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FEASIBILITY OF INDUCTION SKIN HARDENED TAPERED ROLLER BEARINGS FOR ENGINE MAIN-SHAFT AND TRANSMISSION PINION APPLICATIONS

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SKF Industries, Incorporated

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The purposes of this program were (1) to improve the design of ultrahigh-speed tapered roller bearings based on the computer analyses and test results previously reported under Contract DAAJ02-70-C-0047, and (2) to develop the required technology to induction skin harden M-50 tool steel rings of relatively thin cross section for use in the manufacture of aircraft rolling bearings. Experimental bearings were manufactured and some

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brief tests up to ultrahigh speeds were conducted to demonstrate the basic feasibility of the bearing design and lubrication system for application to aircraft gas turbine engine main shafts and transmission pinion support shafts. The program was conducted in three phases.

In Phase I, the bearing designs previously manufactured of carburized 8620 and 52100 steels and tested under Contract DAAJ02-70-C-0047 were reviewed, and design modifications for M-50 steel bearings were made to improve high-speed, high-temperature capabilities based on these earlier analyses and test results. Two designs of an outer-ring flanged bearing were developed for ultrahigh-speed service (2.5 million DN* and higher).

Phase II consisted of two parts. In Part A, a heat-treat method was developed to induction harden a controlled surface layer on relatively thin section bearing inner and outer rings of M-50 tool steel. The method developed can harden the raceway, flange contact and cage guide land areas of a varied contour section M-50 steel ring, such as a tapered roller bearing ring, while maintaining a softer "core" similar to the section of a carburized steel ring.

In Part B, bearings of the designs developed in Phase I were manufactured. A method was developed to ECM (electrochemical machine) small (0 030 in. diam.), accurately located holes in M-50 steel rings for lubrication supply and/or drainage in the flange undercut region of the ring. The successful manufacture of both through-hardened and induction skin-hardened (inner races only) M-50 steel bearings demonstrates that precision aircraft tapered-roller bearings are feasible and can be made from M-50 tool steel material, which has previously been applied only to premium grade bearings of other configurations.

The method developed for induction skin hardening of relatively thin section M-50 tool steel bearing rings was proven to be practical. This should enable the development of long-life, high-temperature bearings with increased resistance to both ballistic damage and catastrophic spalling failures.

In Phase III, an air-turbine-driven test rig was modified to improve its ultrahigh-speed capabilities and to provide improved lubrication, including under-race lubrication, to the test bearing and dynamic stability to the test shaft assembly at speeds in excess of 100,000 rpm (corresponding to 2.5 million DN on the 25mm-bore test bearings). A limited test program was conducted with through-hardened M-50 steel bearings to speeds in excess of 2.5 million DN. These test results indicate than an improved outer-ring flanged tapered-roller bearing was developed.

*Product of shaft speed in rpm and the bearing bore in mm.

EUSTIS DIRECTORATE POSITION STATEMENT

The work reported herein represents an advancement in the speed capability of tapered roller bearings.

It is expected that further effort will be required to demonstrate the life/endurance characteristics of the bearings before they are considered for a gas turbine engine application.

Appropriate technical personnel of this Directorate have reviewed this report and concur with the conclusions and recommendatons contained herein.

Mr. David B. Cale of the Propulsion Technical Area, Technology Applications Division served as Project Engineer for this effort.

DISCLAIMERS

The findings in this report are not to be construed as an official Department of the Army position unless so designated by other authorized documents.

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PREFACE

This report represents the results of a program conducted by S K F Industries, Inc., for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, under Contract DAAJ02-73-C-0009. This report encompasses the development and evaluation effort conducted from October 2, 1972 to July 2, 1974.

Eustis Directorate technical direction was provided by Mr. David Cale, Propulsion Technical Area, Technology Applications Division.

The principal investigators for the program were T. F. Conners, Scientist, Mechanical Test Section, and J. W. Rosenlieb, Research Scientist, Contract R and D Department, who served as project leaders; H. M. Martinie, Scientist, Contract R and D Department; R. E. Maurer, Scientist, Materials Laboratory; and R. A. Nash, Scientist, Mechanical Processing Section. Additional significant contributions were made by L. B. Sibley, Manager, Contract R and D Department; F. R. Morrison, Supervisor, Mechanical Test Section; and C. G. Hingley, Supervisor, Configuration Section.

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I. INTRODUCTION

A. STATEMENT OF THE PROBLEM

Currently, a typical small, 2- to 10-lb/sec airflow, gas turbine engine main-shaft or transmission input pinion shaft support system consists of a thrust-carrying bearing unit and one or more radial bearings. The typical thrust bearing is a split-inner-ring angular-contact ball bearing. Engine designers attempt to minimize the standard operating thrust loads through balancing of the power turbine and compressor loads, but a considerable thrust load exists at some points in the cycle. This can be large enough to require the stacking of thrust bearings, i.e., the use of multiple bearings, an obviously undesirable feature due to difficulties in providing adequate lubrication and load sharing. Such tandem thrust bearing designs are often used on larger turbine engine main shafts and on helicopter transmission pinion shafts.

A survey performed by S K F under Contract DAAJ02-70-C-0047 indicated that the four engine manufacturers interviewed were designing engines with specified DN values for the main-shaft bearings of 1.6 to 2.0 x 10^6, which is within range of current technology. However, trends suggest that bearings capable of 2.5 x 10^6 DN will be required in the next few years and 3.0 x 10^6 DN in the foreseeable future. These higher speeds are desired without having to suffer penalties or trade-offs in bearing life, load-carrying ability, and reliability.

As a result of the increased speed, gas turbine engine and transmission pinion bearings will operate at speeds where the centrifugal force of the rolling elements is a significant addition to total bearing load. In order to reduce the centrifugal loading effect of these bearings, attempts are now under way to produce low-density or hollow rolling elements as a means of limiting centrifugal forces. These efforts are promising but are by no means the only approach to an ultrahigh-speed rolling bearing with longer life and/or higher load capacity. It has long been recognized that line contacts can have a substantially increased load-carrying capability over that of even a closely conforming ball-to-race contact. As the technology of roller crowning and lubrication has been refined in recent years, this basic advantage of roller bearings over ball bearings has become even more pronounced. Comparison of the endurance life results obtained at S K F on several hundred bearings of deep groove ball and cylindrical roller designs of the same envelope dimensions (45mm bore) indicates that life capability of about 35 times the ball bearing life is obtained with the line contact roller bearing when evaluated under similar loading.

Under Contract DAAJ02-70-C-0047, the predecessor to this program, both inner-ring flanged (standard configuration) and outer-ring flanged (advanced configuration) tapered roller bearing configurations were designed, fabricated from 8620 and 52100 steels, and tested. The test results showed that it was feasible to operate up to 2.4 million DN with outer-ring flanged designs and 2.13 million DN with inner-ring flanged designs under the loading and lubrication techniques applied.

In the survey of engine manufacturers noted above, it was also established that three of the four manufacturers expected nominal bearing operating temperatures to increase above the present levels of 250°-$400^{\circ}F$ in the near future. Thus it becomes necessary to use a high-temperature-resistant steel such as M-50 tool steel in the manufacture of advanced tapered roller bearing designs which could find application in gas turbine engines.

In many helicopter gearbox and transmission applications and in some gas turbine applications, integrated (one-piece) parts, such as bearing inner rings and shafts and bearing outer rings and housings or flanges, have been incorporated to reduce weight and increase reliability. To date, however, these integrated parts have been constructed from less than optimum materials. With through-hardened bearing tool steels having proven long life and high-temperature resistance, heavy structural sections must be used to keep impact and bending stresses within tolerable limits for these hard, brittle materials. Available case-carburized bearing steels permit high hardness of the bearing races while retaining a lower hardness and greater ductility throughout the rest of the part to withstand structural and impact loads, but they do not have proven long-life, high-temperature capabilities.

Therefore, the development of a high-speed, high-temperature thrust load-carrying bearing made of case-hardened steel which could be integrated with other component parts would be a major step in solving some of the expected problems in advanced engine and transmission designs while providing improved reliability and resistance to ballistic impact.

B. METHOD OF APPROACH

In order to evaluate the capability of tapered roller bearings to operate at an even higher speed than that shown under Contract DAAJ02-70-C-0047, several improvements in the lubrication system for the bearings were incorporated in the test fixture, and an improved bearing design based on the previous results was developed for limited testing. Through-hardened M-50 tool steel was selected for the rings and rollers to be tested because of its known long-life, high-temperature capability and to show that the manufacturing techniques for fabricating tapered roller bearings of this material were known.

Concurrently, induction heat treating of M-50 tool steel was selected as the means of obtaining a hardened case on a proven high-temperature bearing steel, thus providing a technique which would permit operation at higher temperatures and the integration of the bearing parts with other components.

The program was divided into three distinct phases. The first phase was the development of two tapered roller bearing designs for very high speed service. The second phase was (A) the development of suitable induction skin hardening heat treatment methods for both the inner and outer rings of M-50 tool steel, and (B) the manufacture of test bearings from both through-hardened and induction skin-hardened rings. The third phase was the testing of the through-hardened M-50 bearings. Phase I was completed first and Phases II and III were performed concurrently.

Phase I - the bearing design phase based on previous Contract DAAJ02-70-C-0047 - was completed without difficulty. The outer ring flange type was selected, and two design variations were developed.

Phase IIA - the induction heat treatment development - was initiated using the inner ring, due to the simplicity of the configuration, and progressed to the more complex outer ring. The development effort on induction heat treating encountered unexpected and persistent difficulties. Induction skin-hardening techniques were evaluated (including changes in inductor configuration, heating time, inductor power and frequency, and quenching methods) before a successful combination was obtained. The development of a satisfactory induction skin-hardening process required so much effort that only a few test bearings were manufactured and tested.

Phase IIB - manufacture of two bearings of one selected design - was completed from through-hardened M-50 steel, showing that standard manufacturing techniques for tapered roller bearings were applicable to this material. Bearing groups of both design variations using M-50 steel were partially manufactured with induction skin hardened inner rings. During the manufacturing cycle, technology was developed for ECM (electrochemical machining) of the 0.030-inch-diameter lubricant access or drainage holes in the bearing semifinished outer ring. The ECM method developed is capable of holding a very close hole location tolerance for these very small holes without disturbing the metallurgical structure of the bearing material.

In Phase III, an existing test rig was modified to increase speed capability and to incorporate various methods of lubricating the test bearing.

Five tests were performed utilizing the two fabricated bearings. The first three tests were primarily rig "shakedown" tests to identify and correct rig difficulties and to determine the characteristics of the various lubrication methods available. Tests 4 and 5 were step-speed tests to establish high-speed performance.

II. TEST FACILITY

A high-speed, high-temperature bearing test machine was used for all testing conducted under this contract. A detailed discussion of the design and operation of this test rig is given below.

A. TEST RIG

A layout drawing of the basic test head is presented in Figure 1. The tester mounts two bearings - the test bearing and a turbine-end support bearing (rig bearing) - on the same shaft. The test bearing is mounted in a movable housing through which load is applied, called a load plug, and the turbine-end bearing is mounted on the drive end to act as a slave bearing. The axial load is applied by a calibrated spring system which thrust loads the bearings against each other. Although spring systems are generally less accurate than deadweight loads, it has been found that at the higher speeds, the test machine will develop shaft vibrations which are related to the natural frequency of the deadweight system and the constant nature of the applied load. The use of spring loading provided less vibrating mass and a positive load/displacement gradient under the axial motions characteristic of the disturbances, and thus provided a stabilizing influence. In this manner, it is possible to obtain test speeds in excess of the first shaft critical.

The test rig is driven by an air turbine which provides speed capability in excess of 100,000 rpm. Air is supplied by a 750-cfm, 150-psig air compressor through heaters provided to eliminate the formation of ice at the nozzles and to increase the specific volume of the air, thereby

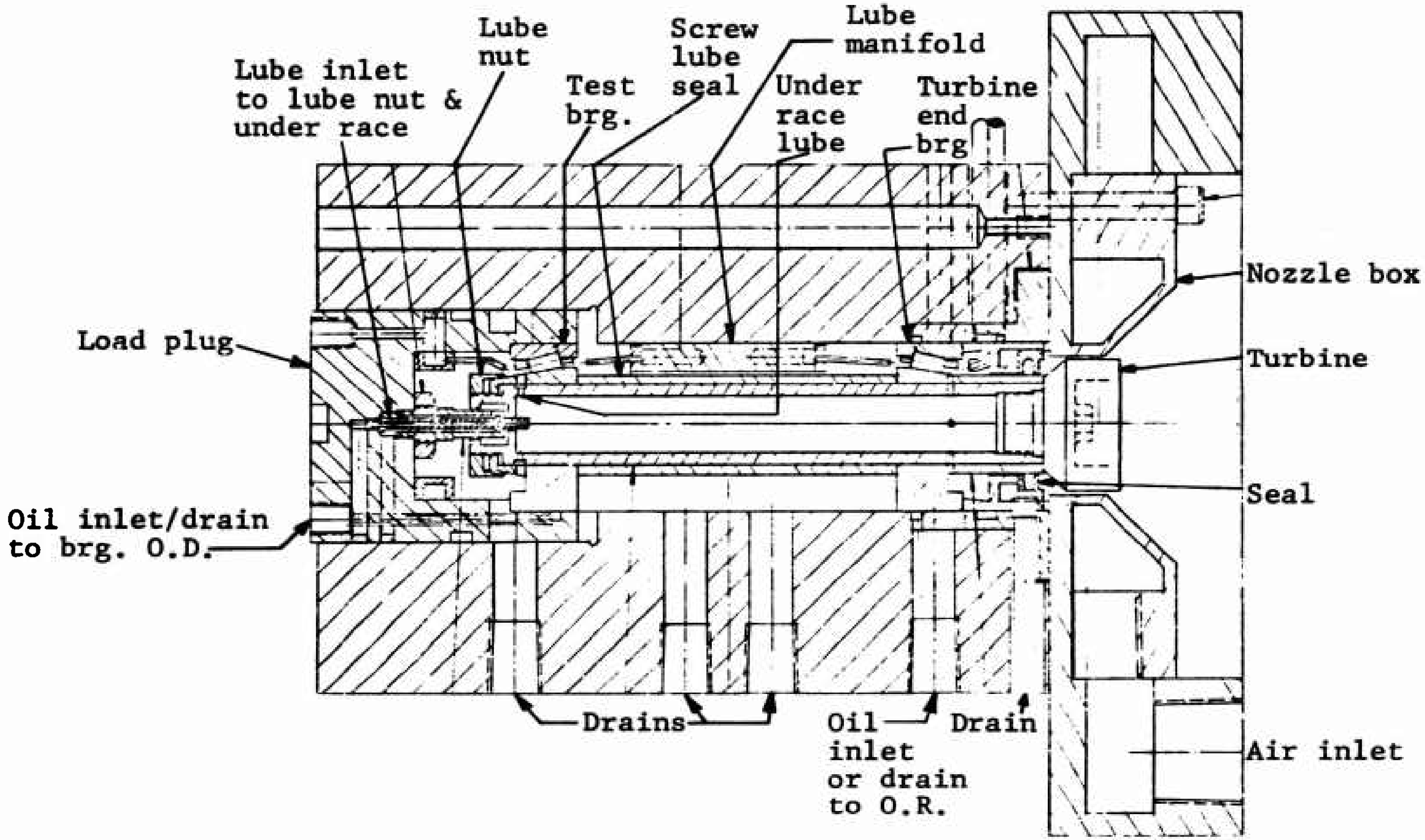

Figure 1. Layout Drawing of High-Speed Tester.

providing greater power output of the turbine.

