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POST TEST INSPECTION OF THREE BRAYTON ROTATING UNITS

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ABSTRACT

The NASA Lewis Research Center has assembled and is testing a closed-Brayton-cycle space electric power conversion system. This system was designed to produce 2 to 10 kW net continuous electric power for at least 5 yr.

The turbine-alternator-compressor power conversion unit in this system is designated "Brayton Rotating Unit" (BRU). Four BRUs have been tested and have an accumulated running time in excess of 9000 hr. These tests have demonstrated that the unit has met or exceeded performance objectives. Yet to be verified are the design objectives for life.

Two BRUs have been disassembled and inspected after substantial running time. One BRU has operated for 700 hr, another for 3000 hr. The bearings ran successfully with no detectable change in bearing operation or performance. However, minor pitting was found in the ball-socket bearing pivots. Dimensional, electrical, and visual inspections revealed no other changes in the units.

In testing two of the BRUs, inadvertent runaway occurred which resulted in bearing seizure at 52,000 rpm (140 percent of design speed) but with no damage external to the BRUs. In fact, damage was limited to the bearing assemblies. With just bearing replacement, both units have passed acceptance test. The failure mode of the bearings and the resulting damage is described.

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THE NASA LEWIS RESEARCH CENTER has assembled and is testing a closed Brayton-Cycle electric power system for space. This system was designed to produce 2 to 10 kW net continuous electric power for at least 5 yr.

The power conversion unit in this system is designated Brayton Rotating Unit (BRU) which has been tested extensively as a single component and in combination with the complete Brayton system. Four identical BRUs have been tested and have an accumulated running time of 9000 hr with 3000 hr on BRU-2 and 5300 hr on BRU-4. These tests have been very successful in that they have demonstrated that the unit meets or exceeds all design objectives, no mechanical problems were detected in the BRU, and there was no measurable degradation of aerodynamic or electrical efficiency.

Three of the four units tested have been disassembled, inspected, and rebuilt. BRU-2 was a scheduled teardown after 3000 hr. BRU-1 and BRU-3 were disassembled to repair damage caused by problems external to the units that resulted in uncontrolled runaways. BRU-1 had operated for 700 hr and BRU-3 for 4 hr.

This paper presents the results of those inspections and shows that at this time there is no apparent limitation to the 5-yr-life objective. Because the bearings contain the only potential wear points in the machine, they were the components of most concern. This paper therefore deals heavily with that area.

BRU DESCRIPTION - The BRU, designed and fabricated by the AiResearch Manufacturing Company is shown in cross section in Fig. 1 and fully described in Ref. 1. The rotating assembly which operates at a design speed of 36,000 rpm includes a radial outflow single-stage compressor, a four-pole modified Lundell alternator and a radial inflow single-stage turbine. The severe axial thermal gradient resulting from the turbine inlet temperature of 1600° F and the compressor inlet temperature of 80° F, the extended range of operation (2.0 to 10 kW_e) and the rotational speed that requires operation between the second and third system criticals presented a difficult problem in the design of the rotor bearing system.

The rotating assembly is supported by two pivoted pad gas journal bearings and a double acting gas thrust bearing. At design speed the bearings operate in the hydrodynamic mode. External pressurization is provided for hydrostatic operation during startup and shutdown. A journal bearing assembly is shown in Fig. 2. Each of the three pads in each journal bearing is pivoted on a lapped ball-socket joint of 0.2500 in. radius and fabricated from solid tungsten carbide (K 96). The ball end of one pivot of each bearing is flexibly supported radially with respect to the bearing carrier on a resilient mount having a nominal spring rate of 2000 lb/in.

The thrust bearing assembly consists of a flat disc runner (integral with the rotor) and a pair of mirror image Rayleigh step stator plates. The thrust bearing assembly is shown in Fig. 3. To facilitate alignment of the stator faces with the thrust rotor, the stator assembly is mounted on a flexure pivoted gimbal. The motion about each gimbal axis is damped by two lightly loaded friction pads shown in Fig. 4.

The dynamic motions of the rotor bearing system were monitored by 20 capacitance probes reading shaft orbital motions at each bearing, pitch and roll motions of the bearing pads, radial displacement of the flex-mounted pivot, thrust bearing gimbal-to-ground motions, and thrust bearing film thickness.

The ball and socket pivots in the journal bearing and the friction pads on the thrust gimbal are the only potential wear points in the BRU. Excessive wear in these contacting areas may possibly limit the life of the unit. The prime objective, therefore, of endurance testing is to verify the integrity of these components.

RESULTS AND DISCUSSION

BRU-2 - BRU-2 was first tested for 1000 hr in a test loop for evaluation of the BRU only. The objective being to evaluate its operation and overall performance as a component. This testing, in addition to covering the complete design operating range, also included off-design testing to determine its operational limitations. The results of this phase of testing, reported in Refs. 2 and 3, demonstrated the mechanical integrity of the unit and showed that it exceeds or meets all performance objectives. The rotor bearing system dynamics were monitored continuously during this test phase. Bearing operation was stable at all times and there was no change in the bearing performance over the period tested.

Reference 2 also describes a pneumatic instability that occurred in the turbine and journal bearing during hydrostatic operation when the pressure ratio - prior to startup - between the bearing hydrostatic supply pressure and the bearing ambient pressure exceeded 10. Figure 5 is the picture of an oscilloscope trace of the radial motion of the flexibly-mounted pad and pivot observed at zero speed and a bearing pressure ratio of 16. The top trace is the turbine end pivot with radial motions of 0.0006 in. and the bottom trace is the compressor pivot with radial motion of 0.0001 in.

This anomaly has not been observed in the three other units that have been tested. It does not present an operational problem because the unit, when operating in the Brayton Engine, is started with the bearing pressure ratio below 10.

After completing the component phase of testing the unit was installed in a flight-type system and tested as part of a complete Brayton system. The test duration in this configuration was almost 2000 hr. No problems were encountered with the BRU and no performance changes were detected during the 2000 hr of operation. However, very early in this test, the capacitance probe signal conditioning equipment became inoperative and the performance characteristics of the rotor bearing system were therefore unknown.

Because of this, the unit was subjected to a cold spin test after removal from the engine and prior to disassembly. The objective being to obtain an accurate evaluation of the dynamic performance of the rotor bearing system after 3000 hr of operation. Dynamic performance can be evaluated by means of the outputs of the capacitance probes since performance degradation would result in changes in the motions of the rotor or bearing components. Oscilloscope traces of the probe outputs photographed early in the component test phase were compared with the traces photographed during the post test cold spin. The results indicate that dynamic performance was unchanged by 3000 hr of hot operation. Typical comparisons are shown in Figs. 6, 7, and 8. All traces were obtained at 36,000 rpm with the bearings operating in the hydrodynamic mode. Figure 6 shows the time trace of the turbine end shaft orbits and Fig. 7 shows the compressor end. The peak-to-peak amplitudes have not changed as would result if for example a change in shaft balance had occurred during operation. Figure 8 shows the time trace of the motions of the leading edge of the turbine flexibly-mounted bearing pad and one solidly mounted pad. Leading edge motion is measured by two probes; one mounted at each corner of the leading edge. Leading edge motions of the flex pads are a combination of pitching and radial translation of the pad and pivot. The peak-to-peak amplitude of the traces shows that no change in the amplitude of pad motions has occurred. A comparison of the wave form of the traces shows that no change has occurred in the performance of the pivot. Stick-slip motion in the pivot, for example, would result in an erratic wave form.

