# Development of Counter-Rotating Intershaft Support Bearing Technology

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**Analytical studies on intershaft cylindrical roller bearings for advanced gas turbine engines configured with counter-rotating shafts showed advantages in fatigue life and internal radial clearance control when the outer ring was mounted on the low-speed rotor and the inner ring on the high-speed rotor. Parametric rig tests on eight bearings showed that the primary drivers on roller end wear were roller-to-guide flange end clearance, outer race preload, and internal radial clearance. Test results showed concentric roller end wear patterns on all test bearings and varying levels of wear. The performance data were used to improve prediction techniques for bearing heat generation and temperatures.**

## Introduction

STATE-of-the-art gas turbine engine mainshaft bearings function successfully up to 2.2 million DN, where DN = bore diameter (mm) × speed (rpm). To meet advanced technology fighter engine performance and thrust-to-weight goals, increases in shaft speed of 20% and above are required. Increased shaft speeds will increase bearing DN levels up to 3 million and higher, which will require technology advancement and bearing design system development to adequately address complex bearing performance/ operational parameters, such as ring fracture margin, roller stability, fatigue life, heat generation, cage slip, and wear.

Advanced engine configuration studies have shown large life cycle cost advantages for an engine with counter-rotating spools and a rotor support system in which the high-speed rotor is straddle mounted (bearings on each end) with an intershaft support bearing at the high-pressure turbine. As a result, advanced engine designs are incorporating intershaft support bearings with operational requirements up to 3.7 million DN (equivalent) and roller rotational speeds up to 170,000 rpm, over twice conventional levels.

The objective of this program was to define key design parameters for an advanced engine configuration, to update bearing design analysis methodology to address intershaft bearings in counter-rotation, and to conduct tests on eight bearing configurations to evaluate the impact of two levels each of four key bearing geometry parameters on bearing performance and durability relative to a baseline bearing.

## Approach

The program consisted of analytical and experimental effort to develop design criteria and obtain operational performance and wear data for load carrying roller bearings mounted between counter-rotating shafts.

An advanced engine configuration was selected as the vehicle for defining operational requirements and geometric constraints for the intershaft bearing investigation. An existing 124 mm bore diameter bearing was selected for this application.

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An existing bearing design analysis computer program, TRIBO-I[1] which had been developed for conventional bearing mounting arrangements (one race rotating, one race stationary) was modified for intershaft mounting applications and was utilized to predict intershaft bearing operation and performance. Program results were then utilized to improve the computer program and operational prediction capabilities.

Results from previous bearing programs on 124 mm and other size bearings were utilized in the selection of four bearing geometric parameters most likely to influence intershaft bearing performance and durability. A statistically derived matrix was generated as the basis for an experimental test program to evaluate, with a high level of confidence, the influence of bearing geometry on bearing heat generation, cage slip, thermal response, and wear. A total of nine bearings, consisting of a conventional geometry baseline and eight configurations employing controlled variations in the key geometric parameters were obtained for experimental evaluation.

The test vehicle, an intershaft support bearing rig, was capable of simulating anticipated advanced engine rotor speeds, bearing loads, lubricant supply conditions (flows, pressures, temperatures), engine mounting arrangements, and bearing misalignment. Extensive instrumentation was incorporated in the rig to monitor both rotating (both rotors) and static bearing operational and performance parameters. Bearing inspections at selected intervals during the test program were conducted and led to wear vs time data.

At the conclusion of the test program, results were analyzed and used to modify computer bearing analysis programs and provided the basic information necessary to select a bearing geometry for advanced engine intershaft bearing operation between counter-rotating shafts.

### Design Studies

Prior to this program, studies were conducted to determine an optimum rotor support system, shaft rotation, and bearing mount arrangement as applicable to twin-spool advanced military fighter engines. A typical advanced engine configuration was used for the studies since it was a technology vehicle having the size and operational requirements projected for future engine requirements.

The rotor support system arrangement selection was based on an analytical study which compared a high-pressure rotor bearing arrangement with an overhung turbine to two straddle-mounted configurations, one with a hot strut and the other with the high spool supported off the low spool through an intershaft bearing. Engine designs were generated for each rotor support configuration and analyzed for running clearances, weight, manufacturing cost, and maintainability. The study results led to the selection of a straddle-mounted configuration with an intershaft support bearing as the optimum based on life cycle costs.

