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Performance of the MAGCOOL Subcooler Assembly for SSC Single Magnet Test Program

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PERFORMANCE OF THE MAGCOOL SUBCOOLER ASSEMBLY FOR SSC SINGLE MAGNET TEST PROGRAM

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ABSTRACT

Thermal acoustic induced heat leak on the order of 150 watts has been eliminated from the circulating compressor installed in the SSC Subcooler assembly located at BNL. As a result, the SSC Subcooler has a greater capacity than previously reported in Technical Note SSC-N-654. The Subcooler can now provide more cooling for the magnets or operate at a lower magnet temperature and it is also faster for the system to subcool a magnet and to recover from magnet quench. The lowest temperatures achieved are 3.0 K at the suction of the cold vacuum pump and 3.2 K at the return of the magnet. These are 0.3 K lower than previously obtained. Since July 1989, the SSC Subcooler has been in service for testing SSC dipoles with great success. This technical note summarizes two years of operating experience and reports the performance of the Subcooler assembly including cold compressors.

INTRODUCTION

The subcooler assembly for the SSC magnet test program incorporates the following three features: 1) a circulating compressor for closed loop circulation of 100 to 150 g/s single phase helium at 5 atm nominal pressure to simulate flow conditions in SSC magnets, 2) a cold vacuum pump for adjustment of operating temperatures between 4.35 K and 3.5 K, and 3) a wet expander to balance the heat caused from compression work of the vacuum pump.

The assembly unit is the first of its kind ever built. BNL is responsible for process design, equipment selection, system integration, and instrumentation and control. Cryogenic Consultants Inc., CCI, is the general contractor of the subcooler and is responsible for the design, construction and heat leak performance of the cold box. CCI is also the supplier of the cold vacuum pump. Construction began in October 1988, finished in April 1989 and the unit became operational in July 1989.

Today, with more than 5000 hours of operating time, the subcooler has proved to be a reliable unit with individual components meeting design expectations. The overall system performs well in both steady state operation and during magnet quench, subcooling, cooldown and warmup.

SYSTEM DESCRIPTION

The flow schematic of the SSC single magnet test system is shown in Fig. 1. It consists of the subcooler assembly, the feed can, the SSC magnet and the return can. The feed can and the return can provide interfaces for cryogenic piping and room temperature penetrations for magnetic field measurements. Both the magnet and a 4.5 K line inside the magnet cryostat are connected to the cans. Thus helium from the feed can flows to the return can and back through the magnet. The volume for this loop of piping is about 220 liters.

Located on the subcooler assembly, the circulating compressor is used for closed loop circulation of single phase helium. Cooling is delivered from the subcooler helium pot to the magnet. Helium temperature is regulated by the saturation pressure inside the helium pot which is connected to the suction of the cold vacuum pump. The arrangement of the piping and instrumentation for the circulating compressor, the cold vacuum pump, the wet expander, and the precooler and subcooler helium pots can be seen in Fig. 1. For the purpose of computer control, the flow diagram is reconfigured on the computer console as shown in Fig. 2. The liquid volume in the helium pots are approximately 100 liter each. The SSC subcooler assembly is connected to the cold end of the 1500 watt HEUB refrigerator through MAGCOOL distribution headers.

The assembly is operated by the CRISP process control computer. Real time pressure, temperature, flow and speed readings are directly available on the computer console. The controllers for wet expander, cold vacuum pump, circulating compressor, and control valves are also shown in Fig. 2. Control valve AOV302 is used to maintain constant liquid helium level in the subcooler pot. The pressure in the circulation loop is controlled by the makeup valve AOV300. The cold vacuum pump could be operated either in manual mode at constant speed or automatic mode to maintain the magnet at a specified temperature. The wet expander could also be operated in manual mode or automatic mode to maintain constant liquid level in the precooler pot.

