PERFORMANCE OF HIGH-TEMPERATURE BEARINGS LUBRICATED WITH SYNTHETIC LUBRICANTS

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ABSTRACT

Data presented from three test programs indicate no need for derating high-temperature bearings lubricated with synthetic oils. Significance of effect of retainer friction on bearing failures is demonstrated.

In the development of any new material to meet advanced requirements, it is advantageous to make use of basic material characteristics and simulating tests in order to avoid the time and expense involved in full-scale testing. Nevertheless, in the end it is necessary to evaluate the results of screening work by the use of full-scale tests in which the environmental effects are controlled as closely as possible to equal the representative conditions to be met in service.

In the development of high-temperature lubricants, all standard chemical laboratory tests have been utilized, standard lubricant evaluation procedures have been used and even novel simulating apparatus have provided a reference scale on which to compare candidate lubricants.

These screening processes have reduced the number of candidate high-temperature lubricants to a quantity which can be accepted for full-scale testing. In order that this testing may be valid, a popular jet engine main shaft thrust location bearing has been selected and set up to operate at speeds equivalent to normal engine speeds. In the first program to be described, the bearing size selected is the MRC 9130-UK-18 bearing which is the No. 2 bearing of the General Electric J-79 engine. This bearing is manufactured of consumable electrode melted M-50 steel balls and rings and fitted with a cage or retainer of forged silicon iron bronze, silver plated. In its production version, an internal clearance of .0050 inches, corresponding to a contact angle of 25°, is incorporated.

Experience with existing MRC research test equipment was utilized as background to the design and construction of a four-bearing test machine as shown in Figures 1 and 2. Since limited quantities of experimental oils were to be made available for this test, a minimum quantity sump and oil circulation system was designed.

To facilitate replacement of test bearings, this machine was designed with separate bearing housings, complete with oil entry and drain ports as well as heater elements and thermocouples. These bearing housings were supported on ways to maintain linearity and still permit the outer portions of the housings to be pushed together under thrust loads, applied by calibrated annular load bellows abutting one external face of each pair of housings. Splined shafts join the two pair of test units together and connect with the drive gear box.

The four motors used in the system (main gear drive unit, test oil supply pump, test oil scavenge pump, gear drive lubricant pump) are wired in such a manner that if any one of the supporting motors fails, the drive motor will cut off. A pressure switch in the drive gear lube system also will shut off the drive motor if lube pressure drops. A pressure switch in the test oil supply system rings a warning bell if pressure in this system drops.

Each bearing package is surrounded by three 2450-watt heaters. Two of each group of three are controlled by a 3-position switch (high-low-medium-off). The third heater of each group is controlled by a Powerstat, permitting control from 0 to 100% of heater output. All of these controls are tied into a Speedomax controller-recorder. The O.D. temperature of each bearing is picked up by a thermocouple at one point and recorded on a Speedomax multipoint recorder.

