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A MINIATURE WET TURBOEXPANDER

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ABSTRACT

A miniature turboexpander capable of operating with the exhaust conditions down to sub critical temperatures and pressures has been developed. The expander shaft is supported in pressurized gas bearings and has a 4.76 mm turbine rotor at the cold end and a 12.7 mm brake compressor at the warm end. The expander has a design speed of 384,000 rpm and a design cooling capacity of 444 watts. A prototype machine has been built and tested in one of the satellite refrigerators at Fermi National Accelerator Laboratory. The gas bearings have demonstrated robust operation under severe system transients, including operation with the turbine exhaust in the two-phase flow regime. This paper describes the machine and the results of the testing.

INTRODUCTION

Distributed refrigerators for cooling superconducting magnets in particle accelerators incorporate one or more dry expanders to increase overall refrigeration capacity and one wet expander to produce liquid helium [1 2 3]. Existing systems, such as the Fermilab Tevatron[2] for example, use reciprocating expansion engines for both the dry and wet expanders. These machines have polymer seals and other sliding components which are subject to wear and call for frequent maintenance. To increase the system reliability, reciprocating wet engines could be replaced with wet turboexpanders in present or future helium refrigerators. In a gas-bearing turboexpander there is no rubbing friction and no wear and there should be no need for maintenance. The life should be of indefinite duration.

A gas bearing turboexpander has been designed to reliably withstand the sudden changes of pressure that can occur, such as those due to magnet quenches or failure of the compressor's power supply. The turboexpander has been designed to be robust, reliable and maintenance free.

DESIGN AND TESTING OF TURBOEXPANDER

Design specifications for the turbine are:

Specification of Wet Turboexpander

Working Fluid	helium
Inlet Pressure	19 x 10^5 Pascals
Inlet Temperature	6 K
Exhaust Pressure	2.2 x 10^5 Pascals
Exhaust Temperature	5.025 K
Mass Flow	50.0 g/s
Isentropic Efficiency	72.2 percent
Turbine Diameter	4.76 mm
Speed	6400 rps (384,00 rpm)
Pressure Ratio	8.6
Useful Refrigeration	444 watts

Design of Flow Passages

The design of the turbine rotor is similar to the one described in Reference 4. The main features of the design are shown in Figure 1. The design approach has been to reduce to a minimum the area of surface exposed to high velocity fluid. The diameter of the shaft is the same as that of the rotor. This feature offers significant advantages in fabrication, assembly and performance. It avoids the need for precise axial location of the shaft and permits the shaft to be fabricated from a single piece of metal.