After disassembly, the rotor and wheels along with the compressor and turbine scrolls were dimensionally inspected, no measurable changes were detected. The alternator windings were given a resistance and dielectric check, no changes were detected.

Those parts exposed to the hot turbine gas were coated with a thin oxide coating that ranged in color from light gray to black. Spectrographic and X-ray diffraction analysis indicated that the prime constituent in the coating was manganese metaborate. Color was a function of the coating thickness; the thicker the coating, the darker the color. The origin of the boron is as yet unknown although boron did exist in the braze material used to braze the headers to the manifold in the electric heat source. The significant aspect of the oxide coatings is that two of the surfaces coated by the oxide had been plated to minimize radiant heat transfer. The seal and shroud assembly shown in Fig. 9 was rhodium plated, and a portion of the back of the turbine scroll shown in Fig. 10 was gold plated. The design objective of these plated surfaces was obviously negated by the oxide film, however, there was no apparent change in performance or internal temperatures of the unit.

Inspection of the bearing pivots revealed identical amounts of minor surface fretting or roughness on all the pivot surfaces. The turbine end flexible pivot however also contained a small localized area of more extensive damage.

Figure 11 shows a typical pivot pair. The fretted areas were uniformly distributed over the total contact area indicating a good mechanical fit and uniform load distribution. To the naked eye, these areas appeared to be surface stains and covered about 25 percent of the total contact area. Figure 12 is a typical Talysurf trace taken on a pivot ball. Looking at the trace, it is difficult to distinguish the fretted surface from the original surface roughness of 2 μ in. It can, however, be concluded that the peak-to-peak roughness was no greater than 4 μ in. Figure 13 is a picture of the turbine end flexible pivot showing the additional damage area in that pivot. Figure 14 is a Talysurf trace taken across that area and shows that the maximum depth of damage was 130 μ in. Comparing the areas under the curve above and below the base line (surface) it can be seen that most of the damaged material has been removed from the surface. The matching area in the socket was similar, indicating that the damage debris worked its way out of the pivot.

Microscopic examination of the material formation at the edges of the damage area indicated that the damage was the result of abnormal loading conditions in the pivot. The observed material deformations were those resulting from relative motions normal to the surface which could occur only with changing loads. Material has been plucked from the surface with no evidence of smearing that would result with sliding. The pivots are designed to operate under constant load with pure sliding motion. There is little question but that the above damage was the result of the severe abnormal motions and loading conditions to which the pivot was subjected in the extensive investigation of the pneumatic instability during the component test phase.

The contacting surface of the friction dampers on the thrust bearing gimbal showed no signs of wear. The unit has been reassembled with a new turbine end flexible shoe and beam mount. It has been cold tested to rated speed prior to reinstallation in the Brayton engine for continued testing.

BRU-1 AND BRU-3 - BRU-1 and BRU-3 were disassembled after overspeed bearing seizures. Both overspeeds were the result of problems external to the BRU that caused sudden removal of the alternator load while under power. The results were almost identical in both cases. The BRUs accelerated to above 50,000 rpm in about 20 sec. Sudden stoppage due to bearing seizure occurred at 52,000 rpm (140 percent of design speed). Inspection, which included a dimensional check of all parts, revealed that the damage in both units was identical and was limited to the shaft journal coatings, the bearings, the labyrinth seals, and eight capacitance probes in each unit (4 orthogonal and 4 thrust). The damage sustained by the journal bearing and seals was minor, with the thrust bearing absorbing the energy of the sudden stop. The limit of the damage suffered by the journal bearings shown in Fig. 15 was demonstrated during the disassembly. After removal of the thrust bearing, external pressurization was applied to the journal bearings and the shaft rotated freely.

Figure 16 shows the turbine side of the thrust bearing; the wear pattern on the compressor side was identical. Damage was contained within the circular band that extends 0.4 in. from the edge of the runner. The resulting debris locked up the shaft by filling the axial clearance (0.0024 in.) between the runner and the stators. Both units have been rebuilt with replacement shafts, thrust bearing stators and runners, journal bearing pads, labyrinth seals, and eight proximity probes. With the exception of the probes, all the damaged parts have been repaired.

EVALUATION OF PIVOT WEAR - BRU-1, having run for 700 hr, provides an additional data point for the evaluation of pivot wear. The depth of the surface roughness in the wear areas of the 3000-hr unit were no greater than that observed in the unit that ran 700 hr. It appears that after an initial "wear in" the wear rate decreases to a point where the additional wear is not measurable after an additional 2000 hr.

The results obtained here are not sufficient to predict the life of the BRU pivots. Additional insight, however, can be obtained by comparing these observations with results of an investigation of gas-bearing tilting-pad pivots conducted by Mechanical Technology Inc. under contract to the NASA (4)*. Included in this investigation were tungsten carbide pivots with the same configuration as those in the BRU. The pivot loads used were the same as those in the BRU but the pivot motions were 10 to 15 times greater. As might be expected, the wear rates were much more severe.

The pivots were tested for 1000 hr and resulted in surface roughness up to 60 μ in. that covered the entire contact area. The difference in pivot motion amplitude precludes a direct comparison of wear rates with the BRU. The significant point of this test is that at the conclusion of the pivot tests there was no significant change in the pivot motions.

CONCLUSIONS

BRU-4 has accumulated in excess of 5300 hr and testing is continuing. This, plus the post test inspections of BRU-1 and BRU-2 after 700 and 3000 hr have verified the mechanical integrity of the design. The results also indicate that there is no apparent limitation to the objective of 5-yr life. The limited wear found in the bearing pivots does, however, point out the need for the continued testing to fully verify 5-yr pivot life.

The conclusion to be drawn from the results of the BRU tests and the pivot wear tests is that surface roughness over the entire pivot surface of as much as 60 μ in. as seen in the pivot tests and localized damage to depth of 130 μ in. as seen in BRU-2 does not impair the motions of the bearing pads.

The fact that the BRU-2 pivot roughness after 3000 hr is less than 4 μ in. and that BRU-4 has operated in excess of 5300 hr with no change in performance has firmly demonstrated a 6-month life capability. It also indicates that BRU life testing should now be continued to at least 10,000 hr (14 months) to provide meaningful new data.

REFERENCES

1. Anon., "Design and Fabrication of the Brayton Rotating Unit (BRU)." AiResearch Mg. Co. of Arizona. Rep. APS-5334-R, 1971.