The air turbine is provided with safety devices to safeguard personnel and test equipment in the event of mechanical malfunctions within the rig. In the event of the loss of oil line pressure, the air supply is automatically vented to the atmosphere, bypassing the turbine assembly. Additionally, if the air supply pressure drops below a preset level, the heaters in the air line are automatically shut off.

To minimize the unbalance forces at high speeds, the shaft assembly was balanced to 0.001 oz-in. for each test setup.

B. LUBRICATION SYSTEM

The bearings tested in this program were lubricated with an MIL-L-23699 specification oil supplied from an external 15-gallon sump. Oil scavenging was performed by two pumps, connected to the inboard and outboard drain lines of each bearing respectively.

Five different inlet oil flow paths were incorporated into the test bearing end of the fixture to permit evaluation of various oil application schemes and to insure, if possible, proper lubrication of the test bearing at ultrahigh speeds. The flow paths as shown in Figure 1 were: (1) through the inboard jets, (2) through the outboard jets, (3) through the hollow shaft and holes in the locking nut under the cage at the small end of the roller, (4) through the shaft end and a second set of holes in the locking nut, and (5) through the holes in the outer ring at the race/flange interface. The holes in the race/flange interface could be used either to supply oil or to drain oil trapped in the flange area.

Scavenge pumps removed the oil from drain holes provided inboard and outboard of each bearing and returned it to the sump, with the oil passing through a heat exchanger in order to maintain temperature stability of the oil supply.

III. BEARING DESIGN

The bearing design developed under this contract is based on work previously reported in USAAMRDL Technical Report 73-46.[1] In this previous work, two basic designs were developed and tested, as described in the following paragraphs.

The conventional inner-ring flanged bearing design was adapted, as shown in Figure 2, to aircraft bearing tolerances and was modified to use a land-riding, silver-plated cage and to supply under-race cooling and lubricating oil to the flange. Analytical studies pointed out the "declutching phenomenon" of this tapered-roller bearing design. The "declutching phenomenon" occurs when the centrifugal force on the rollers overcomes the effects of the applied thrust load. At this point the load on the inner raceway becomes zero and all the load is transferred to the flange contact. This speed value, of course, varies with load. The actual tested conditions for declutching compared very closely with the theoretical calculations.

A second design with an outer-ring flange, shown in Figure 3, was developed to overcome the "declutching phenomenon". The basic dynamics of this bearing design apply a constant load to the inner race with the increasing load due to the

[1] Conners, T., and Morrison, F., FEASIBILITY OF TAPERED ROLLER BEARINGS FOR MAIN-SHAFT ENGINE APPLICATIONS, USAAMRDL Technical Report 73-46, Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, August 1973, AD 771984.

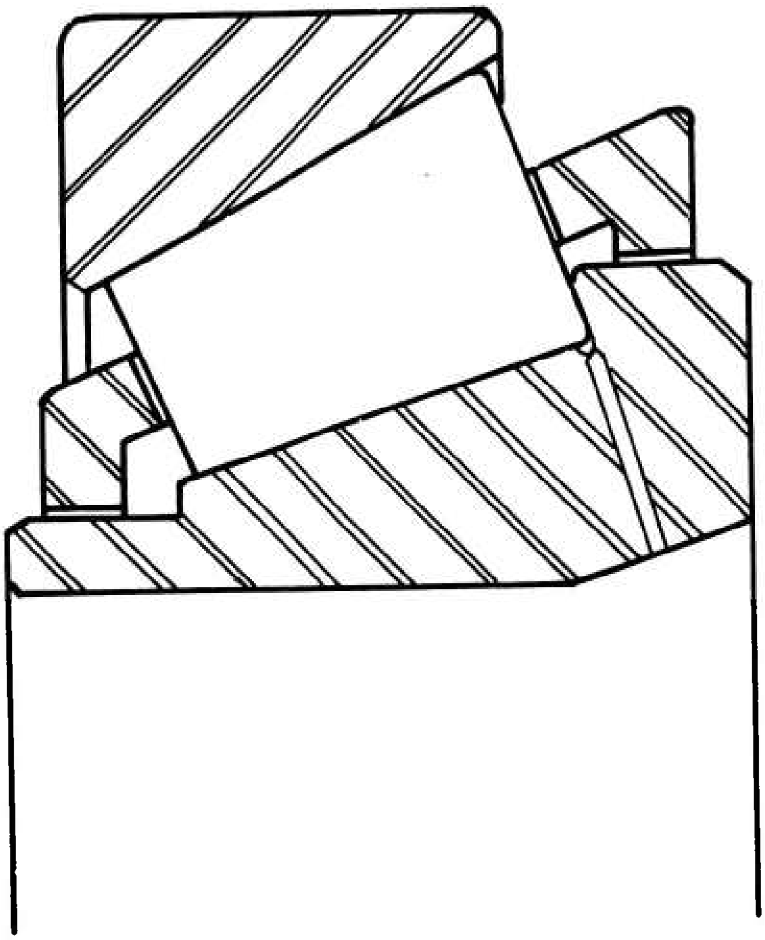

Figure 2. Experimental Inner-Ring Flanged Bearing.

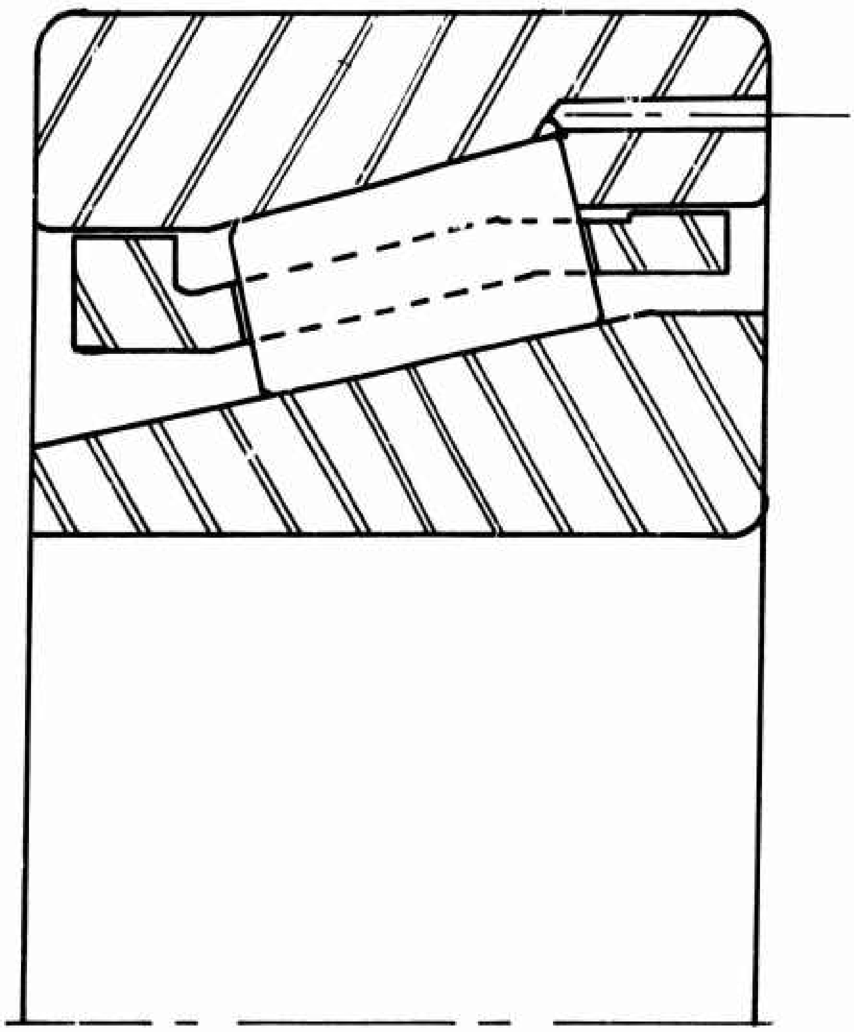

Figure 3. Modified Tapered Roller Bearing Design With Outer-Ring Flange.

speed effect applied to the outer race and outer-race flange. Thus the "declutching phenomenon" does not occur; however, the increasing load applied to the outer race causes a reduction in the theoretical calculated life of the bearing for given conditions.

The trade-offs in the two bearing designs become an engineering judgement. As the present program was not large enough to develop, manufacture, and test both designs, a judgement was made to develop the outer-ring flanged bearing due to its inherent higher speed capability without the instability which could occur at or above the "declutch" speed.

This selection was based on the fact that in the precursor Contract DAAJ02-70-C-0047, outer-ring flanged bearings with 25mm bore were operated at speeds in excess of 90,000 rpm successfully for several hours without showing any signs of failure.

On the other hand, inner-ring flanged bearings were operated successfully in the precursor contract only up to 80,000 rpm, i.e., below their declutching speed (85,000 rpm for the test conditions).

It was also judged that solid results were available on which to base bearing design and testing for 2.5 million DN operation of 25mm-bore outer-ring flanged tapers which undergo no declutching, whereas a combination of bearing, test rotor and stator design would have to be developed in order to assure operation in excess of 2 million DN for inner-ring flanged tapers, which undergo declutching at that speed.

A major goal of the program was to prove the viability of the tapered roller bearing principle at speeds up to 2.5 million DN for 25mm-bore bearings made of M-50 steel. Thus, so as not to divert effort to bearing/rotor configuration development, the outer-ring flanged bearing was considered to be the most promising candidate.

The above statements do not imply that inner-ring flanged tapers are inherently unsuitable for ultrahigh-speed operation. In fact, the flange and track force diagram of an inner-ring flanged taper permits higher external load for a given fatigue life at ultrahigh speed than that of an outer-ring flanged taper.[1] The calculated differences in load-carrying capability are quite substantial. Furthermore, inner-ring flanged tapers are somewhat easier to manufacture, and their performance characteristics are better known than those of outer-ring flanged tapers. Thus, there is a real incentive to extend ultrahigh-speed capability development of the inner-ring flanged design before a final bearing type is selected for incorporation into a gas-turbine main shaft or other ultrahigh-speed equipment.

However, the objectives of the contract were best carried out within the specific state of the art of ultrahigh-speed tapered roller bearing technology.[1,2] The key questions of this technology are:

1. Can a small roller bearing with straight generatrix operate for a long period (200 hours or $1.2 \cdot 10^9$ revs at $2.5 \cdot 10^6$ DN, without

   track surface distress,
   track edge loading and spalling,
   track skid mark failure,
   flange smearing,
   flange wear,
   heat imbalance,
   cage wear or smearing?

2. Can induction skin hardened tool steel be made into small tapered roller bearings of quality sufficient to survive the above tests, and does it offer advantages of high-temperature resistance, dimensional stability, long fatigue life, and lubricant compatibility? What are the required induction heat-treating parameters?

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[2] Crecelius, W. J., and Milke, D. R., DYNAMIC AND THERMAL ANALYSIS OF HIGH SPEED TAPERED ROLLER BEARINGS UNDER COMBINED LOADING, NASA Report CR-121207, National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio, March 1973.

3. Is an outer-inner flanged roller bearing (whether tapered or cylindrical) viable under heavy continuous flange loading?