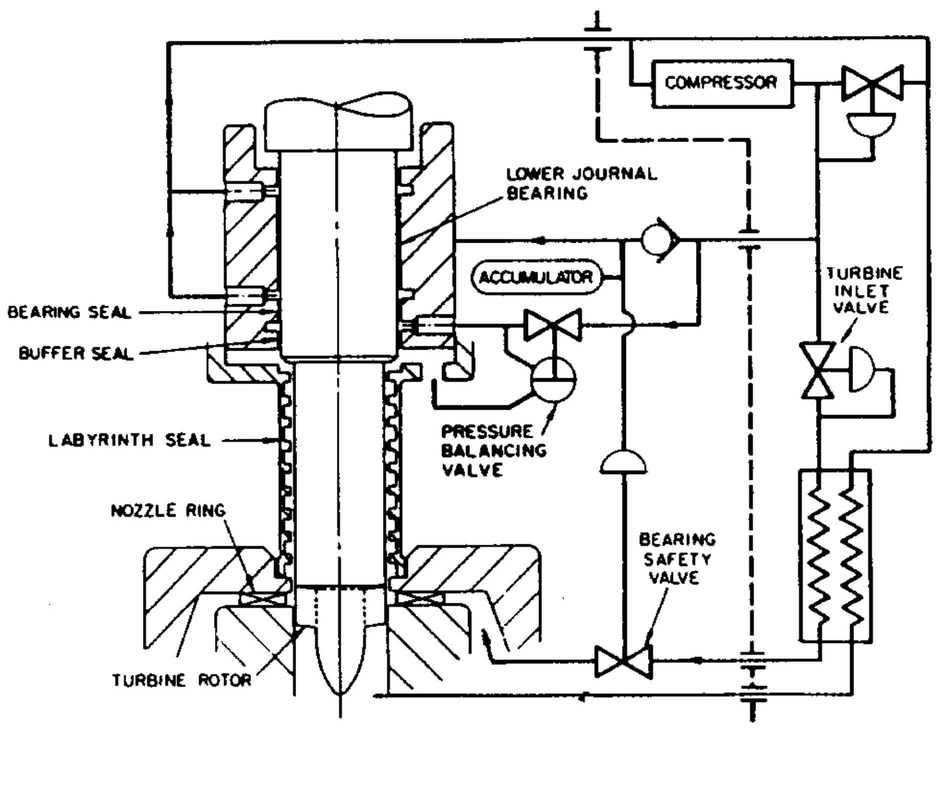

Fig. 1. Cold End of Turbine.

Turboexpander Assembly

An assembly drawing of the turboexpander is shown in Figure 2. The expander was designed to be flange-mounted on the top cover of a cold box with the shaft axis operating in a vertical position. Vertical operation is important to reduce convective heat leaks in the gas spaces separating the warm and cold ends of the machine. The inlet and exhaust connections to the expander are made by means of bolted flanges incorporating static seals[5]. By using flange connections, we eliminate the possibility of introducing contaminants (due to solder fluxes or weldment slag), and provide a quick and easy means of installing and removing the turboexpander. The expander is approximately 365 mm long, 150 mm in diameter, and weighs about 17 kg.

The shaft is made of 6Aℓ-4V titanium alloy. It consists of a 4.76 mm turbine rotor and a 12.7 mm brake compressor wheel integrally machined in it. The shaft is supported by two externally pressurized gas journal bearings and a diaphragm type pressurized thrust bearing. The shaft and bearings are assembled into a removable cartridge and secured into the top of the expander housing by a single nut. In the event of bearing failure, the cartridge can be replaced in a few minutes and operation resumed.

Because of the need to operate the gas bearings close to room temperatures, the expander is divided into upper and lower regions operating at widely different temperatures. The lower region incorporating the turbine rotor, nozzle ring and diffuser operates at cryogenic temperature, while the upper region containing the bearings and brake compressor operates at about 300 K. The large temperature difference causes heat to flow along the shaft and the thin-walled support tube connecting the warm and cold ends of the machine. This heat leak represents a direct loss in cooling capacity and must therefore be reduced by making the support tubes as long and thin as possible. The support tube still needs to be sufficiently rigid to maintain good alignment between turbine rotor and the shroud. The heat leak for the present design is estimated to be 11.5 watts at design point, representing 2.5 percentage points loss in the efficiency of the turbine.

The Labyrinth Seal

A major potential source of heat leak between the warm and cold ends is due to the flow of gas along the shaft. To minimize this leakage, the shaft extension from the lower bearing to the turbine rotor is surrounded by a labyrinth seal. It is essential that any flow of gas, either upward or downward, through this seal should be reduced to a minimum. An upward flow will carry refrigeration out and a downward flow will carry heat in. In either case, there will be a loss of efficiency. To minimize this loss a special buffer seal is provided as shown in Figure 1. The pressure difference across this buffer seal is used to actuate a flow control valve which in turn maintains this pressure differential at an acceptably low level. The pressure difference can be adjusted to within 100 Pascals so that the corresponding flow rate through the seal is negligible.

