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Elimination of Thermal Acoustic Induced Heat Leak in a Centrifugal Circulator for Supercritical Helium

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Abstract

Thermal acoustic induced heat leak on the order of 150 watts magnitude has been identified in circulating compressors installed in the SSC Subcooler and MAGCOOL cold box located at BNL. This heat leak was eliminated by increasing the "warm end" volume of the circulator by 0.7 liter. As a result, adiabatic efficiencies greater than 50% were achieved for these circulating compressors, and the SSC subcooler consequently will have excess capacity for cooling large number of magnets in series. This report documents the diagnostic process involved and the performance of the circulating compressors. The implication of higher cooling capacity of the cryogenic system due to elimination of circulator/compressor heat leak will be given in a subsequent technical note.

I. Introduction

BNL purchased two helium compressors from Creare Inc. for circulating supercritical helium for both SSC and RHIC magnet test programs. These gas bearing centrifugal compressors, often called cold circulating compressors or circulators, are operated at liquid helium temperatures and their performance is crucial to the entire cryogenic system.

The first circulating compressor was installed on the SSC Subcooler which was commissioned in June, 1989. Since then, the circulating compressor has been used for testing SSC dipoles. The machine is well built. It can be started and stopped easily at operating temperatures from room temperature down to liquid helium temperature. It is rugged and can take large pressure transients from a magnet quench. Both power supply and speed controller are well designed and packaged. It has proved to be a dependable machine to circulate supercritical helium through superconducting magnets.

While mechanically reliable, the compressor showed poor thermal performance. There were indications that an unusually large "heat leak" between the room temperature and the cryogenic ends of the machine was occurring. In September 1990, BNL began an extensive effort to investigate the heat leak problem associated with the unit and to examine solutions for a satisfactory correction.

II. System Description

The compressor is mounted on the cold box as shown in Figure 1. The warm end consists of a motor drive and bearing assembly and is external to the cold box. The compressor cold end consists of the shaft, impeller, diffuser, and inlet and outlet piping, and is located inside the cold box. There is a closed cycle water/glycol cooling system in the warm end of the unit for cooling the motor and rejecting waste heat to the air. The cold end piping is connected to the circulating loop of supercritical helium system as shown in Figure 2.