4. What are the friction losses of an ultrahigh-speed tapered roller bearing?

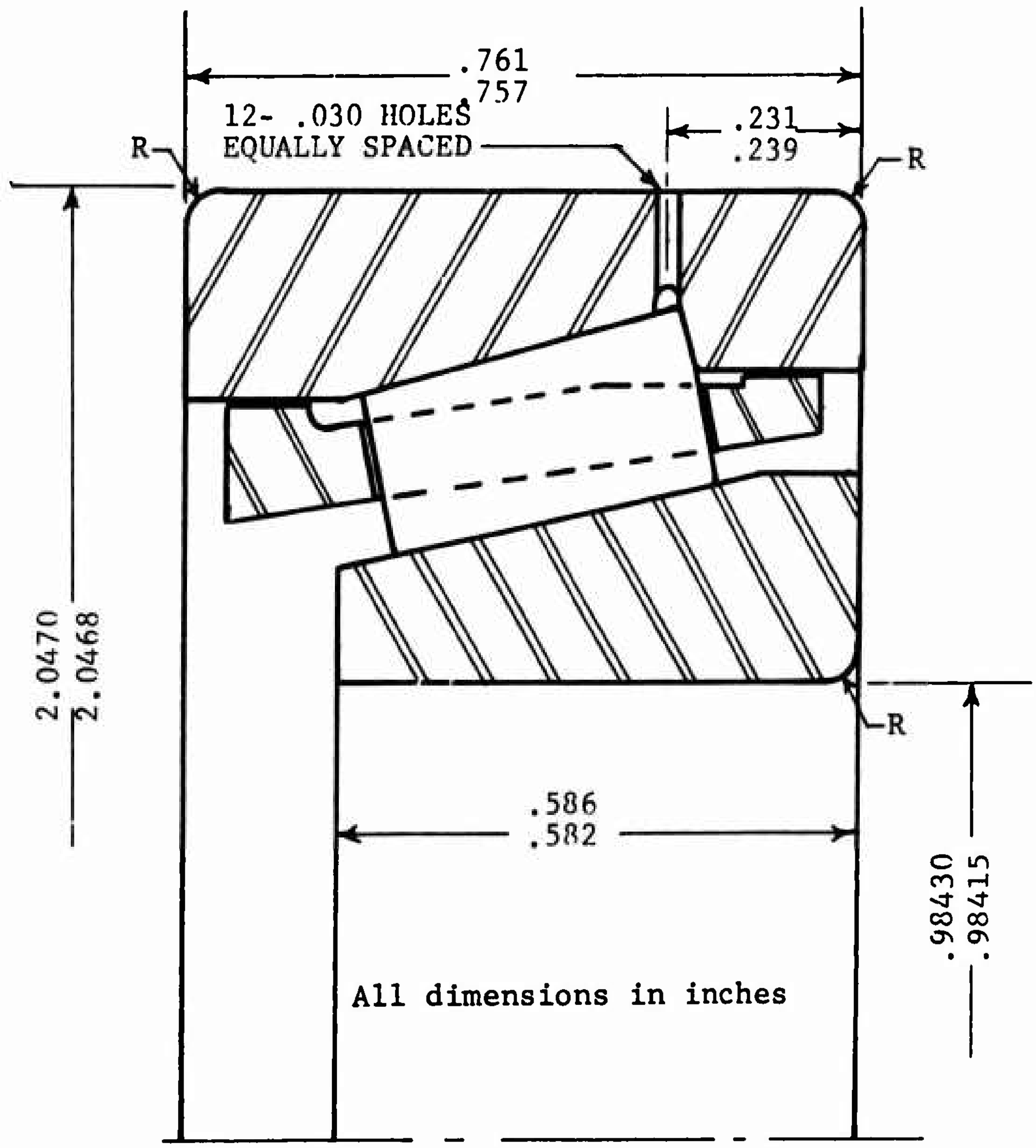
5. What are the inherent stability properties of an ultrahigh-speed thrust-loaded roller bearing? What is its kinematic high-speed operation?

6. How do losses at ultrahigh speed compare for various tapered roller and ball bearing designs?

The basic configuration of the outer-ring flanged bearing (Figure 3) previously tested was adopted for this program.

Review of the previous testing data and tested specimens indicated that modifications to assure better lubrication to the flange face - roller end contacts and cage contacts were desirable. Therefore, design effort was concentrated on this area.

The lubrication holes to the race track flange face were moved from entering the face to entering from the O.D.; see Figure 4. The O.D. location facilitates the connection with the lubricant system and allows for either pressure feed or suction drain as determined to be desirable during testing. The number of holes was increased to 12 to allow



The two variations of this bearing are identical except as noted below.

	Outer Ring Flange Angle	Roller Sphere End Radius	Cage Section Thickness	Cage Pocket Clearance
L42043A	$15^{\circ}0'0''$	70% of Apex	0.060"	0.005"/0.013"
L42043B	$15^{\circ}9'0''$	85% of Apex	0.070"	0.008"/0.016"

Figure 4. Outer Flange Guided High-Speed Tapered Roller Bearings.

better distribution of lubricant to the flange face.[3] Due to the small undercut at the flange face raceway junction, the hole size was decreased to 0.030 inch.

Due to the small openings between the cage and inner ring, it was decided to apply under-race lubrication to the bearing through holes in the inner race. However, due to the critical location of these holes and the possibility of cracks propagating from the holes,[1] it was decided to reduce the length of the inner ring and to use slots in the face of the locknut to serve as lubricant entrance holes, as shown in Figure 1. The use of this method was also more economical from a manufacturing viewpoint. The inside of the cage on the small end was tapered from the edge inward to direct the lubricant from the under-race source toward the inside of the bearing.

Two variations of the basic configuration were designed involving further improvements in the roller end and flange geometry and cage details as shown in Figure 4. One design (L42043A) was identical in these respects to the bearing previously made from 52100 steel. This design was used to evaluate the improvement which could be expected from the above modifications in the lubrication scheme and from M-50 high-temperature tool-steel material compared to the normal

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3 Lemanski, A. J., Lenski, J. W., Jr., Drago, R. J., DESIGN, FABRICATION, TEST, AND EVALUATION OF SPIRAL BEVEL SUPPORT BEARINGS (TAPERED ROLLER), USAAMRDL Technical Report 73-16, Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, June 1973, AD 769064.

52100 bearing steel. The second design (L40243B) incorporated further changes to investigate the large roller end/flange face contacts. The roller end radius was modified to 85% of the apex radius, with the flange face angle modified to 15° 9' to retain the contact at the center of the flange face. The cage pocket clearance also was increased to 0.008/0.016 inch, and the cage thickness was increased to 0.070 inch nominal to avoid pocket edge contact which sometimes was observed[1] in testing bearings of the earlier design.

IV. BEARING MATERIAL

Material for bearing manufacture has evolved through use into two basic types: through-hardening material, principally 52100 steel, and carburizing case-hardening materials such as 4320, 8620 and 3310 steels. In aircraft and other critical applications, through-hardening material of the tool-steel class has come into use due to its ability to operate at elevated temperatures and the increased reliability of bearings made of this material.

Carburizing grade steels for bearing service are heat treated to a hard surface layer (case), i.e., Rc 58+, which is supported by a softer, i.e., Rc 35-45, core material. Case-hardened material will arrest the propagation of a surface layer crack when the crack reaches the softer core material, thus reducing the possibility of a catastrophic failure. Case-hardened material is thus more resistant to ballistic damage in military service.

In addition to minimizing crack propagation, the combination of a hardened surface layer with the remaining material less hard and more ductile permits the integration of bearing and structural components. Bearing contact surfaces must be hardened to obtain the desired life, while structural parts must be more ductile to support the applied forces and impact loads. The integration of bearing inner rings and shafts, and bearing outer rings, and flanges or housings is desirable in aircraft engines, gearboxes and transmissions to cut costs, save weight and improve reliability. The development of induction skin hardened bearing quality surfaces on M-50 tool steel initially heat treated for high ductility and impact resistance would permit these advantages to be incorporated in engine and transmission components where the high-temperature-compatibility characteristics of M-50 steel are necessary.

Induction heat treating of normal through-hardening material (i.e., 52100) is a known and proven method of developing a hardened load bearing area while retaining a softer tough core material. Limited previous testing[4] indicated that this method of hardening could be developed and applied to M-50 tool steel. Due to the significant improvements in the high-speed, high-temperature service of tapered roller bearings which could result from the use of M-50 material, this material and method of heat treating were selected for development to ascertain their feasibility for future aircraft engine and transmission bearings. The rollers, not being subjected to bending stresses, are not as critical as the rings in needing high core ductility; therefore, no induction skin hardening development for the rollers was conducted.

The bearing rings and rollers manufactured for testing in this program were made of through-hardened M-50 steel, and the cages were made from AMS6415 steel, a standard material for machined high-speed aircraft bearing cages, which were silver plated according to standard aircraft practice. Both inner and outer ring blanks were successfully induction skin hardened, and a group of these inner rings was manufactured complete except for a final finishing operation, thus demonstrating the basic feasibility of this important novel bearing ring heat-treatment method.

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[4] Maurer, R. E., et al, SUMMARY REPORT FOR THE FIRST YEAR OF EFFORT ON THE DEVELOPMENT OF DUPLEX METALLURGICAL STRUCTURES FOR AIRCRAFT ROLLING BEARINGS, S K F Report AL69M016, to U. S. Department of the Navy, Naval Air Systems Command, Washington, D. C., July 1969, AD 859393.

V. INDUCTION HEAT TREAT SKIN HARDENING DEVELOPMENT

A. BACKGROUND

Induction hardening is a widely used, well established technique for producing components with regions of localized hardening. The characteristics of an induction hardened region are dependent upon the induction heating parameters (inductor shape, induction frequency, induced power, etc.) and the metallurgical characteristics and configuration of the workpiece. Generally, the higher the operating frequency, the shallower the layer in the workpiece surface that is being heated. For any given frequency, the rate of heating increases with the magnitude of the induced current. Thus, to produce a very shallow case, high frequency is required to restrict the inductive heating to a thin surface layer, and high power is required to heat that thin layer at a rate that significantly exceeds the heat dissipation through the workpiece by thermal conduction. For deeper case depths, a lower frequency and/or a lower power input must be used.

The temperature to which a steel workpiece surface must be heated to effect case hardening is primarily dependent upon the chemical composition of the workpiece; the latter determines the temperature at which the ferrite to austenite transformation is completed. Since the total time at temperature is generally only a few seconds during induction heating for case hardening, the temperatures required are generally slightly higher than those used in conventional furnace heating, so the kinetics of the phase transformations are accelerated to compensate for the reduced soak time. In addition, since only a relatively thin surface layer is inductively heated, the workpiece experiences a temperature profile that decreases from the surface inward. Thus, the surface temperature may be considerably higher than that required for complete austenitization, so the material down to some subsurface point on the temperature profile is above the minimum required temperature to provide the desired case depth. Generally, the frequency and power parameters are adjusted in an attempt to minimize the degree of this "overheating" of the surface.

Tool steels, which exhibit secondary hardening, present a unique set of circumstances to be reckoned with in induction case hardening. These material types contain strong carbide formers, such as molybdenum, vanadium or tungsten, which provide secondary hardening or tempering. In the heat treatment for hardening tool steels, temperatures well above those that will provide for complete austenitization of the matrix material are required to partially dissolve carbides of molybdenum, vanadium and tungsten to obtain secondary hardening. The 1% carbon matrix of M-50 tool steel is completely austenitized at approximately 1550°F, whereas the conventionally used austenitizing temperature for this material is in the range of 2000° to 2100°F. During the tempering stage in the heat treatment of all tool steels, very fine carbides of vanadium, molybdenum and tungsten are reprecipitated to provide high-temperature hardness and strength retention that cannot be obtained with the martensite matrix alone.

Since the tool steel matrix will austenitize at a temperature considerably below the temperature required for limited carbide dissolution, there is a wide range in depth below an inductively heated surface layer that will exhibit appreciable hardness upon quenching. For example, in the case of M-50 steel, all material above 1550°F will be austenitized and transformed to martensite (or be maintained as retained austenite) upon quenching. Material in the temperature range of 1450° to 1550°F, although only partially austenitized, will contain an appreciable volume fraction of martensite after quenching. The greater the volume fraction of martensite, the higher the hardness of the matrix. Whereas most conventional induction case hardening applications, using relatively low alloy or plain carbon steels in which only matrix hardening is required, impose a temperature profile such that the workpiece experiences partial or complete austenitization over a temperature interval of 200° to 300°F, that interval may easily be doubled in the induction case hardening of a tool steel. This particular characteristic necessitates the use of special induction heating techniques to obtain shallow case depths in tool steel components.

The recognition of the need for these special techniques to accommodate tool steels such as M-50, and the development of such,[3] formed the basis for the metallurgical phase of this project.

B. EXPERIMENTAL PROCEDURES

The axial profiles of the inner and outer rings of a flanged outer-ring tapered roller bearing design using service-proven high-performance M-50 material are illustrated in Figure 5. The roller paths, cage-riding surfaces and flange face were to be induction case hardened, while the remaining material was to be of such hardness as to provide a relatively tough, crack-arresting, impact-resistant

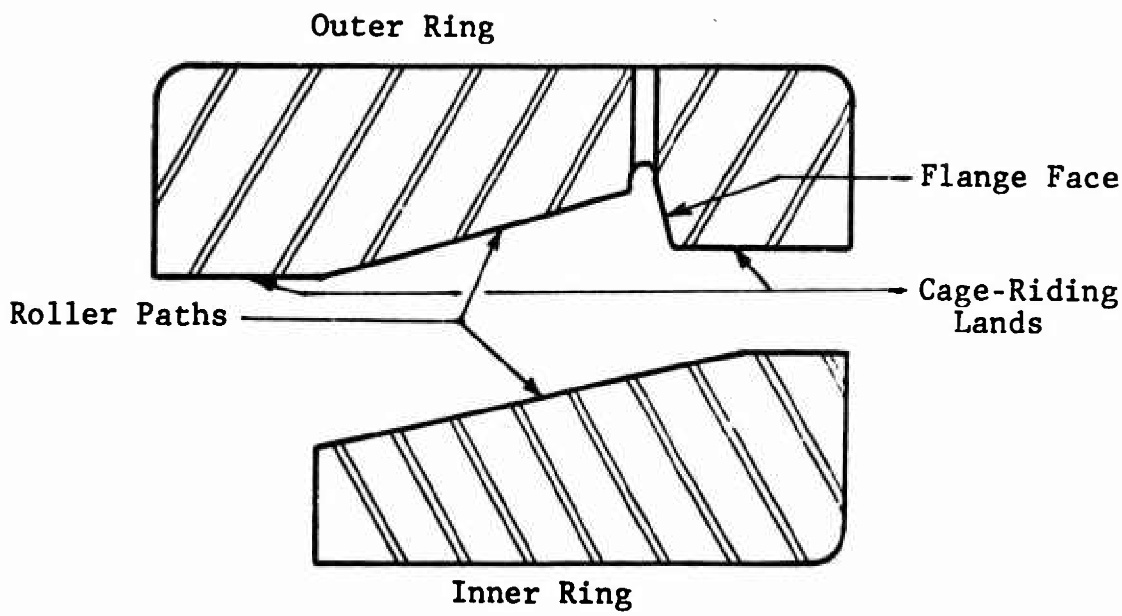


Figure 5. Axial Profiles of Outer-Ring Flanged Tapered Roller Bearing Rings.

