Development of Grease Lubricated Ball Bearings

Course No: M02-048
Credit: 2 PDH

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Introduction

This course deals with the development of grease lubricated ball bearings for aircraft turbine driven auxiliary equipment. It contains a plethora of information on the performance of grease lubricated ball bearings and their peripheral components for any application.

Ball bearings are used primarily to support rotating shafts in mechanical equipment. They can be found in the smallest electric motors to the largest pieces of mining equipment. They are of simple design and can be precision manufactured in mass production quantities. They can support a whole spectrum of loads over a wide speed range and do it virtually free. They come in many different sizes and shapes, are relatively inexpensive, and require little or no maintenance. They have predictable design lives and are truly a valuable asset to the rotating equipment industry.

Ball Bearing Design and Use

A ball bearing consists of an inner ring (IR), an outer ring (OR), a complement of balls, and a separator. The inner ring outer diameter (IROD) and the outer ring inner diameter (ORID) have a groove in which the balls rotate. The groove is commonly called the pathway. The raised surface on each side of the pathway is called the shoulder. The balls are held equally spaced around the annulus of the bearing by the separator.

The basic dimensions of the bearing are the bore (B), outside diameter (OD) and the width (W) (See Figure 1). In most applications, there are two ball bearings supporting a rotating shaft. The shaft can be either straddle mounted or overhung mounted. Loads or forces are imposed on the bearings by the equipment that is driving and/or being driven by the shaft. The loads can be separated into a radial component that acts 90 degrees to the shaft centerline and a thrust component that acts along the shaft centerline (See Figure 2).

Figure 1

Ball Bearing Terminology

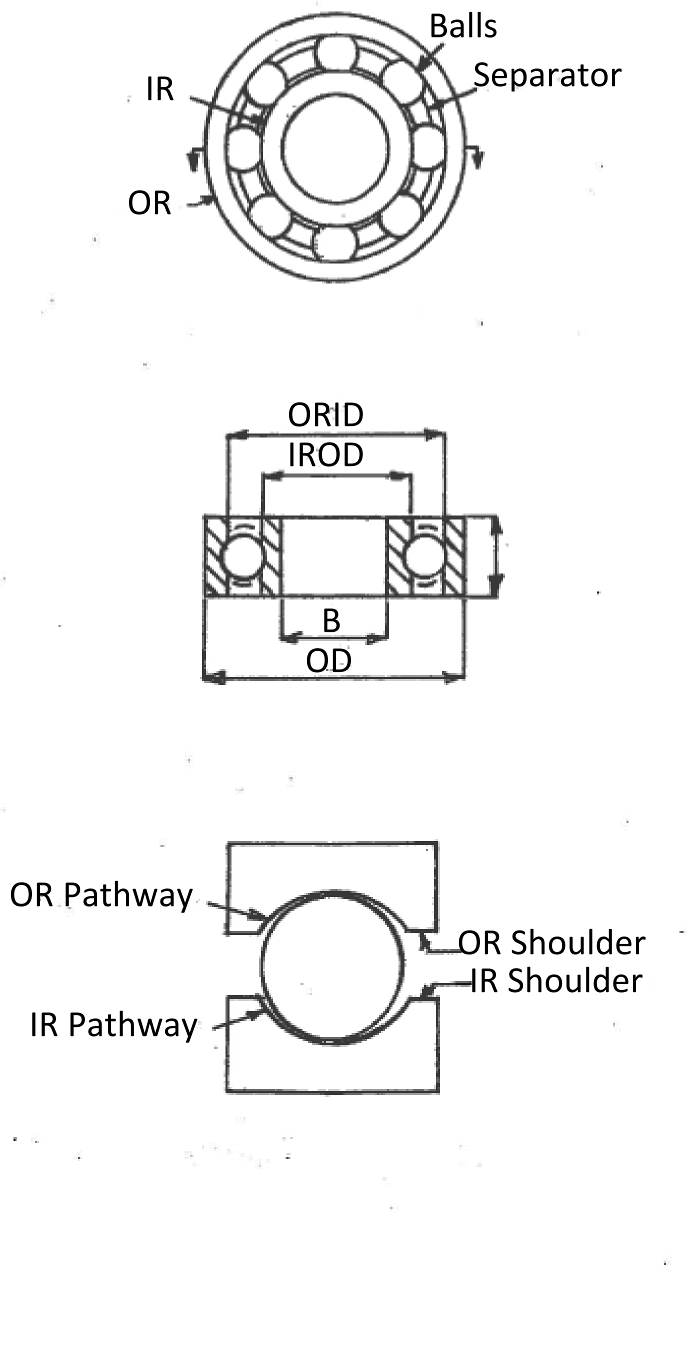
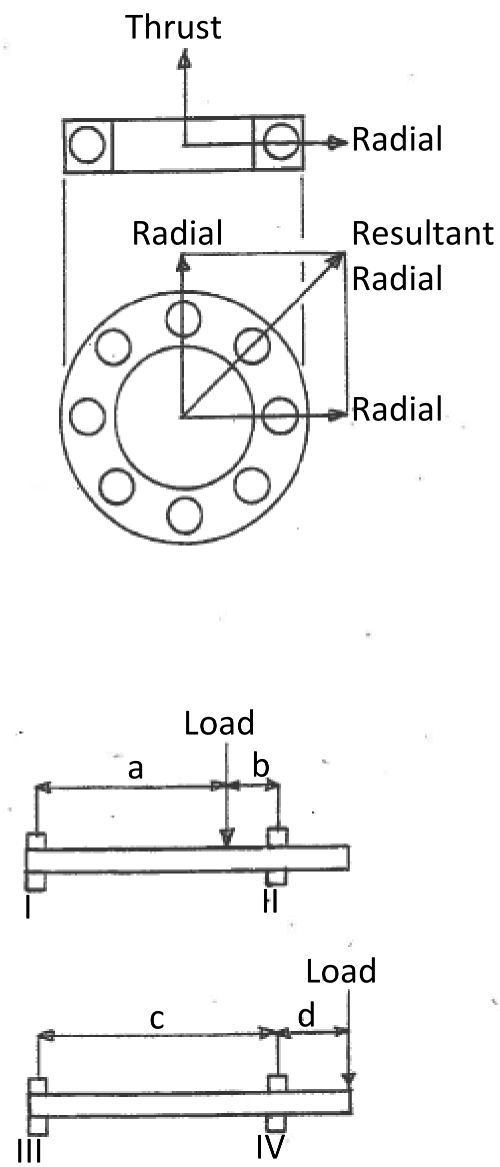


Figure 2

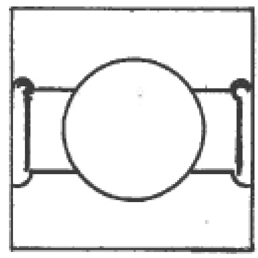
Ball Bearing Loads



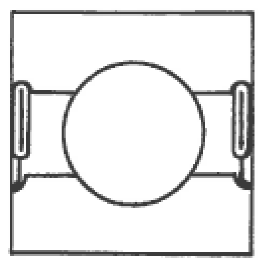
Single row radial ball bearings can be furnished with seals or shields. Seals and shields are closures that are used to retain lubricant inside the bearing while, at the same time, prevent contaminants from entering the bearing. Seals are constructed of a thin layer of rubber covering a thin metal stiffening disc. Seals are assembled on the bearing by snapping into a bearing outer ring inner diameter groove while the inner ring rubber lip firmly contacts an inner ring outer diameter notch. The seal lip is intricately designed to seal in lubricants and seal out contaminants. Some seals have more than one lip to provide even better sealing ability. In some instances shields can be used in a grease lubricated bearing. The function of shields is the same as seals except shields operate without the friction created by the seal's contacting rubber lip. Another important design feature of ball bearings is that they can be provided with an outer ring snap ring for retention in housings (See figure 3). Bearings can be furnished grease lubricated and double sealed providing a very compact and independent package and run maintenance free for their entire design lives. This gives the machine designer a great advantage in not having to provide an external means of lubricating and sealing ball bearings.