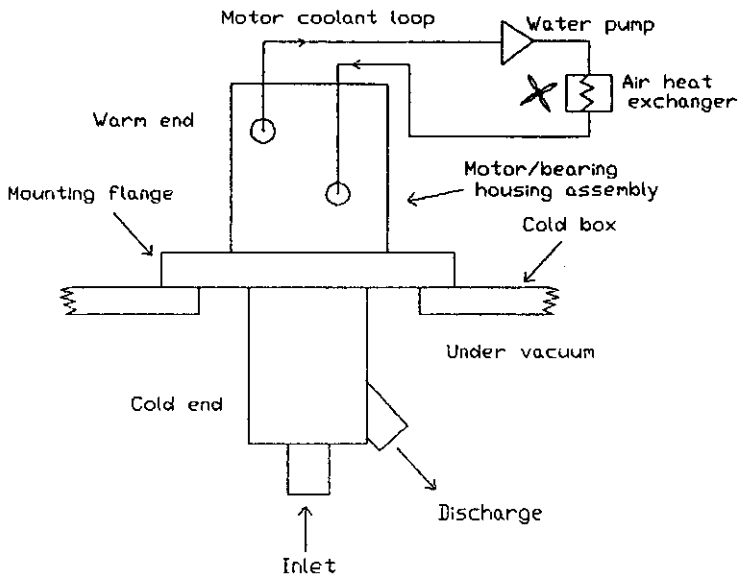

Fig. 1 Cold circulating compressor mounted on the cold box

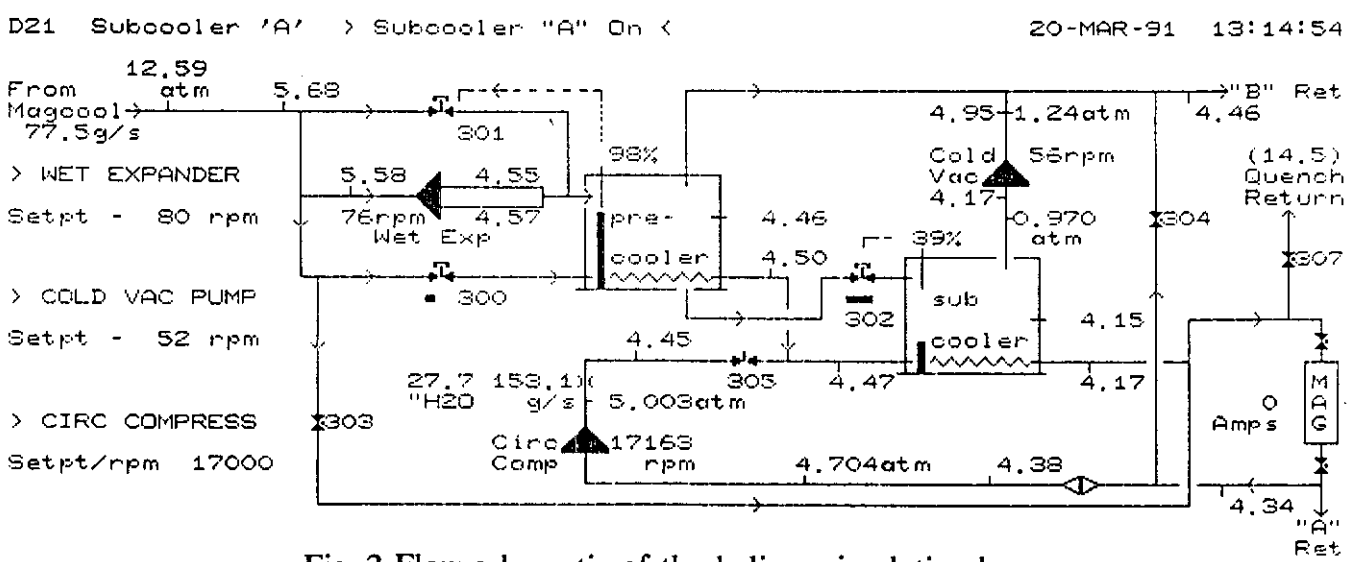

Fig. 2 Flow schematic of the helium circulating loop
(Temperatures are shown without units and are in Kelvin)

A pressure transducer and a temperature sensor are located on the upstream and downstream side of the circulator as shown in Fig. 2. A venturi flow meter is located downstream of the circulator. A large temperature rise across the circulator was the first indication of low compressor efficiency. Near the operating condition, the total enthalpy rise across the circulator was found to be around 250 watts versus the 75 watts nominal design value. Analyses suggested that a high heat leak existed in the compressor.

III. Heat Leak Measurement

Normally, a heat leak on the order of 150 watts in a cryogenic system could be detected on the warm surface of the cold box as a cold spot. However, the warm end housing of the compressor was not unusually cold. In October 1990, with the second circulating compressor installed and tested for RHIC magnets, we found that the majority of the heat leak was contributed by the coolant system of the motor. The water/glycol cooling system was designed to remove heat from the motor, however observation confirmed that the water/glycol loop was being cooled by the circulator due to the heat leak from its warm to its cold end.

Heat input from the water/glycol system and enthalpy rise of helium through the circulating compressor over a period of five days of investigation in October 1990 are given in Figure 3. The results show that heat loss from the water/glycol system is approximately 70% that of the enthalpy rise in the helium passing through the circulating compressor.

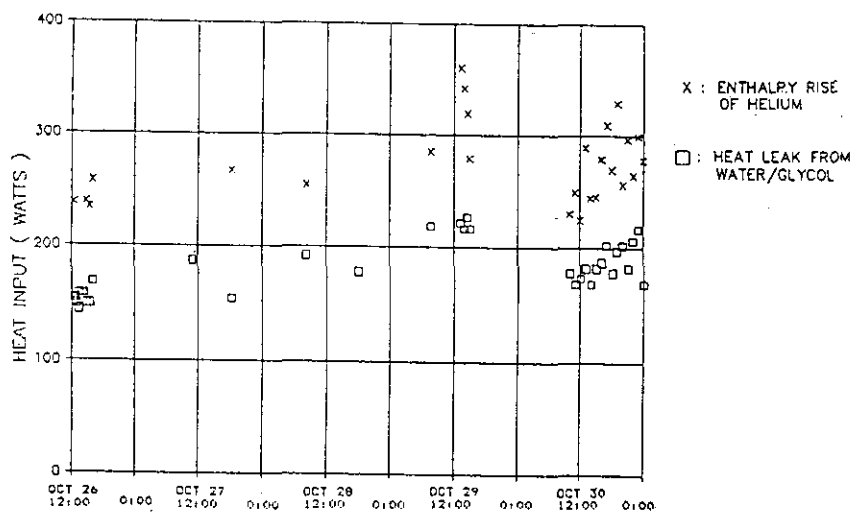

Fig. 3 Heat input from water/glycol versus enthalpy rise of helium

IV. Strategy to Eliminate Heat Leak

When the heat leak problem became apparent, BNL and Creare worked together to examine potential solutions. It was our goal that solutions be simple and could be integrated into the framework of the existing machine. We also hoped that corrections could be made without delaying the magnet testing program.