"core" of sufficient strength to support the induction hardened case.

C. INNER-RING CASE HARDENING

Induction case-hardening experiments were initiated with the inner rings due to the less complex surface geometry and the relative ease of identifying induction heating parameters with an O.D., as opposed to an I.D., heating application.

In an attempt to expedite the carbide solution kinetics, while simultaneously providing an optimized combination of toughness and strength in the "core" regions, the inner rings were conventionally through-hardened and tempered to a hardness of 32-35 Rc. It was anticipated that an induction hardened case could be imposed on the roller path and cage-riding surface while preserving the 32-35 Rc hardness in the core regions. It was also anticipated that the tempered martensitic structure and finely distributed carbides would be more easily austenitized and dissolved than the conventional pre-heat treat, spheroidized annealed structure.

All inner-ring induction hardening experiments (and the initial outer-ring experiments) used 450 kHz frequency and 40 kw maximum power. These experiments were conducted on the induction heat treat facilities of American Metal Treating, Cleveland, Ohio. The inner rings were statically centered in a wrapped copper tubing inductor. Observation of a ring cross section revealed a condition referred to as "inductor patterning"; i.e., the heat imprint corresponding to the positioning of the individual inductor turns manifests itself as discrete hardened zones rather than as a continuous layer of hardened material. This condition was remedied by rotating the ring during induction heating.

Several combinations of heating time, induced power, and inductor design were investigated, with the results that if full hardness was obtained on the roller path and maintained through tempering, the inner-ring bore hardness was in the 50 to 55 Rc range.

Various attempts were made to cool the bore of the inner ring while the O.D. was induction heated to preserve a low core hardness. A fixture designed to provide quenchant cooling of the I.D. is shown in Figure 6. Results indicated that the only observable effect was to increase the heating time; i.e., the bore hardnesses that resulted were still higher than the 32-35 Rc target values.

These experiments were plagued with inconsistencies in heating times, resulting hardnesses, etc., many of which were attributed to imprecise part centering in the inductor and the inability to precisely reproduce wrapped inductors. The high temperatures being used, the extremely close proximity of the workpiece to the inductor, and the small diameter of the copper tubing used for the inductors (which did not permit sufficient water flow for adequate cooling) resulted in frequent inductor burnout. Since each inductor was wrapped manually and the spacing between turns was adjusted depending upon the hardness pattern produced, it was impossible to make each inductor identical to its predecessor.

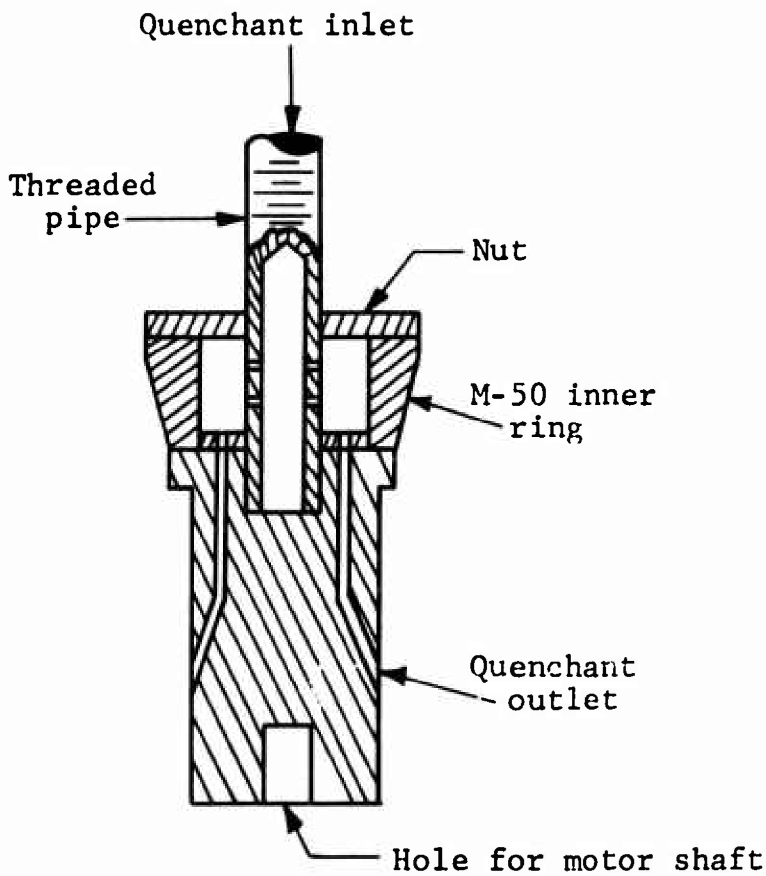


Figure 6. Schematic of Bore Cooling Fixture.

In an attempt to obtain a temperature profile that would be more conducive to preserving the desired core hardness and simultaneously to guard against frequent coil burnout, experiments were performed in which the inner ring and the inductor were completely submerged in circulating quenchant throughout the entire heating cycle. The inner ring was positioned on a plug-type fixture concentric with the inductor, as shown in Figure 7. The intent of this approach was to extract heat from the O.D. as it was being inductively heated in lieu of heat sinking the I.D., which had previously been found to be ineffective in preserving a low core hardness. Extremely erratic and nonreproducible heating rates were experienced under these conditions. It was eventually learned that the quenchant being used (UCON Quenchant "A") has a bulk resistivity of approximately 1.0 ohm when heated to temperatures in excess of $170^{o}F$. Thus, when the inductor was submerged in UCON Quenchant "A" and induction heating was initiated, the quenchant was heated in the vicinity of the inner ring and acted as an effective shunt for the current in the inductor, short-circuiting the inductor and dissipating power intended to heat the workpiece. Coating the inductor with an electrically insulating enamel eliminated the erratic heating behavior.

Twenty rings (32-35 Rc) were induction hardened with good consistency using the coated inductor and submerged heating. Roller path surface hardness was maintained in the range of 60 to 63 Rc through tempering, indicating that sufficient carbide dissolution had occurred to provide for secondary hardening. The bore hardness in the center of the ring height is in the range of 52-56 Rc; whereas in the regions of the ring faces where the positioning fixture

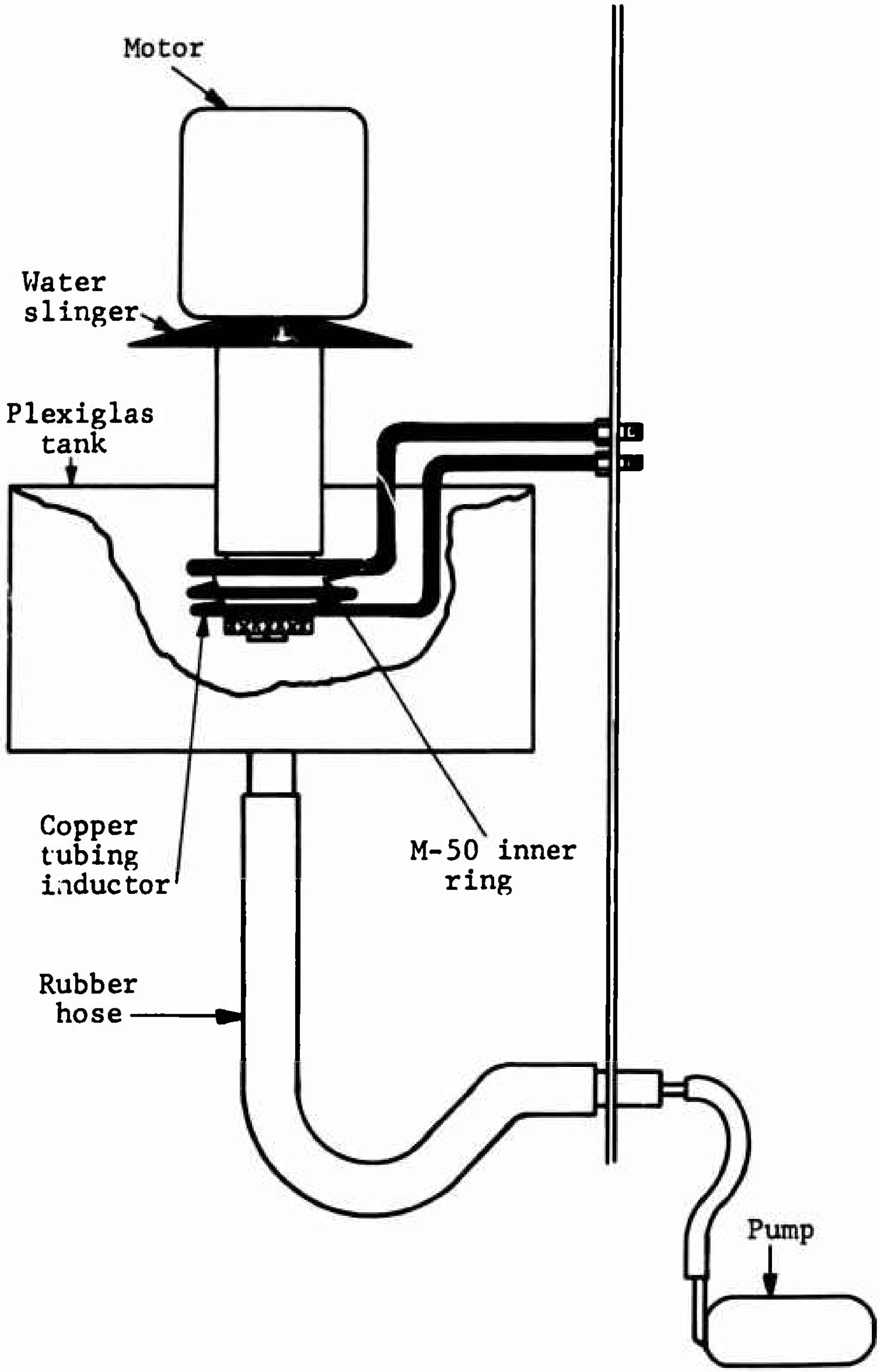


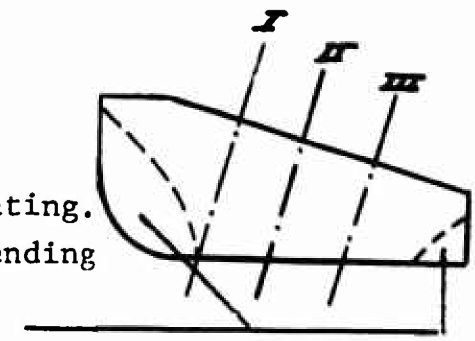
Figure 7. Schematic of Submergent Cooling and Quenching Fixture.

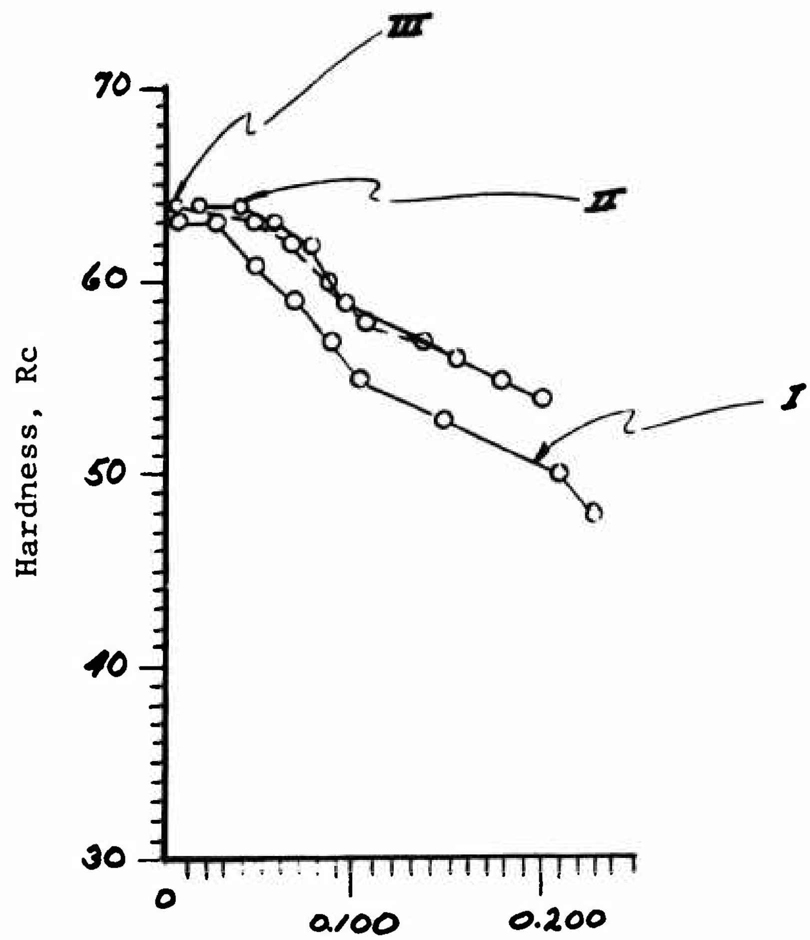
maintained metal-to-metal contact (thereby providing additional heat removal), the original 32-35 Rc hardness was preserved. This is accounted for by the fact that the thermal expansion of the ring during induction heating causes the ring to move away from the bore plug, precluding heat dissipation from the center of the ring bore. Hardness profiles for one of the rings of this group are given in Figure 8.