Figure 3

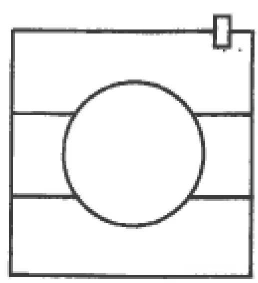
Ball Bearing Attachments



Seals



Shields



Snap Ring

Development of Grease Lubricated Ball Bearing

This course is based on development testing of high speed grease lubricated ball bearings for smaller turbine-driven aircraft accessories such as compressors, starters and pumps. Wick lubrication, employed on many of these aircraft accessories, necessitates carrying a considerable amount of oil-soaked packing in an enlarged housing and running oil feed wick to slingers positioned adjacent to each bearing.

If a compact, high speed grease-lubricated bearing package could be perfected, a substantial cost and weight savings would be realized in the design of these units. It is also anticipated that the same bearing package would have applications in many other areas of engineering because of the never ending universal trend toward cost and weight reduction.

The purpose of the test is to develop an aircraft accessory ball bearing that will meet all design requirements with grease as the lubricant. New improved greases have been developed that are thought to have good potential in providing lubrication for high speed aircraft accessory ball bearing application.

The overall goal of this two-part test program is to develop a grease-lubricated aircraft accessory ball bearing that will operate for 50 hours at one million DN. DN is the diameter of the bearing bore in millimeters multiplied by the inner ring revolutions per minute (rpm).

Test Procedure I: Grease Lubricated Ball Bearing Speed Testing

This first part of the program is to determine the limiting speed of the aircraft version of the basic 3204 ball bearing operating with three different existing design separators and two different greases.

The basic 3204 ball bearing is classified as a light series bearing as opposed to a 3304 size bearing which is classified as medium series, and a 3L04 which is an extra-light series bearing. The 3404 is classified as a heavy series ball bearing but is not commonly used. The 3204 has a bore of 20 millimeters (.7874 inches), an outside diameter of 47 millimeters (1.8504 inches), and a width of 14 millimeters (.5512 inches). The radial capacity of the 3204 ball bearing is 760 pounds which is the radial load on the bearing, which when operating at 500 RPM, is expected to have a survival rate of at least 90% of all bearings tested.

Three different ball bearing separators and two different greases will be tested to determine their limiting speed of operation. The separators tested are ribbon (RB), phosphor-bronze (PB) and non-metallic (NM).

The ribbon is the standard production separator and consists of two stamped thin metal pieces that are positioned on each side of the bearing against the balls. One of the pieces has tabs that clamp to the opposite piece between ball locations. The ribbon separator is classified as a ball-controlled separator, meaning that the radial movement of the separator (which is very small) in the bearing is controlled by the balls.

The phosphor-bronze separator (ATB) is a one piece thin metal stamped design that has radial pockets that contain the balls. The aircraft version of the basic 3204 bearing has one inner ring shoulder removed which allows bearings with one piece phosphor-bronze separators to be assembled with the balls pre-positioned in the separator. The phosphor-bronze separator is an outer ring inner diameter controlled separator.

The non-metallic separator is a one piece molded or machined ring of phenolic resin cotton laminate with radial holes for ball placement. It too is an outer ring inside diameter controlled separator (See Figure 4).

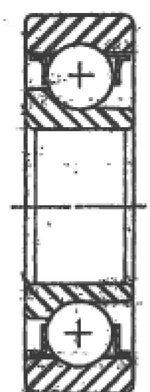
The first grease (G1) is a No. 2 grease with a synthetic hydrocarbon oil, a bentonite clay thickener, and a temperature range of -80°F to +300°F. The second grease (G2) is a No. 2 grease with a mineral oil, a polyurea thickener, and a temperature range of -30°F to +300°F.

All testing is conducted in a laboratory where an air-operated turbine drives two test bearings simultaneously; and, for this test, each under a 50 pound thrust load. Bearing pairs are run at increasing speed increments of 5,000 rpm with outer ring temperatures allowed to stabilize (15 to 45 minutes) at each increment before proceeding to the next higher speed. Bearing failure results in almost immediate loss of operating speed because of the low inertia characteristics of the turbine drive system. Because of this almost immediate shutdown after failure, bearing damage is held to a minimum (See Figure 5).

