A MINIATURE TILTING PAD GAS LUBRICATED BEARING

Herbert Sixsmith and Walter L. Swift
Creare R&D Inc.

ABSTRACT

This paper describes the design and development of a miniature tilting pad gas bearing developed for use in very small turbomachines. The bearings have been developed for cryogenic turboexpanders with shaft diameters down to about 0.3 cm and rotational speeds up to one million rpm. Cryogenic expansion turbines incorporating this type of bearing should be suitable for refrigeration rates down to about 10 w.

INTRODUCTION

In helium refrigeration the adoption of compressors and expanders with gas lubricated bearings should eliminate wear and permit long life continuous operation with little or no maintenance. The present report describes a miniature tilting pad gas lubricated bearing which should be suitable for application in miniature cryogenic expansion turbines. The new design of bearing represents a further development of a tilting pad bearing which is described in References 1, 2 and 3. The major novel feature lies in the design which facilitates the manufacture of bearings down to internal diameters of 0.3 cm or less.

The successful operation of bearings of this size permits reductions in size and corresponding rates of refrigeration for cryogenic turboexpanders down to 10 w without appreciable loss of efficiency.

DESCRIPTION OF BEARING

The layout of the bearing is shown in Figure 1. The internal and external diameters are 0.317 cm and 1.27 cm respectively. The length is 0.33 cm. The pads are supported on elastic hinges the material of which is beryllium copper. The hinges are machined by spark erosion from a solid annulus at each end of the bearing. Each pad is given a bias in the direction of tilt to form a converging film by means of a helical compression spring. Each spring is provided with a shoe to make contact with the pad and a screw to adjust the pressure and the corresponding angle of tilt.

CONSTRUCTION

The various components of the bearing are machined and assembled ready for silver soldering as shown in Figure 2. The material of the outer housing and inner sleeve is leaded bronze. The material of the end discs is beryllium copper. After soldering, heat treating to temper the beryllium copper and cleaning, the bearing is turned to its final dimensions. A small

189

amount of metal is left in the bore which will be removed later by lapping to provide a smooth finish. Four offset slots are cut through the inner sleeve thereby forming four pads. The bore is then lapped out to size. The elastic hinges are then formed by removing unwanted metal by spark erosion. When the bearing has been cleaned, the springs and adjusting screws are inserted and the angle of tilt is adjusted. The bearing is now ready for operation.

THE TEST RIG

A pair of bearings have been tested in a rig as shown in Figure 3. The length of the shaft is 2.095 cm. The radial clearance between the shaft and the bearing along the hinge lines is less than 5 microns. The shaft is driven by a very simple turbine which consists of ten radial holes which break through into a recess 0.170 cm diameter in one end of the shaft. This is surrounded by a nozzle ring as shown in Figure 4.

A thrust bearing to position the shaft and support the thrust reaction from the turbine is located at the other end of the shaft. It consists of a conventional inherently compensated orifice 0.05 cm diameter. It is fed with a stream of compressed air to form a film of air between the end of the shaft and the thrust bearing. The flow restrictor in the outlet line from the thrust bearing serves to stabilize the shaft against pneumatic hammer.

A pair of capacitance probes are located one adjacent to each bearing to indicate the balance and stability of the shaft. A small flat on the shaft adjacent to the upper probe provides a signal once per revolution, which appears as a spike on the oscilloscope trace. The height of the sine wave due to shaft run-out can be compared with the height of the spike to provide an estimate of the run-out.

TEST RESULTS

In the first attempt to run the shaft whirling at exactly half speed commenced at about 5000 r/sec. The bearings were readjusted, with greater spring pressure to increase the angle of tilt of the pads. The adjustment was crude as equipment to measure the angle of tilt is not yet available. The method of adjustment consisted of screwing in the adjusting screws until the shaft would not slide freely in the bearing under the force of gravity. The screws were then backed out slightly until the shaft would again slide freely in the bearing. When the rig was reassembled the shaft was run up to 13,500 r/sec (810,000 rpm). At this speed whirling in the upper bearing commenced. The whirl speed was one quarter that of the shaft and the amplitude was quite small, about one tenth that of the spike. The whirling stopped when the rig was hit with a piece of wood which happened to be the back of a brush. After about an hour the speed was reduced to 12,500 rps (750,000 rpm). As this speed is sufficient for one potential application the rig was left running to demonstrate the absence of wear and to boost our morale. At the time of writing it has completed four weeks of continuous running.

REFERENCES

1. Sixsmith, H.; A HIGH SPEED TILTING PAD BEARING; Proceedings of the Gas Bearing Symposium at the University of Southampton, England, April 25-28, 1967.

