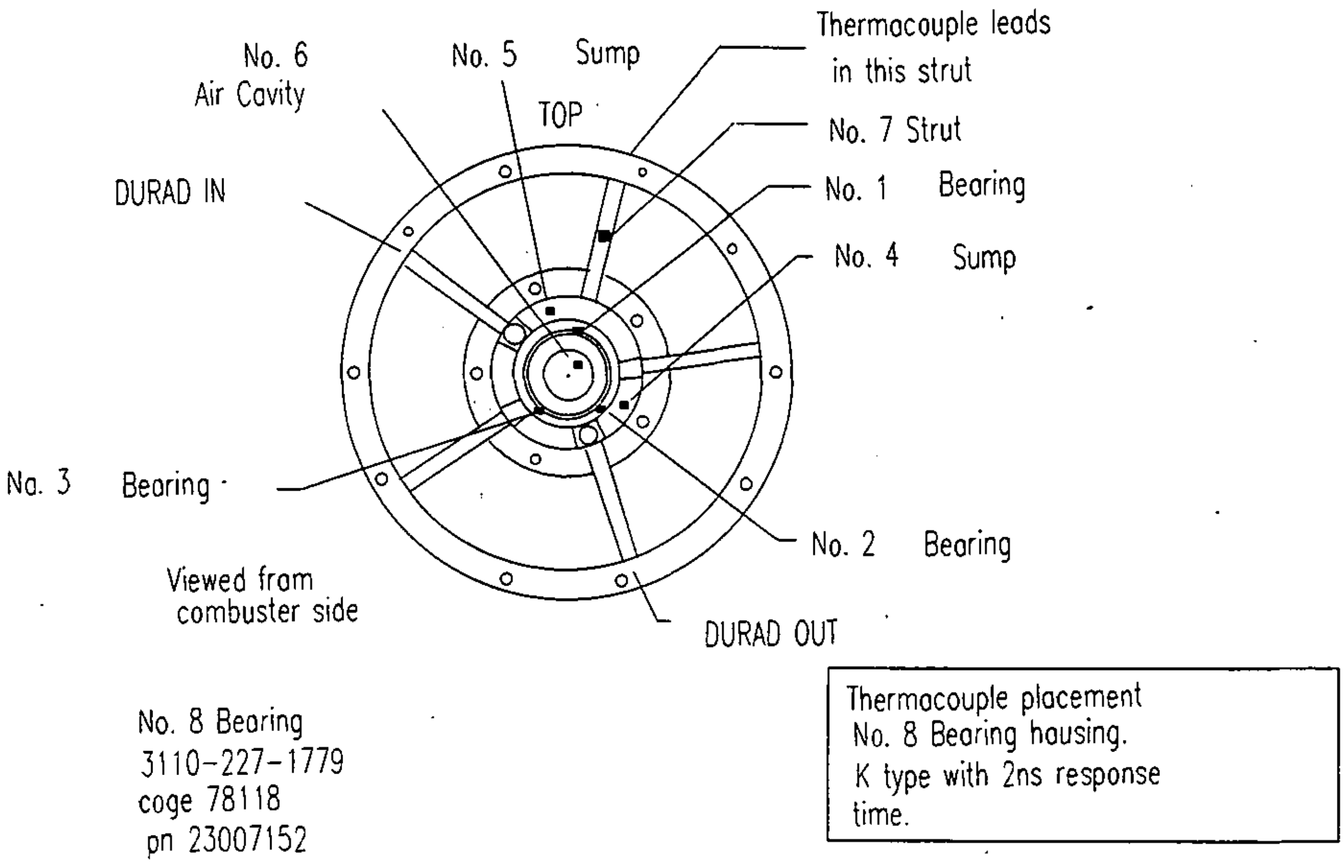

Figure 3 T-63 #8 Bearing Housing Instrumentation

Another of the struts was modified to allow seven K-type thermocouple leads to pass through to the bearing and bearing housing. Three thermocouples were in contact with the bearing outer race. Two thermocouples were placed in the return sump cavities around the bearing to monitor the fluid temperatures after the fluid passed through the bearing but before it exited the bearing housing. Another thermocouple was located in the air cavity between the heat shield and the bearing sump cap. The interior strut temperature was also monitored to determine if the fluid temperature increased prior to reaching the bearing as a result of passing through the strut. Lubricant inlet and outlet temperatures were measured just prior to entering and just after exiting the bearing housing.