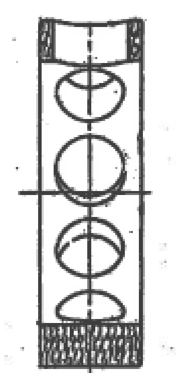
Figure 4

Ball Bearing Separators

Ribbon Separator



Phosphor-Bronze Separator

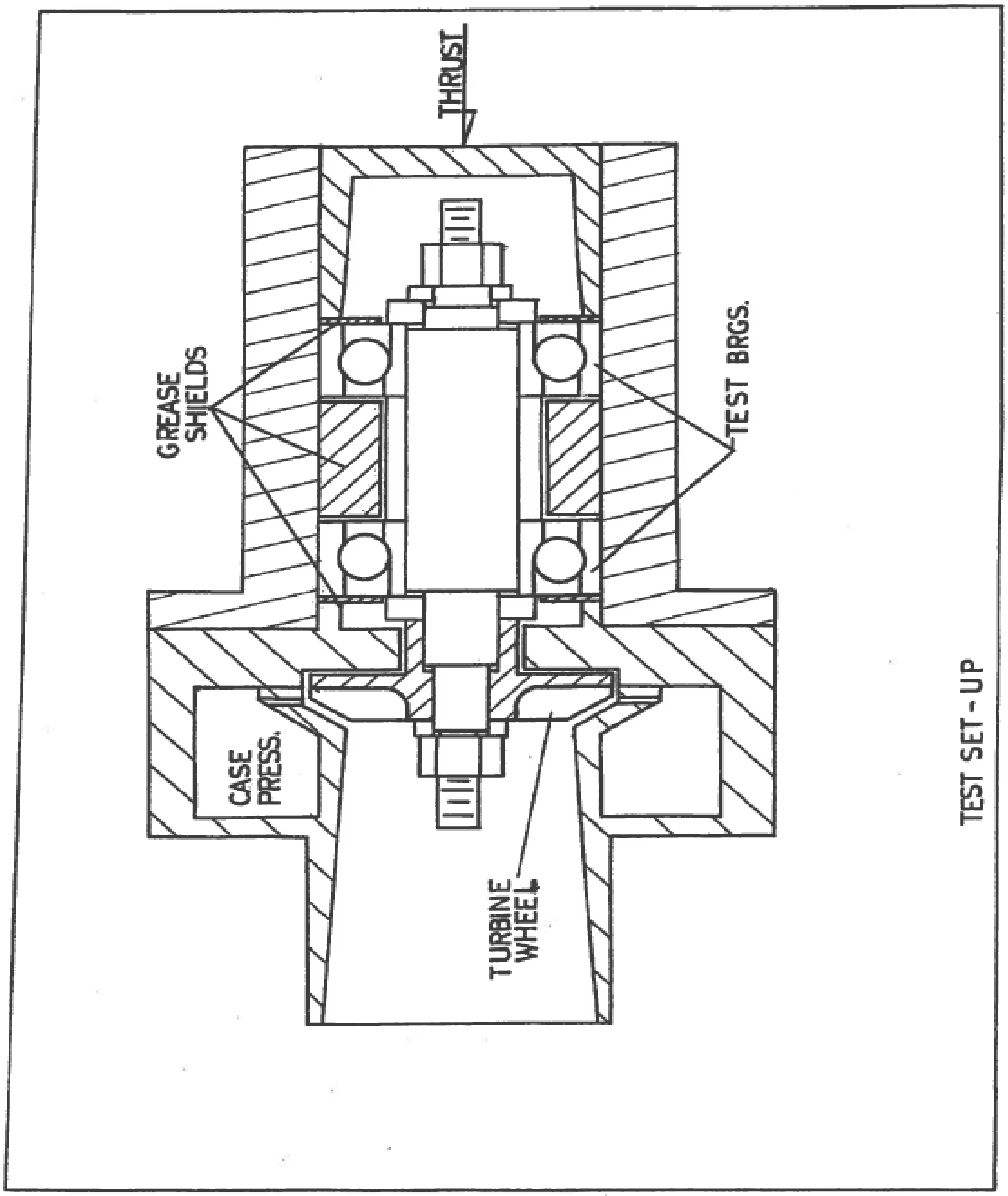


Non-Metallic Separator

For each test, the two bearings are matched for radial play and packed 40% with the test grease. (Radial play is the movement of one ring in the radial direction with respect to the other ring.) The complete rotating assembly, before the addition of grease, is balanced to 90 micro oz-in. This small amount of unbalance results in just one pound of radial load at the maximum operated test speed of 50,000 rpm (one million DN for the 3204 ball bearing with an inner ring bore of 20 millimeters). No provisions are made for assembling seals or shields to the test bearings; therefore, makeshift guards are used which somewhat simulate shields. (See on Figure 5). During testing, continuous recording is made of bearing speed, outer ring temperature, and drive turbine case pressure. Three tests are run for each separator-grease combination.

Figure 5

Test Set-Up



Test Results: Bearing limiting speed were established as the point on the temperature versus speed curve where the plot started to deviate from a straight line and curved sharply upward. This can be seen in Figures 6 through 11 for all separator-grease combinations tested. The results are as follows:

Separator	Grease	Limiting Speed
Nonmetallic	G1	1.20 million DN (60,000rpm)
Nonmetallic	G2	1.05 million DN (52,500rpm)
Ph.-Br. (ATB)	G1	1.05 million DN (52,500rpm)
Ph.-Br. (ATB)	G2	0.95 million DN (47,500rpm)
Ribbon	G1	0.60 million DN (30,000rpm)
Ribbon	G2	0.60 million DN (30,000rpm)

Separator Comparison: Figure 12, which is a summary of Figures 6 through 11, indicates that initially the ribbon separator bearings performed normally; however, after the speed of 30,000 rpm was reached, rapid temperature rise occurred. Post test investigation of ribbon separator bearings indicated that, above the limiting speed of 30,000 rpm, the normally ball controlled separator became eccentric in the bearing contacting the outer and inner rings. This condition appeared to rapidly accelerate ball-separator temperature causing bearing failure.

Figure 6

Non-Metallic Separator – G1 Grease

Performance Evaluation

Bearing Outer Ring Temperature (F)

Versus

Speed (RPM)