Even though the bore hardness is higher than that ultimately desired, the roller path surface is representative of good bearing quality induction hardened M-50 steel. The 8 to 10 Rc points hardness difference from the center of the bore to the roller path is sufficient to provide significant residual compressive stresses in the roller path region. Since the program was also intended to provide bearings of induction skin hardened M-50 steel with which to demonstrate the compatibility with rolling contact loading, it was decided that this group of twenty inner rings would suffice for this purpose despite the higher than desired hardness in the center of the bore. The original hardness being preserved in the bore corners of the rings demonstrates the basic feasibility of this induction skin heat-treating process. It is certain that with an effective heat removal technique, inner rings could be manufactured with fully controllable bore hardness. This group of induction skin hardened inner-ring blanks was manufactured successfully into complete inner rings except for final finishing.

D. OUTER-RING CASE HARDENING

Many of the same problems that were incurred in the inner-





Induction Heating Conditions:

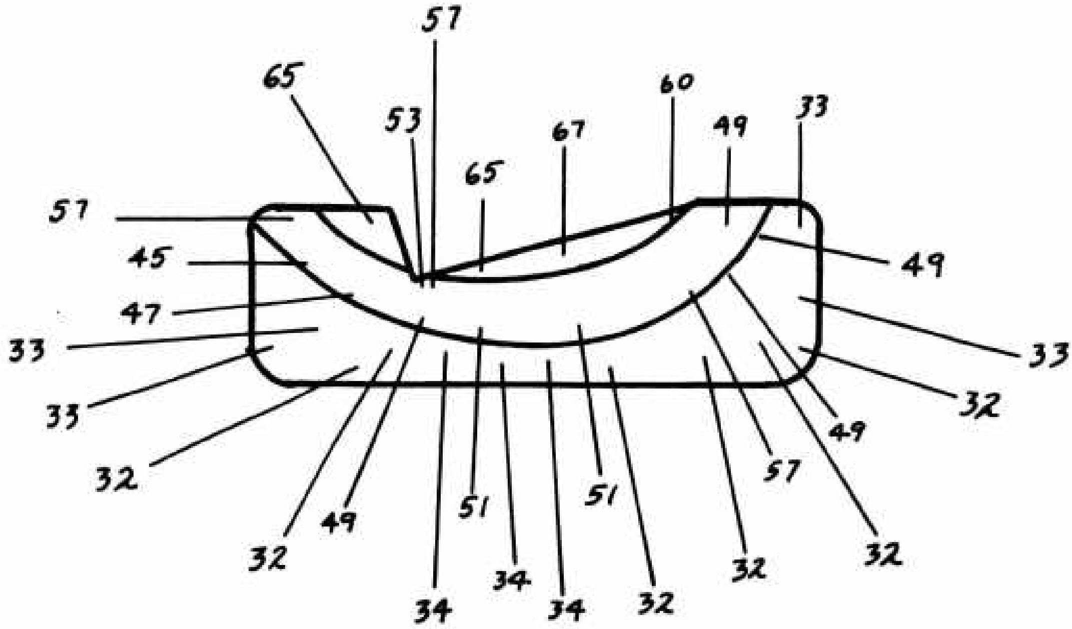
Frequency: 450 kHz
Peak Power: 40 kw
Heating Time: 10 sec
Quench: Part submerged in 16% UCON Quenchant "A" during heating

Figure 8 . Hardness Profiles for Induction Skin-Hardened and Tempered M-50 Inner Ring.

ring skin hardening experiments were experienced with the outer-ring skin hardening, with some additional difficulties introduced by the flanged outer-ring geometry.

An inductor used to heat the outer-ring roller path cannot be any larger than the smallest outer-ring I.D. Since there is an appreciable difference between the smallest I.D. (corresponding to the cage-riding surface on the flange) and that corresponding to the flange face - roller path juncture (see Figure 5), this necessarily implies that the standoff distance between the inductor and the outer ring will be considerably different at these two areas. Since the coupling efficiency between the workpiece and the inductor becomes an increasingly more sensitive function of standoff distance as frequency is increased - a relatively high frequency (450 kHz) was being used - it became apparent that the region of the flange face - roller path juncture was not being sufficiently inductively heated. Even when power levels sufficient to melt the flange I.D. were generated, full hardness could not be obtained at the juncture. Hardness patterns such as that shown in Figure 9 are typical. When the inductor input power was reduced so that the heating time was increased sufficiently to permit conductive heating of this region, the resulting ring was through hardened. Through hardening was most pronounced at the region of the flange face - roller path juncture since this region represents the thinnest section of the ring.

At this point in the heat-treat development, the services and facilities of the Tocco Division of Park-Ohio Industries were employed.

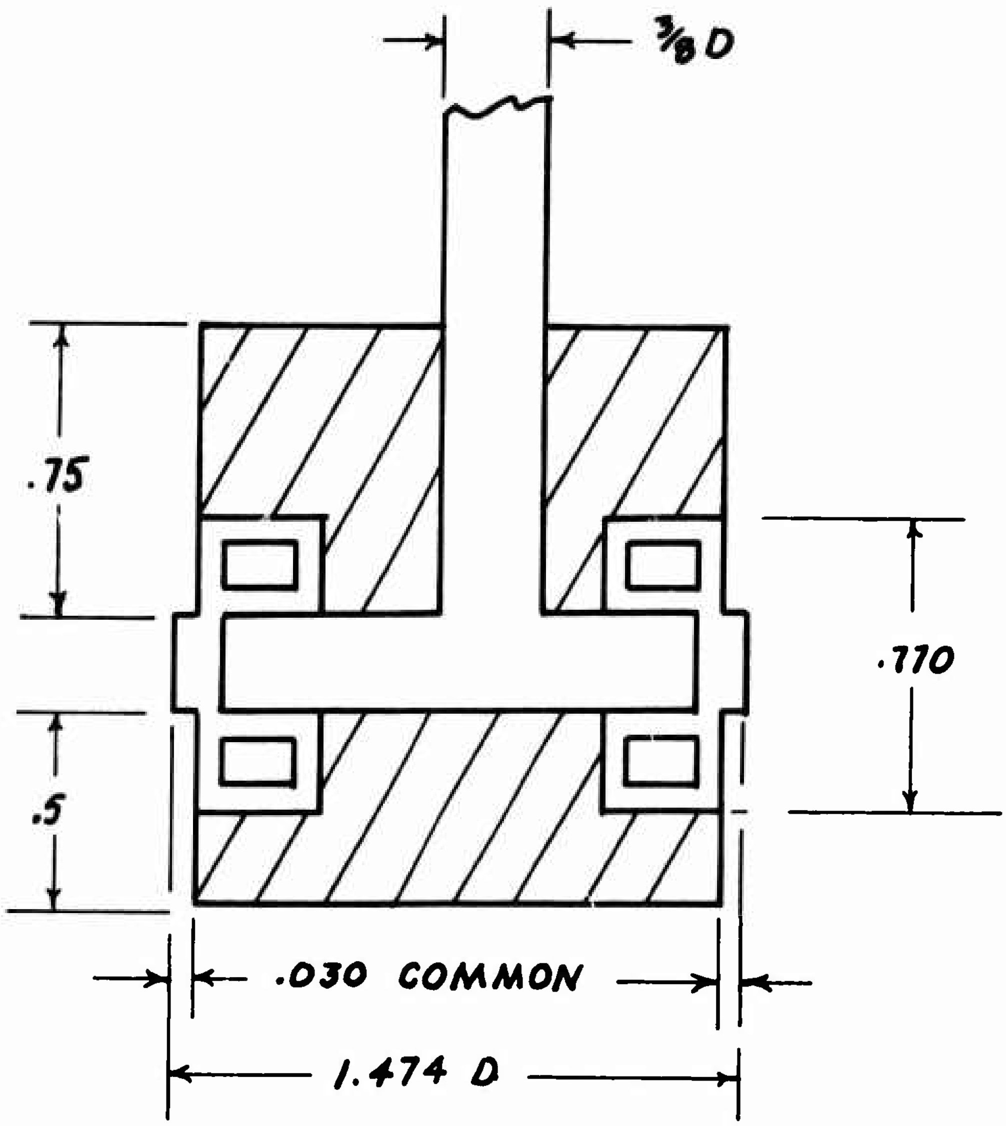


Note: Readings in Rockwell "C" Hardness (untempered)

Figure 9. Hardness Data for Induction Case-Hardened M-50 Steel Outer Ring.

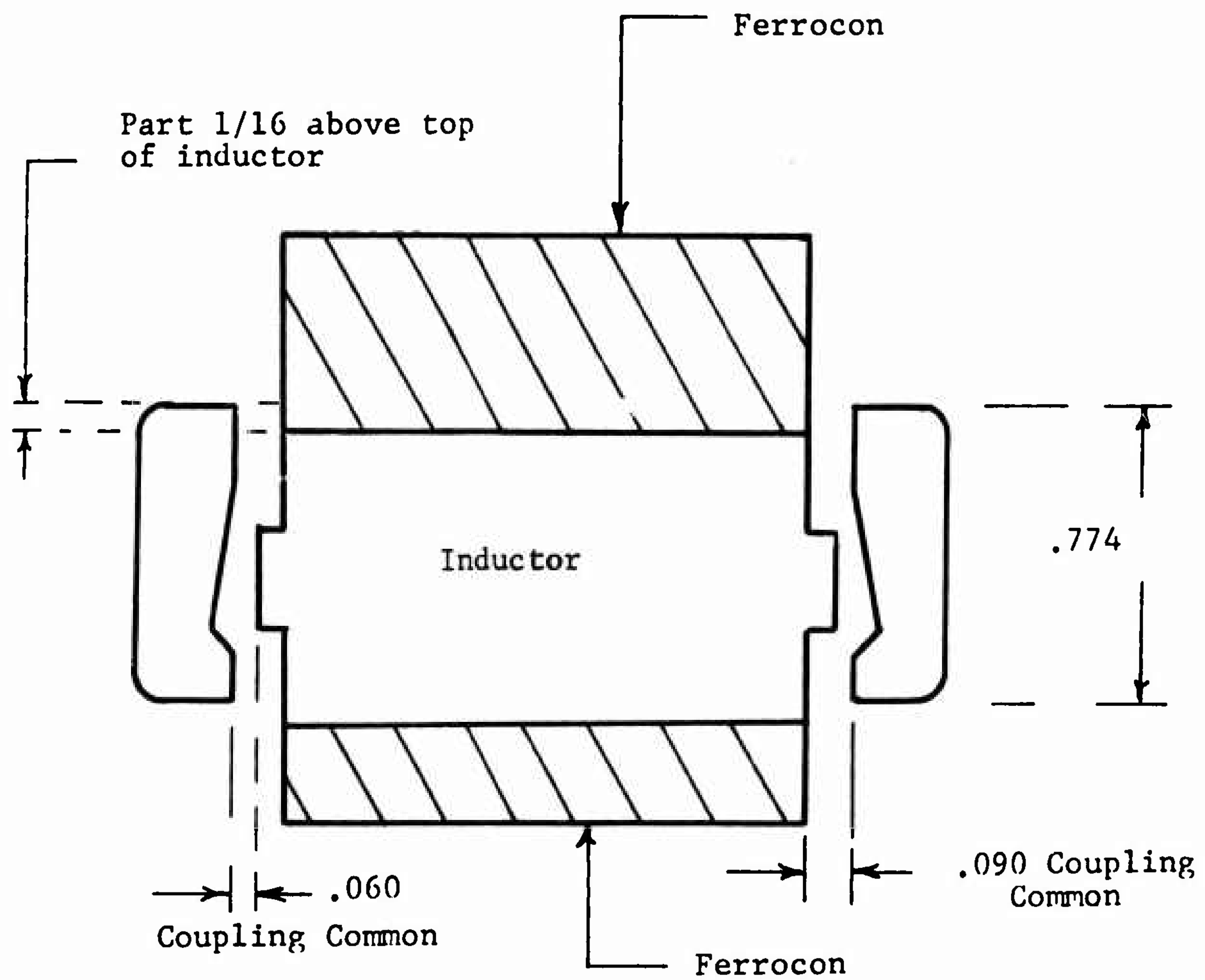
Tocco fabricated a thick-walled machined inductor whose O.D. could be machined to provide the optimum contour for hardening the outer-ring I.D. Sketches of this inductor are shown in Figures 10 and 11. The inductor was filled with a material called Ferrocon, which provides a high permeability path for the magnetic field, thus minimizing coil losses. Rigidly constructed locating and positioning devices were also tailored for accommodating the flanged outer rings. Instead of 450 kHz, a frequency of 10 kHz was used in the experiments at Tocco. The Tocco personnel believed that the inductor-workpiece standoff variation was much too great for radio frequency heating. Experiments were performed with input power as high as 90 kw to heat the surface rapidly, with workpiece and inductor submerged in quenchant as well as in air. The results were very similar to those previously obtained at American Metal Treating; i.e. the flange face - roller path juncture was not being inductively heated, and when conductively heated, the ring would through harden in the juncture region.