Prior to using VPL in the engine, the bearing temperatures using the conventional oil lubrication system were determined. The engine was operated under various load conditions (ground idle, flight idle, and full throttle at maximum continuous operation) to completely characterize engine performance and to have sufficient data to compare conventional lubrication operation with VPL.

After the baseline oil tests were accomplished, the #8 bearing was isolated from the engine oil system by coupling the #8 oil inlet line to the exit line which completed the oil system. The oil intake line was replaced with a line from the mister. The mister, an Alemite Model 4955, was attached to the test rig. A tertiary-butylphenyl phosphate (TBPP) lubricant, DURAD 620B, was chosen as the lubricant for this test. The lubricant was preheated to 93°C to reduce its viscosity and allow better misting. The mister used 0.00066 m^3/s of shop air at 1.72 bar (gage) and was set to supply 13ml of lubricant per hour, according to research done during the High Speed Bearing Rig tests. To increase the flow of the mist to the bearing, the highly restrictive oil nozzle in the bearing compartment was removed and replaced by one with a larger diameter (3 mm) vapor injection nozzle which permitted greater mist. After passing through the bearing, any unused lubricant was passed directly into the engine exhaust. Toxicology studies conducted at the Armstrong Lab determined that DURAD 620B is nontoxic in liquid form. Tests of decomposition rates were conducted at Wright Laboratory. At vapor temperatures representative of exhaust gas temperatures, slow decomposition rates were measured, resulting in low toxicity at these temperatures. Venting the vapors to the engine exhaust is safe since the quantity of lubricant, 13 ml/hr, is small and consequently poses no significant environmental impact when diluted in the exhaust air stream.

THEORY

Tests with the oil lubrication system provided the base line data for comparison with the vapor-phase test. In the liquid lubricant tests it was determined the bearing operated at a steady state condition of 83°C at the ground-idle condition. Knowing this temperature and the temperatures of the other thermocouples in the bearing compartment, it was possible to develop a model to predict what the bearing temperature would be using a VPL system at the same engine operating conditions. Knowing the temperature difference of the oil across the bearing (inlet and exit), the flow rate of oil, and the specific heat of the oil at the average temperature, the total energy absorbed by the oil can be calculated:

$$\dot{Q}_{oil} = \dot{m} C_{p_{oil}} (T_{exit} - T_{inlet})$$

The heating rate was found to be 582 W at ground-idle conditions. This value was used to calculate a characteristic heat loading coefficient, h_{load}. This coefficient is simply the energy transfer rate to the oil divided by the difference between the temperature of the air cavity surrounding the bearing and the temperature of the bearing race:

$$h_{load} = \frac{\dot{Q}_{oil}}{(T_{bearing} - T_{surrounding})}$$

The energy transport mechanism for the vapor-phase flow is not as great as in the liquid case. Energy transport is expected to be a function of both the thermal conductivity and heat capacity of the fluid. Comparing VPL to oil lubrication, the thermal conductivity and the specific heat of air are less than that of oil ($k_{oil}/k_{air} \sim 10$, $C_{poil}/C_{pair} \sim 2$). As a result, the energy transport from the bearing to the fluid is less for VPL than for the liquid lubrication case. In reality, h_{load} is expected to be a lower value. It was optimistically assumed that h_{load} would be the same for the VPL case as that of the oil. With this assumption it was possible to develop an equation to predict the temperature of the bearing. The energy pickup in the VPL case is given as follows:

$$\dot{Q}_{vap} + \dot{Q}_{air} = h_{load} (T_{bearing} - T_{surrounding})$$

The energy absorption due to vaporization was found to be several orders of magnitude smaller than the energy transfer rates caused by the air and therefore is considered negligible in the remainder of the calculations. The mass flow of air was calculated as 0.00081 kg/s from the specifications set forth in the mister user's manual. The VPL bearing temperature was predicted using the following equation:

$$\dot{m} C_{p_{air}} (T_{exit} - T_{inlet}) = h_{load} (T_{bearing} - T_{surrounding})$$