2. H. A. Klassen, C. H. Winzig, R. C. Evans, and R. Y. Wong, "Mechanical Performance of a 2-to-10-Kilowatt Brayton Rotating Unit." NASA TM X-2043, 1970.

3. D. G. Beremand, D. Namkoong, and R. Y. Wong, "Experimental Performance Characteristics of Three Identical Brayton Rotating Units." NASA TM X-52826, 1970.

4. M. B. Peterson, B. F. Geren, E. B. Arwas, S. Groz, S. F. Murry, J. W. Lund, and F. F. Ling, "Analytical and Experimental Investigation of Gas Bearing Tilting Pad Pivots." Mechanical Technology, Inc. Rep. MTI-69-TR-32, NASA CR-72609, September 1969.

*Numbers in parentheses designate References at end of paper.

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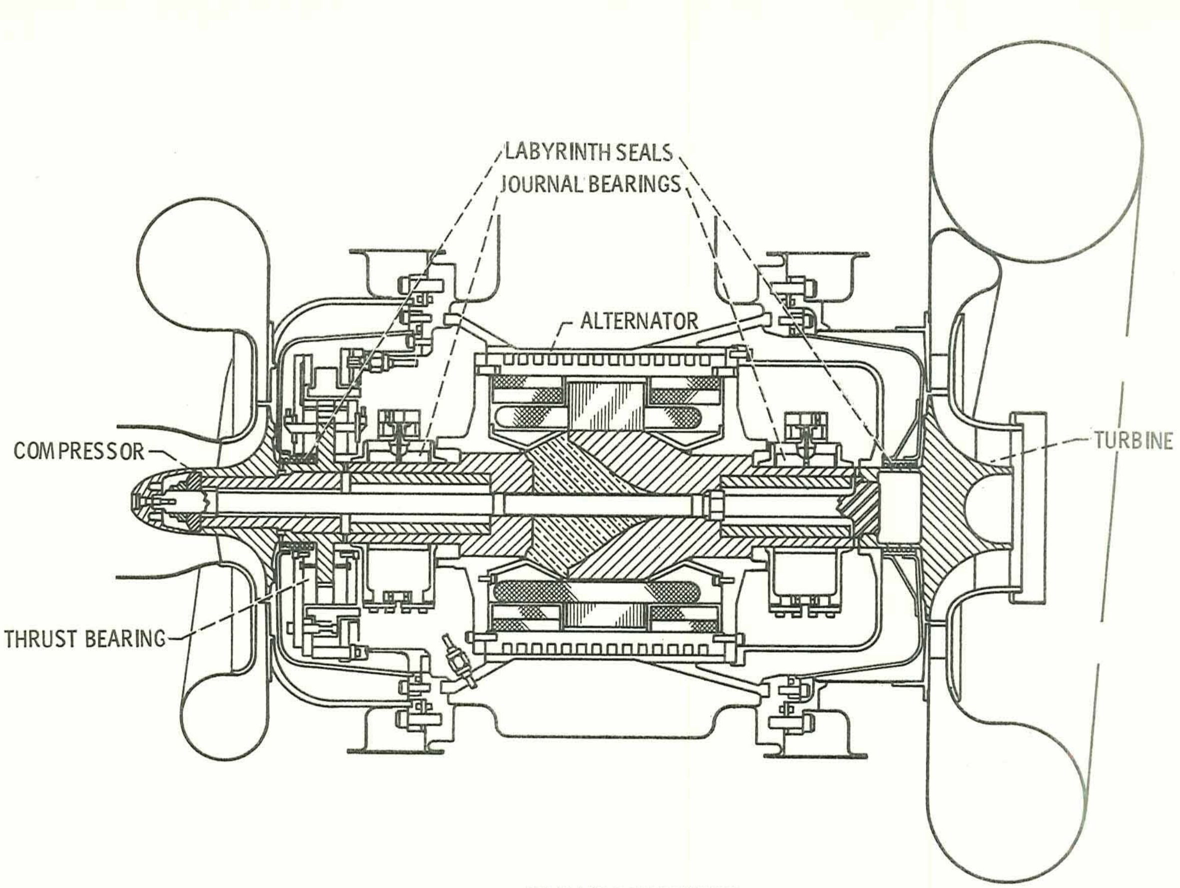
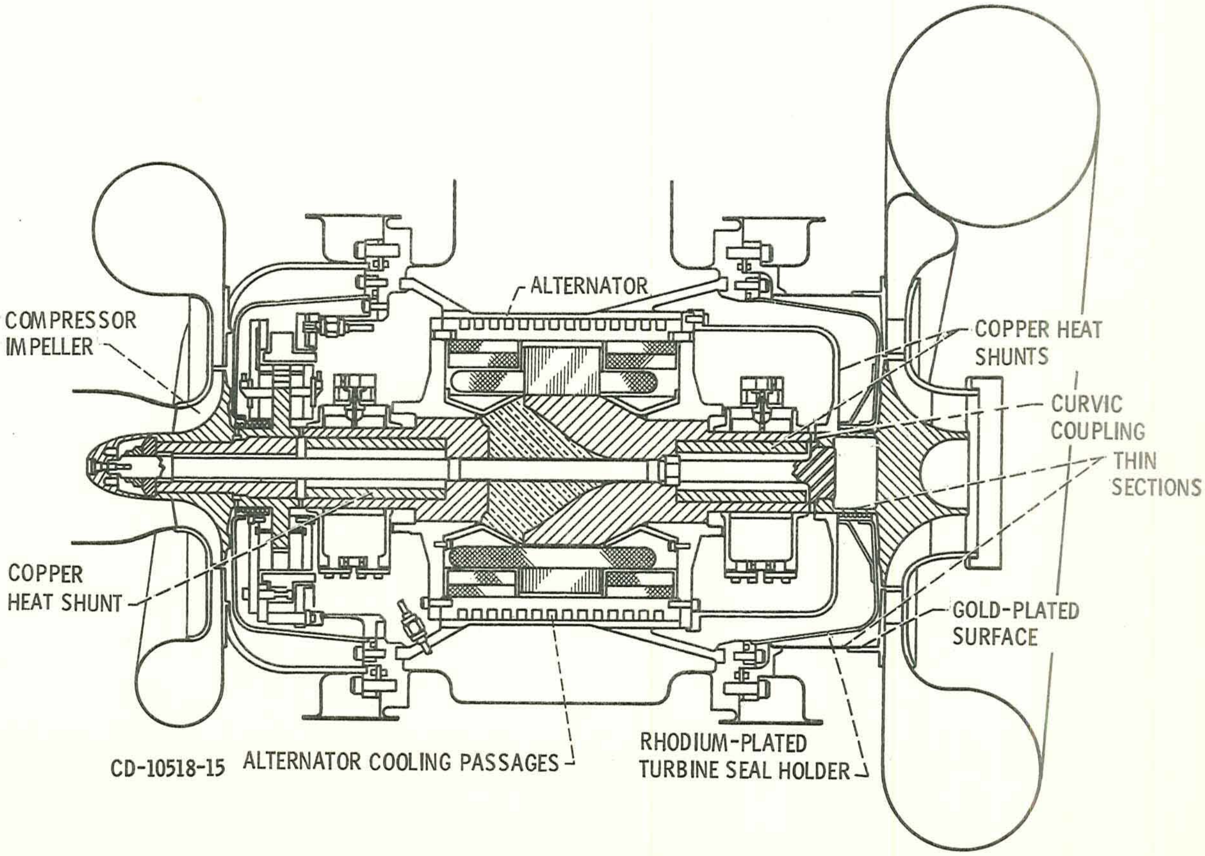


(A) MAIN COMPONENTS.