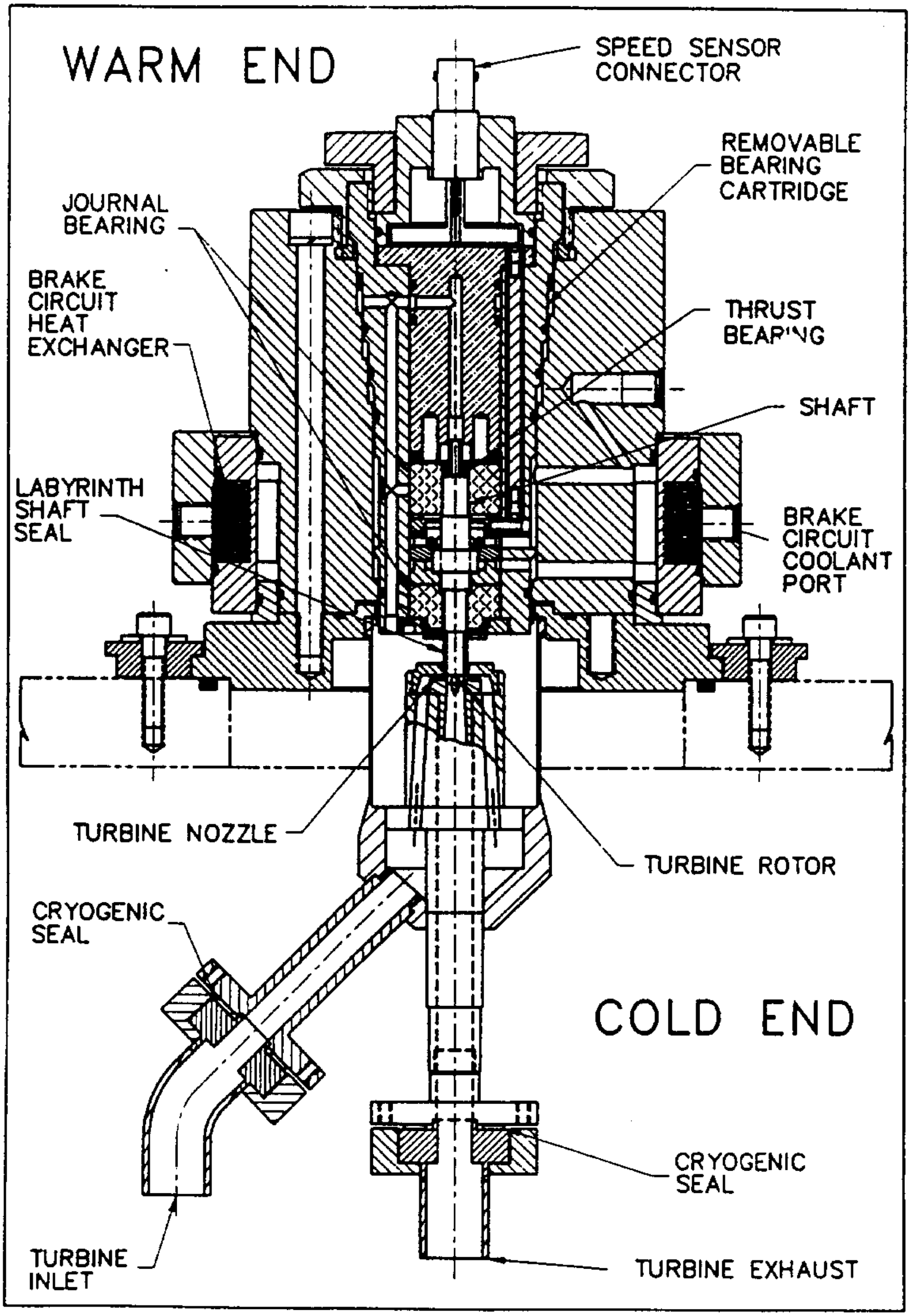

Fig. 2. Turbine Assembly.

Journal Bearings

The journal bearings are externally pressurized. They incorporate damping against shaft whirl. The theory of the bearings is is described in Reference 4. Shaft seals are located a short distance from the ends of each bearing as shown in Figures 1 and 3. The narrow grooves between the seals and the ends of the bearings are connected to the low pressure return line. These seals serve to isolate the journal bearings from the pressures in the labyrinth seal, the brake circuit and the thrust bearing. Sudden changes in any of these pressures can have no effect on the journal bearings.

The Thrust Bearing

The shaft is located axially by the pressure in a film of gas between the upper end of the shaft and a stationary pad which is supported on a flexible diaphragm. The diaphragm and support structure are provided with flow control resistances and volumes which provide damping against pneumatic hammer [6 7]. The axial load and the corresponding film pressure can be varied between 0.05 and 0.63 times the bearing supply pressure.

Because the thrust bearing is unidirectional, a minimum axial load is needed to raise the shaft towards the thrust pad, against gravity and the force due to the pressure at the exit orifice. To provide this load the diameter of the shaft seal above the brake rotor is greater than the diameter of the shaft seal below the brake rotor. The pressure in the brake circuit acting on the area of the annulus between the diameters of the two seals provides an upward thrust to hold the shaft against the thrust pad. To prevent this pressure from falling below the minimum value needed to support the shaft, a minimum pressure control valve should be provided as shown in Figure 3. A minimum of about 4.0×10^5 Pascals in the brake circuit is needed to support the shaft.