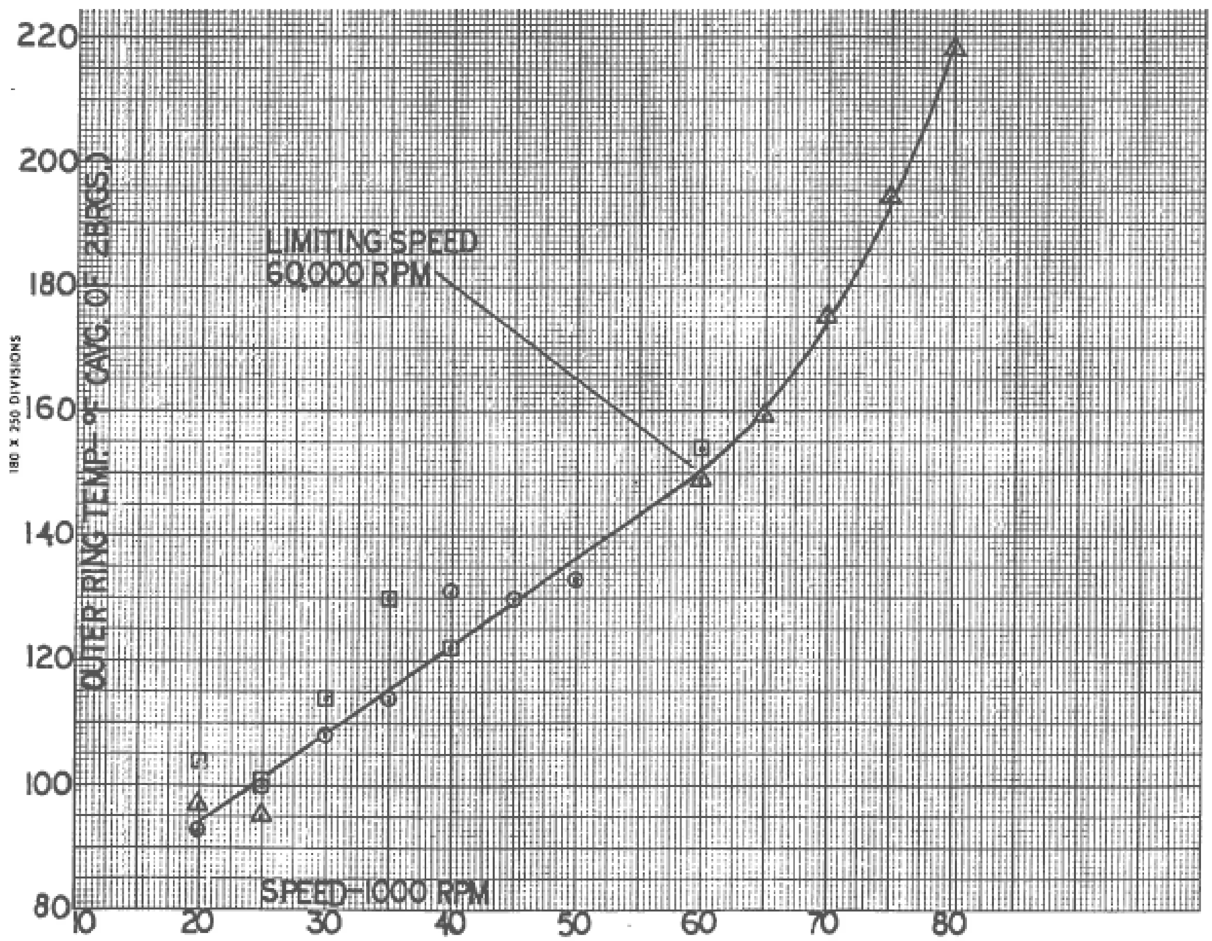


Figure 7

Non-Metallic Separator – G2 Grease

Performance Evaluation

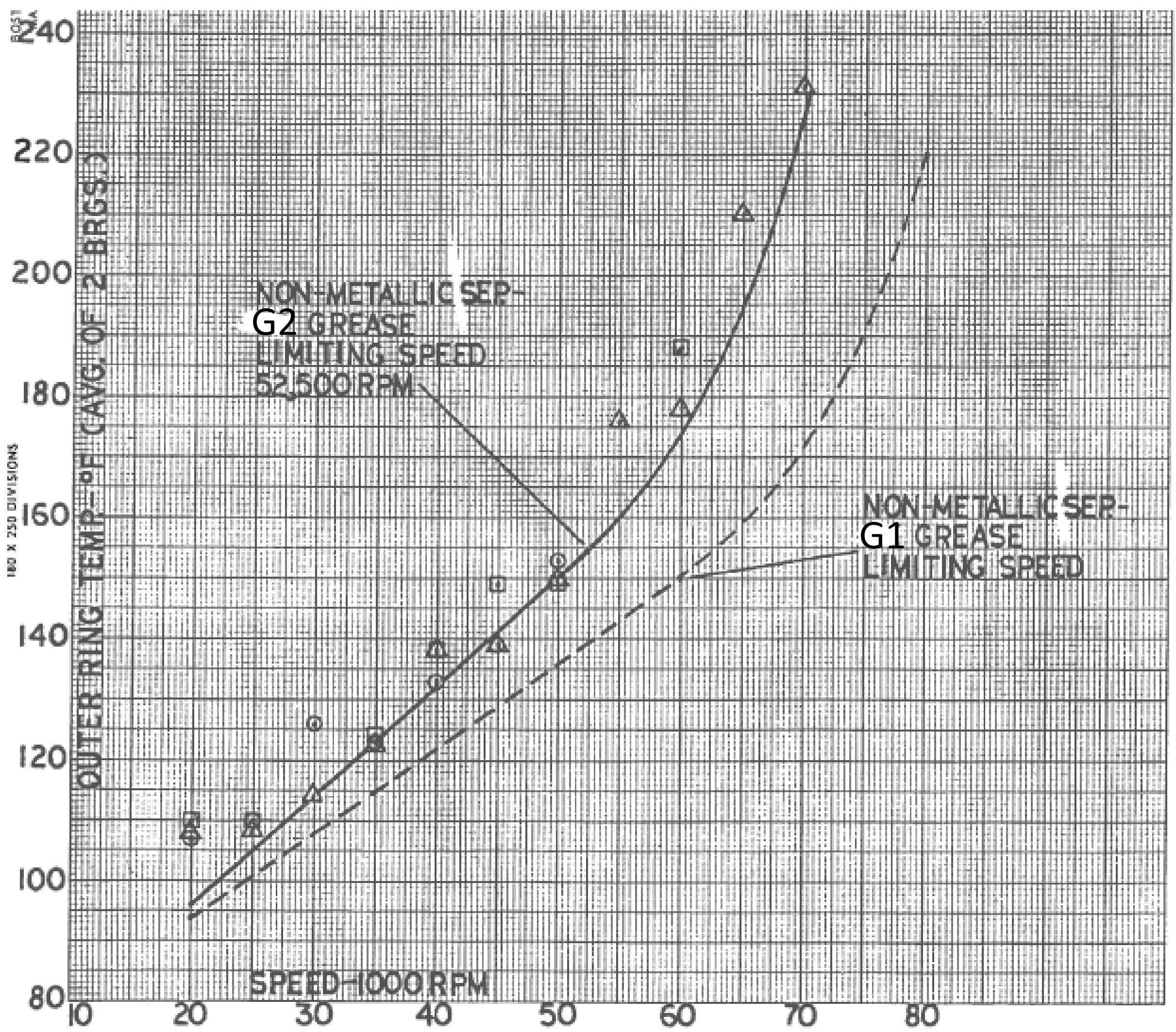


Figure 8

Phosphor-Bronze – G1 Grease

ATB Separator-G1 Grease Performance Evaluation

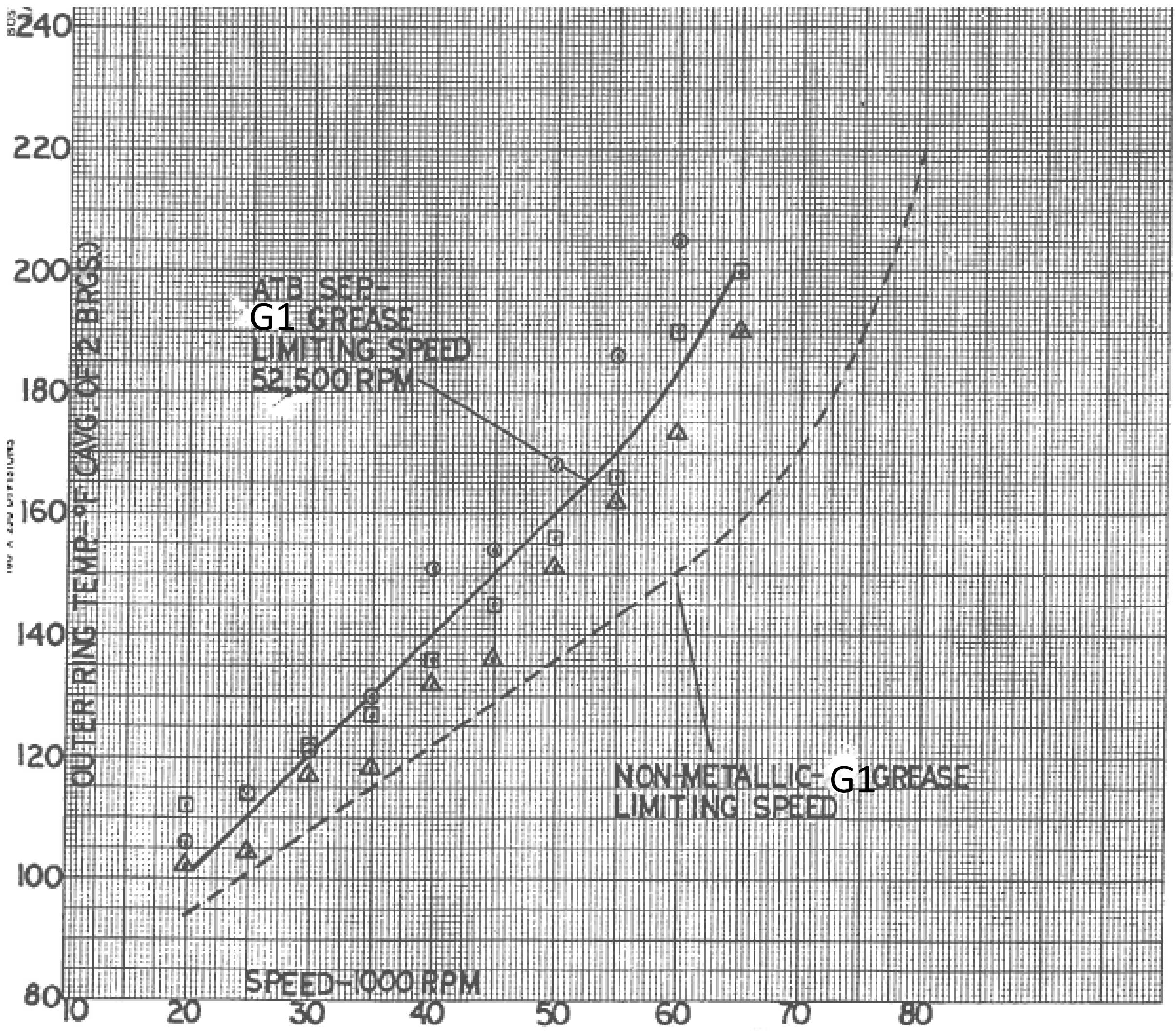


Figure 9

Phosphor-Bronze – G2 Grease

ATB Separator-G2 Grease Performance Evaluation

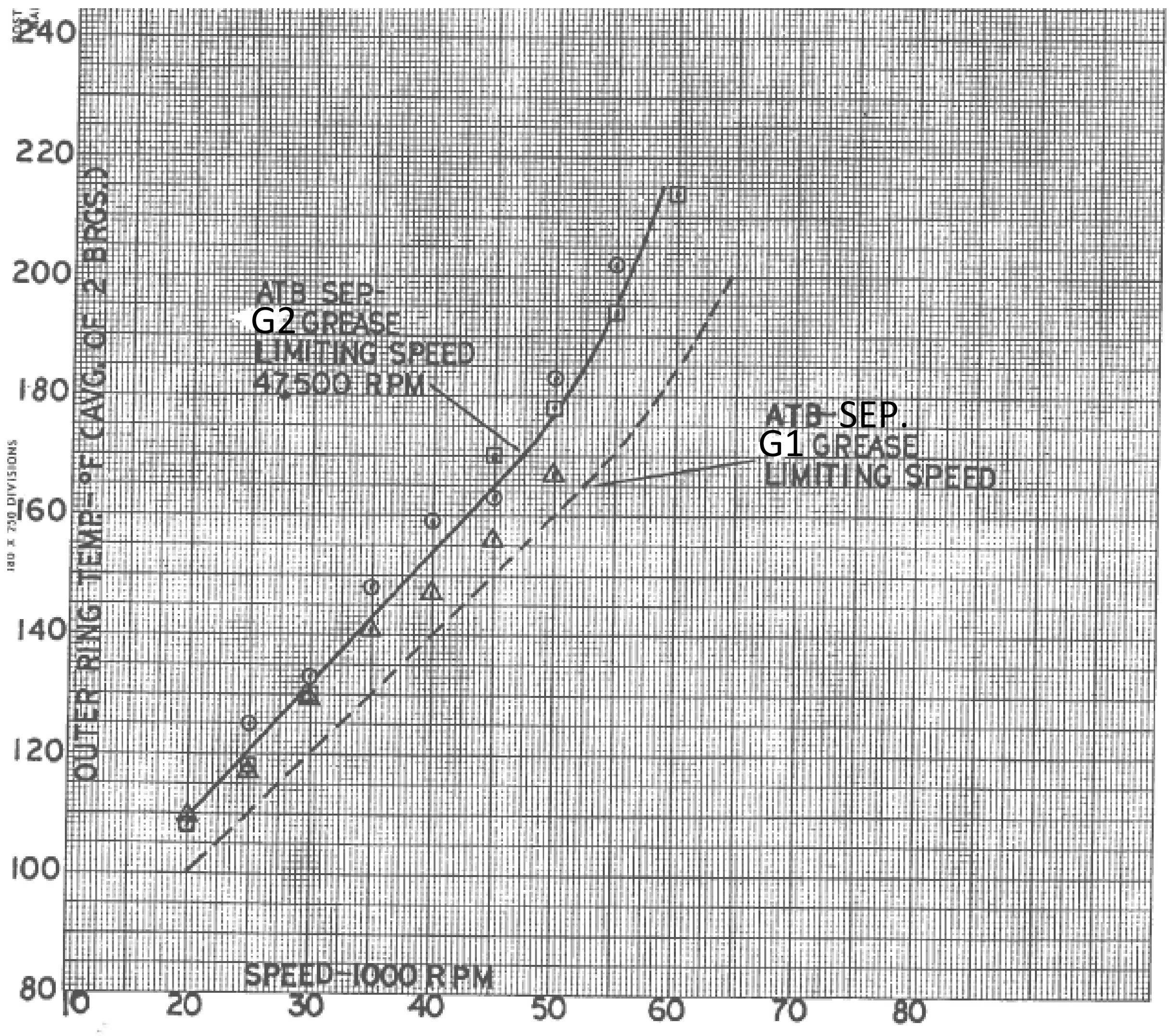


Figure 10

Ribbon Separator – G1 Grease Performance Evaluation

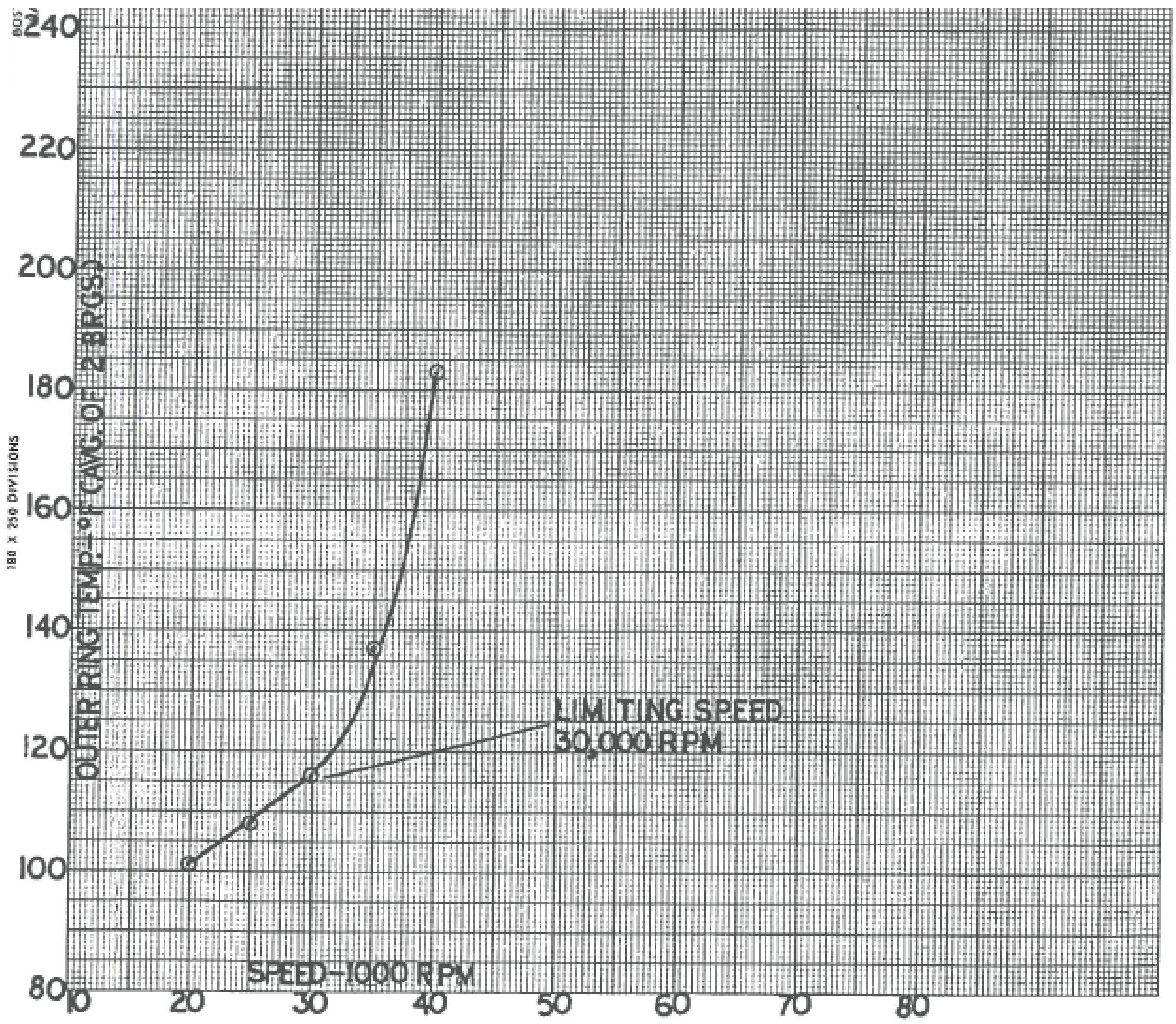


Figure 11

Ribbon Separator – G2 Grease Performance Evaluation

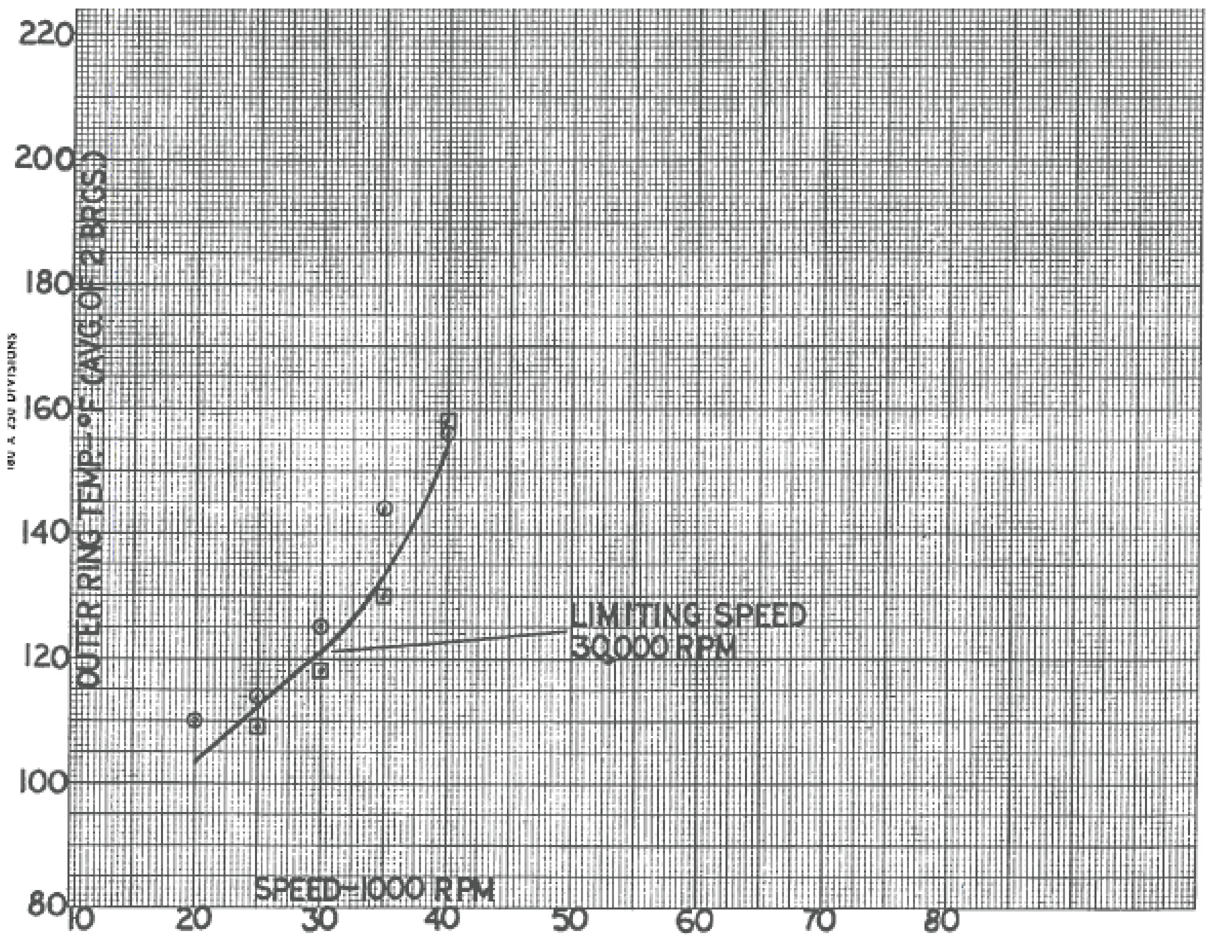
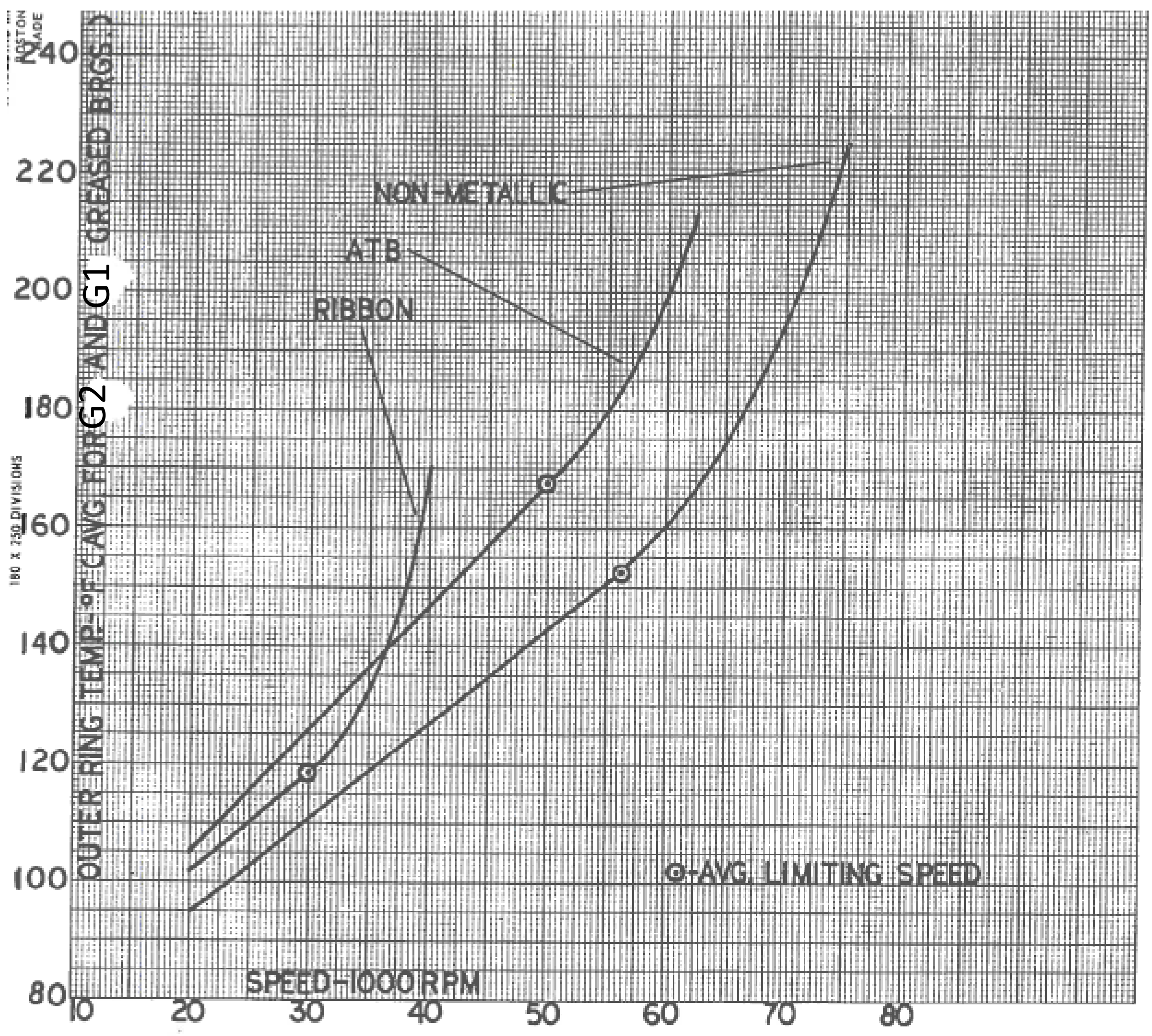


Figure 12

Separator Comparison

Non-Metallic VS ATB VS Ribbon



All nonmetallic and phosphor-bronze separators looked normal after testing. Slight wear was visible at the ball contact surfaces and at the separator contact surfaces. Ribbon separators exhibited more severe distress