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Ceramic Roller Bearing Performance in a Gas Turbine Engine

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The program tests conducted on a modified J402 turbine engine were to demonstrate the feasibility of using ceramic bearings for high speed gas turbine applications with reduced or no lubrication to the bearing.

Six engine tests were conducted, each test complementing the next to establish baseline parameters of the test to follow. Test No. 3 was the first to demonstrate the ceramic bearing operation unlubricated at 39,000 rpm

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ABSTRACT (Cont'd.)

and minimum lubrication of 10cc per minute at 41,200 rpm. Engine Test No. 4, which was a repeat of Test No. 3, was successfully completed and demonstrated 33 minutes of unlubricated operation at 39,000 rpm.

All engine tests were conducted with the bearing and bearing housing instrumented for temperature read-out during the test. Engine Tests 3, 4, 5 and 6 were also instrumented for lubrication flow pressure and bearing cavity pressure in order to monitor lubrication to the bearing as a function of pressure differential.

FOREWORD

This report covers activites carried out by Teledyne CAE, Toledo, Ohio 43612, under U.S. Navy Contract N00140-76-C-1104, which incorporates U.S. Army MIPR Number AMRDL-75-5.

The government technical manager for this program was Raymond Valori of the Naval Air Propulsion Center, Trenton, New Jersey 08628 (telephone 609/882-1414). Daniel Pauze of the Applied Technology Laboratory, USARTL (AVRADCOM) Fort Eustis, Virginia 23604, provided technical direction for the U.S. Army.

The project was conducted by Teledyne CAE under the direction of Glenn W. Hamburg and William F. Prusaitis.

Acknowledgement is provided by Wray Johnson of Norton Company, Worcester, Massachusetts and Paul Cowley of Federal Mogul Corporation, Ann Arbor, Michigan for their assistance and consultation during this program.

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SUMMARY

The program tests conducted on a modified J402 turbine engine were to demonstrate the feasibility of using ceramic bearings for high speed gas turbine applications with reduced or no lubrication to the bearing.

Six engine tests were conducted, each test complementing the following test to establish baseline parameters of the test to follow. Test No. 3 was the first to demonstrate the ceramic bearing operation unlubricated at 39,000 rpm and minimum lubrication of 10cc per minute at 41,200 rpm. Engine Test No. 4, which was a repeat of Test No. 3, was successfully completed and demonstrated 33 minutes of unlubricated operation at 39,000 rpm.

All engine tests were conducted with the bearing and bearing housing instrumented for temperature readout during the test. Engine Tests 3, 4, 5 and 6 were instrumented for lubrication flow pressure and bearing cavity pressure in order to monitor lubrication to the bearing as a function of pressure differential.

CERAMIC BEARING FINAL REPORT

I. INTRODUCTION

In response to increasingly stringent demands being placed upon metallic bearings, Teledyne CAE, under U.S. Navy Contract N00014O-76-C-1104, has conducted a series of ceramic roller bearing tests using a bailed J402-CA-400 production engine. The ceramic bearings were made of Norton Company's NC-132 hot pressed silicon nitride material.

The unique characteristics of ceramic makes it a viable bearing material. The corrosion resistant nature of the ceramic material avoids the necessity of inspection and retrofitting of engine bearings due to environmental corrosion throughout both storage and operational life. The combination of corrosion resistance and higher hardness of silicon nitride compared to steel prevents fretting corrosion and false brinnel indentations in the bearings that can be caused by handling and shipping, and by storage aboard ship and air launch vehicles. These potential problems have been identified in both laboratory engine tests, and from field experience with missile engines carried aboard launch aircraft.

Current metal bearings for gas turbines must be lubricated to maintain rolling contact element temperatures within the range established by the load carrying requirements of the bearing. The upper limit of this range is about 600^{o} F even for the tool steels currently used in mainshaft applications. The ability of ceramic bearings to maintain hot hardness and capacity well in excess of 1000^{o} F complemented by a low coefficient of friction, provides a substantial potential payoff through the elimination of the lubrication system, results in fewer parts, simplifies the engine assembly, increases reliability, and substantially reduces cost.

II. DISCUSSION

Ceramic rolling contact bearings have potential advantages over conventional metal bearings for use in the mainshaft support positions in gas turbine engines. Some of these advantages are listed below:

1) The low coefficient of thermal expansion of silicon nitride alleviates the problem of loss of internal clearance due to a thermal differential across the races. Skidding may occur in conventional roller bearings which have excessive initial internal clearance to compensate for the anticipated clearance loss under maximum thermal operating conditions. (However, it must be recognized that to accommodate the low coefficient of thermal expansion of silicon nitride, a means of providing thermal compatibility between the metal shaft and the ceramic bearing inner race must be incorporated.)

2) The lower density of silicon nitride rolling elements results in lower centrifugal body forces, particularly in high speed bearings, and provides a potential residual capacity to carry higher applied external loads or operate at increased speeds for the same load levels.

3) The lower density also minimizes the tendency of the bearing to skid during speed transients by decreasing the mass moment of inertia of the rollers and cage assembly.

The low coefficient of friction and high hot hardness of silicon nitride may allow operation with no external lubrication; self lubricating cages could be utilized to extend life under these conditions. A substantial potential cost and weight benefit results from the elimination of external lubrication requirements. The extent of the cost benefit would depend on the complexity of the lubrication system eliminated:

1) The ability to operate at high temperature (2000° F) without loss of hardness and capacity allows relaxation of cooling and lubrication requirements. In engines such as the J402-CA-400 (HARPOON and VSTT) bearing cooling is provided by compressor discharge bleed air. Minimizing the required amount of cooling bleed air improves the aero thermal cycle efficiency.

2) The ceramic's inherent oxidation and corrosion resistance precludes deterioration due to heat and gaseous corrosive substances. Salt water immersion of drone type aircraft would not require post-recovery replacement of the bearing normally necessitated by salt water attack.

3) Long term storageability, and operational readiness (especially critical to missile type applications such as the J402-CA-400 HARPOON engine) is enhanced by the potential resistance to false brinneling and by the virtual immunity to corrosive attack provided by the extremely hard, chemically inert ceramic material. In addition, if the bearing is proven consistently capable of operating dry, deletion of the lubricant eliminates concern for its long term storage stability.

In view of the potential advantages that characterize the use of silicon nitride as a rolling element bearing material, particularly in military oriented applications, various branches of the military have sponsored research and development programs related to such usage.* As a result of these programs, full-scale silicon nitride roller bearings have been manufactured and successfully tested in rigs at "DN" values exceeding 2×10^6 (the "DN" value is a relative speed indicator, which equals, bearing bore diameter in millimeters times the speed in rpm).

The purpose of this program was to extend previous successful silicon nitride roller bearing operating results to the realistic environment of the gas turbine engine. The engine employed in testing the silicon nitride bearings was a combination of the J402-CA-400 (Navy HARPOON) engine (Figure 1) and the J402-CA-700 (Army-VSTT) engine, which is a longer life derivative of the J402-CA-400. Both engines run at relatively modest rear bearing "DN" values (0.7×10^6) but at maximum operating conditions, provide a hostile thermal environment for the rear bearing. Compressor discharge air under Sea Level Static (SLS) conditions provides an ambient bearing cooling stream of 400° F to 450° F, but under certain high Mn number hot day conditions, the cooling air temperature reaches 500° F to 660° F with resultant 600° F to 800°F bearing operating temperatures.