(B) TEMPERATURE CONTROL SYSTEM.

Figure 1. - Brayton rotating unit cross section.

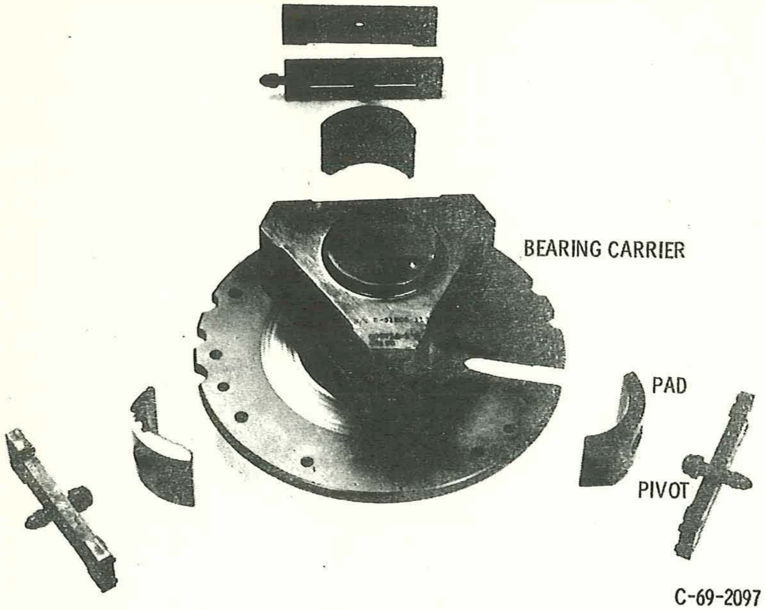


Figure 2. - Journal bearing assembly.

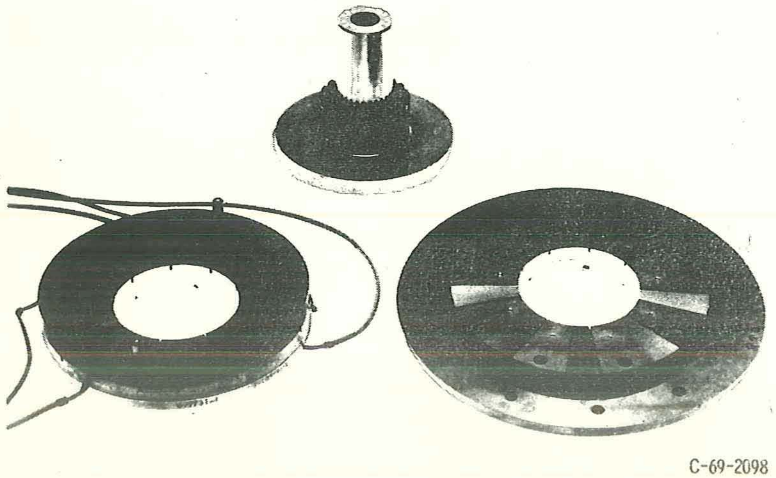


Figure 3. - Thrust bearing stators and runner.

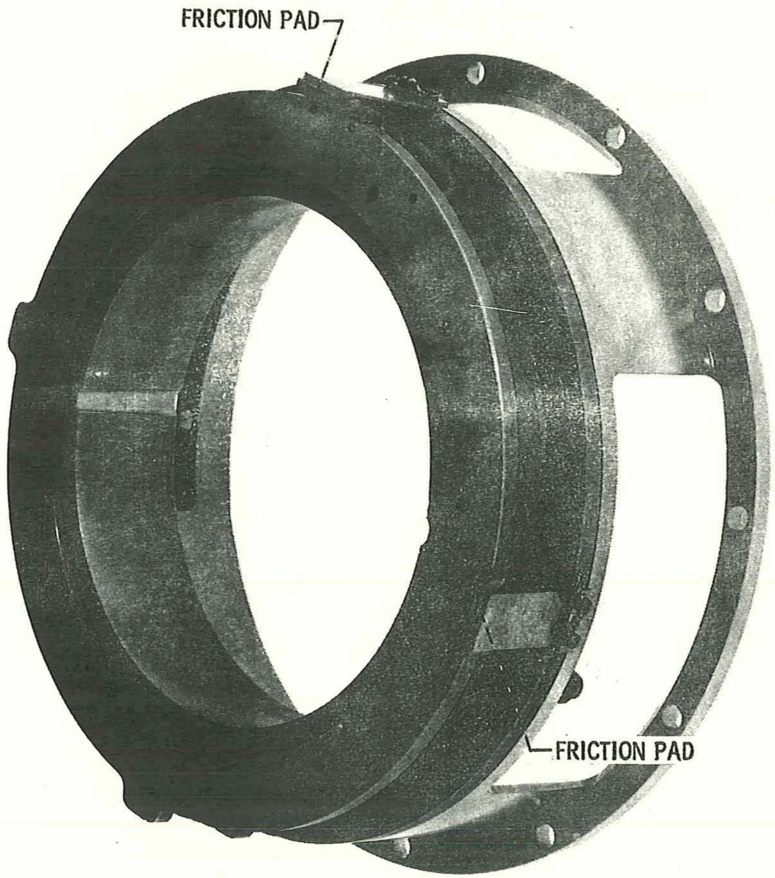


Figure 4. - Flexure pivoted gimbal assembly.