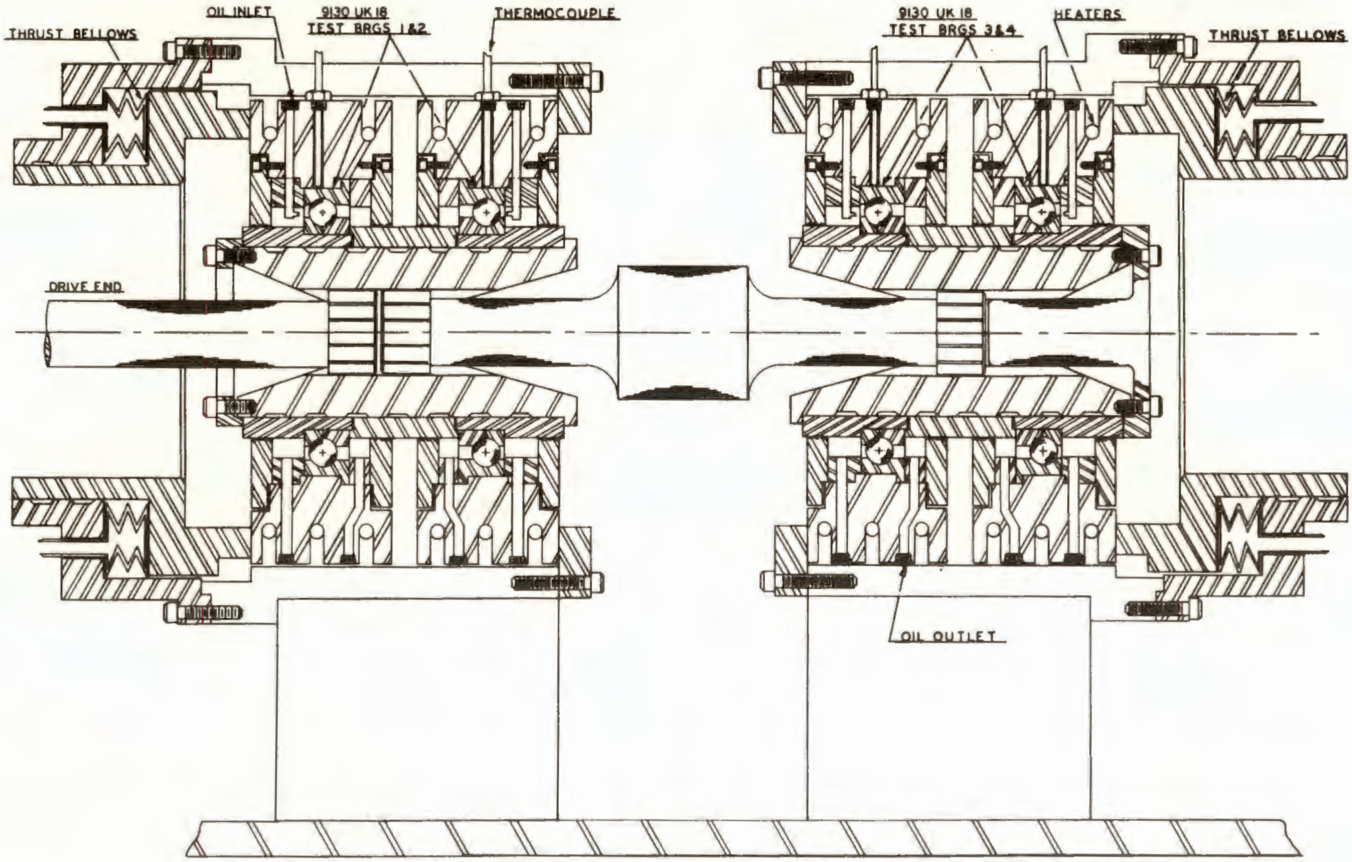

FIGURE 1. CROSS SECTION OF BEARING TEST RIG

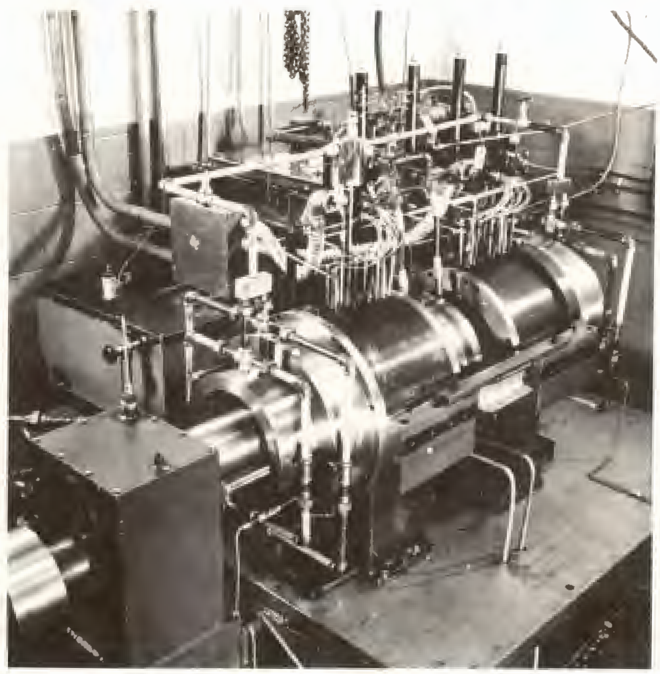

FIGURE 2. PHOTOGRAPH OF BEARING TEST RIG

To avoid local hot spots, the test oil is heated indirectly by two 2000-watt heaters in heat exchange oil surrounding the test oil sump. These heaters are controlled by individual 3-position switches in a control circuit monitored by a Speedomax recorder-controller. Oil sump temperature, temperature of surrounding oil bath, and temperature of oil-in to one bearing are recorded. In addition, the temperature of the oil as it leaves each bearing is also recorded. It should be noted that oil enters each bearing from one side but is drained from both sides. Thermocouples are installed so that oil temperatures are measured on the side opposite to oil input.

Vibration of various components of the rig has been monitored by a single-channel IRC Vibrotron. This instrument is basically a velocity pickup or accelerometer which measures amplitude and frequency of vibration.

Test conditions initially selected were based on the engine speed of 8000 rpm, with a thrust load of 13, 000 pounds, calculated to result in a B-10 life of 33 hours. At that time it was considered reasonable that the temperatures and lubricants to be used warranted a reduction in life to 40% of catalog rating life, or 13 hours. It was assumed that a test cut-off at 65 hours would provide sufficient failures to permit valid statistical analysis.

Oil samples were taken frequently, and checked for viscosity and acid number changes.

When started, this equipment demonstrated many problems, some of which are of significance to designers of bearing mountings as well as test equipment.

Bear in mind that the MRC 9130-UK-18 bearing, in its production version, operates quite satisfactorily in the J-79 engine. Post runnings examination of bearings from this engine indicates that the ball path location is normal for the thrust applied. There is no evidence of decrease in internal clearance due to thermal effects.