We experimentally found the heat leak to be speed dependent and therefore not due to conduction or radiation. It had be convective or thermal acoustic related. Since the circulating compressor employs a clearance shaft seal rather than a labyrinth-type seal commonly used on similar machines, one solution under consideration was the addition of a labyrinth seal. This was believed to be a solution if the source of the problem was due to either of the above causes. It was later concluded that the required high precision machining would have created scheduling problems.

So it was decided instead to implement the Creare proposal to increase the effective volume of the "warm end" of the machine to change the resonant frequency of thermal acoustic oscillations that might be occurring.

V. Thermal Acoustic Oscillation

The warm end volume of the circulating compressor was enlarged by connecting it to an external variable volume cylinder through a penetration on the cover plate and 3/8" copper tube. Two isolation valves and a pressure transducer were installed between them.

Tests were performed when the cold end of the circulator reached the design condition of 4 K and 5 atm. When the warm end was isolated from the external cylinder, a pressure. oscillation of nearly 2 psi at a frequency of 68 Hz was observed on an oscilloscope. With the warm end connected to a cylinder volume of about 0.45 liter, the oscillation completely disappeared. The heat leak from the water/glycol into the compressor disappeared. At the same time, the temperature rise across the circulator became small. The reduction of the thermal acoustic induced heat load also resulted in a substantial reduction in the speed of the cold vacuum pump since the speed of the cold vacuum pump is in direct proportion to the heat load of the system, as shown in Fig. 2. The system response is fast. In a matter of minutes, the cryogenic system was settling to a new steady condition.

Further increase of the warm end volume has no effect on the circulator performance. Since the critical volume slightly depends on the speed of the circulator, a 0.7 liter volume was chosen as a permanent add on volume to the compressor.

Surprisingly enough, we also found the fluctuation of the shaft speed read out from the capacitance probe, located on the journal bearing, is also caused by the thermal acoustic oscillation. When thermal oscillation ceases, the speed read out becomes stable too. This is due to the axial oscillation of the shaft caused by the thermal acoustic effects.

VI. Compressor Performance

With the success of eliminating undesirable heat leak, data was taken for performance evaluation. The following results were obtained when the circulating compressor was operated near 5 atm and 4 K with an unpowered SSC magnet installed in the helium cooling loop. The loop resistance was adjusted by a manual valve located at the suction side of the circulator. A differential pressure gauge, with a full scale range of 300 inch water, was installed across the circulator for head measurement. A venturi flow meter, corrected to helium density according to measured pressure and temperature, was used for flow rate measurements. The compressor performance was evaluated both at different speeds with constant loop resistance and at different loop resistances with constant speed.

With the manual valve fully open for the least loop resistance under the existing configuration, the compressor speed was set at 14000, 17000 and 21000 rpm. The manual valve was then throttled to a design loop resistance, and the compressor was set at speeds of 14000, 15000, 17000, 18500, 21000 and 22000 rpm. In this test, 15000, 18500 and 22000 rpm would produced the design head of 0.23, 0.36 and 0.52 atm at design flow rates of 100, 125 and 150 g/s.

A second test was performed at different loop resistances with the compressor speed set at 21000 rpm. Seven loop resistances were selected between the minimum and maximum allowable loop resistance. Beyond the maximum allowable loop resistance, surge phenomena were observed.

The head, defined for incompressible conditions as pressure rise over density, as a function of flow rate is given in Fig. 4. Indicated are the compressor speed and adiabatic efficiency, which is the ratio of ideal enthalpy rise over actual enthalpy rise. The adiabatic efficiencies ran between 31 and 56 % in proportion to the head rise. The measured efficiencies at three design conditions of 100, 125 and 150 g/s are 36 %, 48 % and 56 % which agree well with the 37 %, 47 % and 53 % predicted by Creare even without all the measurement errors taken into account. Preliminary analyses suggest that the error associated with the measurements is small. At a typical magnet testing condition, the efficiency is 46 %.

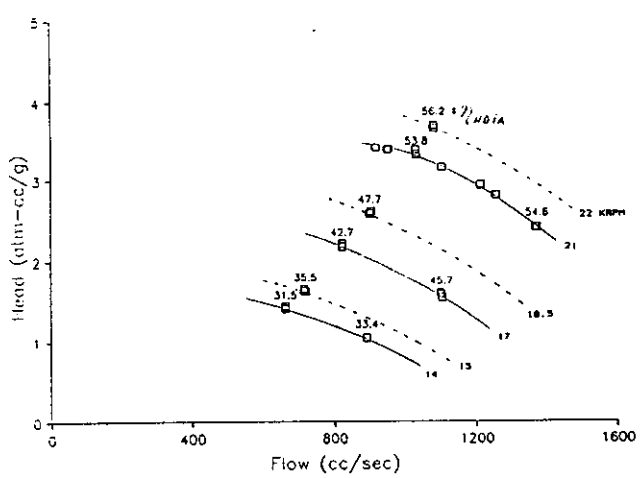

Fig. 4 Circulator performance curve

The aerodynamic efficiency of a compressor is another important parameter. It is the efficiency of the compressor minus heat leak and can be obtained by subtracting the heat leak from the present data. The heat leak of the circulator is estimated to be 20 watts. The aerodynamic efficiency of the compressor at 21000 rpm with seven sets of loop resistance are given in Fig. 5. The peak aerodynamic efficiency is about 72 % at 1300 cc/s flow rate.