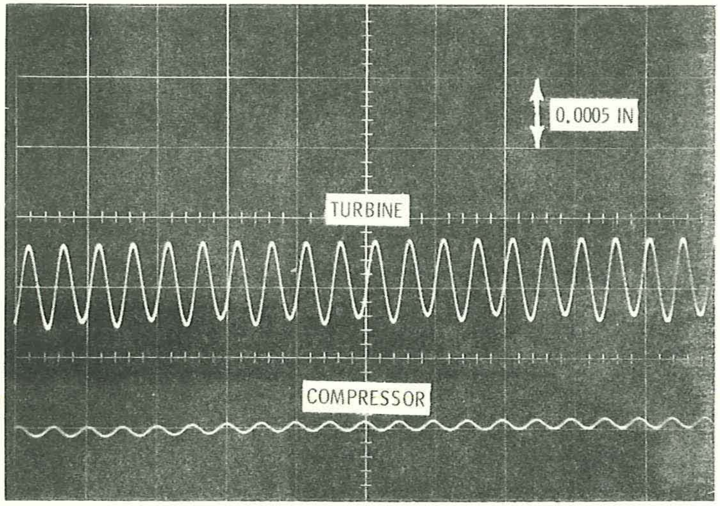
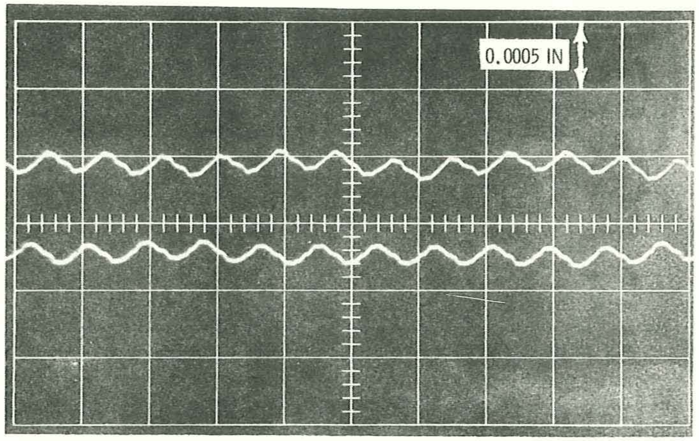
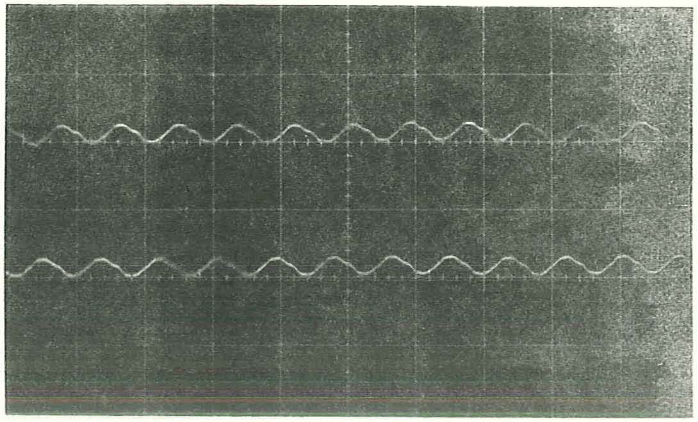


Figure 5. - Radial motions of flexibly mounted pads and pivots; turbine and compressor journal bearings. Pneumatic instability; bearing pressure ratio, 16; zero speed. Externally pressurized.

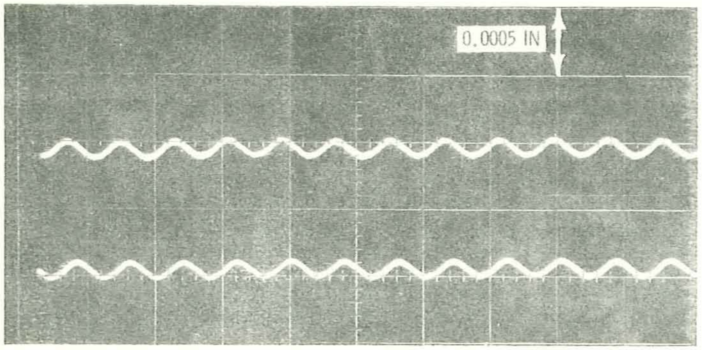


(A) POST TEST COLD SPIN.

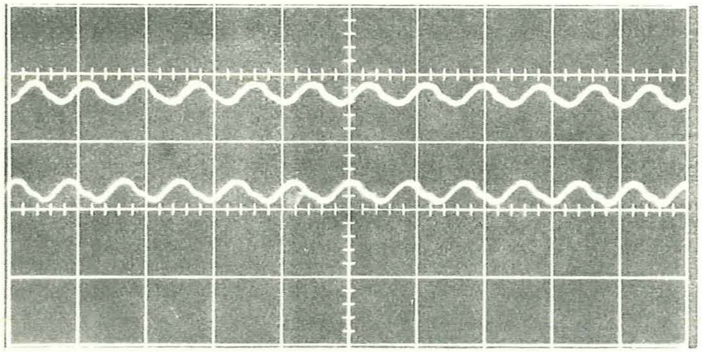


(B) COMPONENT TEST PHASE.

Figure 6. - SHFT motions at turbine bearing 36 000 rpm; self acting.

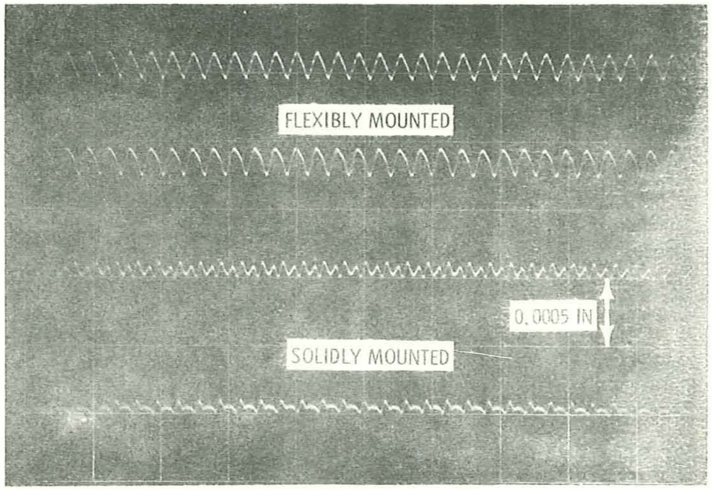


(A) POST TEST COLD SPIN.

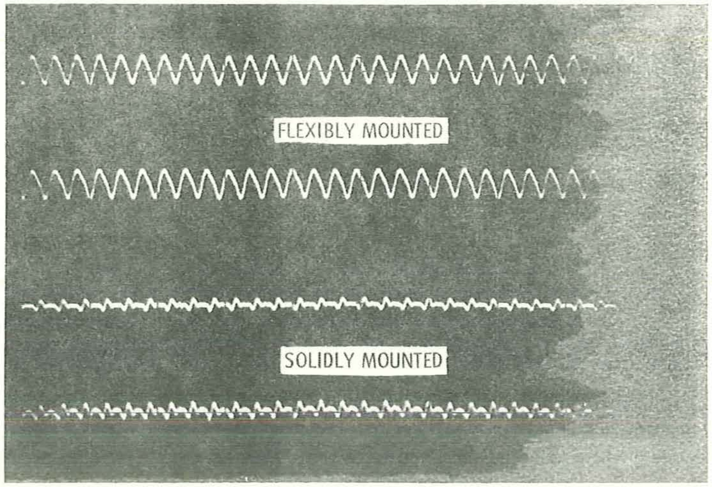


(B) COMPONENT TEST PHASE.

Figure 7. - Shaft motions at compressor end 36 000 rpm; self-acting.