Further studies were conducted to determine whether advanced engine performance and durability requirements were best satisfied with corotating or counter-rotating shafts. The study led to the selection of counter-rotation for advanced engines due to improved performance, reduced weight, and improved intershaft bearing fatigue life. The improved bearing life results from the very low orbital cage speed with counter-rotation resulting in reduced roller centrifugal forces and tighter internal radial clearance (IRC). Figure 1 compares 124 mm bearing operation for conventional base, co- and counter-rotation applications, the latter two with an increase in rotor speed.

Maintaining minimum growth in IRC, in addition to the life advantages, is essential to maintaining bearing spring rate and controlling shaft critical speed vibration modes. With both inner and outer races rotating, for the application analyzed, IRC increases as rotor speeds increase along an engine operating line. If the smaller diameter race is mounted on the higher speed rotor, this growth in IRC is kept to a minimum and is manageable from a rotor stiffness standpoint. Schematic mounting sketches for high- and low-rotor-mounted inner races are shown in Fig. 2. The low-shaft-mounted inner race is undesirable from the standpoint of IRC control.

In conclusion, the design studies showed that a straddle-mounted turbine high rotor supported through an intershaft bearing onto the counter-rotating low rotor, with the bearing inner race mounted on the high rotor, can yield significant life cycle cost advantages over conventional rotor support systems. Such a rotor was selected for advanced fighter applications.

The high (170,000 rpm) roller speeds, coupled with very low cage speeds in an environment where both shafts are rotating and "communicating" with each other through the bearing presented the bearing designer with a set of challenging conditions which were addressed in this program.

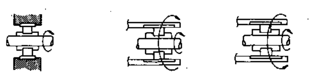
| | Conventional | Co-Rotating | Counter Rotating |
|---|---|---|---|
| |  | | |
| Outer Ring, rpm | 0 | 13,400 | 13,400 |
| Inner Ring, rpm | 15,050 | 16,100 | 16,100 |
| Fatigue Life | Base | 0.12 | 1.9 |
| Element Speed, rpm | 87,000 | 15,600 | 169,000 |
| Cage Speed, rpm | 6,900 | 14,700 | 73 |
| Change in IRC, in. | Base | 0.0058 | 0.0024 |

**Fig. 1 Comparison of conventional, co- and counter-rotational bearing applications.**

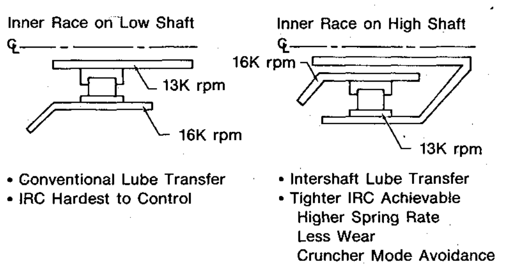



**Fig. 2 Comparison of inner race mount on high shaft vs low shaft.**

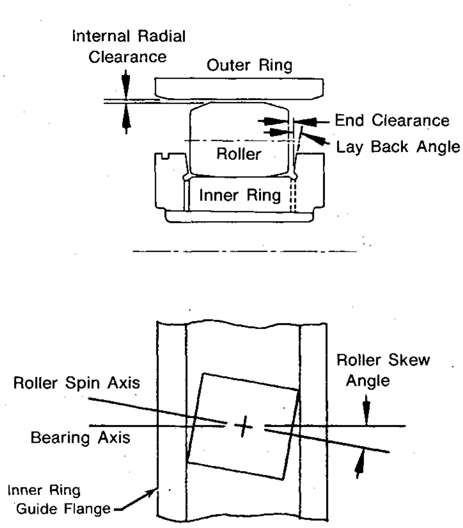



**Fig. 3 Counter-rotating bearing nomenclature.**

## Experimental Program

### Selection of Bearing Geometry Variables and Test Matrix

An existing 124 mm bore roller bearing was selected as the bearing most representative of size and load-carrying capabilities of a typical advanced technology engine. Selection of this bearing affords a great degree of commonality between current production engines and proposed advanced technology engines.

Studies and extensive test programs conducted on this bearing at $3 \times 10^6$ DN have provided a relative ranking of over 30 bearing geometric variables.[1] Based on past experience the primary drivers for wear are roller-to-guide flange end clearance (EC), guide flange layback angle, internal radial clearance (IRC), and outer race out-of-round (preload). The relative importance of these variables has been assessed for conventional bearings, but only recently has their importance begun to receive attention for counter-rotating intershaft bearing applications.