However, when these bearings are mounted in the relatively rugged housings of our test machine, we find that the internal clearance is definitely marginal even using outer race pre-heating, gradual loading, and step acceleration to speed. During the rig shakedown period, the internal clearance of the bearing was increased by reduction in ball size until the bearing temperature could be stabilized and the ball path maintained in its proper location in the race. In order to accomplish this, it was necessary to reduce the ball size .005" below that of the production bearing version. This represents an increase in internal clearance of .010". Even with this extraordinarily large internal clearance, it would be possible to cause the bearings to become radially preloaded by the introduction of full load without pre-heating the outer races, if the unit were brought up to speed and load too quickly.

After an extended shakedown period, testing was started, utilizing oil supplied by WADD, coded as GTO-765. During this phase, oil-in temperature was held at 400°F, and bearing O. D. temperature at 525°F. Only one 19. 6-hour bearing test was conducted using this oil. It was found that more than 50% of the oil was lost due to volatilization, and acid number and viscosity determinations indicated severe deterioration of the oil. The test bearings revealed no sign of distress or impending failure after this 19. 6 hours of operation.

The next oil tested was coded as GTO-790. Four tests were conducted on this lubricant with various bulk oil and bearing temperatures. It was found that, even with a reduced bulk oil temperature of 375°F and bearing temperature of 475°F, rapid deterioration of the oil took place and more than 50% was lost due to volatilization in less than 28 hours of operation. On the basis of this experience, all further tests of this oil were conducted using a bulk oil temperature of 365°F and a bearing temperature of 475°F. Make-up oil was added, as needed, every 5 hours of operation. No bearing fatigue was experienced in any of the four tests.

Under these revised test conditions, one 65-hour bearing test was run on GTO-790 lubricant. During this 65-hour test run, a total of 7. 8 gallons of oil was added and it was estimated that 56% of the total was lost due to volatilization. At the end of the test run, examination of the test bearings still revealed no indication of impending failure. In view of this experience, in an attempt to promote fatigue failures, a new set of temperature conditions were prescribed, involving an oil-in temperature of 400°F with a bearing temperature of 525°F.

The next oil to be tested was received under codes GTO-861, GTO-885, and GTO-915. Two 65-hour bearing tests were conducted with GTO-861 lubricant. During these tests 12.0 gallons of oil was added and approximately 60% of the total oil was lost due to volatilization. The third 65-hour bearing test was conducted with GTO-861 and GTO-885, mixed. Fourteen gallons of oil were added and approximately 73% of the total oil was lost due to volatilization. During this testing, one spalled inner race failure was obtained at 66.9 hours, and three complete catastrophic failures were encountered. These catastrophic failures were traced to marginal internal clearance, due to temperature and lubricating properties of this oil.

Up to this point, the test procedure had demonstrated its ability to deteriorate oil but had failed to produce sufficient fatigue life data. Therefore, it was decided to increase the thrust load from 13,000 pounds to 18,000 pounds, corresponding to a catalog B-10 life of 12 hours. An oil-in temperature of 400°F, with this load, results in oil-out temperatures in excess of 625°F, and bearing temperatures, without external heat added, of approximately 550°F.

The next test lubricant was coded GTO-915. The first test run on this oil resulted in simultaneous failure of all four test bearings in 1.3 hours. Again these catastrophic failures were traced to marginal internal clearance, aggravated by the load increase. To correct this condition the internal clearance of the bearing was increased .002 inches, to a total of .017 inches.

Further, on GTO-915, one 80-hour and two 65-hour oil tests have been conducted. As is to be expected from the increased bearing and oil-out temperatures due to increase in load, an increase is noted in volatilization. During the 65-hour test, 24.2 gallons of oil were added and a total of 30.5 gallons of oil were added for the 80-hour test.

It was found at the end of the 80-hour test that there was an extreme oil sludge built up in the rig and on the bearing, necessitating a complete cleaning of rig and bearing. In view of this sludging tendency, in subsequent testing the oil was changed and the rig system completely cleaned every 65 hours.

During the testing described above with GTO-915, four spalling failures were accumulated out of a total of 13 bearings tested.

In addition to the four spalling failures, one retainer failure was encountered, and five tests were discontinued because the bearing temperature became gradually uncontrollable. Examination of these bearings indicated that heavy wear had occurred in the ball pockets and on the guiding lands (Fig. 3). Evidence was found that a curious ball condition was being promoted by high retainer friction, evidenced by the heavy wear. This ball condition is shown in Figure 4. This shallow failure is definitely different from classical fatigue at the layer of maximum subsurface shear. It is a failure of .002 - .003" thick skin layer, such as would be produced by skidding.

FIGURE 3. RETAINER FROM TEST 475

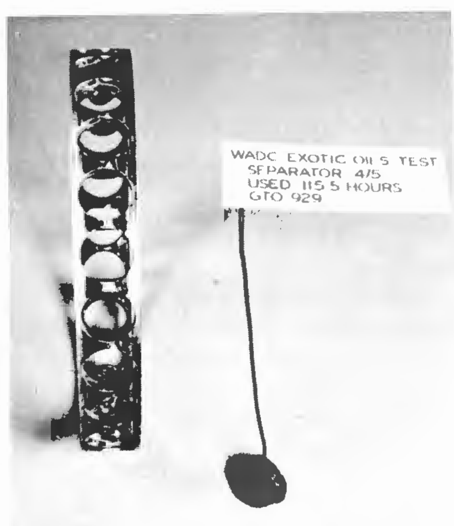

FIGURE 4. BALLS FROM TEST 477

The next oil tested was coded GTO-929. This oil demonstrates considerably less volatilization and, therefore, oil consumption than previous oils. Despite the fact that the oil-out temperature of 635°F is somewhat higher than the other oils tested, we have not experienced any appreciable sludging. Due to high room temperature viscosity, we have experienced some difficulty in pumping and are now heating the oil to 160°F before starting circulation.