By using the circulating compressor in a closed loop, flow rate, pressures and temperatures can be controlled independent of the refrigerator. The system is easy to operate. During the magnet quench, a large amount of heat could be absorbed in the loop. Meanwhile the circulating compressor continuously delivers cooling from the precooler and subcooler pots into the loop. Helium will be vented through valve 307 out to warm recovery only if the loop pressure exceeds the set pressure. Because only a small portion of helium in the loop is vented, the quench recovery is more efficient than other cooling schemes.

In all tests performed, the subcooler provides cooling for the dipole magnet, feed can, return can, transfer lines and about 1 g/s of helium for power leads. A liquid nitrogen shield is used in the system, but the 20 K shield is not connected.

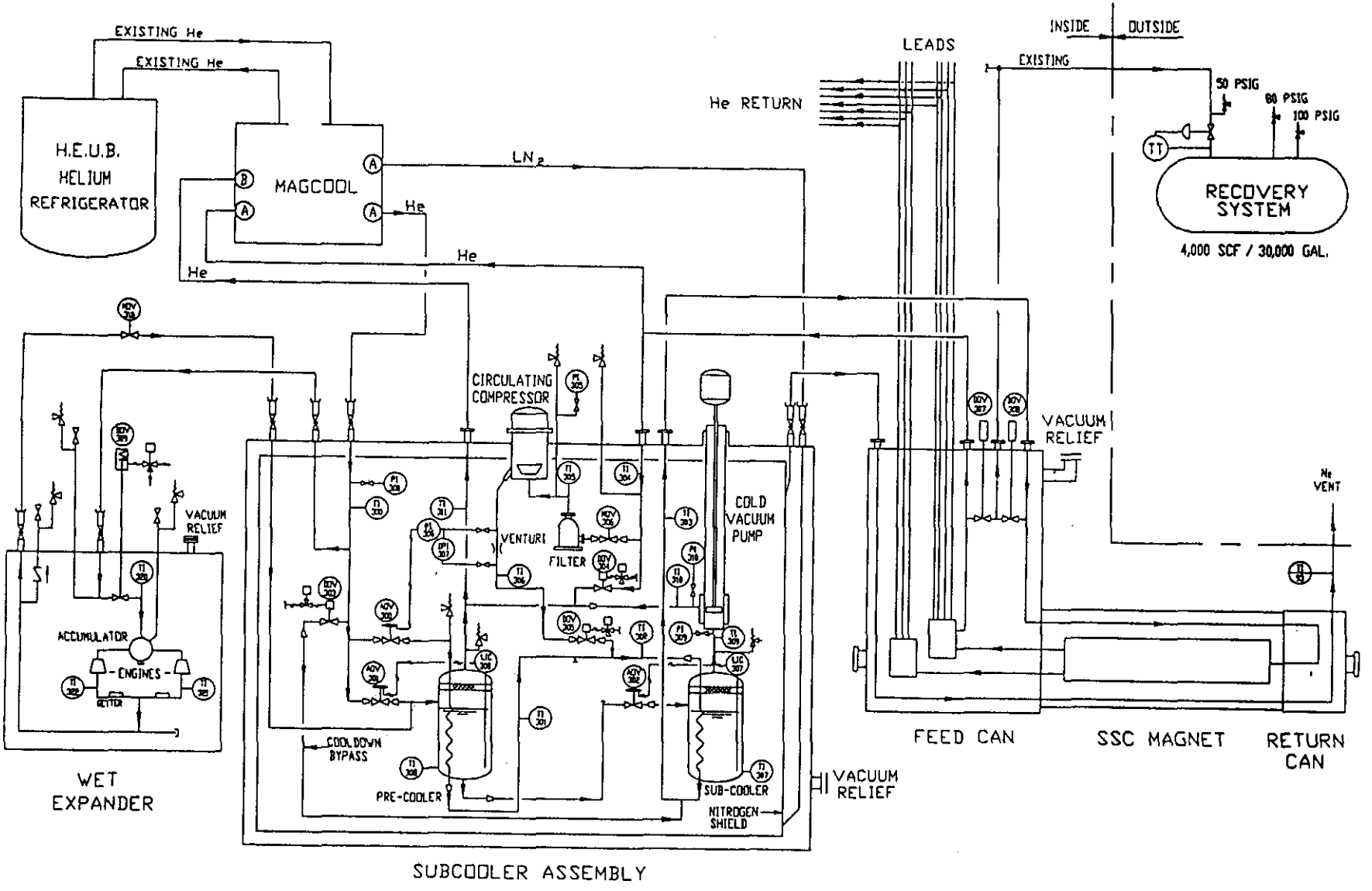

Fig. 1. Flow schematic for SSC single magnet test program

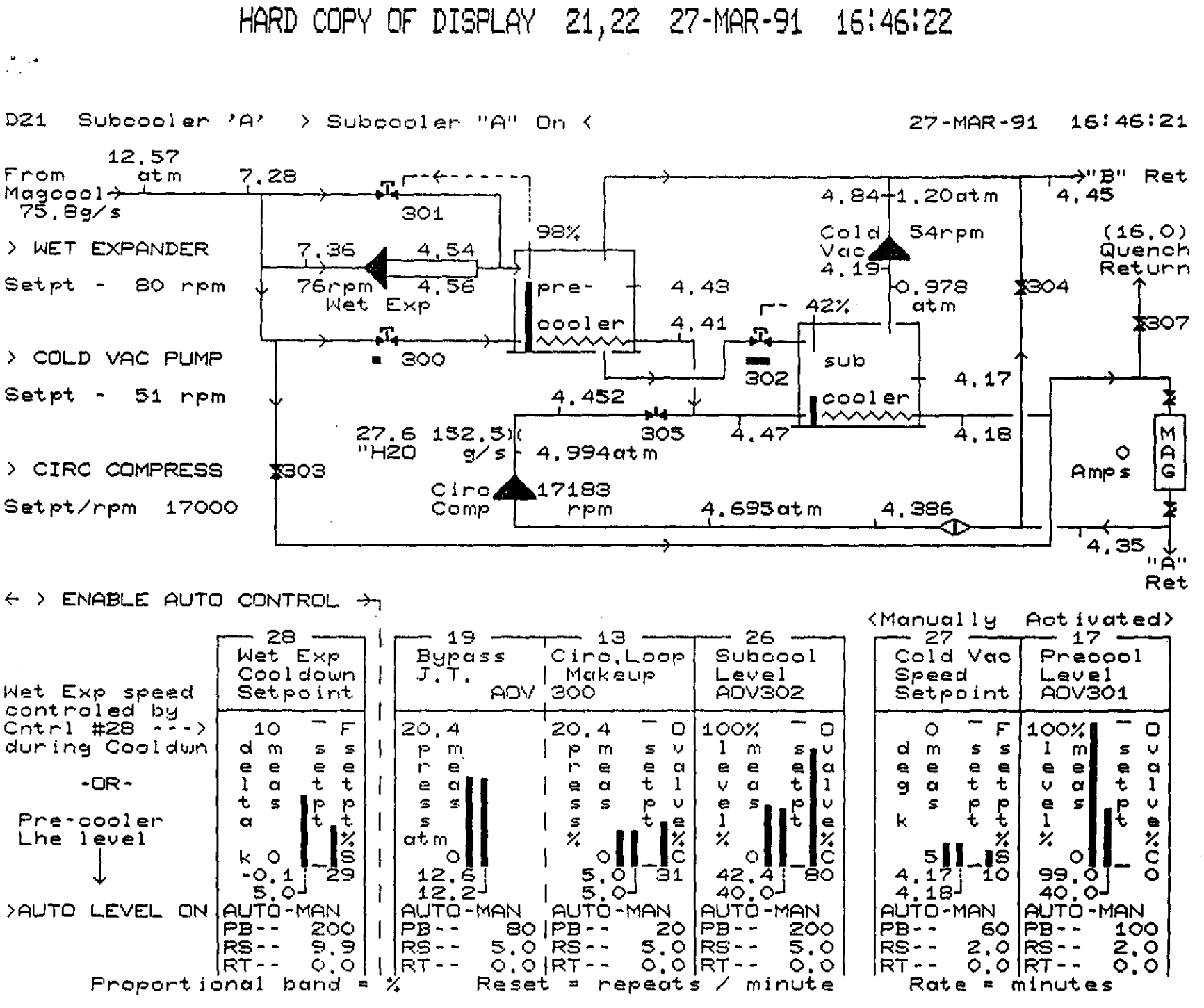

Fig. 2. Flow schematic and process controllers for SSC subcooler

EQUIPMENT DESCRIPTION

The cold box is 54 inch O.D. and 84 inches high and nitrogen shielded. Cold compressors, valves and bayonets are located on the top of the cold box. The wet expander is housed in a separate cold box. The circulating compressor, the cold vacuum pump and the wet expander are the three major items of equipment in the subcooler assembly.