---

*Military sponsored programs include:
1. NAVAIR - N00019-72-C-0299
2. NAVAIR - N00019-73-C-0193
3. NAVAIR - N00019-74-C-0157
4. NAVAIR - N00019-75-C-0197
5. ARMY - DAAG 46-74-C-0055

These engines have innovative lubrication systems,(Figure 2) designed to save weight, volume, and cost. The J402-CA-400 rear bearing has a 30-minute life requirement and is grease lubricated. However, the J402-CA-700 rear bearing has a 30-hour life requirement, and therefore is lubricated by a fuel mist system. The latter system was used to provide baseline data relative to engine operation of ceramic rolling element bearings and to provide a convenient means of controlling lubricant flow to the bearing. Following successful demonstration of ceramic bearing operation in a fuel lubricated mode, the bearing fuel lube system was modified to permit fuel lube flow modulation.

III. CERAMIC BEARING CONFIGURATION

The bearings were manufactured by the Federal Mogul Corporation from Hot Press Silicon Nitride billets supplied by the Norton Company. Figure 3 describes the bearing's salient features. The inner race mounting rings are conical at the ceramic inner race interface to accommodate the thermal differential growth between the steel shaft and the ceramic inner race, both axially and radially. This conical innerface maintains the inner race centering requirements throughout the operating temperature range of the ceramic bearing. The bearing outer race is supported in a damping ring which absorbs the shaft vibrations within the bearing housing. This damping ring is identical to the production ring with the exception of the ceramic bearing outer race anti-rotation pin slot. A number of design modifications were implemented after the first test resulted in a bearing failure due to axial movement of the outer race caused by the disengagement of the outer race retaining ring from its groove. These modifications are summarized below:

1) The fit between the bearing O.D. and the vibration isolator ring was tightened to partially offset the differential thermal expansion between the ceramic outer race and the steel bearing housing's higher coefficient of expansion.

2) A slot was added to the bearing outer race to engage a dowel pin to preclude outer race spin.

3) The bearing inner race rear mounting ring was widened to enhance the race stability by increasing the length over diameter (L/D) ratio, improving the inner race runout control. The new rear mounting ring also reduced the assembly time necessary to achieve the required inner race runout.

The alterations to the original bearing design were subsequently incorporated in all the bearings used in the tests.

Photographs of the bearings are shown in Figure 4.

IV. TEST CONFIGURATION

The ceramic bearing capability to operate successfully in a gas turbine was established by installing the bearing in a modified J402-CA-700 engine. The bearing was lubricated identically to its production engine bearing counterpart. The fuel lube, tapped from the engine centrifugal fuel pump discharge port, flows through a flow restrictor and is finally jet impinged into the bearing. At maximum shaft speed, this lube flow is approximately 60cc per minute.

Following demonstration of successful ceramic bearing operation in a lubed configuration, the engine was modified to provide fuel lube flow modulation. The production engine fuel pump discharge passage was blocked and replaced with an external source of fuel lube (Figure 5). Fuel lube flows of from 60cc to zero were established as a function of pressure (P) on a hydraulic flow bench (Figure 6). An air pressurized fuel lube tank provided the lube pressure, and an electrically actuated motorized valve varied the flow control orifice. In engine operation the desired fuel lube flow was established by actuating the motorized valve to obtain the required P between the valve discharge pressure and the pressure in the bearing cavity.

V. INSTRUMENTATION

In addition to the instrumentation normally employed in production engine acceptance test, e.g., fuel flow, thrust, level of vibration, etc., the rear (ceramic) bearing outer race temperature was monitored via two thermocouples; a third thermocouple monitored the bearing housing temperature. Oscillograph recording data were obtained; Figure 7 shows a typical data sheet.

VI. TEST HISTORY

A summary of the ceramic bearing test history is shown in Figure 8.

Test Number 1 - Date: 7/19/77

The test objective was to establish the capability of a ceramic roller bearing to operate in a gas turbine. After 17 minutes of a scheduled 50 minute test the bearing failed. The bearing failure was caused by disengagement of the bearing retaining (Spiralox) ring from its groove, which allowed the outer race and roller assembly to move axially forward over the edge of the inner race. The resultant high edge load and stress concentration caused chipping and subsequent cracking which eventually resulted in complete bearing failure (Figure 9). Loosening of the snap ring in its groove was initiated by spinning of the bearing outer race and end shield. The spinning occurred as a result of increased O.D. clearance at operating temperature due to the large difference in coefficients of thermal expansion between the silicon nitride bearing race and the steel housing, and was further accentuated by the extremely smooth bearing O.D.

For subsequent tests, an anti-rotation slot was ground in the outer race, and a longer, more stable inner race mounting ring was incorporated in the bearing assembly. Modifications to the bearing housing consisted of introducing a bearing outer race anti-rotating pin and deepening the retaining ring groove. The Spiralox wound ring was replaced by a "True-Arc" type with substantially greater retention capability.

Test Number 2 - Date: 11/9/77

This test's objective was the same as Test No. 1, to determine the ceramic bearing ability to operate in a gas turbine application. After successfully accumulating 63 minutes of testing, the bearing was visually inspected in situ and found to be in excellent condition (Figures 10 and 11). Testing was resumed to the J402-CA-700 (VSTT) endurance schedule; after completing four hours of the scheduled five hours run, a turbine rotor blade failure occurred at maximum shaft speed. The bearing had been operating normally prior to turbine blade separation; post test examination revealed the bearing inner race was destroyed and the outer race was cracked in several areas (Figure 12).

In this test the bearing was continuously fuel lubricated by approximately 60cc per minute flow. Bearing temperature was 400 F at maximum shaft speed.

Test Number 3 - Date: 3/14/78

The objective of this test was to determine the bearing ability to operate at a reduced/zero lube flow.

The bearing demonstrated the capability to operate unlubricated at 39,000 shaft rpm by running for 15 minutes with the bearing temperature stabilized at 625° F. Successful operation at Harpoon 100 percent shaft speed (41,200 rpm) with fuel lube flow reduced to 10cc per minute was also demonstrated (bearing temperature 575° F). However, an attempt at operating at this shaft speed with zero flow resulted in a rapid bearing temperature rise beyond 675° F. Prompt throttle reduction and reapplication of fuel lube failed to save the bearing.

Test Number 4 - Date: 3/31/78

The purpose of this test was to repeat the objective of Test No. 3. The test was successfully completed with the bearing in excellent condition as shown in Figures 13 and 14. Harpoon 100 percent shaft speed (41,200 rpm) was sustained for 22 minutes with lube flow of 10cc per minute (bearing temperature 627° F) and for 4 minutes with flow of 0 to 10cc per minute (bearing temperature 670° F). Shaft speed was subsequently reduced to 39,000 rpm, the fuel lube was shut-off and the engine was operated for 33 minutes with the bearing temperature stabilizing at 573° F.

Test Number 5 - Date: 5/11/78

The intent of this test was to run 5 hours under the VSTT endurance schedule (Figure 15) with zero lube flow.

The engine was operated at 39,600 rpm (100 percent VSTT shaft speed) with the bearing lubricated. Lube flow was gradually decreased to zero. After the bearing temperature stabilized at 575° F, the endurance run was begun. Approximately 3 minutes into the endurance cycle, the bearing temperature began to rapidly increase, eventually peaking at 825° F while the shaft speed was simultaneously reduced to 30,000 rpm. On disassembly, inspection showed the outer race had numerous fractures, and the inner race had been destroyed.