The total mass flow rate through the bearings and shaft seals is approximately 0.7 g/s, which puts an extra load on the main compressor of approximately 1.4%.

This thrust bearing has demonstrated exceptional robustness against sudden changes in load which can occur as a result of rapid pressure transients during start–up, shut–down, system upsets, magnet quenches and power failures.

Shaft Stability and Speed Sensor

To provide a shaft stability and speed signal, a small proximity probe is located between the brake rotor and the upper bearing. The clearance between the probe and the shaft is about 0.1 mm. An electronic circuit which is sensitive to small changes in capacitance provides a signal which is approximately proportional to small displacements of the shaft towards or away from the probe. The signal is displayed on the screen of an oscilloscope. Radial displacements from the position of equilibrium down to about 0.2 micron can be observed.

To provide a speed signal, a small flat on the shaft generates a sharp spike each time the flat passes the probe. An oscilloscope with a calibrated sweep provides a display of both shaft stability and shaft speed.

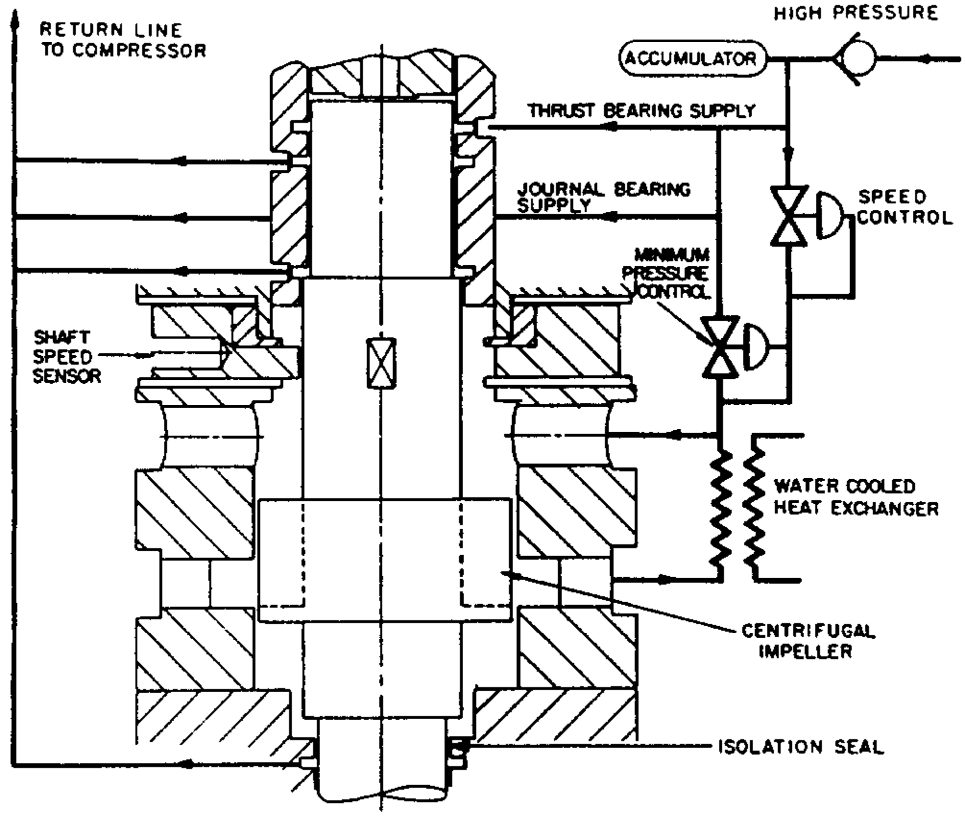

Fig. 3. Warm End of Turbine.

Shaft Speed Control

The power generated by the turbine is used to drive a centrifugal impeller which acts as a brake, as shown in Figure 3. The gas is circulated through a water cooled heat exchanger where the heat generated by the impeller is removed. The power needed to drive the impeller at a given speed is proportional to the density of the fluid and, when the fluid is a gas, the density is proportional to the pressure. The pressure in the brake circuit can be varied from a lower limit of about 4.0×10^5 Pascals to the full supply pressure from the compressor which was 20×10^5 Pascals. This corresponds to a density range of 5.0. The corresponding range in shaft speed for a given value of the torque generated would have a minimum to maximum ratio of about 2.2.