The effects of the four variables on the performance and durability parameters of interest—roller wear, skew angle change, and roller weight loss—were evaluated; layback angle and end clearance combine to determine skew angle (Fig. 3). The initial skew angle has been shown experimentally to correlate with roller wear. Internal radial clearance was selected because computer analyses, recent analytical studies, and test results have shown that the clamping effect of IRC can significantly influence wear. The outer race out-of-round provides an internal load or preload which results from the diametral "pinch." Bearing preload controls skidding which has been shown in test results to affect bearing wear and fatigue life.

A statistically designed parametric test matrix was developed and tests were conducted on the eight bearings after an initial baseline test. The results were examined with mathematical methods, such as regression analyses. Using this technique, it is possible to isolate and rank the effects produced by each variable on roller end wear.

Figure 4 shows the parameters varied for the eight parametric bearings. This matrix permits the study of the main effects of each of the four design parameters. Parameters which were not studied in this program were maintained at the same level used in the baseline bearing. This matrix provides the maximum amount of statistical data with the limited number of tests. Because of the number of tests, the results of the statistically oriented tests will give trends for the effects of high or low levels, but will not establish exact or optimum levels. Regression analysis was used to develop a mathematical model for predicting bearing wear and skew angle change for the levels of the variables to be tested. The criteria used for ranking the performance of the series of test

| Internal Radial Clearance | Preload | Roller End Clearance Low, Layback Angle Low | Roller End Clearance Low, Layback Angle High | Roller End Clearance High, Layback Angle Low | Roller End Clearance High, Layback Angle High |
|---|---|---|---|---|---|
| High | High | 4* | | | |
| High | Low | | | | 7 |
| Low | High | Baseline | 6 | 1, 3 | 2, 5 |
| Low | Low | 8 | | | |

*Parametric Bearing Number

*Nominal Dimensions for High and Low Bearing Geometric Variable Values*

| | End Clearance (in.) | Layback Angle (min) | IRC (in.) | Preload (in.) |
|---|---|---|---|---|
| High | 0.0038 | 40.0 | 0.007 | 0.022 |
| Low | 0.0021 | 15.0 | 0.004 | 0.000 |

**Fig. 4 Counter-rotating intershaft bearing test matrix.**

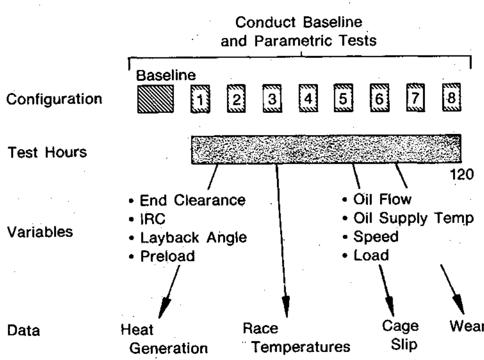



**Fig. 5 Test plan.**

bearings were roller weight loss, change in skew angle, and roller end wear width and pattern.

The objective of the test program was to experimentally evaluate a baseline and a series of parametric bearings. The baseline bearing was of a geometry identical to bearings used in the fixed outer race application in an operational military gas turbine engine and was tested to establish a reference rig data base on which to improve through the parametric test series.

The test program (Fig. 5) shows the experimental portion of the program, test and bearing variables, and output data categories.

**Rig and Facility**

The baseline and parametric tests were conducted in a twin-rotor high-speed test facility which has the capability of conducting corotational, counter-rotational, fixed inner race, and conventional fixed outer race testing up to 24,000 rpm on each rotor.

The intershaft support bearing rig (Fig. 6) consists of two rotors. The high rotor drives the test bearing inner race and the high-speed shaft support bearings. The low rotor drives the low-speed shaft support bearings and the test bearing outer race. The test rig is instrumented to measure bearing and oil temperatures, oil flows and pressures, vibrations, shaft speeds, and test bearing radial load. Slipring rotating instrumentation monitors test bearing as well as slave bearing operation.

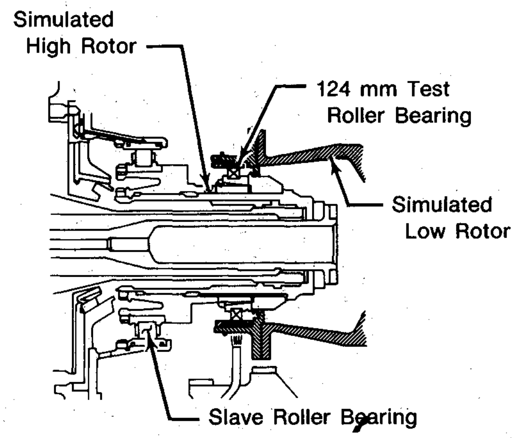



**Fig. 6 Counter-rotating intershaft support bearing rig test section.**

**Test Program**

A test program was designed to experimentally evaluate the bearings under a range of variable conditions, such as rotor speed combinations, oil supply temperatures, flow rate, and radial load. Rotor speed combinations simulated an advanced engine speed schedule while oil supply temperature, oil flow rate, and radial load gave nominal and off-design data for design system modifications.