It was finally decided that since the flange face - roller path juncture could not be inductively heated and since conduction from the adjacent inductively heated material had to be relied upon for heating, the ring section thickness at this location had to be substantially increased to provide an efficient heat sink. Outer rings with an additional 0.50 inch of stock on the O.D. (i.e., an increase in section thickness of 0.25 inch) were manufactured and conventionally hardened and tempered to 32 Rc. These rings were heated in air while the O.D.'s were sprayed with an oil quenchant, and the rings were submerged in oil at the completion of heating. It was intended to keep the hardened case as



All dimensions in inches.

Figure 10. Schematic of Machined Inductor.



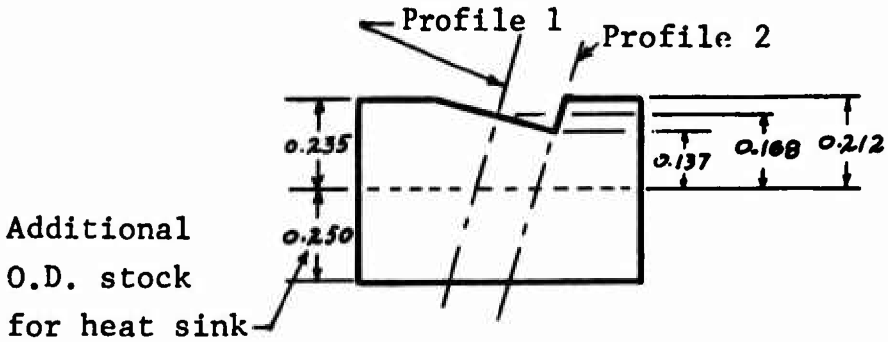
All dimensions in inches.

Figure 11. Orientation of Machined Inductor and Bearing Outer Ring.

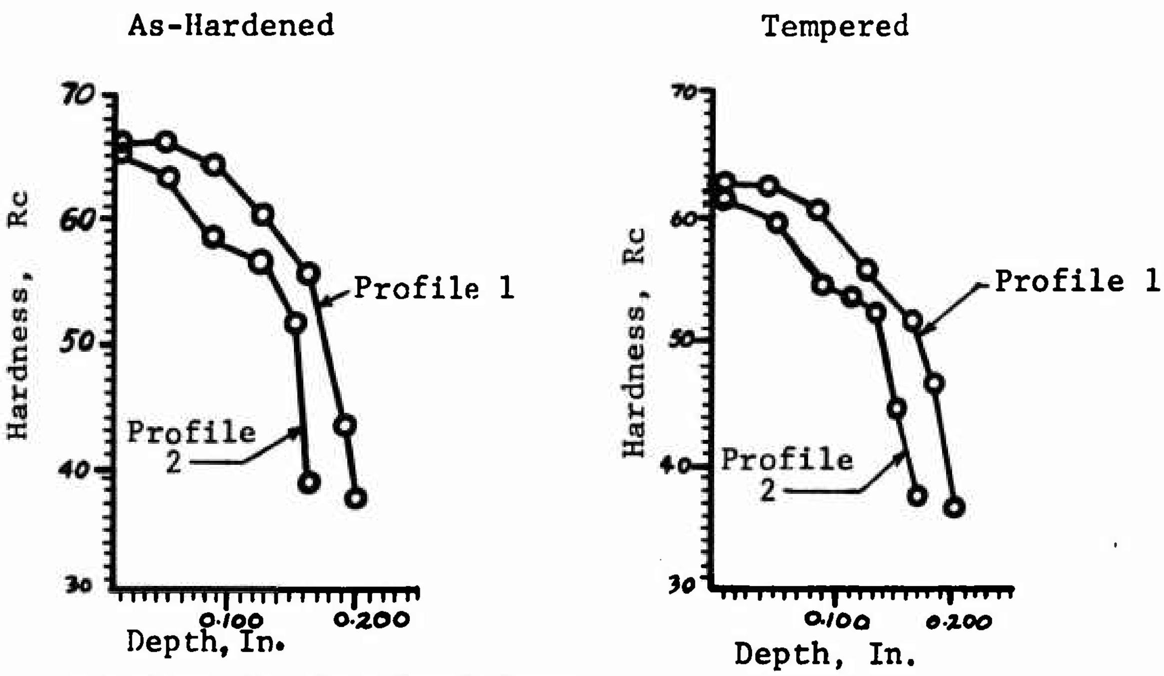
shallow as possible so that all of the extra O.D. stock could be removed while preserving a reasonable case-core ratio in the finished rings.

Figure 12 shows hardness profile data indicating hardness and case depths in the regions of the center of the roller path and the roller path - flange face juncture. It can be seen from this data that full hardness was maintained through tempering in both regions. Within 0.030 inch of the extra stock, the hardness drops below 40 Rc, again demonstrating the basic feasibility of induction case hardening M-50 steel, and highlighting the fact that the geometry of these particular rings imposes the most severe limitations on obtainable case depths, and not the capability of M-50 steel to be induction hardened. Attempts at reducing the case depth by reducing the heating time resulted in incomplete hardening of the flange face - roller path region. A higher power input in conjunction with a shorter heating time may have provided a more shallow case, but this approach was restricted by the equipment limitations.

Figure 13 contains photomicrographs of the microstructure of the outer ring for which the hardness profiles are given in Figure 12. The matrix is martensitic and free of soft constituents that would be indicative of incomplete austenitization. The microstructural characteristics and coarseness are comparable to those normally observed in conventionally through-hardened M-50 steel.



All dimensions in inches.



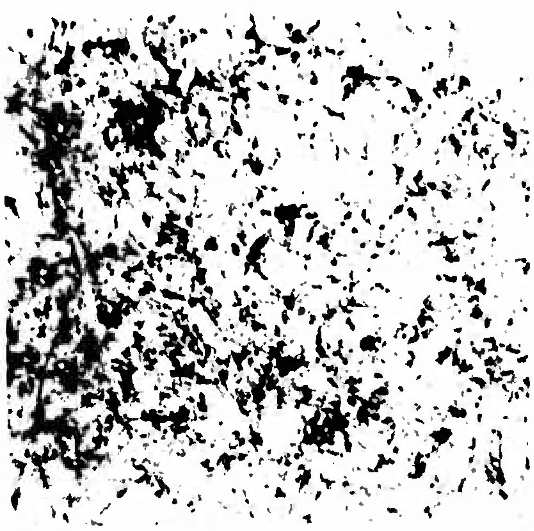
Induction Heating Conditions

Frequency: 10 kHz
Peak Power: 86 kw
Heating Time: 14 sec
Delay Before Quench: 6 sec

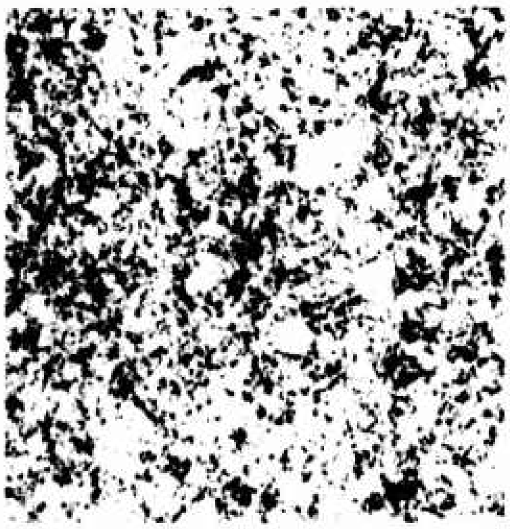
Figure 12. Hardness Profiles for Induction Skin-Hardened M-50 Flanged Outer Ring.

0.010 inch beneath induction
hardened surface (Rc 62)
Nital etch - 500X

0.025 inch beneath induction
hardened surface (Rc 62)
Nital etch - 500X



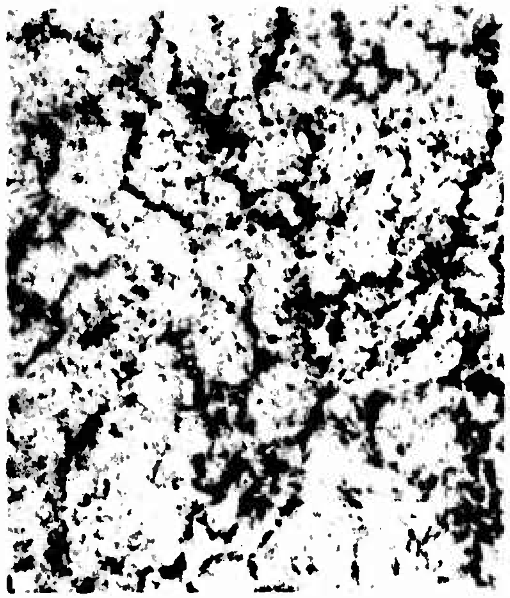
0.050 inch beneath induction
hardened surface (Rc 62)
Nital etch - 500X



0.080 inch beneath induction
hardened surface (Rc 62)
Nital etch - 500X

Figure 13. Photomicrographs of Induction Skin-Hardened
Outer Ring Sections.

0.120 inch beneath induction hardened surface (Rc 60)
Nital etch - 500X



0.180 inch beneath induction hardened surface (Rc 51)
Nital etch - 500X

0.240 inch beneath induction hardened surface (Rc 36)
Nital etch - 500X

Figure 13. (Continued).

VI. BEARING TEST RESULTS

Prior to testing, the test equipment was modified to increase the speed capability of the rig. A new shaft, oil insertion system and turbine wheel were designed and manufactured. The improved shaft design provided stability and eliminated the critical speed problems encountered in previous tests. The methods of lubrication included inner ring, outer ring, lubrication nut, inboard jet and outboard jet to the test bearing. The new turbine wheel was designed to obtain higher efficiency and thus provide increased torque at the same airflow rate.

Five tests were performed using the same bearings of L-42023A design. This is the design using through-hardened M-50 steel rings and rollers in the improved lubricant supply scheme described previously in this report under Bearing Design, but with the same roller end and flange geometry and cage configuration as the 52100 steel bearing tested previously.[1] A thrust load of 240 pounds was applied to the test bearing through the load plug and reacted through the rig bearing and shaft. During the first four tests, oil was supplied to the test bearing through all five flow paths provided. During the fifth test, the outer ring lubricant flow path was used as a drain. Thermocouples were employed to determine the test and rig bearing outer-ring temperatures and the oil inlet and outlet temperatures.

The first three tests were performed at low speeds. Tests 1 and 2 were "checkout" tests to determine the operating characteristics of the modified rig and to permit correction of minor rig problems encountered. In Test 3 the shaft was slowly accelerated to 22,000 rpm, and alternative lubrication methods and flow rates were evaluated. The shaft was then accelerated to 33,000 rpm and operated for approximately one hour before the test was terminated.

The effect of changing the lubricant flow rates through each path was determined by supplying a set flow rate through each path and then modifying the flow through a single independent path by first shutting it off and then increasing the flow to a higher value than normal. The effects on shaft speed and bearing temperature were noted during each change. The flow rates were established from monitoring the inlet line pressure and converting to a flow rate in accordance with pretest calibration values. The test results are presented in Table 1.

TABLE 1 - EFFECTS OF LUBRICANT FLOW RATE CHANGE ON BEARING TEMPERATURE AND SHAFT SPEED

Lube Supply Method	Standard Setting Pressure (psi)	Standard Setting Flow Rate (gpm)	Modified Setting Pressure (psi)	Modified Setting Flow Rate (gpm)	Change from Std. Speed (rpm)	Change from Std. Temp. (^{o}F)
Outboard Jet	10	0.15	0	0	No Change	No Change
	10	0.15	20	0.25	-5000	-15
Inboard Jet	10	0.25	0	0	No Change	No Change
	10	0.25	20	0.35	-2000	-5
Lube Nut	10	0.33	0	0	-1000	0
	10	0.33	26	0.72	-1500	-5
Inner Ring	10	0.098	0	0	+3000	+5
	10	0.098	32	0.25	-3500	-10
Outer Ring	10	Not	0	Not	+3500	+35
	10	Calibrated	32	Calibrated	-1000	-5

It is noted that the results are inconsistent; i.e., in some cases a decrease in flow resulted in no change in temperature or torque (indicated by speed change of the turbine-driven rig at constant turbine airflow), while an increase of a similar flow increment produced an appreciable change. In another case the effect was essentially the same for an increase and decrease in flow rate. This condition is attributed to the mixing of the oil from the various paths within the bearing. Therefore, it was concluded that the effects of varying the flow rates through the various paths could not be catalogued to provide a guide to making adjustment during high-speed tests. Changes would have to be made and the effect noted at each speed where changes in lubrication supply appeared to be necessary.

While running at 35,000 rpm, speed variations as great as 5000 rpm were noted, thus indicating the sensitivity of the air turbine to torque variations. After test termination, the rig was disassembled for inspection to determine the reasons for speed fluctuations. All components were in good condition, with some minor dents in the bearing raceways. It was believed that the passage of small metal chips of unknown origin had caused the fluctuations. The rig oil system was thoroughly recleaned and reassembled.

Test 4 was the first high-speed attempt. Speeds were gradually increased and conditions observed. A speed in excess of 90,000 rpm was attained for approximately 15 minutes, with an ultimate speed of 100,000 rpm (2.5×10^6 DN) being reached for a brief time. The recorded bearing and oil temperatures at the high-speed conditions are presented in Table 2. The test rig operated exceptionally well at these speeds, with much less vibration than was encountered in the previous rig.