2. Sixsmith, H.; A CENTRIFUGAL COMPRESSOR WITH GAS LUBRICATED BEARINGS; Proceedings of the 5th International Cryogenic Engineering Conference at Kyoto, Japan, May 7-10, 1974.

3. Sixsmith, H., Schlafky, A., and Swift, W.L.; A SMALL CENTRIFUGAL PUMP FOR CIRCULATING CRYOGENIC HELIUM; Presented at 1981 CEC/ICMC Conference, San Diego, CA, August 1981.

Question 1:   Ralph C. Longsworth, Air Products & Chemicals, Allentown, PA
                What is the anticipated efficiency of the 1/8" diameter expander?

Answer: We expect an adiabatic efficiency between 62% with a sub-critical shaft, and 70% if we can get a shaft to run above the first critical in bending.

Question 2:   J. Zimmerman, National Bureau of Standards, Boulder, CO
                What is the lowest power at which you can imagine a turbine expander being practical?

Answer: We believe that 10 watts is about the lowest practical power limit for expansion turbines. One can always go a bit smaller, but it gets more difficult and the efficiency decreases as the size is reduced.

Question 3:   L. Lombardini, Officine Galileo, Florence, Italy
                What is the material of the shaft?

Answer: Austenitic stainless steel.

Question 4:   Robert D. Averill, NASA Langley Research Center, Hampton, VA
                What pressure do the gas bearings operate at?

Answer: The bearings are self acting and not externally pressurized. Because the viscosity of gases is practically independent of pressure they will work satisfactorily over a wide range of pressures, from about half an atmosphere up to hundreds of atmospheres. In the test rig the pressure surrounding the bearings is about one atmosphere.

Question 5:   G. Claudet, CEA, CENG/SBT, Grenoble, France
                What is the start-up procedure for a dynamic gas bearing?

Answer: Because of the small size, gravity loads are very small and wear? due to friction at start up is negligible. All one has to do is to open the turbine inlet valve.
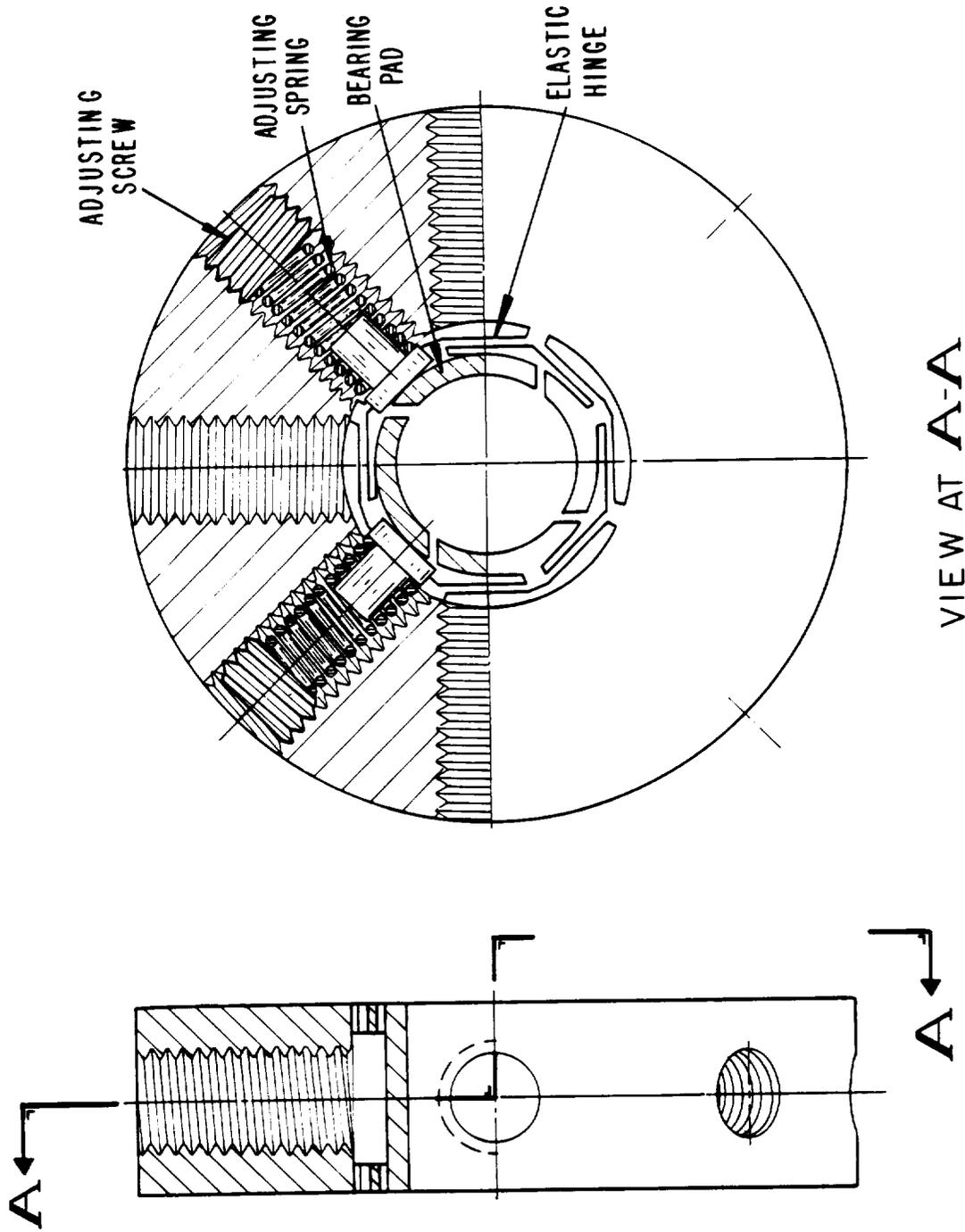
ADJUSTING SCREW