To date 22 bearings have been tested in this oil, and 4 fatigue failures recorded. In addition to the four spalling failures, 9 tests were discontinued because the bearing temperature became gradually uncontrollable.

We found we could prevent ball damage by changing to new retainers at the first sign of temperature instability. In this way, testing was continued to 250 hours, cut-off.

It has long been felt that a dual function is required of a lubricant; reduction of friction and cooling. In the first of these, reduction of friction between the rolling element and cage or separator is required to prevent regenerative wear and friction heat. The material of the cage, and its plating, influence this requirement of the lubricant. The cages used in this test are standard in the MRC 9130-UK-18 bearing and in the No. 2 bearing of the GE J-79 engine. Visual examination does not indicate that this cage is deteriorated by the temperatures of these tests, increased though they are, over the J-79 engine environment. In those tests which have been terminated due to ball surface deterioration, the lubricant has not fulfilled the vital function of controlling the friction between the rolling elements and the cage. However, this program is continuing, to provide comparison of candidate lubricants on the basis of classical bearing fatigue.

The results of these tests are summarized in Figures 5, 6, and 7. On the basis of these data, it is indicated that the B-10 life in GTO-915 is approximately three times the catalog rated life and that in GTO-929 is over seven times the catalog rated life.

In addition to these highly encouraging tests, we have additional data from two other test programs which were run under engine conditions using MIL-L-7808C oil. Both of these programs were run under the sponsorship of the General Electric Company.

In the first of these programs, duplex pairs of MRC 9142-UD-100-T bearings were operated at 5000 rpm under a thrust load of 44,500 pounds per pair. This MRC 9142-UD-100-T bearing is manufactured of consumable electrode melted M-50 steel balls and races. The bearings tested incorporated cages or retainers of nonplated S-Monel or silver-plated forged silicon iron bronze. The test rig utilized for this program is shown in Figures 8 and 9.

The oil-in temperature maintained through this program is 300°F and the O. D. temperature of the bearings averaged 375°F. Figure 10 presents the results of this program. It will be noted that only two failures were achieved and that these were relatively early failures. Not only are the two failures statistically abnormal but a posthumous metallurgical inspection indicated metallurgical irregularity in the failed balls. We would be entitled to treat these data on discontinued tests more optimistically but even with the treatment indicated in the figure, we note that the B-10 life is approximately twice the AFBMA life, which is based on the use of mineral oil lubrication with conventional steels and at low temperature.

The second such test was programmed to demonstrate the reliability of a production engine thrust bearing. In this test, MRC 9109-UK-4 bearings were operated at 26,000 rpm under a thrust load of 1075 pounds, combined with a radial load of 260 pounds imposed by deliberate shaft unbalance (Figs. 11 and 12). The MRC 9109-UK-4 bearings are manufactured with rings and balls of consumable electrode melted MHT steel and are equipped with retainers of silver-plated forged silicon iron bronze. The bearings were lubricated with one gallon per minute of MIL-L-7808C oil supplied at 250°F. The bearing outer race temperature was 335°F. Since this program was established to demonstrate reliability, a cut-off point of twice the B-10 life was invoked. Of a group of 24 bearings, 20 were run to 160 hours without failure. The remaining four we allowed to run on to a duration of approximately 700 hours, without failure. Figure 13 presents these data, and demonstrates one method of statistical analysis which indicates a B-10 life of 6.88 times the AFBMA predicted life, which is based on mineral oil lubrication at conventional temperatures, with bearings of SAE 52100 steel (balls and races). All three of these programs result in lives