The bearing was one of the two remaining unused bearings. Prior to this test, these two bearings had been returned to Federal Mogul to be reworked to the configuration implemented after Test No. 1, as follows:

1) Slot the outer race.
2) Reduce the bearing outer race O.D.
3) Grind the inner race O.D. to increase the bearing internal clearance to 0.0020 - 0.0025 to bring the bearing internal clearance to within the range of those bearings which previously ran successfully.

Subsequent to this test, the one remaining new bearing, which had been modified concurrently with the failed bearing, was returned to Norton for examination.

An investigation by Norton using Optical and X-ray Diffraction analysis disclosed that regrinding of the inner race bearing surface substantially altered the microstructure of the silicon nitride race material and may have contributed to the bearing failure. Quoting from the Norton report "Conclusions":

> "It is virtually certain that the regrinding of the inner raceway altered the surface of the NC-132 base material. Extreme temperatures were generated that resulted in a very low material removal rate. It is not unusual for wheel property variations within a given specification to occur, causing some wheels to cut at much different rates. Enough heat could have been generated to:
>
> A) Oxidize the NC-132 silicon nitride to SiO_2 quenched to a glassy deposite by the coolant.
>
> B) Break down the wheel vitreous bond and deposit it on the race surface.
>
> Either of these is likely, given the apparent glassy structure of the deposits. This altered surface structure cannot be expected to have superior properties in rolling wear of contact fatigue demonstrated by pure NC-132 surfaces. Assuming the same surface was present on the failed raceway the anamalous results could be attributed in part to the non-homogeneous surface."

In light of the report's conclusion that the inner race rolling contact surface was not suitable, the decision was reached to discard this inner race and replace it with a newly manufactured inner race.

Test Number 6 - Date: 10/28/78

The purpose of this test was to run a 5 hour VSTT endurance (Figure 15) with zero lube flow. This test is a repeat of Test No. 5.

The engine was operated at 39,600 rpm (100 percent VSTT shaft speed) with the bearing lubricated initially as in Test No. 5. The lube flow was gradually decreased to 10cc per minute, and held until bearing temperature stabilized at 549° F. The lubrication was gradually reduced to zero when the bearing temperature began to rise rapidly. The engine speed was then reduced, which in turn reduced the bearing temperature. After the bearing temperature once again stabilized, the speed was increased gradually. The bearing temperature began to rise rapidly and the engine speed was again reduced. This time the bearing temperature continued to rise as the engine speed was reduced. With a maximum bearing temperature observed at 790°F, the test was aborted and the engine was shut down. Disassembly of the engine revealed a bearing failure. The outer race had numerous fractures and the inner race had only pieces left.

The primary cause of failure has been identified as loss of inner race radial support during rapid thermal and speed transient conditions. Two edge rings, supplied as part of the bearing assembly, support the bearing inner race through conical surfaces on the inner race end faces. The conical support rings are designed to maintain positive contact under differential thermal growth rates by allowing the shaft supported rings and bearing race to slide along their common conjugate surfaces. Clearance is provided between the inner race bore and the shaft journal to avoid inducing excessive race hoop stresses by the greater thermal growth of the metal shaft. The clearance was calculated to provide positive radial contact at an estimated bearing operating temperature of 900° F. Since steady state bearing operating temperatures did not exceed 700° F, the bearing race was always supported on the edge rings during testing.

This system was extremely sensitive to clamping loads and axial run-out of the clamping members. Each bearing/shaft assembly was accomplished only after numerous torquing iterations involving clamping nut selections and rotary indexing of various components in the clamping path to obtain the required inner race run-out, (0.0002 in.). Bearing race run-out varied from 0.00015 in. to 0.0040 in. with various combinations of indexing and locknuts. The locknut was tack welded to the shaft after proper bearing race run-out was obtained to assure maintenance of the clamping load.

During thermal and speed transients, the static clamping friction between the conjugate mounting faces caused unequal, slip-stick friction motion of one end of the race with respect to the other, resulting in cocking of the race with consequent intolerably high stress concentrations. Under conditions of zero lubricant flow no fluid was available to form an elastohydrodynamic film between the roller/race surfaces, and race deterioration rapidly escalated into failure in several instances.

An inner race mounting system capable of continuous, uniform radial and axial support, while accommodating the relative thermal differential expansion between race and shaft is required to make the unlubricated hot section bearing a reliable gas turbine component.

VII. CONCLUSIONS

Engine tests in a modified J402 gas turbine engine has demonstrated the feasibility of operating a ceramic roller bearing without lubrication for a sustained period of time. The duration of the unlubricated bearing operation was 33 minutes at 39,000 rpm. At an elevated speed of 41,200 rpm a minimum lubrication feed to the ceramic roller bearing of 100cc per minute was provided. This compares to the standard M-50 tool steel bearing lubrication requirement of 60cc per minute.

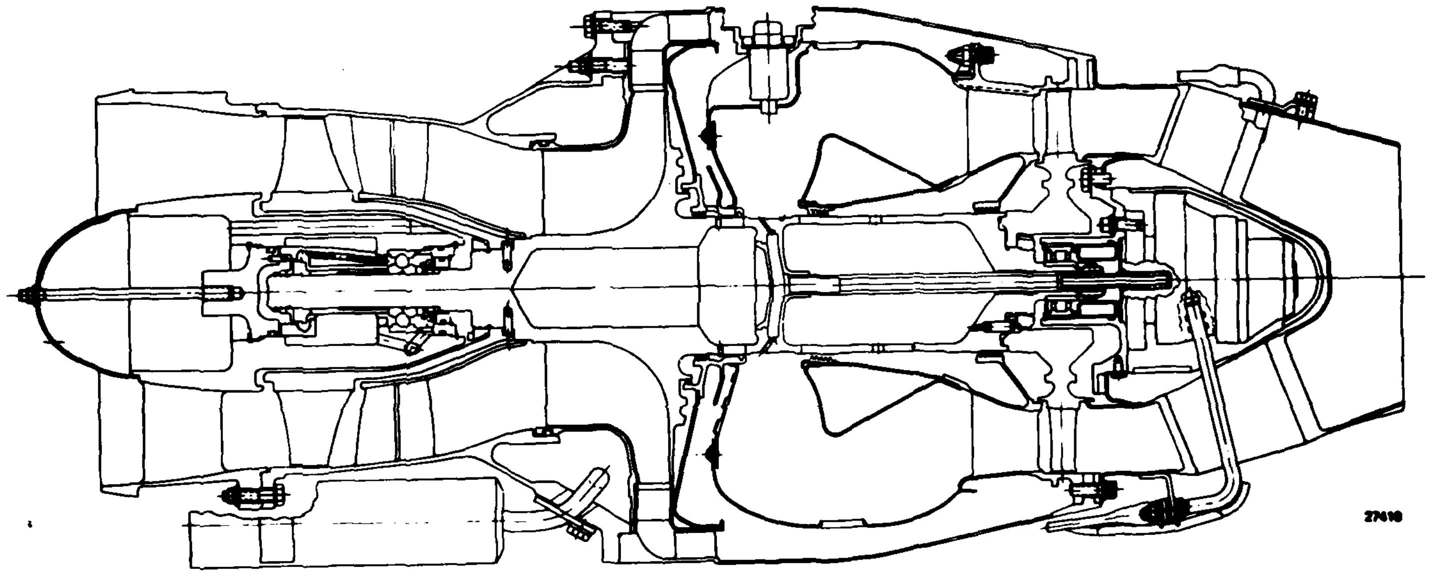
With the exception of the first two runs, the primary cause of failure in those tests which did not achieve the desired test objectives is believed to be due to the inner race mounting arrangement which may be incapable of accommodating the transient speed and thermal conditions imposed during engine operation, particularly in the non-lubricated mode.