(A) POST TEST COLD SPIN.



(B) COMPONENT TEST PHASE.

Figure 8. - Motions of leading edges of two turbine journal bearing pads - 36 000 rpm; self acting.

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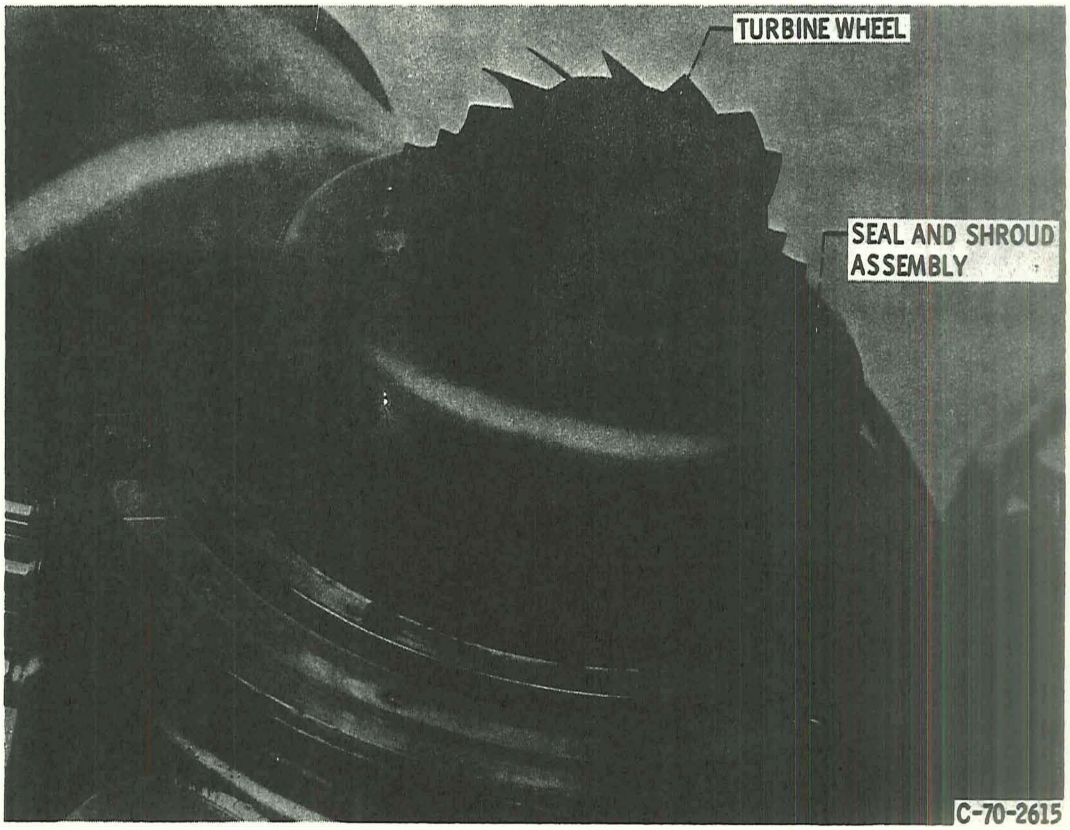

Figure 9. - Seal and shroud assembly.

Figure 10. - Turbine scroll.

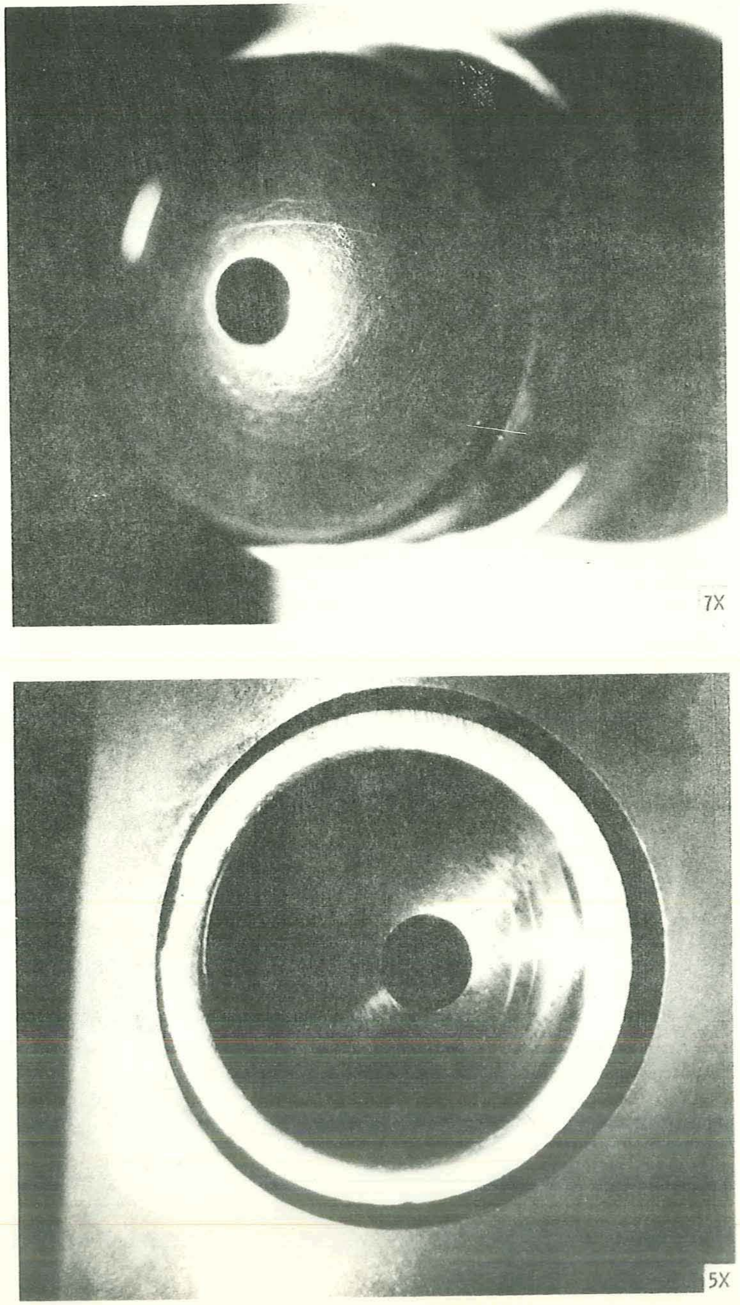

Figure 11. - Photomicrograph of rigid mounted ball and socket showing minor surface damage areas.