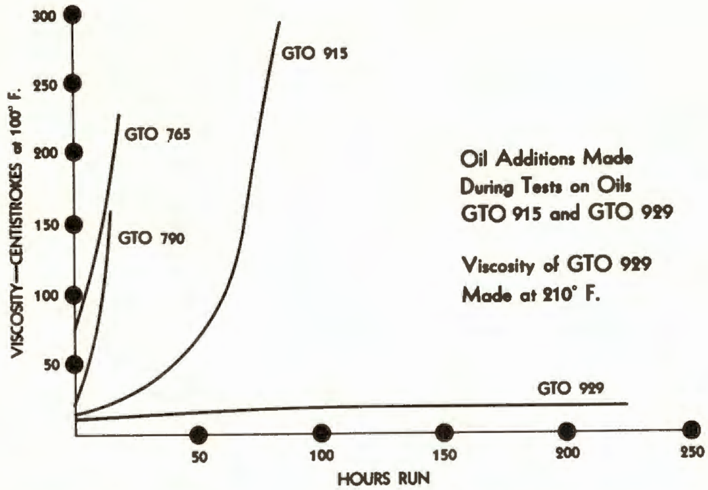

FIGURE 5. EFFECT OF RUNNING ON OIL VISCOSITY

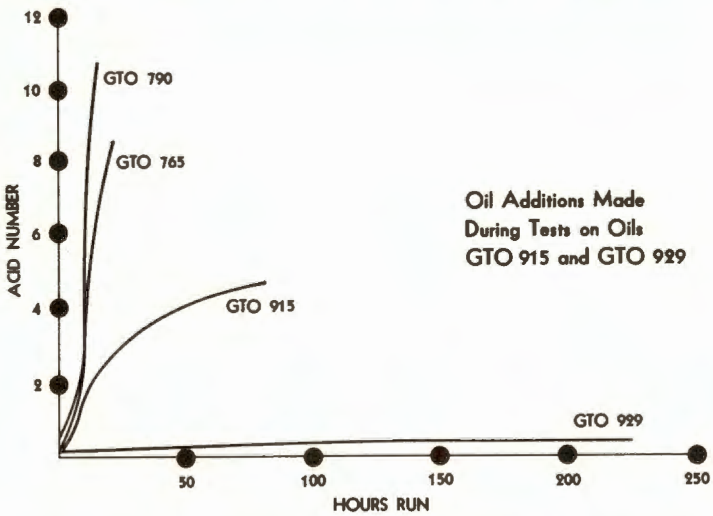

FIGURE 6. EFFECT OF RUNNING ON ACID NUMBER OF OIL

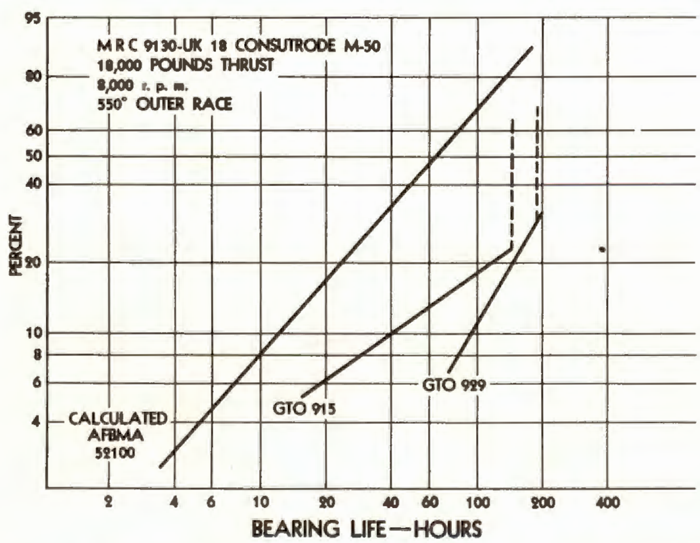

FIGURE 7. WEIBULL PLOT FOR GTO-915 AND GTO-929

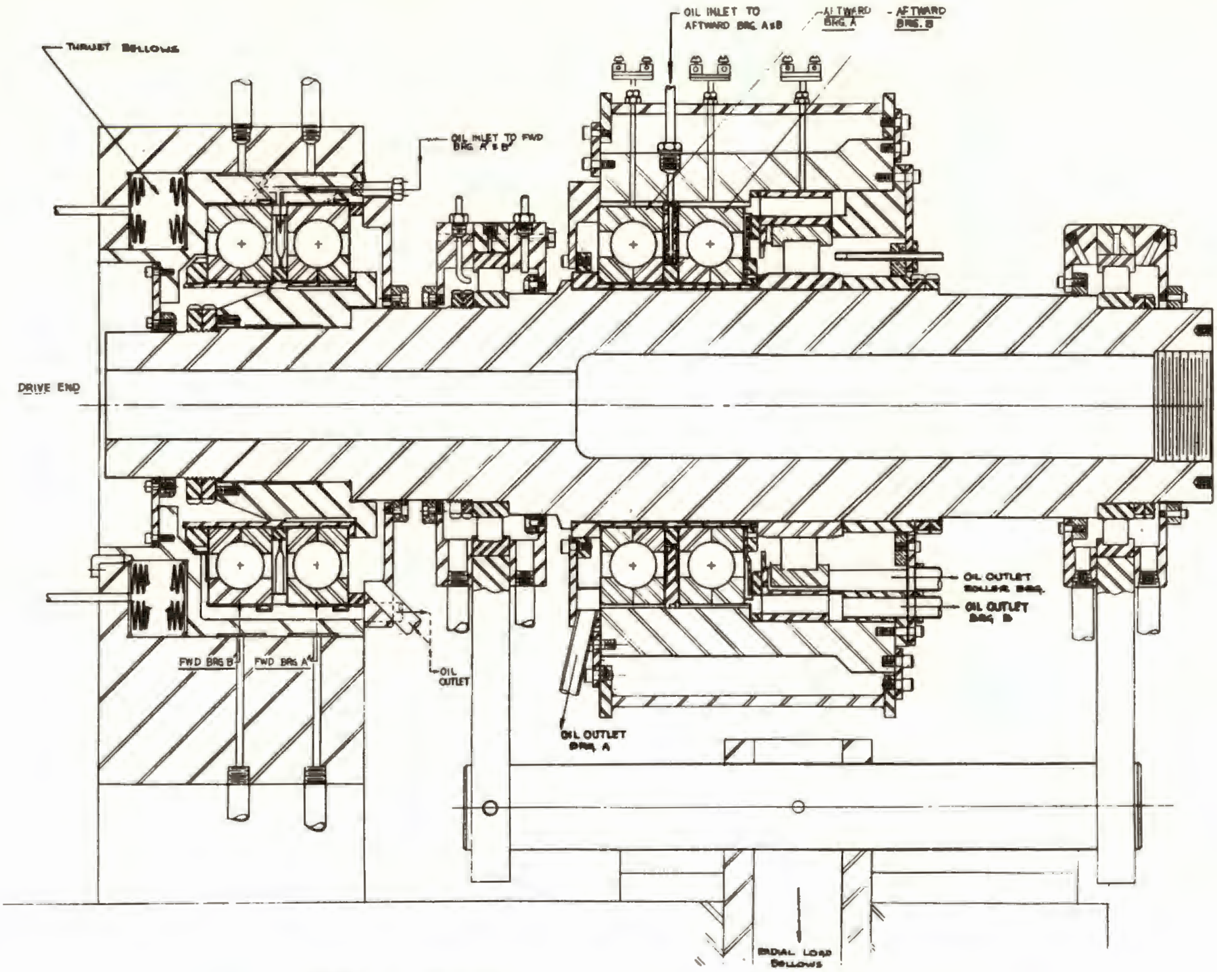

FIGURE 8. CROSS SECTION OF BEARING TEST RIG

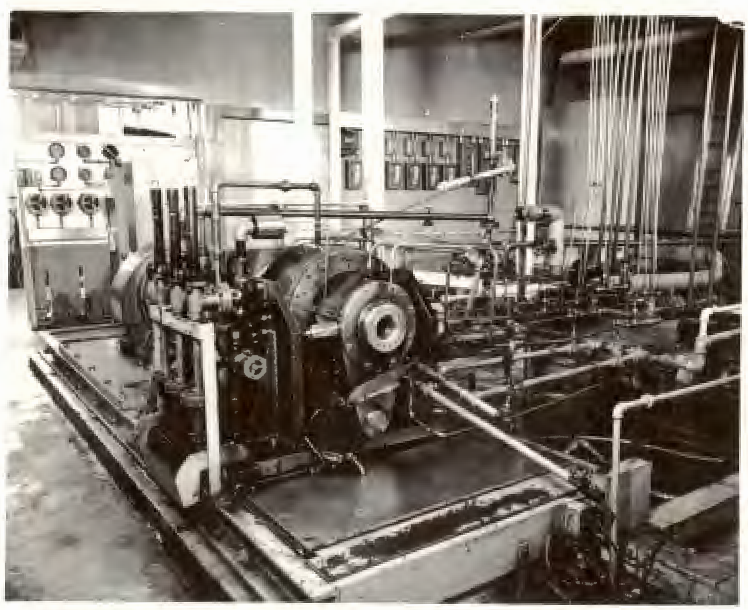

FIGURE 9. PHOTOGRAPH OF BEARING TEST RIG

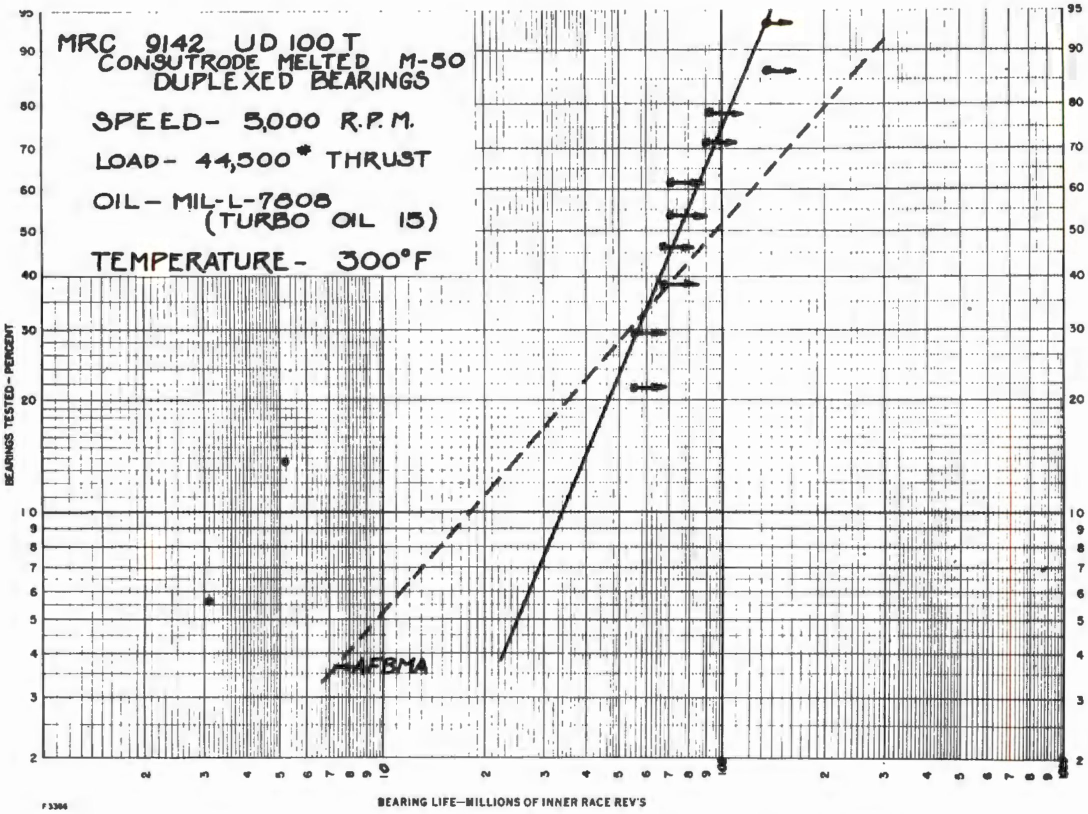

FIGURE 10. WEIBULL PLOT FOR MIL-L-7808 OIL