VIII. RECOMMENDATIONS

It is recommended that development effort be continued with particular emphasis on evaluating improved bearing mounting configurations and bearing designs under a full range of engine operating conditions (i.e., endurance, cold soak, shock, vibration and rapid starts).

J402-CA-400 HARPOON MISSILE ENGINE

LIMITED LIFE (< 1 HR)
STORAGE LIFE GOAL: 5 YR'S IN CORROSIVE (SALT WATER) ENVIRONMENT

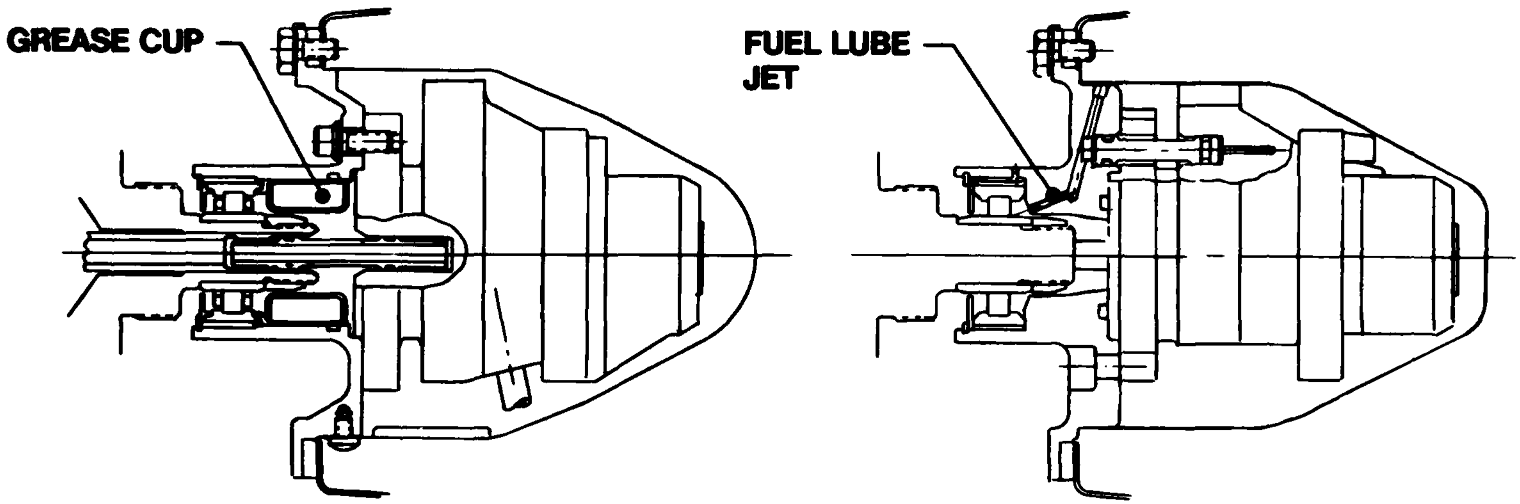


THRUST:	**660 LBS**
SHAFT SPEED:	**41,200 RPM**
TURBINE INLET TEMP.:	**1900°F**
EXHAUST GAS TEMP.:	**1550°F**

28429

FIGURE 1.

J402 PRODUCTION ENGINES
REAR BEARING LUBE SYSTEMS



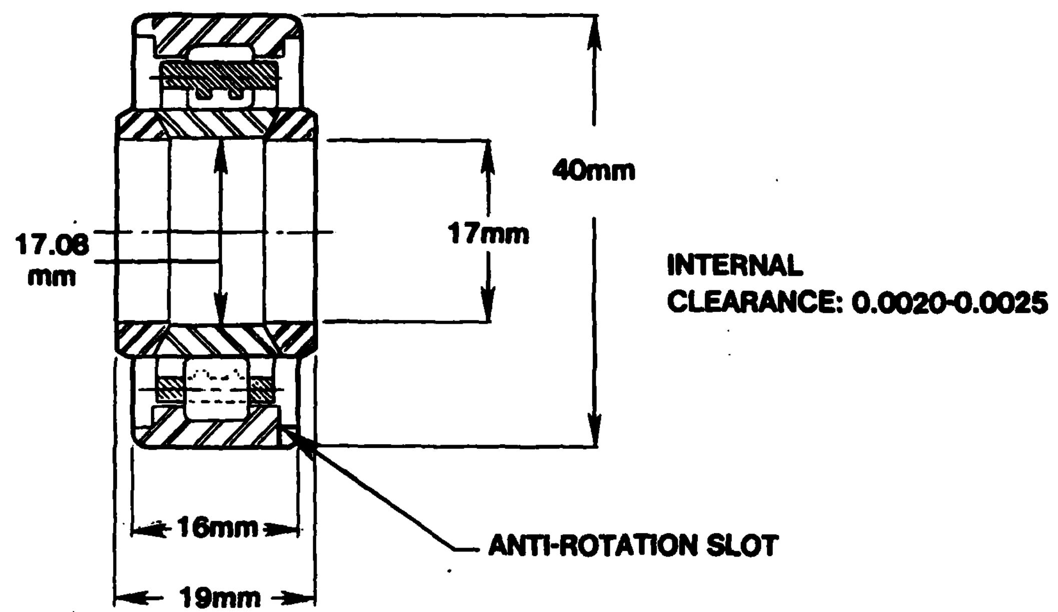
<u>HARPOON</u>
REAR BEARING LUBRICATION:
GREASE CUP
15cc KRYTOX 240AC

<u>VSTT</u>
REAR BEARING LUBRICATION:
FUEL LUBED
60cc/MIN

28437

FIGURE 2.

CERAMIC BEARING CONFIGURATION

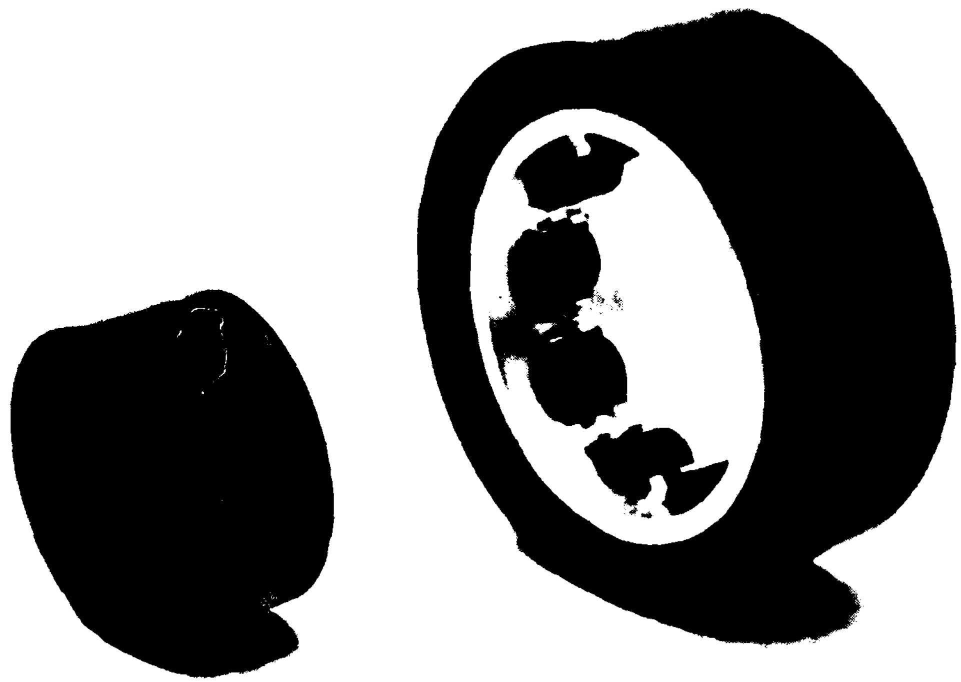


	MATERIAL	REMARKS
RACES & ROLLERS	SILICON NITRIDE NC-132	10 ROLLERS: 6 x 6 mm
CONED SPACERS	440C STST DIMENSIONALLY STABLE TO 900°F HARDNESS R_C 33-40	0.0000-0.0004 TIGHT FIT ON SHAFT
ROLLERS SEPARATOR	AMS 6415(4340 ST) HARDNESS R_C 28-35	SILVER PLATED PER AMS 2412

28455A

FIGURE 3.