A sample test program is shown in Table 1. Each parametric test bearing was scheduled for at least 15 hours of testing with sufficient time at each test point to allow for thermal stabilization of the rig.

Detailed bearing inspections were conducted before test, at intermediate points, and after test completion on all bearings. Parameters which were inspected before and after tests included: guide flange layback angle, roller end clearance, skew angle, roller end wear bandwidth and pattern, and roller weight.

Parameters which were monitored during the test included oil flows, temperatures and pressures, inner and outer race temperatures, high and low rotor speeds, cage speed, rig vibrations, radial load, wear metal in oil, and scavenge oil metallic debris.

## Test Results

The program test plan consisted of a 100 hour baseline bearing test and eight 15 hour parametric bearing tests. The parametric program was designed so that iterative feedback of test data could favorably enhance and alter the test program.

**Baseline Test Bearing Results**

With limited industry experience in the testing of high-speed counter-rotating bearings, the test program proceeded with caution. The bearing achieved 95% full speed on both rotors with all recorded parameters within anticipated ranges. When the rig was advanced to full speed, severe bearing distress occurred, which triggered an immediate shutdown. Post-test analysis of continuous data recording tapes indicated that the outer race temperature had exceeded 600°F, and postrun inspection of the inner race indicated that it reached temperatures comparable to the outer race.

Post-test inspection of the baseline bearing indicated heavy wear on the inner race guide flange with metal smeared over the guide flange, roller running path, outer race, and rolling elements. (During an interim inspection conducted prior to the bearing distress it was noted that none of the rollers showed any evidence of distress or unstable, eccentric roller end wear.) Six cage crossrails were fractured. The remaining crossrails were bent and cracked.

Roller element temperature traditionally had been assumed to be at the mean between the inner and outer race temperatures. Due to the high roller element speed (~170K rpm

Table 1 Typical parametric test program

| Test point No. | Low shaft speed, rpm | High shaft speed, rpm | Oil flow to test bearing, ppm | Oil supply temperature to test bearing, °F | Applied radial load, lbf | Approximate run time, min | Total time, h |
|---|---|---|---|---|---|---|---|
| 1 | 6,200 | 11,600 | 10 | 200 | 1,000 | 30 | |
| 2 | 6,200 | 11,600 | 10 | 200 | 500 | 30 | |
| 3 | 6,200 | 11,600 | 10 | 200 | 250 | 30 | |
| 4 | 6,200 | 11,600 | 10 | 200 | 125 | 30 | |
| 5 | 6,200 | 11,600 | 10 | 200 | 50 | 30 | |
| 6 | 6,200 | 11,600 | 10 | 200 | 0 | 30 | |
| 7 | 12,000 | 14,500 | 10 | 200 | 1,000 | 30 | |
| 8 | 12,000 | 14,500 | 10 | 200 | 500 | 30 | |
| 9 | 12,000 | 14,500 | 10 | 200 | 250 | 30 | |
| 10 | 12,000 | 14,500 | 10 | 200 | 125 | 30 | |
| 11 | 12,000 | 14,500 | 10 | 200 | 50 | 30 | |
| 12 | 12,000 | 14,500 | 10 | 200 | 0 | 30 | |
| 13 | 13,340 | 16,100 | 10 | 200 | 1,000 | 30 | |
| 14 | 13,340 | 16,100 | 10 | 200 | 500 | 30 | |
| 15 | 13,340 | 16,100 | 10 | 200 | 250 | 30 | |
| 16 | 13,340 | 16,100 | 10 | 200 | 125 | 30 | |
| 17 | 13,340 | 16,100 | 10 | 200 | 50 | 30 | |
| 18 | 13,340 | 16,100 | 10 | 200 | 0 | 30 | |
| Teardown for inspection | | | | | | | 9 |
| 19 | 13,340 | 16,100 | 15 | 200 | 1,000 | 30 | |
| 20 | 13,340 | 16,100 | 15 | 200 | 500 | 30 | |
| 21 | 13,340 | 16,100 | 15 | 200 | 250 | 30 | |
| 22 | 13,340 | 16,100 | 15 | 200 | 125 | 30 | |
| 23 | 13,340 | 16,100 | 15 | 200 | 50 | 30 | |
| 24 | 13,340 | 16,100 | 15 | 200 | 0 | 30 | |
| 25 | 13,340 | 16,100 | 10 | 250 | 1,000 | 30 | |
| 26 | 13,340 | 16,100 | 10 | 250 | 500 | 30 | |
| 27 | 13,340 | 16,100 | 10 | 250 | 250 | 30 | |
| 28 | 13,340 | 16,100 | 10 | 250 | 125 | 30 | |
| 29 | 13,340 | 16,100 | 10 | 250 | 50 | 30 | |
| 30 | 13,340 | 16,100 | 10 | 250 | 0 | 30 | |
| Teardown for inspection | | | | | | | 6 |
| Total test time | | | | | | | 15 |