as the inside of some ball pockets were blackened from heat and, at several places, showed a small amount of metal transfer. Bearings with all three separators exhibited a form of banding on a few of the rings and nearly all of the balls. Analysis of this by a chemical laboratory revealed it to be metal oxidation caused by high contact area temperature.

With the phosphor-bronze separator bearings, a small amount of superficial pitting was evident on most of the rings and balls. This appeared to be associated with grease failure which occurred more severely with the phosphor-bronze separator bearings then with the nonmetallic separator bearings.

The most serious damage of all the bearings was microscopic “herring-bone” marks found around the pathway and on several balls of one phosphor-bronze bearing after test. Examination revealed that the damage was most likely the result of debris indenting, with the source of the debris being unknown.

As can been seen from Figure 12, the lowest temperatures and highest limiting speeds were attained with the nonmetallic separator bearings. The reasons for their improved performance are as follows:

1) Coefficient of friction: Based on calculations involving hardened steel running against both phosphor-bronze and phenolic gears, it can be estimated that the coefficient of friction of hardened steel on phenolic is 25% lower than the coefficient of hardened steel on phosphor- bronze.

2) Control surface: Post test examination indicated that the nonmetallic separator with a molded or machined OD presents a much more uniform control surface than the phosphor-bronze separator which has a stamped sharp-edged OD control surface. The advantages of OD control over ball control can be seen when considering the previously mentioned

problem of the ball controlled ribbon separator losing control and contacting the outer and inner rings.