The circulating compressor was supplied by Creare Inc., Hanover, New Hampshire. It is a high speed electric motor driven centrifugal pump specifically designed for circulating large flows of single phase helium with a small head rise. The shaft is supported by a pair of self-acting tilting pad radial gas bearings and a permanent magnet axial thrust bearing. A thrust bearing override bumper/damper is incorporated to prevent bearing system damage during severe pressure transients. The circulator is driven by a three phase variable frequency AC motor and has proved to be reliable.

The cold vacuum pump is built by Cryogenic Consultants Inc. It is a reciprocating pump with a close fitting labyrinth seal between the piston and the cylinder. Both intake and exhaust valves are check valves so the pump does not need cams. The unit has one large cylinder with 4.375 inch bore, 3 inch stroke and 45 cubic inch displacement. Driven by a 1-1/2 hp DC motor, the pump can be operated anywhere between 50 and 380 rpm. Below 50 rpm, the pump is difficult to operate and above 380 rpm, vibration becomes noticeable. This wide speed range provides a wide flow and pressure operating range which is especially important for the magnet testing program. Suction pressures as low as 0.25 atm and suction temperature as low as 3.0 K have been achieved. The use of a non-contact seal design has resulted in a unit that is highly reliable and can accept two phase helium at the suction.

The wet expander is a Koch Process System Model 1600 wet reciprocating engine. It has two engines operating 180 degrees out of phase. Each engine has a 3-inch bore and 2-inch stroke. An engine consists of a piston, cylinder, crosshead and inlet and exhaust valves. It has a single flywheel with two cams on each side to operate the intake and exhaust valves. An electrical