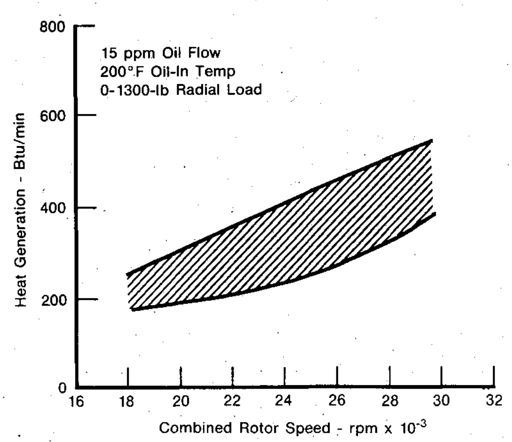



**Fig. 7 Parametric test bearing heat generation vs combined rotor speed.**

about its axis for counter-rotation), it was concluded that the rollers in counter-rotating bearings were operating at temperatures closer to that of the outer race. This may be due to oil churning at the interface of the oil and the roller surface.

The baseline bearing had an initial minimum roller end clearance of 0.0008 in. Analysis of the test results indicated that roller end clearance reduction to zero initiated the bearing distress. A reduction of axial clearance due to retaining nut torque, tight radial fits (between the inner race and the hub), and centrifugal forces reduced the axial clearance by 0.0002 in. The major component in the reduction of end clearance was the thermal growth of the rollers due to a higher than anticipated roller temperature. This thermal growth reduced the end clearance another 0.0006 in., which eliminated the remaining clearance between the rollers and the inner race.

### Parametric Test Program Results

The eight roller bearings were experimentally evaluated at simulated advanced engine operating conditions for a total of 120 h. Data from the program included heat generation, race temperature, cage slip, and roller wear.

#### *Heat Generation*

Bearing heat generation is a measure of the energy imparted to the oil from bearing rolling friction, oil churning, windage, and other effects. Excessively high bearing heat generation results in greater heat rejection system requirements on engines, such as fuel/oil and air/oil coolers. Heat generation is calculated by measuring oil scavenge temperature minus oil supply temperature and multiplying that temperature difference by the fluid specific heat and by the mass flow rate of the oil.

Figure 7 shows the band of bearing heat generation data as a function of combined rotor speed (sum of both rotor speeds). Heat generation data from all the parametric bearings tested was approximately the same as current gas turbine engine main shaft bearings of the same size. Heat

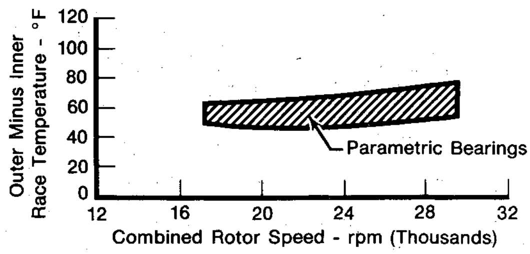



**Fig. 8 Parametric bearing race $\Delta T$ vs combined rotor speed.**

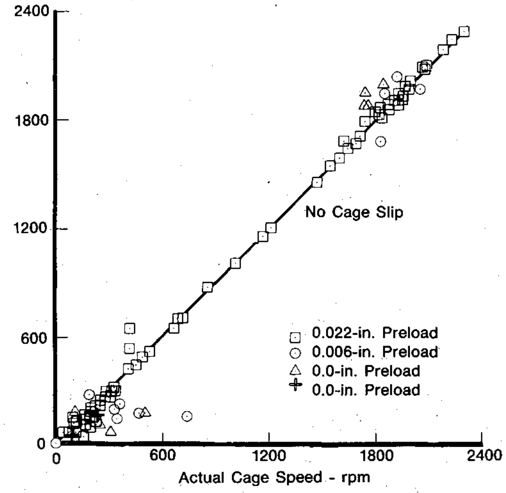



**Fig. 9 Theoretical vs actual cage speed.**

generation increased with oil flow rate and was not significantly affected by applied radial load.