TABLE 2 - TEST 4, HIGH-SPEED TEMPERATURE DATA			
Time	13:00	13:45	15:00
Shaft Speed (rpm)	66,200	79,100	97,400
Rig Brg. Temp. (^oF)	275	370	455
Test Brg. Temp. (^oF)	357	395	453
Oil In Temp. (^oF)	122	143	150
Rig Brg. Oil Out Temp. (^oF)	198	208	244
Test Brg. Oil Out Temp. (^oF)	187	208	293

During the run, the sensitivity of the speed to oil flow was noted. The oil flow rates were adjusted to minimize churning losses while allowing the bearing outer-ring temperature to reach values between 400^o and 450^{o}F, which is well within the capability of the M-50 material. The bearing had a decided tendency to pump oil through the outer-ring lubrication holes at the high speed. Definite oil pressures were noted in these lines even when the oil supply was shut off. This pressure increase at the outer race - flange interface indicated that the holes located at this junction are better utilized as drain holes than lubricant feed holes to prevent excessive heating and churning of trapped oil.

This test was terminated due to excessive oil inlet temperature and a high leakage rate in the rig piping system.

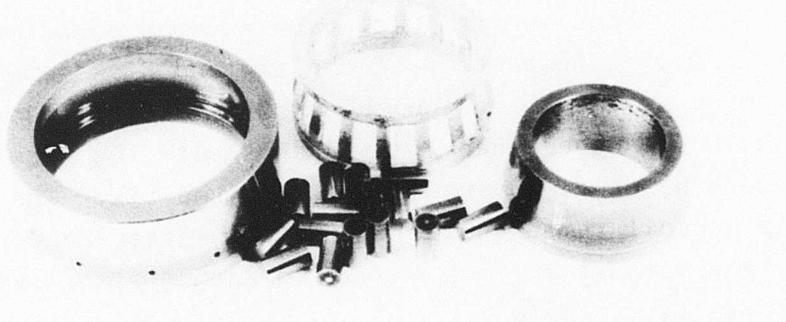
Examination of the bearings after this test revealed that they were in very good condition. The roller ends showed some minor wear, and the flange had developed a straw color; however, no smearing had occurred. The cages were in excellent condition, with very minor polishing on the

O.D. land riding surface and minor wear in the pocket. Post-test photographs of the test bearing components are presented in Figure 14.

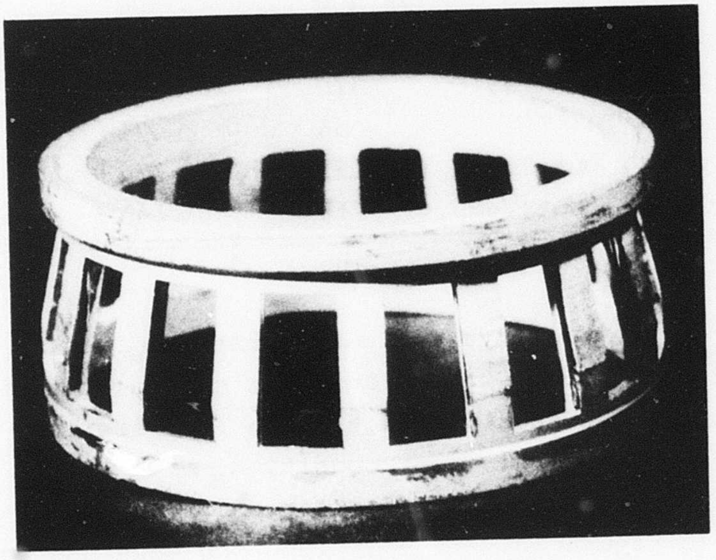
In preparation for the next test, the rig piping system was modified to eliminate the leakage problems which had been encountered. Also, the piping through the heat exchanger was modified to provide more efficient cooling of the inlet oil, and a scavenge pump was connected to the oil outlet holes on the outer ring to provide a positive drain at this location.

The rig was assembled with the same two bearings, and the fifth test run was started. Lubrication was supplied through the lubricator nut and through the outboard and inboard jets. The outer ring holes were used as a drain, and the inner-ring supply was turned off. The supply of lubricant through the lubricator nut at the small end of the inner race proved to be the best method of supplying lubricant for temperature control with the lowest churning losses. Due to the comparatively low torque of the air turbine, a small change in lubricant flow had a relatively large effect on speed of the shaft.

The speed was increased in steps to 92,000 rpm, with the oil flows being adjusted to maintain the bearing test temperature in relatively low range. At this point the air supply to the turbine was at a maximum; in order to increase the speed, the oil levels were adjusted to sacrifice some bearing temperature for increased speed. During this adjustment the speed increased rapidly to 100,000 rpm. A small adjustment of the air control caused a drop in



Disassembled test bearing



Close-up view of test bearing cage

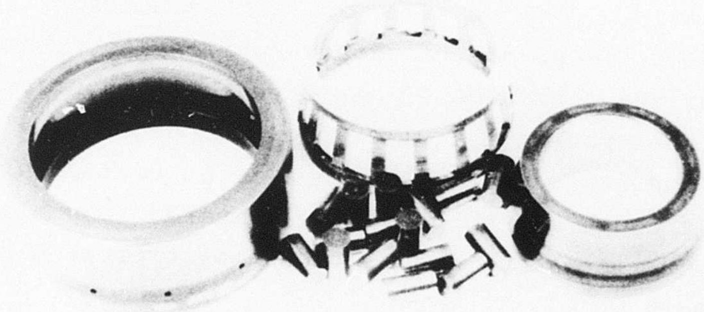
Figure 14. Photographs of Test Bearing Components, Showing Excellent Condition After Test 5 (Appearance Unchanged Since Test 4).

speed to 66,000 rpm. The speed was being brought back up when it began accelerating very rapidly, reaching 108,000 rpm; at that point a noise was heard and the speed dropped to zero. During inspection of the rig and disassembly, it was found that the blades on the turbine had all been sheared from the hub. Other than limited damage to the nozzle box, there were no other adverse effects on the rig. The recorded bearing and oil temperatures at the high-speed conditions, which were maintained long enough to permit stabilization, are presented in Table 3.

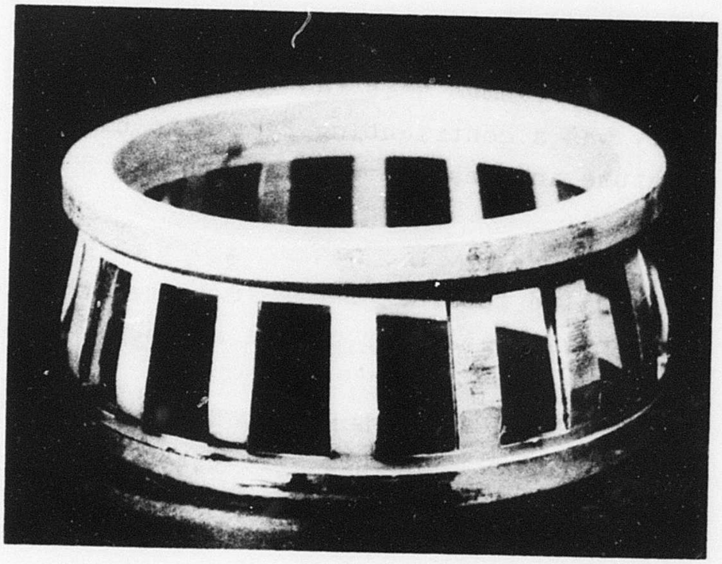
TABLE 3 - TEST 5, HIGH-SPEED TEMPERATURE DATA

Time	11:20	11:43	11:53
Shaft Speed	43,400	74,500	92,400
Rig Bearing Temp. (°F)	305	360	400
Test Bearing Temp. (°F)	310	360	-
Oil In Temp. (°F)	64	71	86
Rig Bearing Oil Out Temp. (°F)	151	174	176
Test Bearing Oil Out Temp. (°F)	115	140	147

Inspection of the bearings indicated that the test bearing was in good condition, with no apparent change occurring during Test 5. The turbine end bearing (rig bearing) was found to be in relatively poor condition, with slight smearing and wear on the large roller end flange contact surfaces; see Figure 15. In addition, the cage had operated in an unbalanced condition and had worn excessively. A detailed inspection of both the test and rig bearing contact surfaces was performed with both light microscopy and under high magnification using a scanning electron microscope (SEM).



Disassembled rig bearing



Close-up view of rig bearing cage

Figure 15. Photographs of Rig Bearing Components, Showing Slight Flange and Cage Land Wear After Test 5.

This inspection further revealed the generally poor condition of the rig bearing (it was not as well lubricated as the test bearing, which was in excellent condition) and emphasizes the importance of proper lubrication. The detailed surface condition of the cage pocket contact surfaces, inner raceways, roller ends and outer-ring flanges of the test and rig bearings is presented in Figures 16 through 20, which can be compared to published SEM and light microscopy documentation of typical bearing surface damage.[5,6]

Due to the unknown effect of rig space limitations, the lubrication through the shaft to the lubricator at the small end of the inner race was provided only to the test bearing. This method of lubricant supply proved to be the best method for control of the bearing temperature at high speed. Thus it is presumed that the lack of the inner-race lubricant supply was a contributing factor to higher bearing temperature and resulting damage to the turbine end rig bearing.

A review of the failed turbine pieces indicated that a thin, rehardened layer of material was on the surface. This

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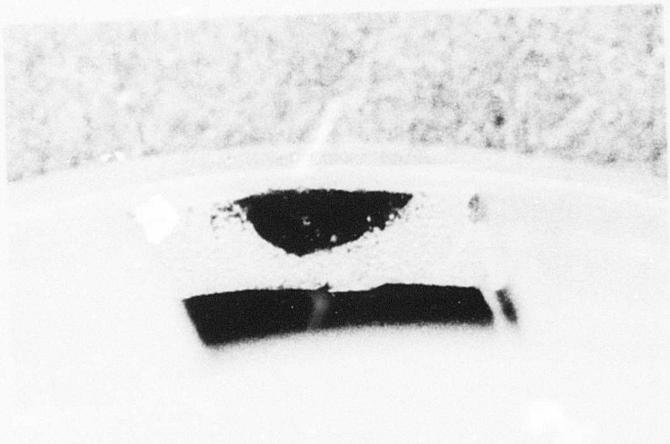
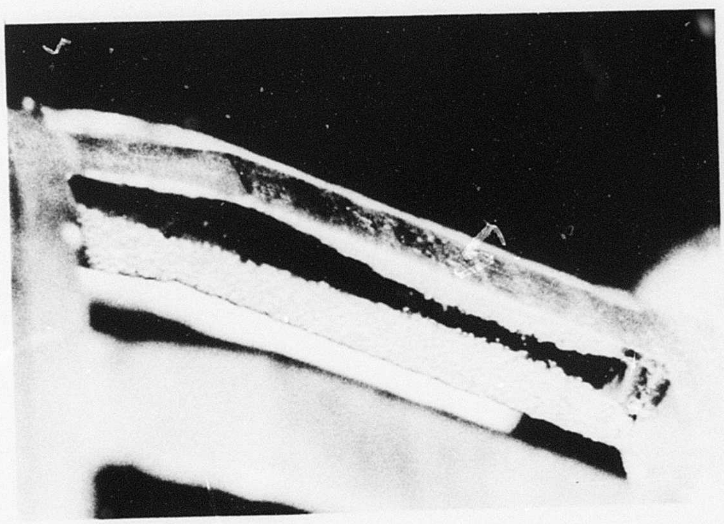
[5] Beerbower, A., MECHANICAL FAILURE PROGNOSIS, USAAMRDL Technical Report (Draft), Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, 1974.

[6] Tallian, T. E., et al, ROLLING BEARING DAMAGE, A MORPHOLOGICAL ATLAS, Library of Congress Card No. 74-81983, S K F Industries, Inc., Philadelphia (1974).



10X

Photomacrograph of polished silver plate on cage rail at large end of roller pocket (test bearing)



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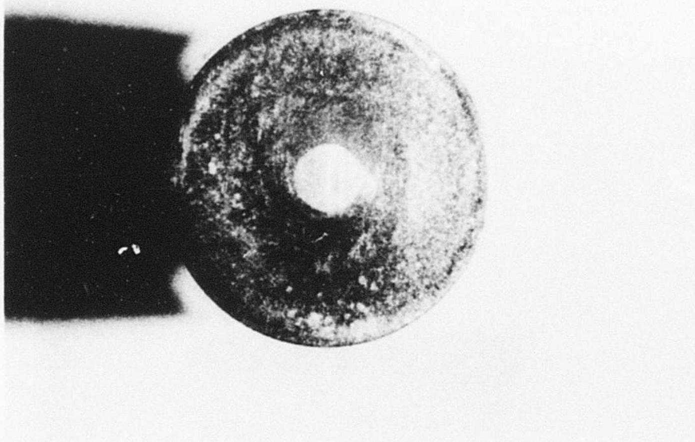
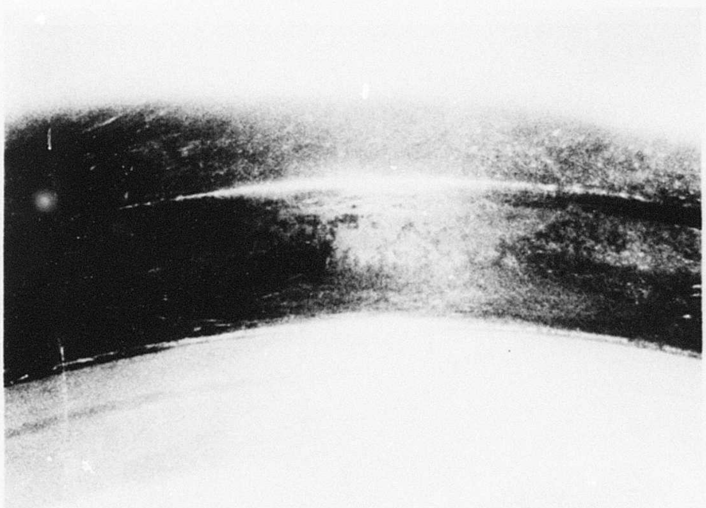
Photomacrograph of polished silver plate on test bearing cage bar

Figure 16. Detailed Wear Pattern in Test Bearing Cage Pocket.