T_{inlet} and T_{exit} are the temperatures of the vapor as it enters and leaves the bearing assembly. The value of T_{inlet} was assumed to be the same as the temperature of the lubricant in the mister, 93°C. From the oil system data and the equation for energy pickup for the VPL system with air only, it was possible to make estimates for T_{exit} and $T_{surrounding}$. In the oil lubrication tests, T_{exit} was found to be approximately the mean of the bearing temperature and T_{inlet} at each operating point. Likewise, $T_{surrounding}$, the temperature of the area surrounding the bearing, was found to be approximately the mean of the turbine intra-stage gas temperature, T_{TS}, and the compressor bleed air temperature, T_{T3}, which is used to cool the bearing housing. In predicting bearing temperature under VPL operation, $T_{surrounding}$ and T_{exit} are therefore defined as:

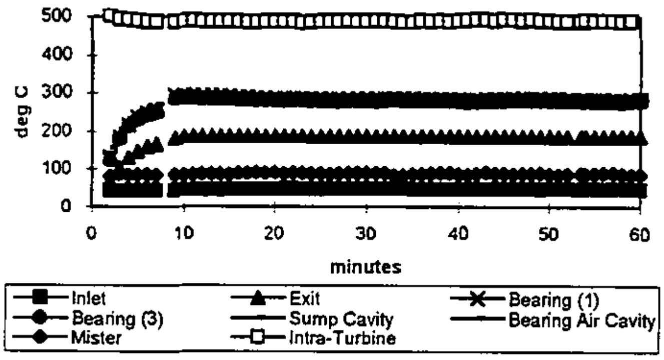

Figure 4 T63 Temperature Time Histories

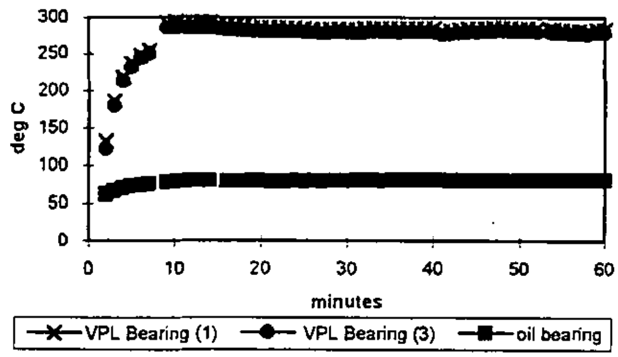

Figure 5 VPL and Oil Bearing Temperature Comparison

$$T_{surrounding} = \frac{T_{T5} + T_{T3}}{2}$$

$$T_{exit} = \frac{T_{bearing} + T_{inlet}}{2}$$

Using the data from the oil system test, a bearing temperature of 306°C for the ground-idle condition was predicted for VPL operation. This temperature was well below the 379°C operating limit determined from the bench tests, so it was concluded that limiting engine operation for the VPL test to ground idle would permit a safe test with reasonable risk. Although differences in the transport mechanisms involved with liquid and vapor lubrication were not considered in the h_{load} value, the $T_{bearing}$ calculated is relatively insensitive to differences in h_{load}. If h_{load} is decreased by 83% the calculated temperature of the bearing increases by only 2%. This provided confidence in the predicted bearing temperature, despite the uncertainty in the assumptions.