ADJUSTING SPRING

BEARING PAD

ELASTIC HINGE

VIEW AT A-A

Figure 1   FINISHED BEARING SHOWING ANGLE-OF-TILT ADJUSTING SPRINGS

A

A

A

A

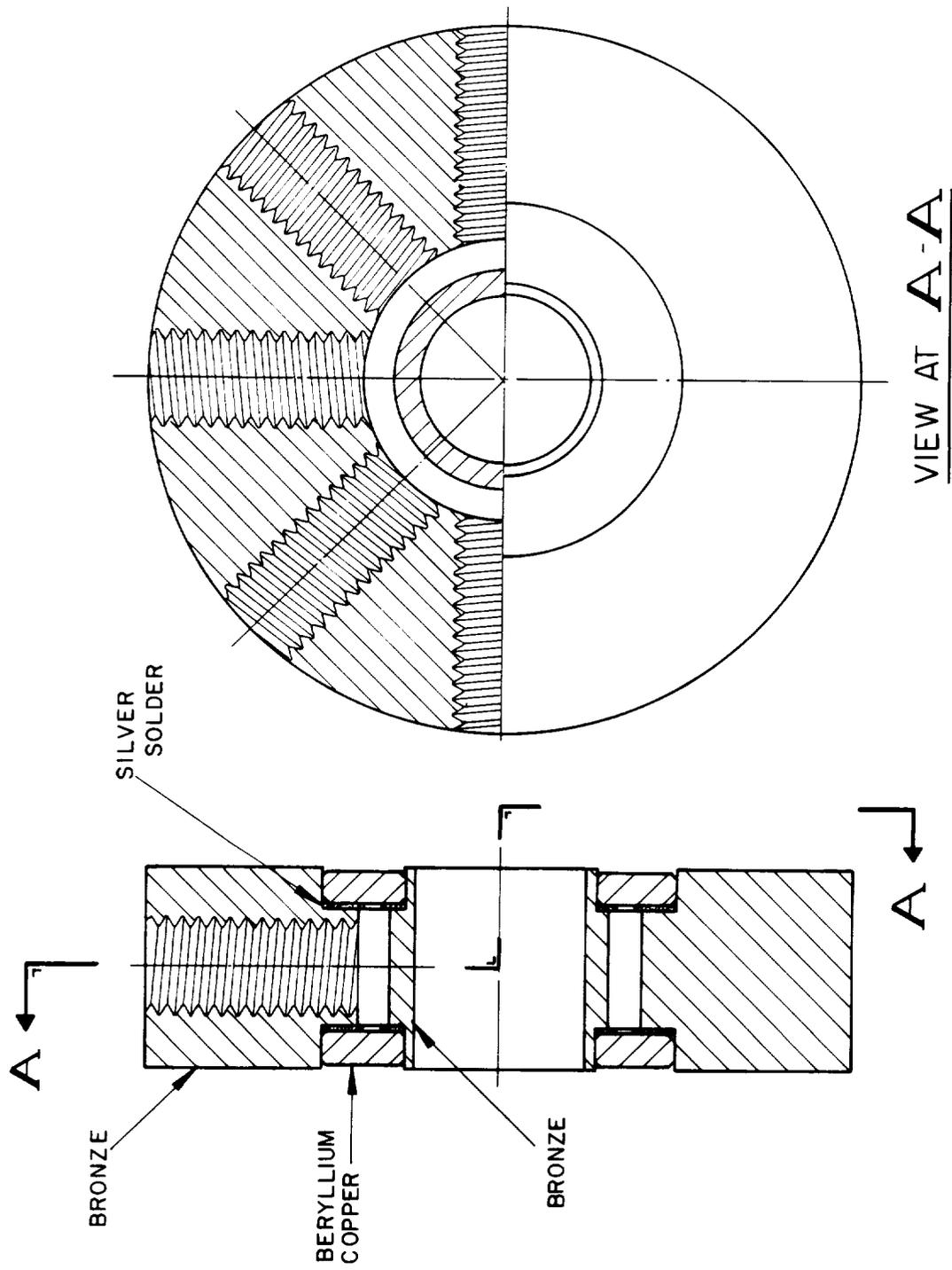SILVER SOLDER