*Race Temperature*

Control of race temperature is essential to the maintenance of bearing IRC which in turn affects the system spring rate. Figure 8 shows how race $\Delta T$ varied with combined rotor speed. Race temperature differences of 40-80°F are manageable with proper lubrication system design. Data from rig parametric tests are used to determine lubrication requirements for engine applications by utilizing heat generation data and race temperatures to determine the amount of cooling required to maintain IRC control.

*Cage Slip*

Cage slip is of major importance when the rollers lose traction and cage speed approaches either of the shaft speeds. All the test bearings experienced negligible slip. Actual vs theoretical cage speed is shown in Fig. 9. For counter-rotating bearings on the selected speed schedule, cage speed decreases as shaft speed increases. At maximum rotor speeds the highest cage slip was 11 ft/s, assuming all slip occurs at the outer race contact. This assumption is based on an increase in the cage speed which rotates in the same direction as the inner race. For comparison purposes with a conventionally mounted 124 mm roller bearing (rotating inner race and a stationary outer) experiencing a 10% cage slip, the slip velocity would be approximately three times higher than for the counter-rotating bearing. Such a cage slip in a conventionally mounted bearing is considered acceptable for most applications. Excessive cage slip can result in surface damage to rolling contact areas. None of the parametric bearings exhibited this type of damage.

*Roller Wear*

Wear generation plots (Figs. 10-12) illustrate average roller weight loss, average roller end wear bandwidth, and average change in roller skew angle vs calculated total integrated roller

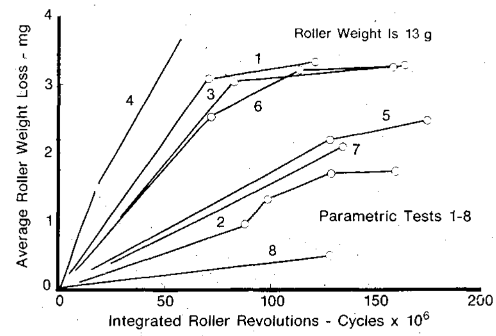



**Fig. 10 Average roller weight loss vs integrated roller revolutions.**

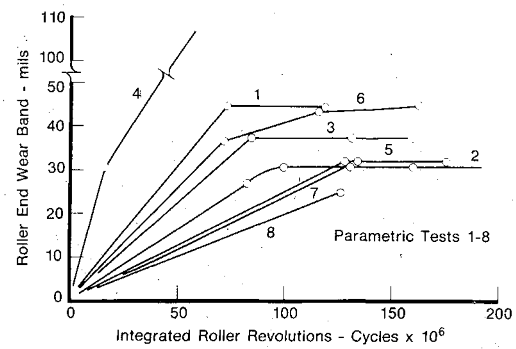



**Fig. 11 Average roller end wear bandwidth vs integrated roller revolutions.**

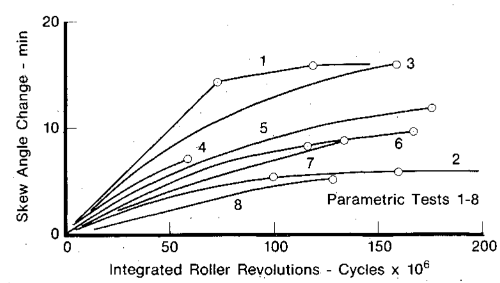



**Fig. 12 Average roller skew angle change vs integrated roller revolutions.**

spin revolutions. Roller revolutions are calculated using roller geometry and shaft speeds. The total number of roller revolutions about their axes were chosen as the abscissa rather than test hours in order to normalize the wear generation from test to test. All wear plots show a similar trend and indicate that the rollers go through a break-in period after which they achieve a stabilized low wear rate. A comparison of the roller end wear pattern of all parametric test bearings is shown in Fig. 13. It can be noted that all roller element end wear patterns are concentric.

**Statistical Evaluation of Wear Results**

A statistical regression analysis of the test data was performed in order to determine the effects of preload, IRC, layback angle, and roller end clearance on roller wear. These factors were considered controlled variables and were varied over eight tests. At least two levels for each variable were preselected. The main roller bearing wear parameters considered as dependent are change in skew angle, roller end wear bandwidth, and roller weight loss.