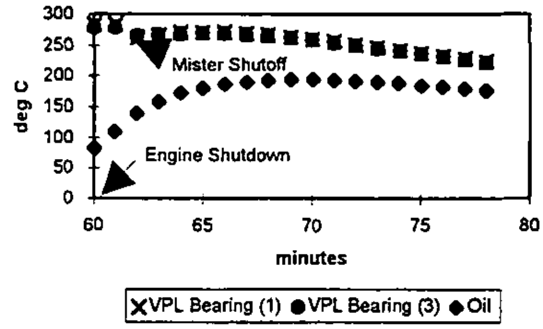

Figure 6 T63 #8 Bearing Soak Back Temperatures

EXPERIMENTAL RESULTS

Engine Test Results

The first steady state test of a VPL system in a turbine engine was conducted on 19 March 1996 in the test cell of the United States Air Force Academy Aeronautics Laboratory. The test lasted 60 minutes and was conducted with the T63 engine at ground-idle. The results are given in Figure 4. The first 13 minutes of the test showed the bearing temperature rising steadily as it reached equilibrium. The VPL bearing reached a state of equilibrium at 283°C, a temperature somewhat below that predicted by the model. Figure 4 compares the temperature profiles of the bearing race, the bearing sump cavity, the bearing compartment air cavity, vapor inlet and exit ports, and the intra-turbine temperature, T_{T5}. This data showed that, as expected, the bearing operated at temperatures very close to the temperature of the bearing air cavity. The bearing air cavity and assembly are actively cooled by compressor bleed air, maintaining a temperature below T_{T5}. Figure 5 compares the bearing temperature of a liquid and vapor lubricated bearing. Once equilibrium was achieved, the bearing temperature remained virtually constant for the remainder of the test. A comparison of the VPL system with the liquid lubrication system showed little change in other engine operating parameters. Table 2 gives a comparison of important measured parameters in the #8 bearing at virtually the same operating condition. The small differences in torque and T_{T5} are not considered significant due to deviations in ambient air temperatures between runs.

PARAMETER	OIL	VPL
Torque (N-m)	35.2	34.8
TT5 (deg C)	497	491
Average #8 Bearing (degC)	83	283
#8 Bearing Oil Sump (deg C)	91	294
#8 Bearing Air Cavity (deg C)	67	274

Table 2 Comparison of oil and VPL systems at ground idle

Another item for consideration in the bearing is the soak back performance. Soak back occurs after the engine is shut-down and the lubricant no longer carries the energy away from the bearing housing. For this reason, normal turbine engines experience a large rise in bearing temperature for about 10 minutes after engine shut-down.