3) Weight: The nonmetallic separator weighs 3.6 grams compared to 7.3 grams for the phosphor-bronze separator. This weight, along with separator radial displacement, produces an inherent unbalance in the bearing which cannot be removed by balancing. The unbalance at 50,000 rpm is calculated to be 9 pounds for the nonmetallic separator and 15 pounds for the metallic phosphor-bronze separator. This amount of unbalance becomes a factor to consider when using high speed bearings.

4) Outside contour: Figure 13 shows that the nonmetallic separator ran with more grease retained than the phosphorus-bronze and ribbon separators. It is suspected that this is attributable to the smooth outer contour of the nonmetallic separator as opposed to the sharp-edged outside surfaces of the phosphorus-bronze and ribbon separators which tended to sling grease from the bearing onto the grease shields. It is believed that a greater amount of grease retained inside the bearing adds to coolness of operation and increased endurance life.

5) Restart Capability: With nonmetallic separator bearings, limiting speed testing of bearing pairs sometimes took two days. When this occurred, it was found that restart capability of grease-packed high-speed bearings is poor. In three of four cases, when testing took two days, failure occurred the second day at a speed that was below the first day’s high. Investigation revealed that, after one day’s running, the grease in the bearings was darkened and stiff. It was suggested by a grease manufacturing company that high-speed ball bearings can act like a centrifuge separating oil from the grease and allowing it to run freely from the bearing. This may point out the need for seals, rather than seal shields in high-speed bearings, in order to retain the fluid portion of the grease and enable resulting good life and restart capability.

Figure 13

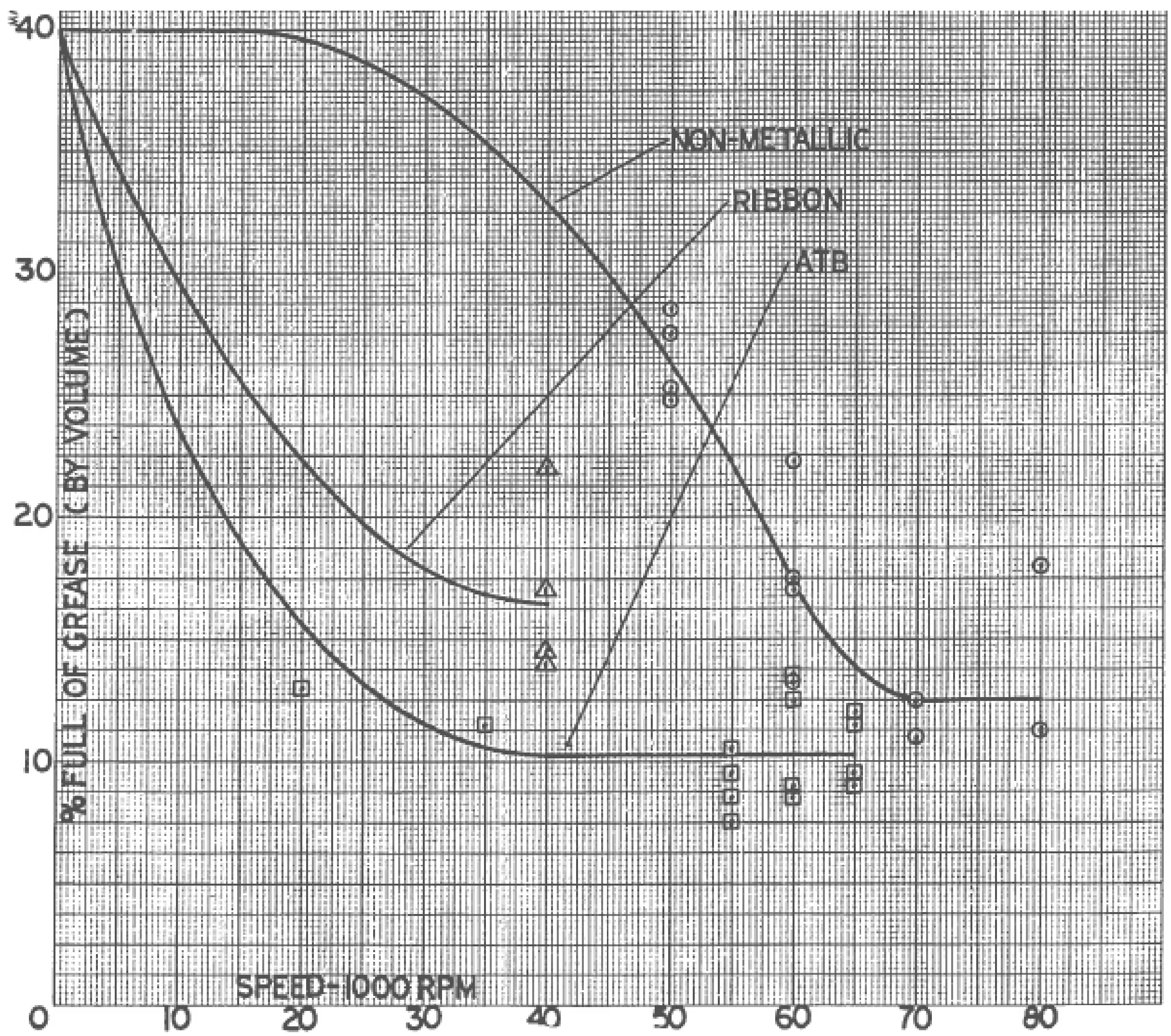
Grease Retention

Per Cent Full of Grease by Volume

Versus

Speed in RPM

Non-Metallic VS ATB VS Ribbon



Test Procedure II: Grease Lubricated Ball Bearing Endurance Testing

The second phase and overall goal of the program is to life-test the best candidates from the first phase of testing in order to develop a grease lubricated ball bearing package that will operate at one million DN for 50 hours. The top four candidates from the phase I testing (listed below) were selected for endurance testing:

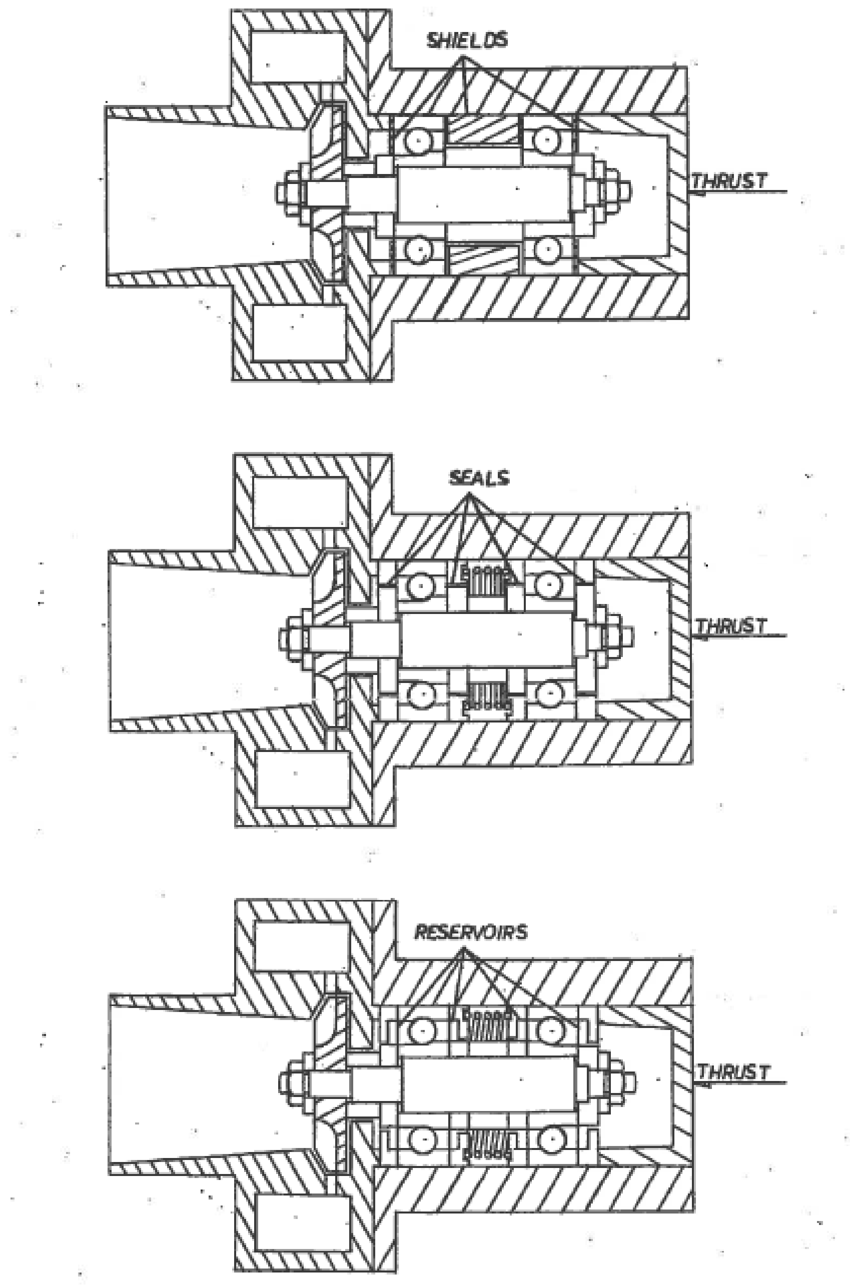
Separator-Grease	Short Term RPM	Limiting Speed
NM-G1	60,000	1.20 Million DN
NM-G2	52,500	1.05 Million DN
PB-G1	52,500	1.05 Million DN
PB-G2	47,500	0.95 Million DN
RB-G1	30,000	0.60 Million DN
RB-G2	30,000	0.60 Million DN