Bearing Safety Valve

The turbine must not be run without pressure in the bearings. To avoid any risk of this occurrence a turbine inlet valve is provided, as shown in Figure 1. The valve is forced shut by a spring and is opened by bearing gas pressure on a diaphragm acting against the spring. The spring can be adjusted so that the turbine cannot be run unless there is adequate pressure on the bearings. The spring acts promptly and it never forgets.

Transient Tests at Creare and at Fermilab

In order to determine the behavior of the thrust bearing and journal bearing system under rapid pressure transients, a transient test was performed by (nearly) instantaneously changing the valve settings that govern the turbine inlet and exhaust pressures. The test simulates a magnet quench in the ring at Fermilab by rapidly raising the load on the thrust bearing to the level that would occur during a quench. Turbine exit pressure went from 1.05×10^5 to 5.5×10^5 Pascals in 5 seconds. As expected, no visible disturbance of the shaft was observed. The time constant of the pressure-compensating mechanism in the thrust bearing is of the order of a few milliseconds. For an upset to disturb the thrust bearing, it would have to be either outside the load capability of the bearing or have a rise time significantly faster than a few milliseconds.

During the initial testing of the expander at Fermilab the bearing supply was turned off by accident with turbine running at 2000 r/s. When the supply was turned on again the start-up transient went from 0 to 2000r/s in about 1.5 seconds. Later the turbine was turned off abruptly and allowed to coast down. Then it was turned back on abruptly. The pressure went from 2×10^5 Pascals to 19×10^5 Pascals in about one second. The turbine accelerated from 0 to 5000 r/s in less than 2 seconds.

Two exhaust pressure transients were observed repeatedly during testing. One was a 1.4×10^5 Pascals to 3.1×10^5 Pascals pulsation with a 5-6 second period. The other was a $+/- 0.34 \times 10^5$ to $+/- 0.68 \times 10^5$ Pascals pulsation with a 1/3 second period. No disturbance of the turbine was observed.

Thermodynamic Performance Tests

Thermodynamic performance testing of the wet expander was performed at Fermilab in their "ER" refrigerator (one of the satellite refrigerators specifically set up for component testing). The inlet and exhaust temperatures were measured with carbon-glass resistors. The inlet flow was throttled with a cryogenic valve which allowed the operator to control the turbine flow. The wet expander was placed into Fermilab's cryogenic loop via U-tube transfer lines in a manner similar to the normal connection of their wet engines. Control of the wet expander was achieved through both Fermilab's control panel and a control panel supplied by Creare. Tests were performed by controlling the inlet temperature and pressure to the cryostat with Fermilab's control system, adjusting the mass flow with the throttling valve, and controlling the turbine speed by adjusting the brake circuit pressure in the local control panel. The machine was installed and operated down to 5.1 K exhaust temperature where it produced subcooled liquid. The machine was also operated with the exhaust in the two-phase region for several hours. No upset of the turbine was observed.

The mass flow rate through the turbine was much less than expected, being slightly less than half the design value. Measured efficiencies were also much less than expected, being about 50% isentropic. Inspection of the machine after testing showed that the turbine nozzles were partially clogged by debris released from the piping between the filter and the expander. This would account for the low mass flow rate and the lower than expected efficiency. Further testing with clear nozzles is needed to determine the isentropic efficiency.

CONCLUSIONS

A miniature cryogenic turboexpander capable of operating with liquid or two-phase helium at the discharge has been developed. The turboexpander incorporates an extremely robust pressurized bearing system which is immune to rapid pressure upsets at the inlet or outlet of the expander. The turboexpander can safely withstand radial loads resulting from operation with a wet discharge, as well as pressure upsets associated with magnet quenches or other system transients. Two-phase capability and bearing system robustness make this turboexpander ideally suited for replacing J-T or wet engine stages in laboratory sized helium liquefiers. A liquefier incorporating a wet turboexpander would have higher efficiency than one with a J-T stage and would require less maintenance than one with a wet engine.

ACKNOWLEDGEMENTS

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