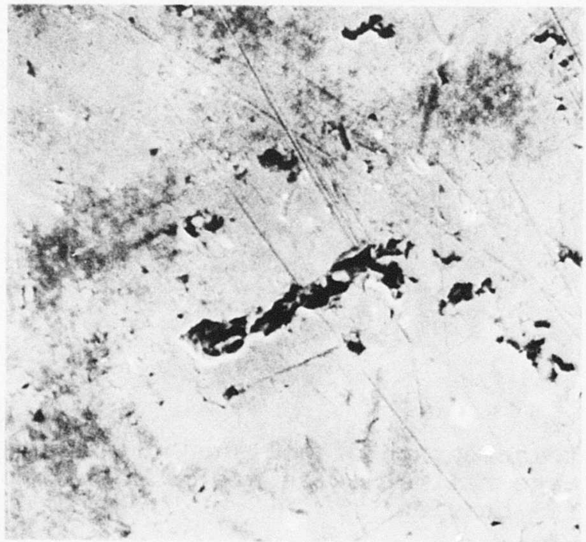


10X
Photomacrograph of contact surface on large end of roller



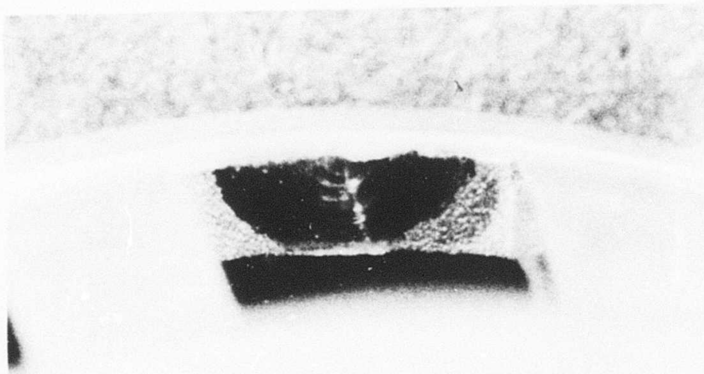
10X
Photomacrograph of contact surface
on outer-ring flange

Figure 17. Detailed Surface Damage on Test Bearing Roller End and Flange.

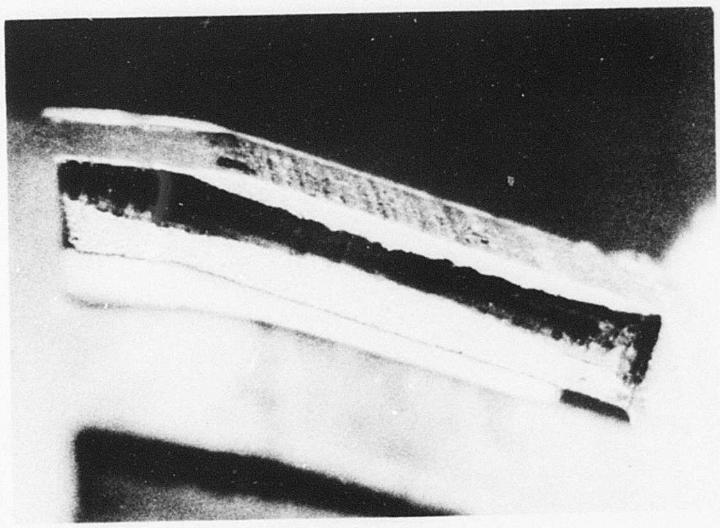


1000X
SEM photomicrograph of typical micropitting on flange surface

Figure 17.(Continued).

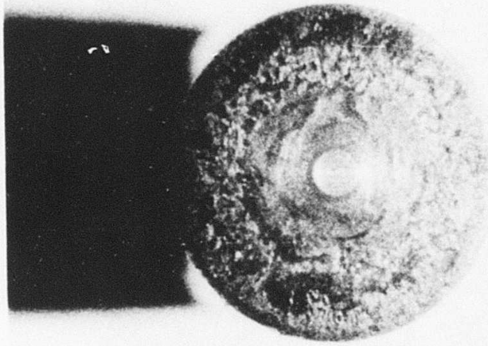


10X
Photomacrograph of polished silver plate on cage rail at large end of roller pocket in rig bearing

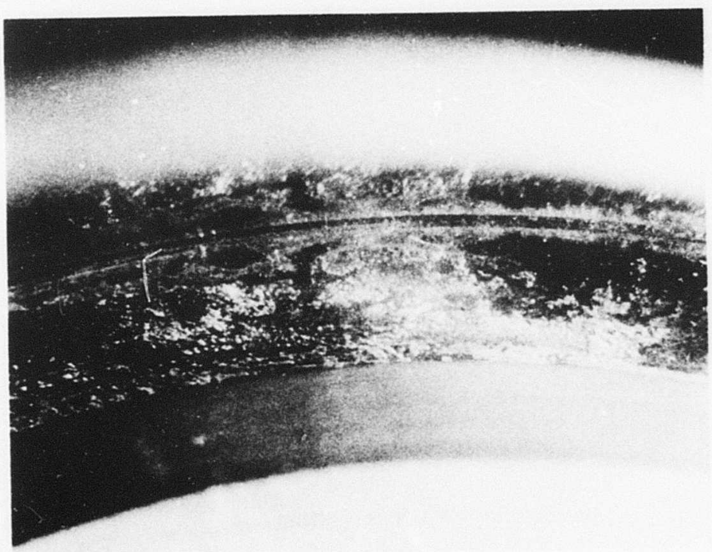


10X
Photomacrograph of polished silver plate on rig bearing cage bar

Figure 18. Detailed Wear Pattern in Rig Bearing Cage Pocket.

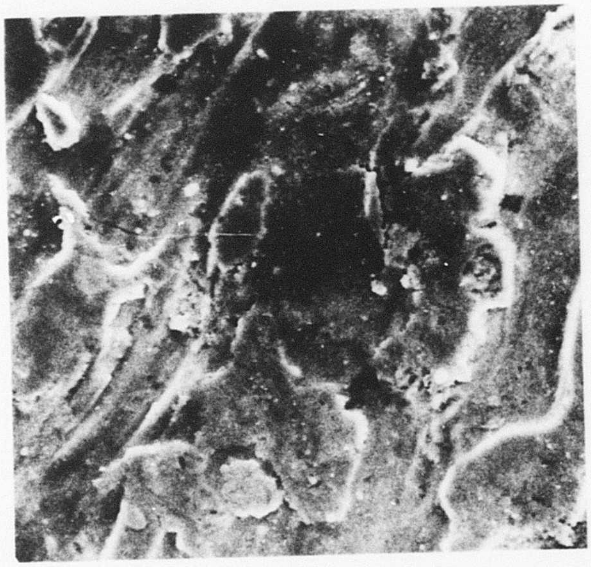


10X
Photomacrograph of smearing on large end of roller



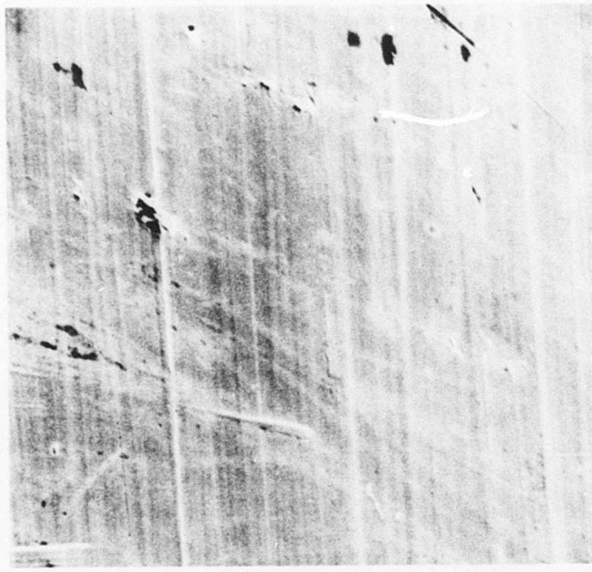
10X
Photomacrograph of smearing on outer-ring flange

Figure 19. Detailed Surface Smearing Damage on Rig Bearing Roller End and Flange.

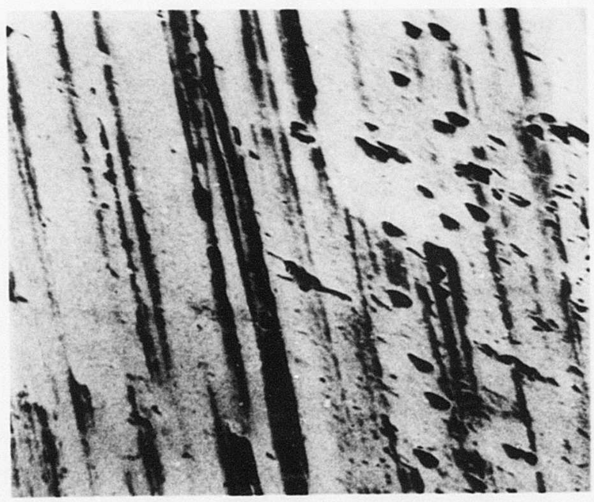


1000X
SEM photomicrograph of typical smeared surface on flange

Figure 19.(Continued).



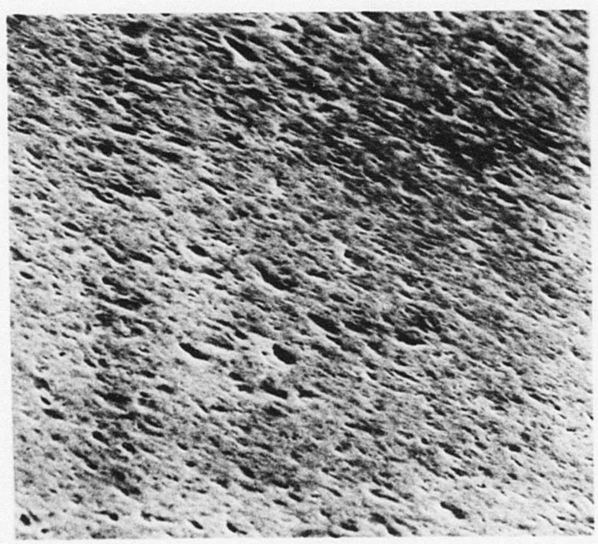
1000X
Typical noncontact surface, showing finishing marks



1000X
Test bearing contact area, showing minute denting and smoothing of finishing marks

Figure 20. SEM Photomicrographs of Inner Ring Tracks on Test and Rig Bearings.

1000X
Near center of rig bearing track, showing surface distress and lack of finish marks



1000X
Discolored band at large end of rig bearing track, showing dents and wear from flange debris

Figure 20. (Continued).

layer remained after the electrical discharge machining of the component, indicating that the electro-polishing of the component was not adequate to remove this layer. Such a micro rehardened layer is known to be prone to micro-cracking, which could have been a contributing factor to the turbine failure.

Although only limited testing was performed, the results of Tests 4 and 5 showed the feasibility of using special-design tapered roller bearings in ultrahigh-speed, high-temperature gas turbine engine and transmission applications where both thrust and radial load are applied.

VII. CONCLUSIONS

1. Tapered roller bearings of an aircraft precision outer-ring flanged design are capable of operating successfully to over 2.5 million DN when manufactured of M-50 steel and adequately lubricated. All contact surfaces of the rings, rollers and cage were in excellent condition on a bearing tested briefly at these extreme speeds, thus demonstrating the basic suitability of the design, which included a critical roller end radius as low as 70% of the bearing apex radius.

2. Induction skin hardening of M-50 steel rings of relatively thin section can supply a hard "case" section while maintaining a relatively soft core. An induction frequency of 10 kHz at 86 kw power was most successful, using a quenchant spray during heating followed by submersion in quenchant after a brief delay.

3. Lubrication applied through the shaft to the small end of the roller (under the cage) is the most effective method of lubricating the bearing and thus controlling temperature. In future test rig designs or in aircraft applications, this technique of lubricating a tapered roller bearing should be incorporated if possible.

4. Lubrication access holes located at the outer-ring race and flange interface are more effective in obtaining proper lubrication of the roller end/flange contact when used as drain paths than when used as oil supply paths. The use of a scavenge pump to drain oil through the holes improved the lubrication of the roller end flange sliding contact.

VIII. RECOMMENDATIONS

1. The development effort to date has shown that small-size (25mm bore) tapered roller bearings can be manufactured of M-50 tool steel and operated to over 2.5 million DN. Further testing is warranted to optimize the configuration by comparing the performance of different tapered roller bearing design features with current ball bearing designs and to determine the long-term wear performance and life of the bearings under extended high-speed service with the type of lubrication systems required for future aircraft applications.

2. This development effort proved the feasibility of induction heat treating M-50 tool steel bearing rings to produce a hardened outer layer on the ring contact surfaces while retaining a softer core structure. This development has significant possibilities for improving aircraft bearings and other engine, gearbox and transmission components. The resistance to shock, both ballistic and equipment induced, plus the resistance to crack propagation .ill allow the use of this superior material for both bearings and multicomponent parts. It is recommended that bearing fatigue tests be carried out to determine the benefits of the induction skin hardened M-50 material.

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