The object of the second test is to develop a separator-grease combination that will satisfactorily complete 50 hours of endurance testing at 50 pounds thrust load and one million DN (50,000 rpm). High speed bearing seals were to be used instead of shields as closures to prevent oil bleed-out problems experienced during limiting speed testing; however, at the suggestion of an industrial seal development group, because of the extremely high inner ring surface speed of 15,150 fpm at 50,000 rpm, hydrodynamic seals were recommended and then used instead. Hydrodynamic seals utilize the pumping action generated by rotating an externally threaded member inside a close clearance hole in a stationary member. The final configuration employed separate hydrodynamic seals operating at the pitch circle on each side of each of the two bearings. The purpose of the hydrodynamic seals is to pump displaced grease back into the bearing (See Figure 14). Also, after a preliminary test indicated that bearing temperatures were running too high, the standard grease fill was reduced from 40% to 25%.

Under the conditions imposed for this test, not one of the four candidates attained the 50,000 rpm, 50 hour life that was desired.

Figure 14

Test Stand Configurations



Results at one million DN are as follows:

Candidate	No. Runs	Avg. life-Hrs.
NM-G2	2	19.0
NM-G1	2	9.2
PB-G2	2	6.2
PB-G1	2	0.6

It can be seen from the above chart that none of the candidate separator-grease combinations reached the desired life of 50 hours. The non-metallic separator outperformed the phosphor-bronze separator; and, in a turn of events from the previous limiting speed tests, G2 grease outperformed G1 grease.

It was decided to repeat the test at 0.8 million DN for the two G2 grease candidates and, as suggested by a lubricant company representative, operate with a 100% grease fill and reservoirs added adjacent to each test bearing, to catch displaced grease and lower the hydrodynamic seal operating point to the inner ring outer diameter as shown in Figure 14.

Results at 0.8 million DN are as follows:

Candidate	No. Runs	Avg. Life-Hrs.
NM-G2	2	43
PB-G2	1	49

It should be noted that one of the two NM-G2 grease bearing combinations reached the desired goal of 50 hours. This test shows that reducing the speed by 20%, increasing the grease fill to 100%, and adding reservoirs adjacent to each bearing to capture displaced grease, results in a very significant improvement in bearing performance which nearly reached the goal of 50 hours of operation for both bearings.

This test was then repeated with new grease. G3 is a No. 2 grease with a high viscosity premium grade mineral oil and an especially effective anti-oxidant rust inhibitor agent.

Results at 0.8 million DN are as follows:

Candidate	No. Runs	Avg. Life-Hrs.
NM-G3	2	50 Hrs. (both runs)
PB-G3	2	1

At 0.8 million DN, the life of new G3 grease and non-metallic separator reached the desired goal of 50 hours in both runs. Because of the poor performance of new G2 grease with phosphor-bronze separators, it is recommended that it not be used with phosphor-bronze separators because of an apparent reaction between the copper in the separator and the new grease.

It was then decided to increase the speed to one million DN and conduct supplementary life tests on bearings with non-metallic separators constructed of new low-friction polyimide material and new G3 grease and again with 100% grease fill and reservoirs adjacent to each bearing.

Results at one million DN are as follows:

Candidate	No. Runs	Avg. Life-Hrs.
Polyimide-G3	2	208

It can be seen that development of a one million DN grease lubricated ball bearing package seems quite feasible based on preliminary testing of low friction polyimide non-metallic separators and new 100% fill G3 grease packed in the bearing with adjacent grease reservoirs. Failure appeared to be associated with loss of grease through the closures which may have been caused by the effects of the 100% bearing fill.

The data indicates that outer ring temperature is a better indicator of limiting speed potential than of life potential. This may point out the fact that grease is a package of a number of different ingredients and, while the use of one grease type may result in lower heat generation and a higher short term limiting speed, another may be less prone to viscous heating resulting in a longer life.

As can be seen from Figures 5 and 14, test bearings are run in pairs supporting a shaft with an overhung turbine drive wheel. All testing resulted in the fan bearing (bearing away from the turbine) running hotter than the turbine end bearing. At one million DN, the temperature variation between the two bearing outer rings ranged from 21°F to 33°F. (Temperatures given in the data tables are the average of the two bearings.) The air driving the turbine enters the turbine axially and exhausts radially providing more cooling for the turbine bearing than the fan bearing creating the temperature differential between the two.

Bearing failure, in all cases except one, occurred with the fan bearing indicating the sensitivity of high grease bearing performance to operating temperature. All failures occurred suddenly and without any advance warning on torque and temperature tracings. At failure, outer ring temperature traces spiked and bearing speed was immediately lost. This immediate loss of speed caused by the low driving torque characteristics of the turbine kept bearing damage to a minimum.

Post-test analysis suggested that bearing failure was precipitated by grease failure since grease in failed bearings was found blackened and dried out while bearing components were relatively free of damage. Subsequent loss of EHD lubricating film in failed bearings was evidenced by dark rings on the balls and races. Chemical laboratory analysis revealed that the dark areas are iron oxide formed by high temperature metal-to-metal contact.

The fact that bearing failures were associated with high temperature corrosion and not normal fatigue, suggests that stainless steel might be used to good advantage for high-speed grease lubricated bearing components. Besides having superior corrosion resistance, 440C stainless steel maintains a hardness of 57Rc while exposed to a temperature of 400°F for 1000 hours while normal 52100 bearing steel maintains a hardness of 57Rc while exposed to only 300°F for 1000 hours.

It was found that high bearing grease fills accompanied by external grease reservoirs result in the best performance. Hydrodynamic seals, if used, should have a very fine low-lead thread which will tend to ooze displaced grease back into the bearing, rather than a larger thread which is thought to shoot grease back into the bearing and possibly causing torque spikes and speed fluctuations of the low inertia drive turbine.

This project indicates that one million DN grease lubricated ball bearings operating for a minimum of 50 hours is quite attainable with low friction non-metallic bearing separators and a lubricant with high viscosity oil. Special consideration should be given to the use of stainless steel for bearing rings and balls because of its anti-corrosion and hardness properties that are advantageous over conventional bearing steel for high-speed, high temperature operation.