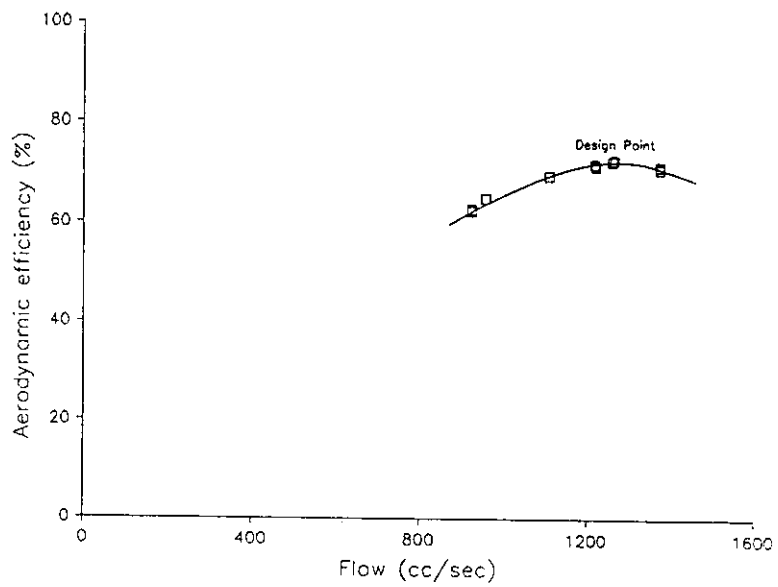

Fig. 5 Aerodynamic efficiency versus flow rate

The head coefficient and flow coefficient are parameters conventionally used for the description of turbomachinery. They are defined here as

$$\psi = \Delta P / \rho U^2$$

$$\varphi = \dot{m} / \rho U A$$

where
ψ is the head coefficient
φ is the flow coefficient
ΔP is the pressure rise across the compressor
$\dot{m}$ is the mass flow rate
ρ is the fluid density (assumed constant)
U is the tip velocity of the impeller
A is the exit area of the impeller

The head and flow coefficient curve for the present circulating compressor are given in Fig. 6. As can be seen, the data for different speeds collapses to a single curve when expressed in this manner.

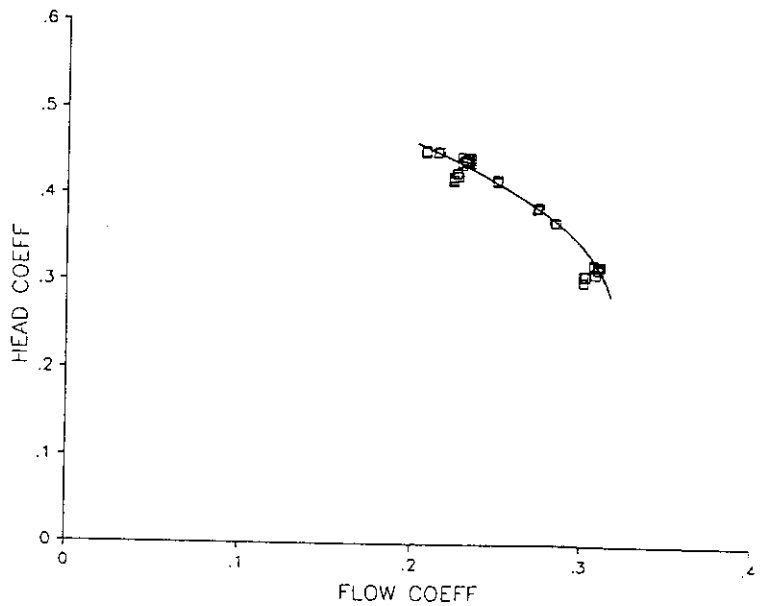

Fig. 6 Head coefficient versus flow coefficient for the circulator

VII. Conclusion

With the warm end volume increased by 0.7 liter, the thermal acoustic oscillations have been eliminated and the compressor performs reliably with high thermodynamic efficiency. The circulating compressor unit meets the expectations for circulating supercritical helium for magnet testing programs.