Multiple regression analysis was employed to develop a mathematical model for the prediction of the wear levels of the three dependent variables. The data set selected consists of all the parametrics excluding the baseline. The baseline was excluded because of its axial pinch distress. The analysis showed that change in skew angle is significantly affected by

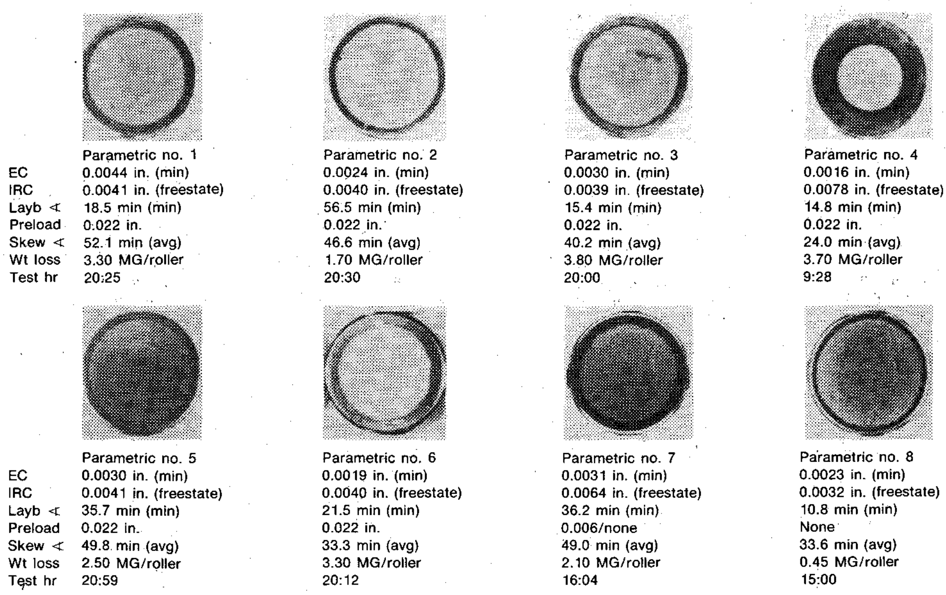



**Fig. 13 Bearing roller end wear pattern (all wear concentric).**

| | End Clearance | IRC | Preload | Layback Angle |
|---|---|---|---|---|
| Change in Skew Angle | | | | |
| Roller End Wear Band | | | | |
| Roller Weight Loss | | | | |

**Fig. 14 Statistical analysis of wear results.**

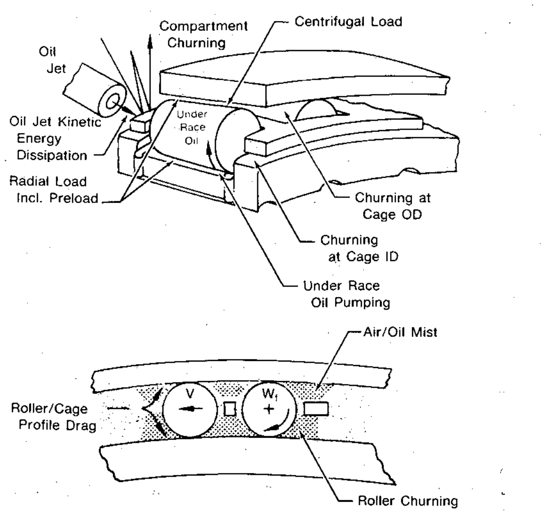



**Fig. 15 Bearing heat generation model identifies nine heat sources.**

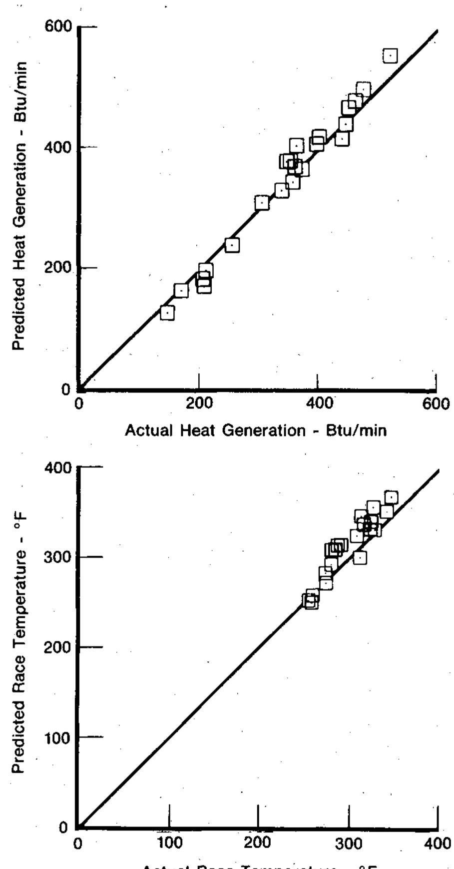



**Fig. 16 Data correlation for counter-rotating bearings.**

roller end clearance (statistical significance evaluated at the 0.95 confidence level). Roller end wear is significantly affected by internal radial clearance. The final variable, roller weight loss, is significantly affected by preload.