REAR MAINSHAFT CERAMIC BEARING FOR J-402 ENGINE SHOWING SILICON NITRIDE RACES AND ROLLERS WITH SILVER PLATED STEEL CAGE



METRIC 1 2 28463

FIGURE 4.

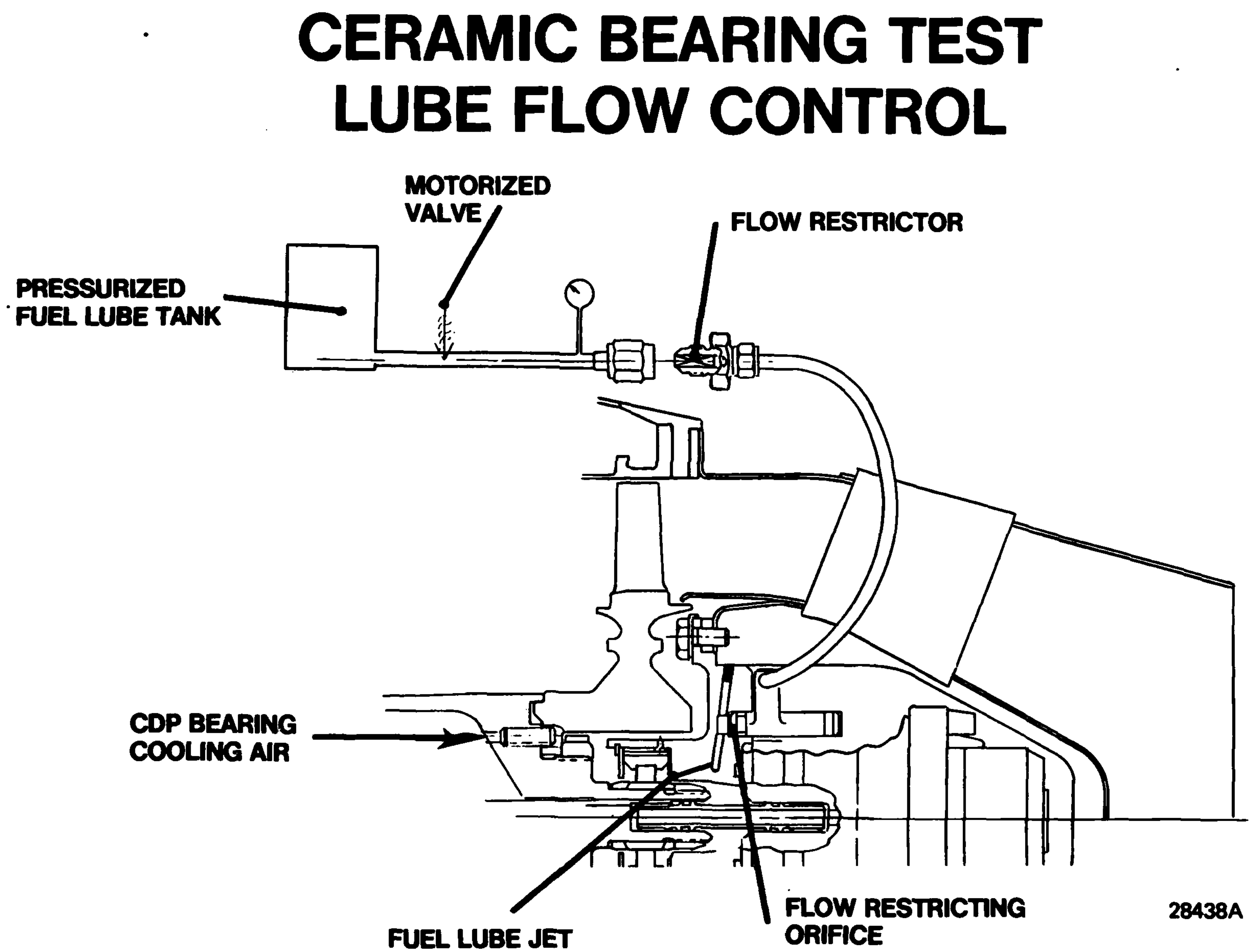

CERAMIC BEARING TEST
LUBE FLOW CONTROL
MOTORIZED VALVE
FLOW RESTRICTOR
PRESSURIZED FUEL LUBE TANK
CDP BEARING COOLING AIR
FUEL LUBE JET
FLOW RESTRICTING ORIFICE
28438A

FIGURE 5.