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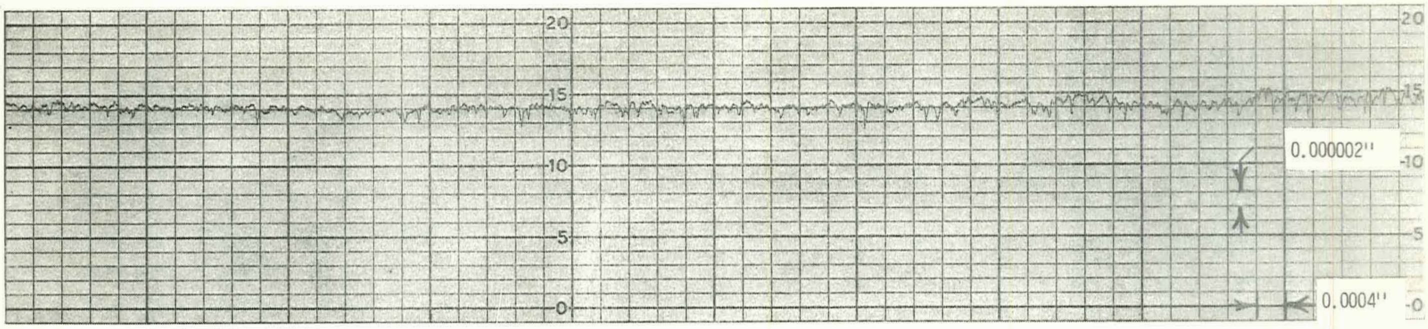

Figure 12. - Talysurf trace across rigid mounted ball pivot.

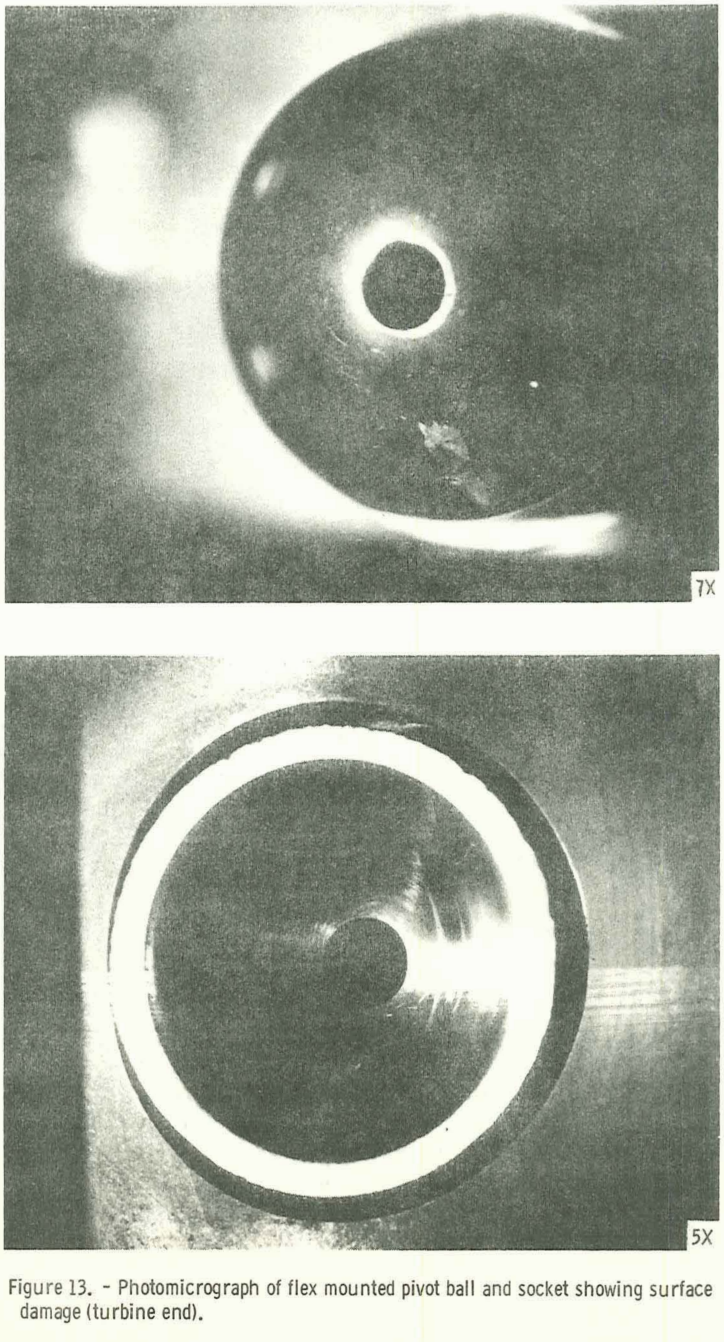

Figure 13. - Photomicrograph of flex mounted pivot ball and socket showing surface damage (turbine end).

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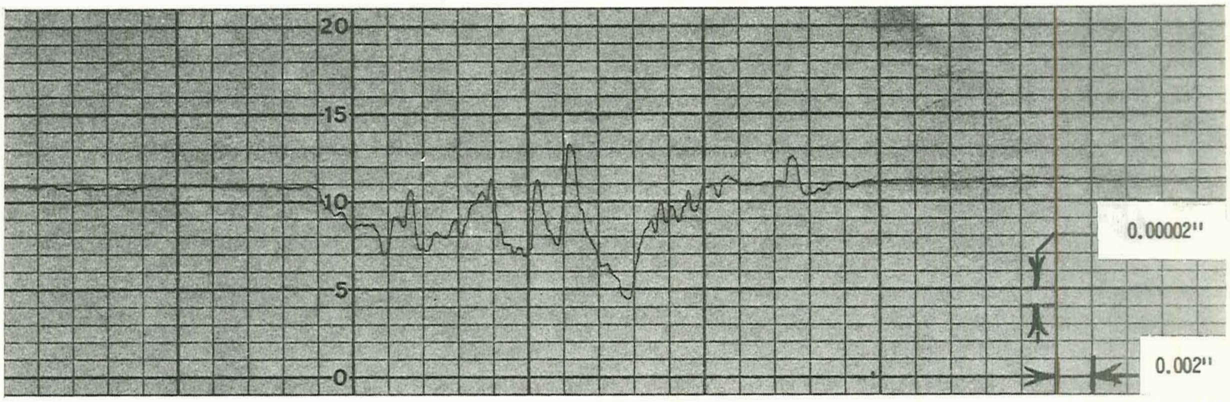

Figure 14. - Photomicrograph and Talysurf trace of surface damage on flex mounted ball (turbine end).