BRONZE

BERYLLIUM COPPER

BRONZE

A

A

VIEW AT A-A

A

Figure 2 COMPONENTS OF BEARING ASSEMBLED READY FOR SILVER SOLDERING
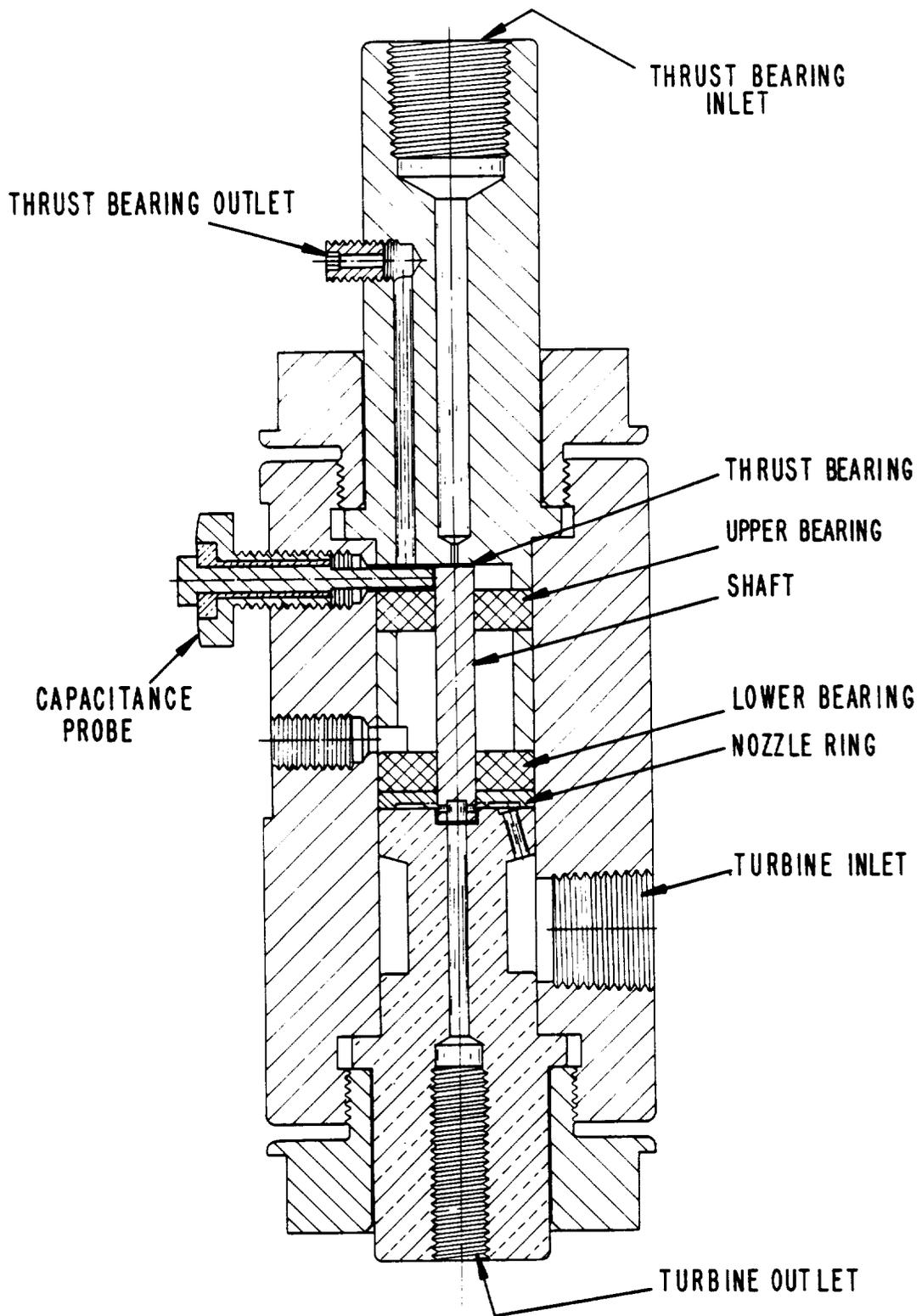
Figure 3 LAYOUT OF BEARING TEST RIG