Figure 14 schematically shows how the dependent variables (wear) were affected by the independent variables (bearing geometry). A surprising result was that guide flange layback angle was not a strong influence on wear.

When roller end clearance is under 0.002 in. for the mounting arrangement of the counter-rotating bearing, the amount of end clearance governed the amount of wear. If end clearance is reduced further, then an axial pinch occurs leading to a bearing distress. When end clearance is over 0.002 in., preload, both IRC and roller end clearance influence roller and guide flange wear.

## Design System Modifications

Typically at the beginning of a component technology program, design studies are conducted to analytically predict performance and durability of the concept. These studies use design systems which are based on analytical modeling and empirical data. It is imperative that design systems be iteratively updated as data become available to provide an improved analytical prediction of new concepts.

In this program the results of the baseline and parametric test series were used to modify an existing design system. A comprehensive state-of-the-art computerized roller bearing design program[1] was used as the basis for the modifications. This program, called TRIBO-1, is capable of predicting race temperatures and heat generation based on input parameters such as bearing geometry and operating conditions. The bearing heat generation model is based on the identification of nine heat generation sources within the operating bearing, as shown in Fig. 15.

A data correlation program, which is used to set up the equations in TRIBO-1, was modified based on conclusions drawn from data from this test program. Equations and coefficients were changed to more accurately predict heat generation and race temperature for counter-rotating applications. TRIBO-1 was based on data from conventionally mounted, fixed outer race configurations; because of this, the initial predictions did not match data well. The selected equations and coefficients were modified to correlate to the present counter-rotating test data. An excellent match between predicted and actual data was attained. This is shown for heat generation and race temperature in Fig. 16. Average scatter between the revised predictions and actual values for all the parametric tests was less than 10%.

## Conclusions

The bearing arrangement design study results led to the selection of a straddle-mounted turbine rotor configuration with an intershaft support bearing for advanced fighter engine applications. Counter-rotation was selected since it offered improved engine performance, reduced weight, and increased bearing fatigue life. Also, the intershaft counter-rotating bearing with the inner race mounted on the high rotor was selected due to internal radial clearance control considerations.

Significant findings on the effect of the geometrical variables were that: 1) accurate temperature prediction for the roller elements is critical in setting end clearance above a critical level at which axial pinch of the rollers would occur, 2) reducing bearing preload and maintaining a small running internal radial clearance reduce roller wear, and 3) guide flange layback angle did not have a large impact on roller wear.

Statistical regression analysis was utilized to relate the dependent bearing geometry variables to roller wear. Significant findings were that: 1) initial roller end clearance was the strongest driver on change in skew angle and as roller end clearance increased wear increased; 2) IRC was the strongest driver on roller end wear bandwidth and, as IRC increased, roller end wear bandwidth increased; and 3) preload was the strongest driver in roller weight loss and, as preload was decreased, roller weight loss decreased. All bearings experienced a low stabilized wear rate and concentric roller end wear patterns after an initial wear-in period.

Parametric bearing test findings for all geometries indicated low heat generation and very low cage slip down to zero applied load for both preloaded and nonpreloaded bearings, and race $\Delta T$ comparable to conventional bearings.

The data correlation portion of the bearing design program, TRIBO-1, was revised to provide excellent agreement with experimental data for prediction of heat generation and race temperature difference for intershaft roller bearings in counter-rotation.

## Recommendations and Future Plans

1) Develop a mathematical model for bearing heat generation prediction applicable to all bearing rotational configurations (inner race rotation, outer race rotation, corotational, and counter-rotational).

2) Develop a technique for measurement of roller element rotational speeds for counter-rotational mode to define roller-to-race slip magnitude.

3) Evaluate effects of contamination and oil film interruption for counter-rotating bearings to determine configuration tolerance to operational variables.

4) Conduct long-term (> 1000 h) endurance testing on a series of bearings to determine influence on intershaft bearing durability and performance.

## Acknowledgments

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[1] "Mainshaft High-Speed Cylindrical Roller Bearings for Gas Turbine Engines," Naval Air Propulsion Center, Report NAPC-PE-60C, Nov. 1980.