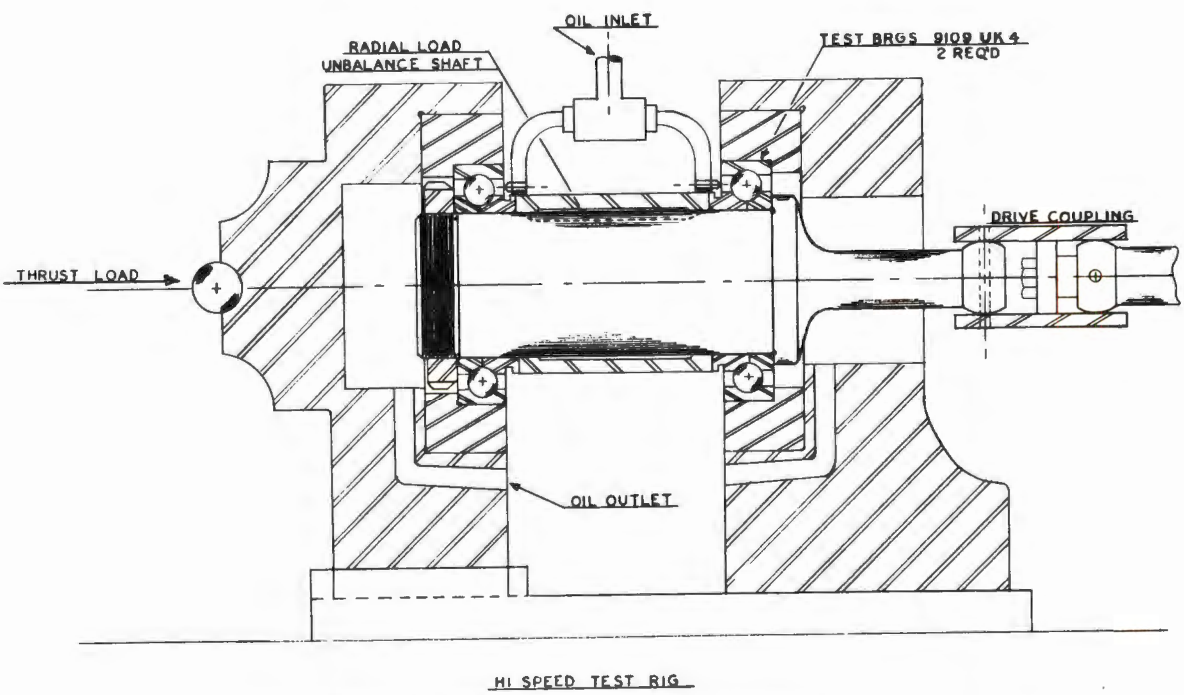

FIGURE 11. CROSS SECTION OF BEARING TEST RIG

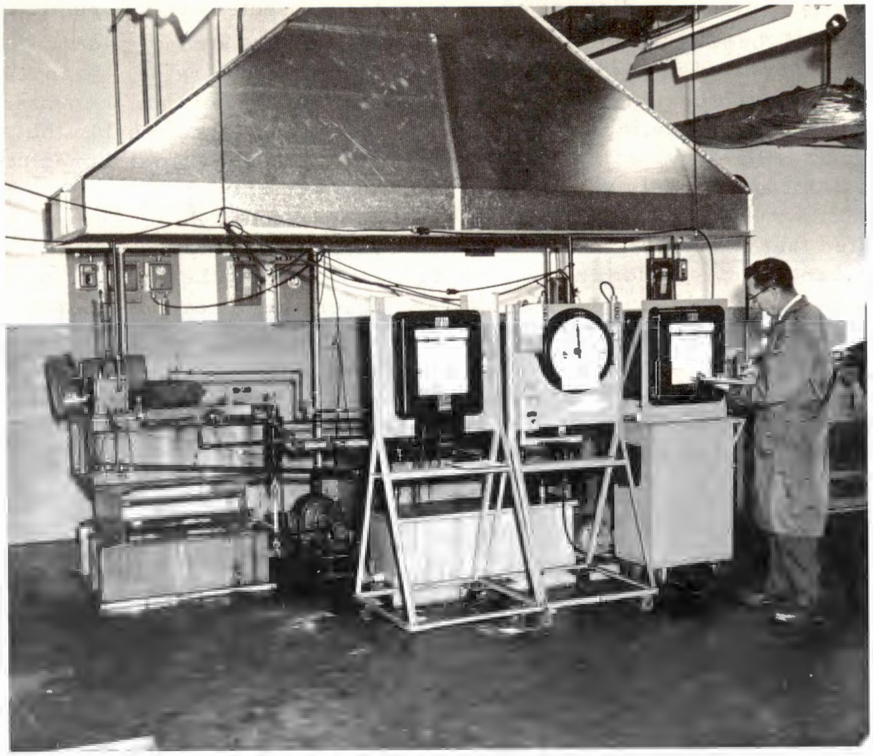

FIGURE 12. PHOTOGRAPH OF BEARING TEST RIG

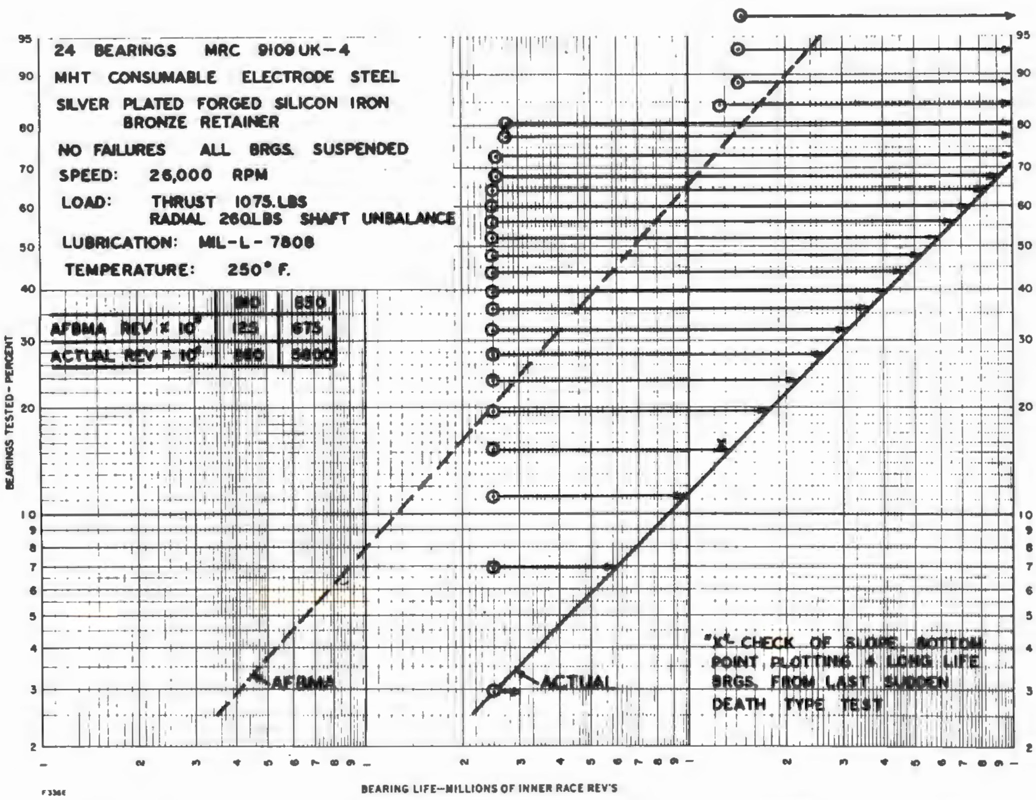

FIGURE 13. WEIBULL PLOT FOR MIL-L-7808 OIL

greater than the bearing catalogs would predict. Some of this is to be expected from the relatively high speeds at which all three tests were run. However, there is strong evidence to indicate that high-temperature bearings operated in synthetic lubricants need not be derated. It appears that the mechanism of failure of such bearings may be entirely different from that encountered in conventional bearing tests, on which catalog ratings are based. If this is so, it is probable that the distribution of failures, scatter or reliability, will also bear no fixed relationship to the predicted scatter for more conventional bearings and test conditions.

It is indicated that more work needs to be done to define the design criteria for bearing materials and lubricants. It is becoming increasingly obvious that the retainer or cage in high-temperature bearings is a highly critical member and deserves increased research and development attention in order to decrease friction for longer periods of operation.

Retainer-ball friction promotes bearing failure, often in the form of spalling. Therefore, it is an important requirement of the lubricant that it be able to prevent regenerative friction and wear in the ball retainer contact as well as in the area of the contact between the balls and races.