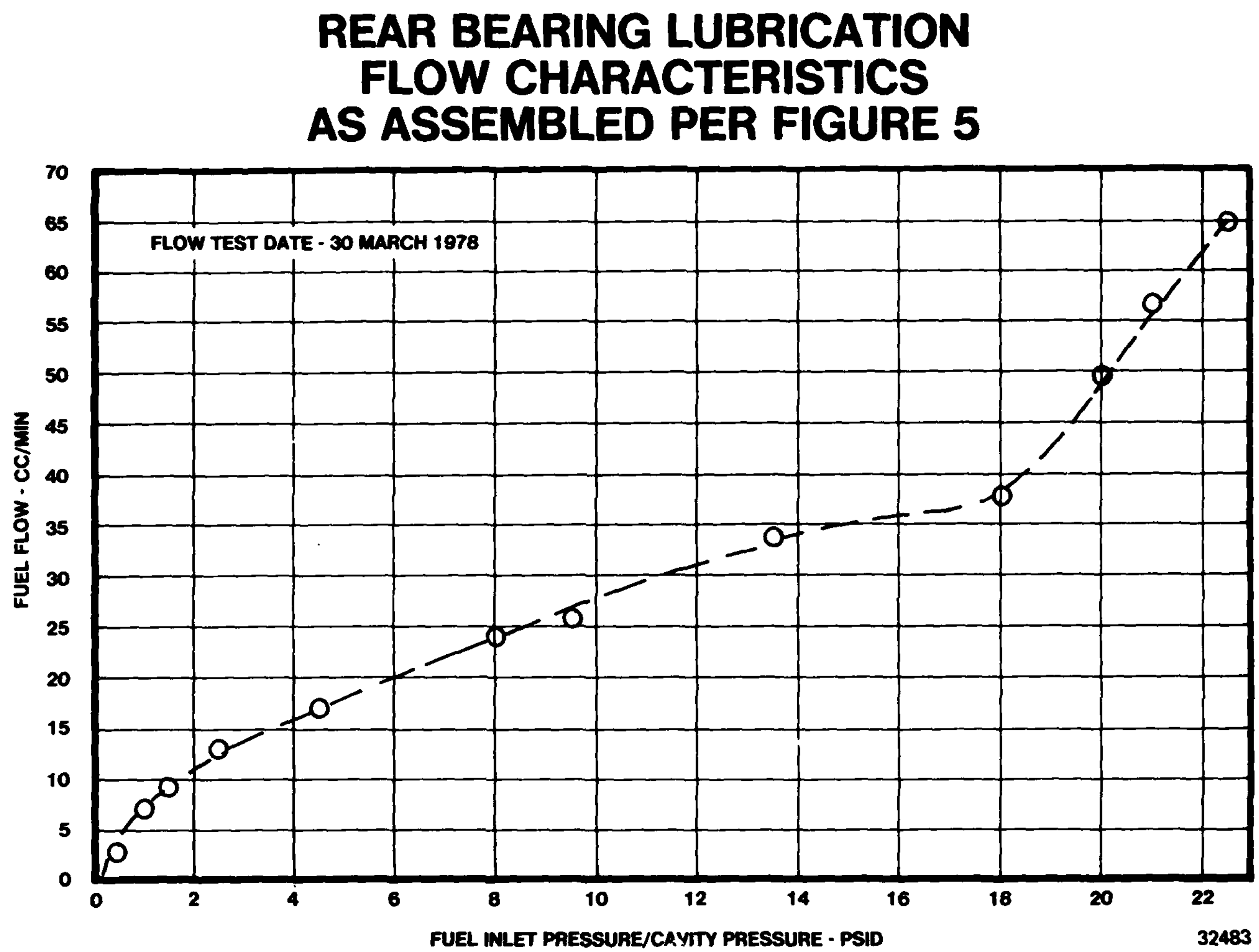

REAR BEARING LUBRICATION
FLOW CHARACTERISTICS
AS ASSEMBLED PER FIGURE 5
FLOW TEST DATE - 30 MARCH 1978
FUEL FLOW - CC/MIN
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
0
2
4
6
8
10
12
14
16
18
20
22
FUEL INLET PRESSURE/CAVITY PRESSURE - PSID
32483

FIGURE 6.

OSCILLOGRAPH RECORDING DATA

Date ______________________ Location Cell #10

Test Ceramic Brg. Test Recorder 5-134

Technician ______________________

Chan. No.	Variable Measured	Range	Zero Ref.	Span	Disp/ Inch	Trans. Type	Cal. Sig.	Galvo No.	Remarks
4	Rear Brg Temp	32° - 782°F	.3	5	150°F	CA	17.10 mv	339	
5	PLA	0.30 VDC	.4	6	5 VDC		10 VDC	315	
7	CDP	0.200 "Hgg	.3	5	40 "Hg	S/N 4627	22.94 K	315	
8	Wf	0.1200 HZ	.1	5	240 HZ	S/N	1200 HZ	338	
10	Ne	0.42K RPM	0	6	7K RPM		11.9 KHZ	338	T/B 3.5294
11	IV	0-1 AMP	0	5	.2 AMPS		1 VDC	315	
12	Metal Temp	32° 782°	.5	5	150°F		17.10 mv	315	
13	Rear Vib	0-5 Mils	5.0	1	5 Mils		5 Mils	326	
14	EGT Tele.	0-5 VDC	5.2	--	400°F			315	OVDC = 5.2" - 2000°F 5VDC = .95" = 300°F
15	Front Vib	0-5 Mils	4.0	1	5 Mils		5 Mils	326	
16	NG Tele.	0-42K RPM			7K RPM			315	+.333 VDC = 6.2" +5VDC = .2"
18	EGT Avg	150° 2150°	.5	5	400°F		45.37 mv	315	

FORM T-697 8/72

FIGURE 7.

CERAMIC BEARING TEST HISTORY

TEST NO.	DATE	LUBE FLOW	TEST TIME	RESULTS
1	8-19-77	60cc/MIN.	17 MIN.	BEARING FAILED @ 34,000 RPM (RETAINING RING FAILURE)
2	11-11-77	60cc/MIN.	5.1 HRS.	VISUAL INSPECTION AFTER 1 HR: BEARING IN EXCELLENT CONDITION CONTINUED TESTING - TURBINE ROTOR FAILED AFTER 4 HRS. OF SCHEDULED 5 HR. RUN
3	3- 9-78	60cc TO ZERO	2 HRS.	SHAFT @ 39,000 RPM - 15 MIN. - ZERO LUBE SHAFT @ 41,200 RPM - 3 MIN. - 10cc LUBE ATTEMPTED 41,200RPM WITH ZERO LUBE; BEARING TEMP CLIMBED STEADILY - TEST ABORTED - OUTER RACE FAILED ON RUNDOWN
4	3-31-78	60cc TO ZERO	1 HR. 41 MIN.	SHAFT @ 41,200 RPM - 22 MIN. - 10cc LUBE BEARING TEMP 627°F SHAFT @ 41,200 RPM - 4 MIN. - 0-10cc LUBE BEARING TEMP 670°F SHAFT @ 39,000 RPM - 33 MIN. - ZERO LUBE BEARING TEMP 573°F BEARING IN EXCELLENT CONDITION
5	5-11-78	600cc TO ZERO	35 MIN.	SHAFT @ 39,600 RPM - 3 MIN. - ZERO LUBE BEARING TEMP CLIMBED STEADILY TEST ABORTED - INNER RACE DISINTEGRATED REMAINING NEW BEARING RETURNED TO MANUFACTURER FOR RACE SURFACE ANALYSIS
6	10-20-78	60cc TO ZERO	45 MIN.	SHAFT @ 39,600 RPM - LUBE @ 10cc REDUCING TO ZERO - BEARING TEMP CLIMBED STEADILY TEST ABORTED - INNER RACE DISINTEGRATED

FIGURE 8.