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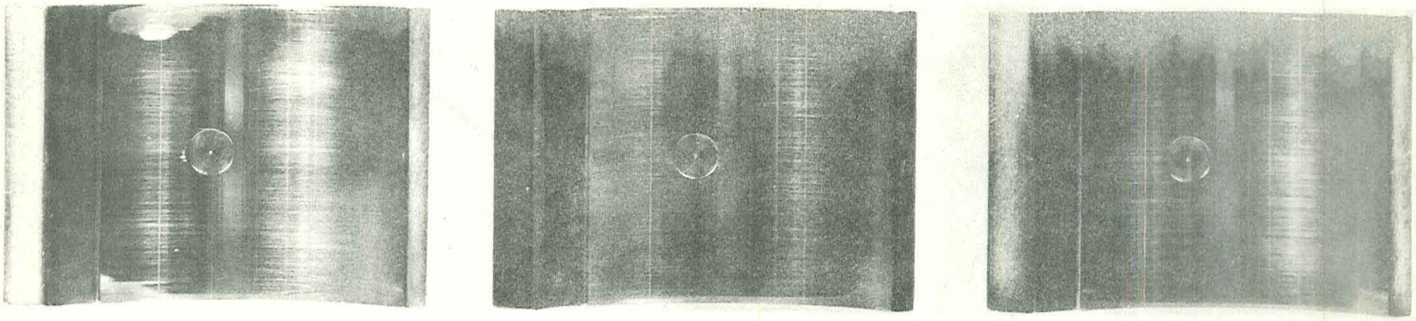
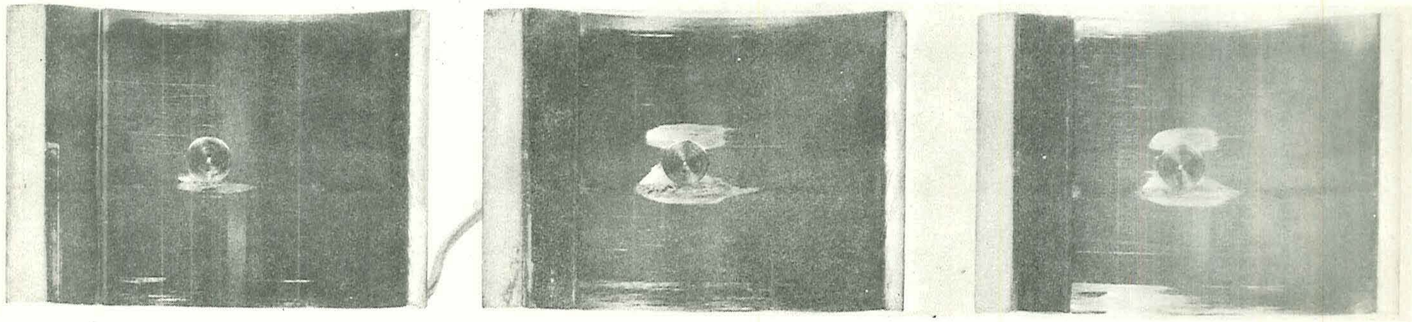
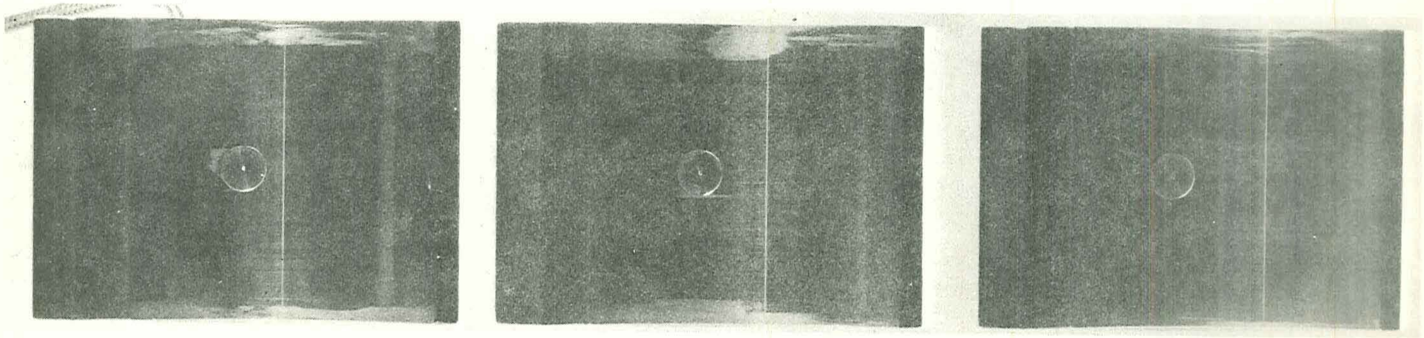


TURBINE END

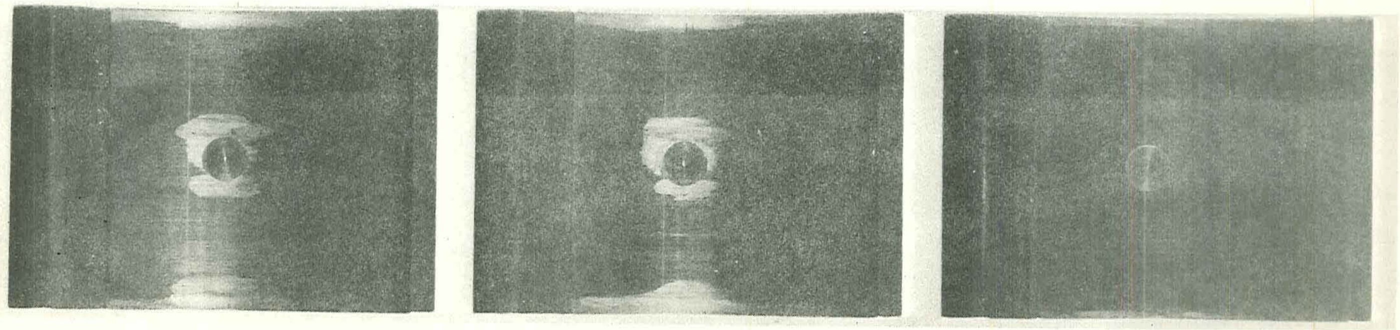


COMPRESSOR END

(A) BRU - 1.



TURBINE END



COMPRESSOR END

(B) BRU - 3.

Figure 15. - Overspeed damage to journal bearing pads.

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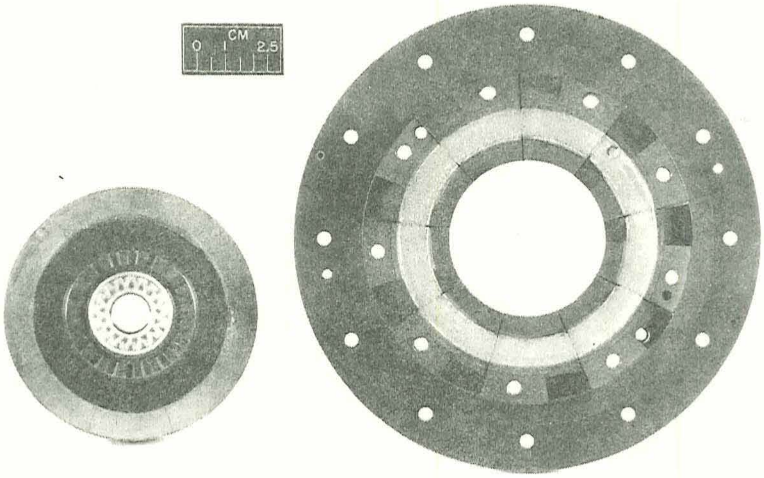

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Figure 16. - Overspeed damage to thrust bearing.

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