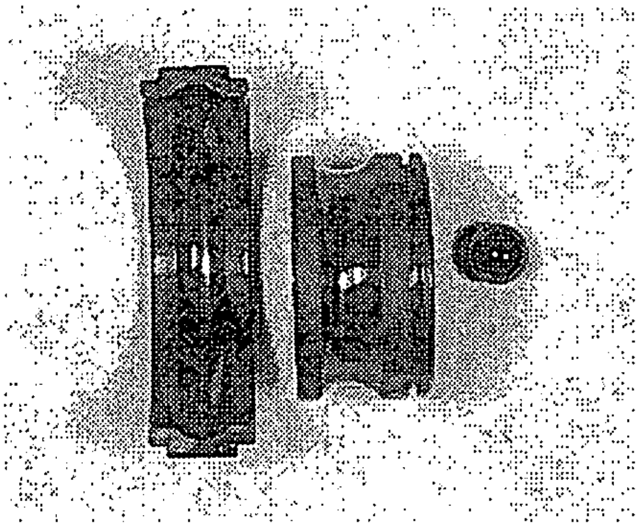

Figure 7 T63 #8 Bearing Outer Race, Inner Race, and Ball

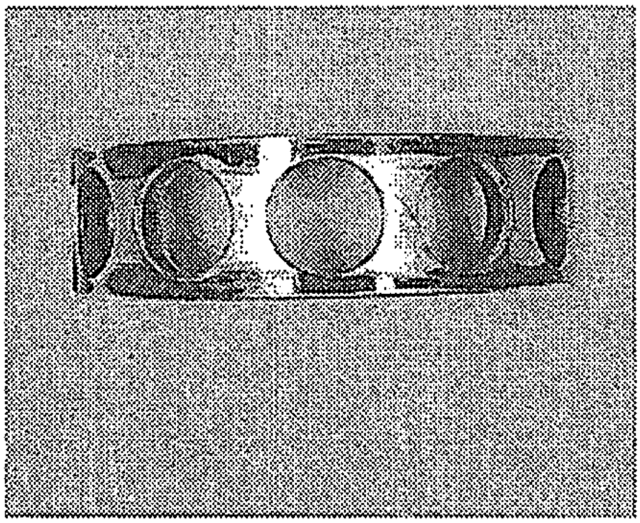

Figure 8 T63 #8 Bearing Cage Outer Land

Soak back affects the formation of deposits from the lubricant in a liquid lubrication system. For expendable engines, soak back is irrelevant since the engine is only intended to be used once, for short duration. Since the VPL bearing operated at extremely high temperatures, there was only a small thermal gradient between the bearing and its surroundings. Therefore, soak back was not experienced. Figure 6 shows the soak back for the #8 bearing after the oil and VPL tests. The VPL bearing shows a steady decrease in temperature with time, indicating it was in thermal equilibrium with its surroundings. The small dip in the VPL curve at the beginning of the soak back curve is due to the mister, which was still supplying some cooling air, before being turned off.

Bearing Analysis

After the engine test was completed, the bearing was removed from the engine and the outer race cut in half so the bearing could be disassembled and the components inspected. Three levels of inspection were performed. First, each component was visually inspected for defects at 1x to 10x magnification, according to overhaul shop procedures (ref. T.O. 44B-1-15, T.O. 44B-1-102). This was to determine if they would be considered serviceable in an operational aircraft engine. Next, each component was viewed at 120x to 600x magnification with a Scanning Electron Microscope (SEM) to evaluate surface finish and the geometry of any defects found. Finally, Energy Dispersive X-ray (EDX) measurements of surface elements were made to check for the formation of a deposition film, as indicated by the presence of phosphorous.

When the engine was disassembled for bearing removal, it was discovered that the tip of the vapor injection nozzle was worn. It appeared that thermal expansion of the nozzle had caused it to come in contact with the retaining nut on the bearing inner race. In general, this did not appear to affect bearing performance, but minor defects found in the balls and races have been attributed to nozzle debris entering the bearing.

The balls, outer race and inner race all passed visual inspection (Figure 7). They appeared smooth and light brown in color, with a slight wear track visible, which is normal. The land surface of the outer race showed signs of cage rubbing, but was still smooth and uniform. The only visible defect was a light scratch in the ball track of the inner race, which ran for one-third of the circumference. The scratch was not detectable when a 0.762 mm radius scribe was dragged across it, so the race was still serviceable. Under the SEM, at 150x magnification, the scratch revealed a series of indentations, characteristic of debris damage, as from the nozzle debris. The outer race and balls showed similar indentations in the wear track under the SEM. Phosphorous was present in the raceways of both races, indicating the formation of a deposition film.

The cage showed signs of wear at the outer land riding surfaces and in the ball pockets (Figure 8). The wear was limited to smearing of the silver plating. This is expected at marginal lubrication conditions at these areas of sliding contact. The outer land was worn for the entire circumference on the forward and aft sides of the ball pockets. The ball pockets were worn around their entire circumference for one half of the cage thickness. A definite ridge could be felt with a 0.762 mm radius scribe between the worn and unworn areas. Although the silver plating on the cage performed its designed function, the cage would not be acceptable for reinstallation in an operational engine. The EDX measurements of the worn areas revealed that a substantial amount of silver was still present. Phosphorous was also present, indicating the formation of a deposition film on the land and ball pocket surfaces.

CONCLUSIONS

This experiment demonstrated, for the first time, the ability to run a VPL system in an operating turboshaft engine and represents the first known tests in which a gas turbine engine was run without a liquid lubricant. VPL is a possible option for future use in the design of modern turbine engines, offering significant weight and cost savings, as well as significant improvements in engine performance due to the increase in temperature capability. For expendable class engines used in air-launched cruise missiles, the weight reduction of a vapor lubrication system compared to a liquid lubrication system is approximately 15 percent of the turbine engine weight. A similar reduction in engine cost could be expected.

Future VPL work will include the development of a self-contained mister system using engine bleed air. Tests will also include operation at conditions above ground idle, requiring careful monitoring of bearing temperatures so as not to exceed the 379°C temperature limit determined by the High Speed Bearing Rig tests. Future tests will also include alternative bearing materials allowing operation above this temperature limit. There is also a need to demonstrate the application of VPL in a bearing that operates at a colder temperature, such as the #1 bearing in the T63.

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