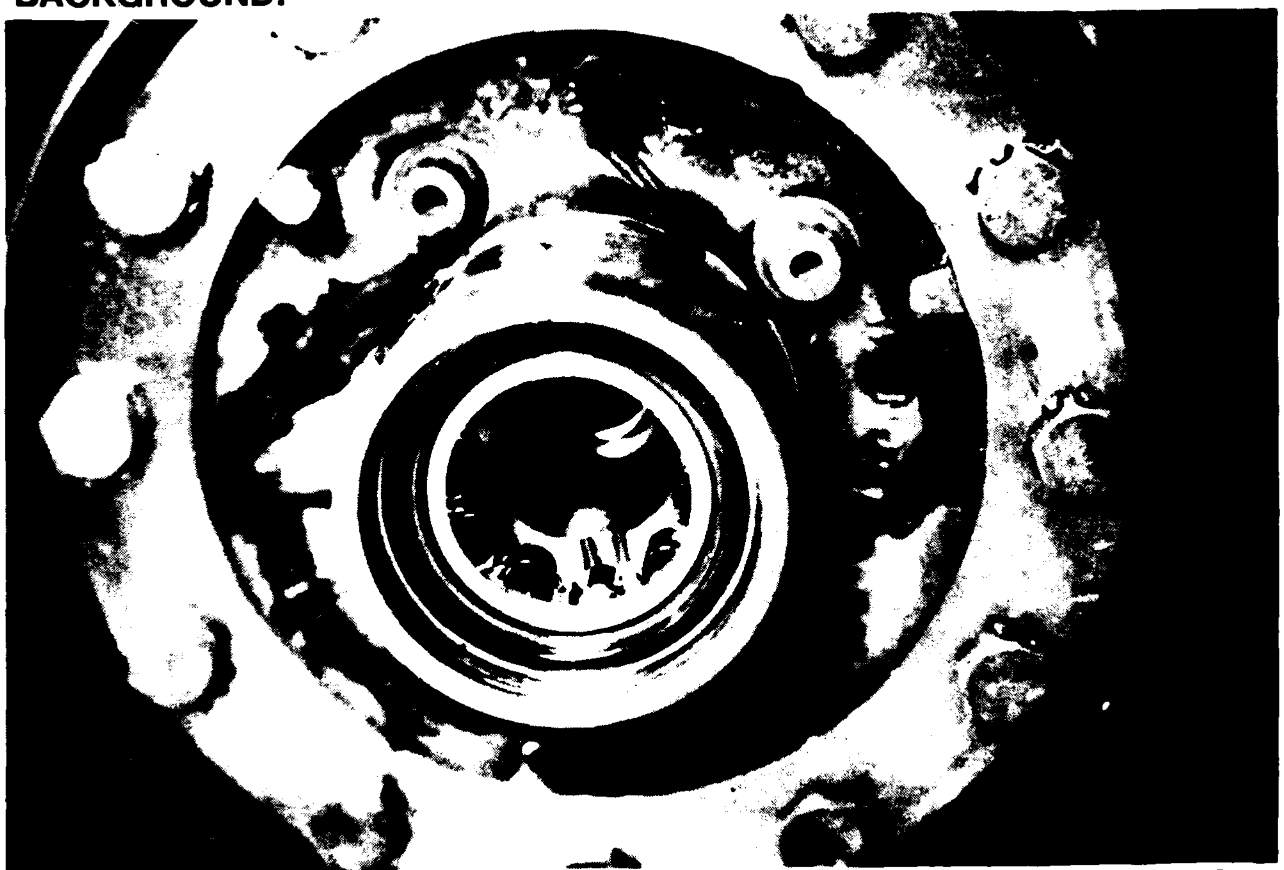
AFTER TEST 1
CERAMIC BEARING FAILURE DUE TO LOSS OF OUTER RACE AXIAL RESTRAINT AND SUBSEQUENT MOVEMENT OF ROLLER ASSEMBLY BEYOND EDGE OF INNER RACE (TOTAL TIME, 17 MIN.)

T-21586 28462

FIGURE 9.

AFTER TEST 2(a)

SILICON NITRIDE ROLLER BEARING (PN 309467) OUTER RACE, CAGE, AND ROLLERS SHOWING EXCELLENT CONDITION OF ROLLER SURFACES AFTER ONE HOUR OF FUEL-LUBED OPERATION IN J-402 ENGINE. ANTI-ROTATION SLOT IN BEARING HOUSING SHOWN IN BACKGROUND.



28431

FIGURE 10.

AFTER TEST 2(a)

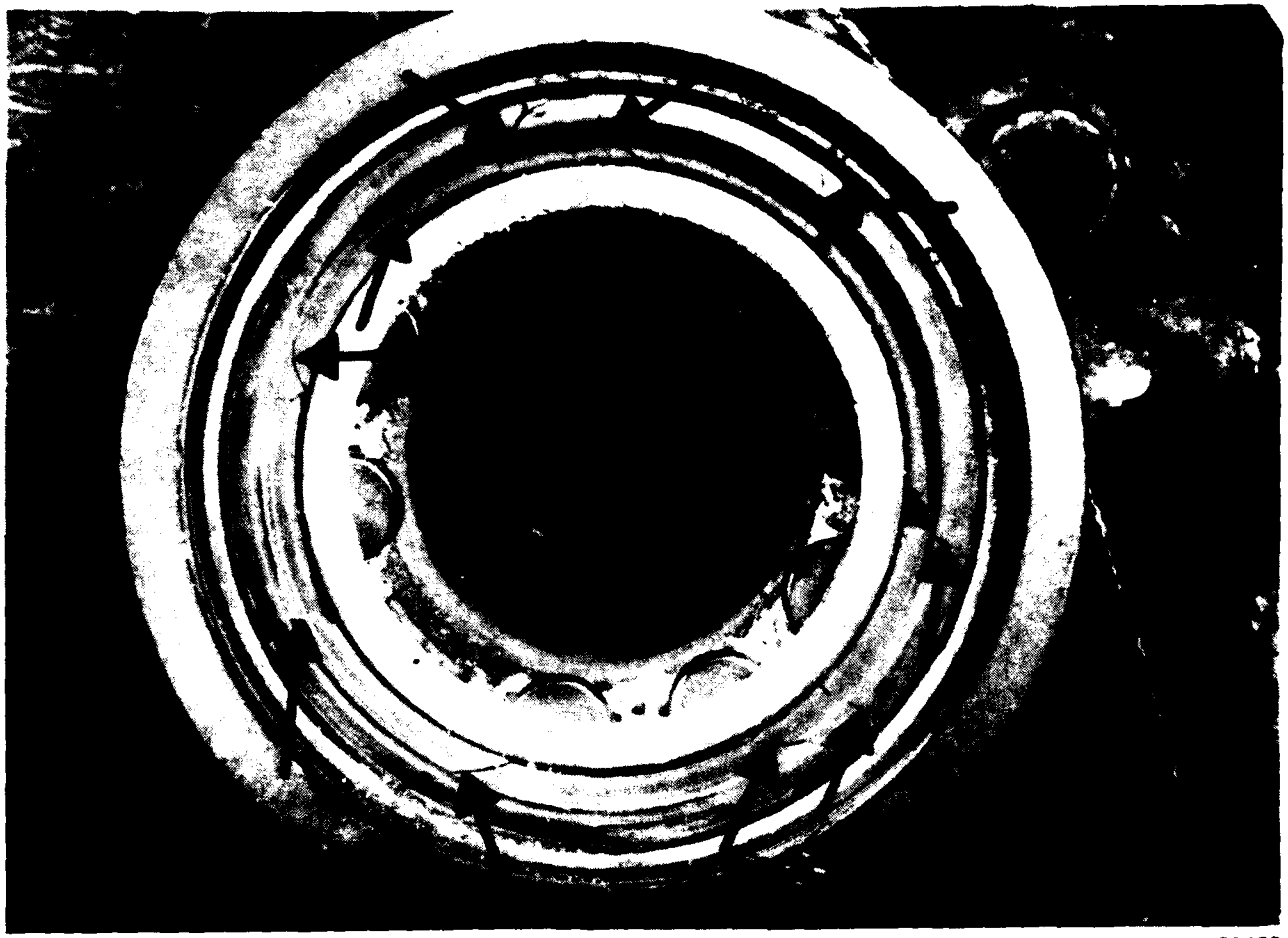
SILICON NITRIDE ROLLER BEARING (PN 309467) INNER RACE MOUNTED ON J-402 ENGINE SHAFT AFTER ONE HOUR OF FUEL-LUBED ENGINE OPERATION. EXCELLENT CONDITION OF RACE IS SHOWN IN SMALL CIRCULAR MIRROR.

28430

FIGURE 11.