THRUST BEARING INLET

THRUST BEARING OUTLET

THRUST BEARING

UPPER BEARING
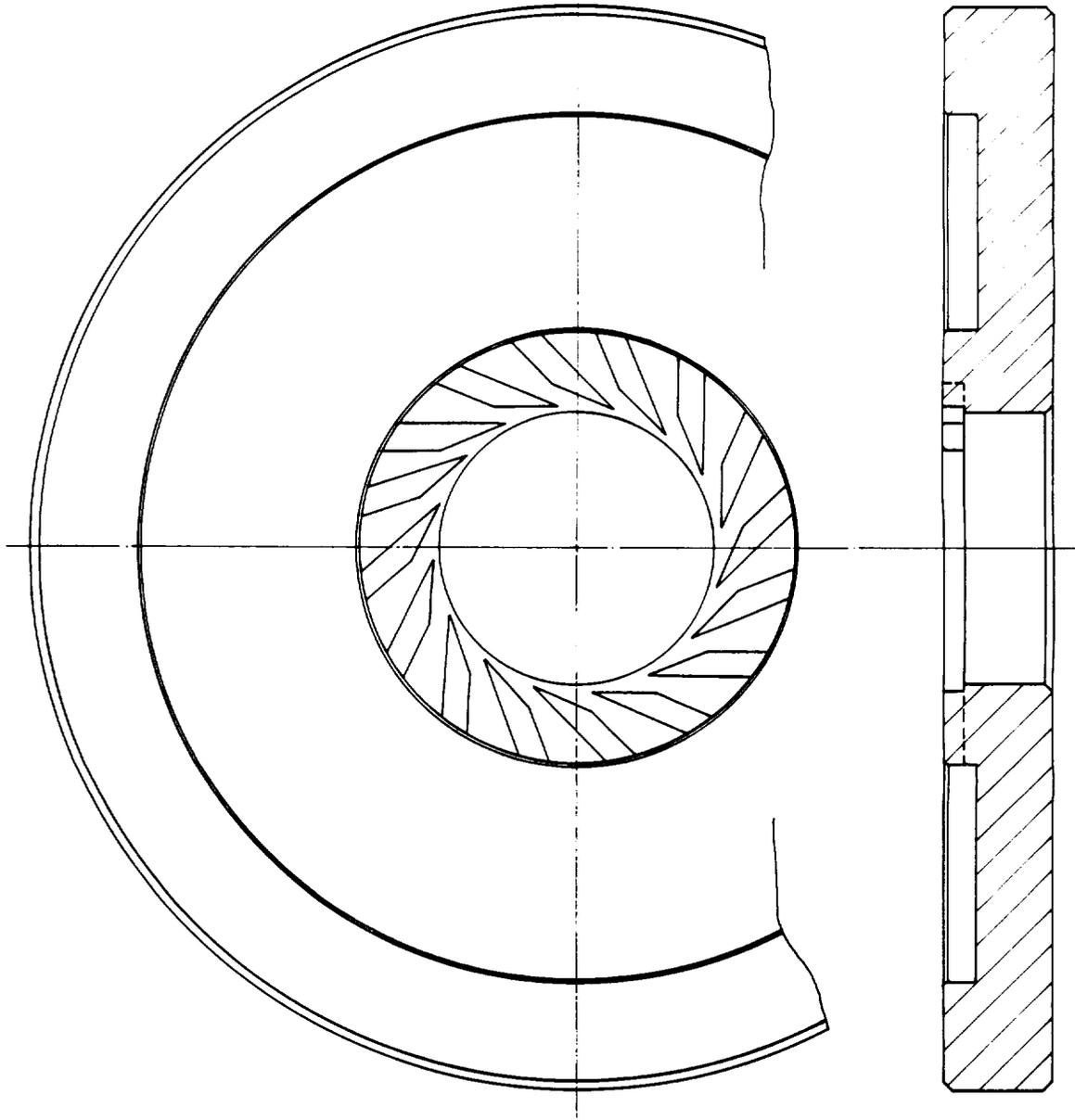
SHAFT

CAPACITANCE PROBE

LOWER BEARING

NOZZLE RING

TURBINE INLET

TURBINE OUTLET

Figure 4 LAYOUT OF NOZZLE RING