AFTER TEST 2(b)

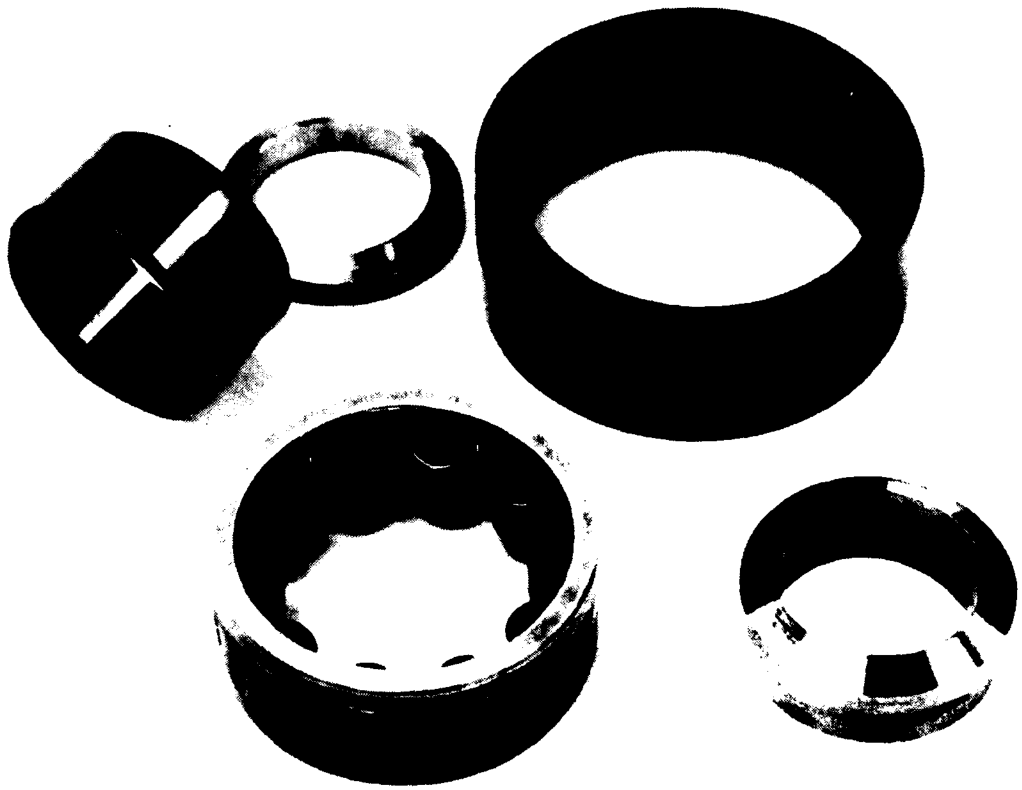
REAR VIEW OF SILICON NITRIDE BEARING IN HOUSING SHOWING OUTER RACE, CAGE AND ROLLERS. ARROWS LOCATE OUTER RACE CRACKS WHICH OCCURRED DURING FAILURE.



28439

FIGURE 12.

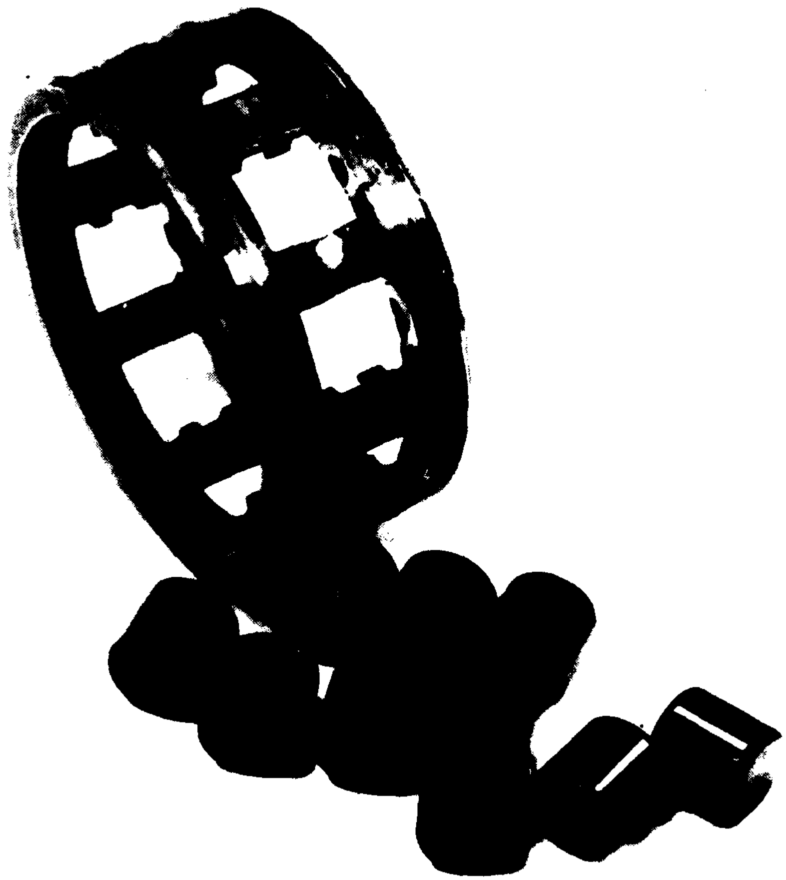
AFTER TEST 4
CERAMIC BEARING AFTER TWO HOURS OPERATION
½ HOUR UNLUBED, 1½ HOURS LUBED WITH 0-10cc FUEL/MIN.



28435

FIGURE 13.

AFTER TEST 4
CAGE AND ROLLERS FROM CERAMIC BEARING. ROLLER SURFACES ARE IN EXCELLENT CONDITION. CAGE GUIDING SURFACE (O.D.) SHOWS WEAR OF SILVER PLATING DUE TO LACK OF LUBRICATION.



28460

FIGURE 14.

ENDURANCE TEST SCHEDULE CYCLE

MODEL 372-2A ENGINE

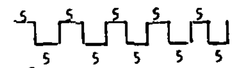
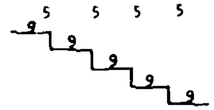
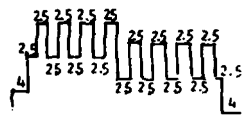
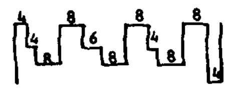
RUN LETTER	NAME OF RUN	MINIMUM OBSERVED RPM	MINIMUM OBSERVED PARAMETER	DESCRIPTION OF RUN	GRAPHIC DESCRIPTION OF RUN	TIME OF RUN	READING TIME	DATA TO BE TAKEN
A	Maximum Continuous - Idle Thrust Run	39,600 29,700	EGT 1475°F	Alternate five minute periods		50 Minutes	Second Min. After Reaching each RPM	Standard
B	Incremental Rotational Speed Run	39,600 37,620 35,640 33,660 29,700	Run to Speed	Nine minute period at each speed.		45 Minutes	Second Min. After Reaching each RPM	RPM, thrust, EGT fuel flow
C	Thrust Transient Run Mission 1, 2, & 3	39,600 37,200 35,500 34,700 33,100 29,700	Run to Speed	2½ minute period at each speed during cycle part of run.		50 Minutes	Second Min. After Reaching each RPM	RPM, thrust, EGT fuel flow
D	Thrust Transient Run Mission 4	39,600 38,700 36,400 29,700	Run to Speed	4 to 8 minute period at each speed as shown.		70 Minutes	Second Min. After Reaching each RPM	RPM, thrust, EGT fuel flow
E	Extended Maximum Continuous	39,600	1475°F	Steady state run	30 Minutes	30 Minutes	At tenth and twenty fifth minutes	Standard
F	Normal Thrust (97% N. Max)	38,400	1370°F	Steady state run	15 Minutes	15 Minutes	At tenth minute	Standard
G	Short Maximum Continuous	39,600	1475°F	Steady state	15 Minutes	15 Minutes	At tenth minute	Standard
H	Normal Thrust (97% N. Max)	38,400	1370°F	Steady state	10 Minutes	10 Minutes	At fifth minute	Standard
I	Short Maximum Continuous	39,600	1475°F	Steady state	15 Minutes	15 Minutes	At tenth minute	Standard

FIGURE 15.

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