alternator is used to absorb energy removed from the gas and to control the speed of the engines. This type of wet expander has proven to be highly reliable and is widely used in helium systems. Our experience agrees with that of other users and we did not evaluate its thermal performance.

PERFORMANCE OF SUBCOOLER ASSEMBLY

The subcooler assembly is physically located next to the feed can for optimal thermal performance. With one SSC dipole, the pressure drop is about 60% of the design head rise of the circulating compressor. The maximum single phase helium which can be circulated through the loop is more than twice the 100 g/s base design value for SSC dipole magnets. The lowest temperatures achieved are 3.0 K at the suction of the cold vacuum pump and 3.2 K at the return of the magnet. The subcooler assembly has ample capacity. It takes 20 to 30 minutes to subcool a dipole from 4.3 K to 3.5 K. During a magnet quench, we found valve 307 open only for a few seconds to vent helium into warm recovery after the loop pressure exceeds 15 atm. The system takes about one hour to recover. At 3.5 K, the system can carry a SSC dipole with an excess capacity of 150 watts for cooling additional magnets.

A typical operating condition for the magnet is given in Fig. 3. The circulating loop is maintained at about 5 atm and 4.2 K. The speed of the circulator is 17,000 rpm for a flow of 152 g/s helium and a pressure rise of 0.22 atm. The pressure and temperature inside the subcooler helium pot are 0.98 atm and 4.16 K. The speed of the cold vacuum pump is 55 rpm. The wet expander is operated at 75 rpm, near its lowest operable speed.

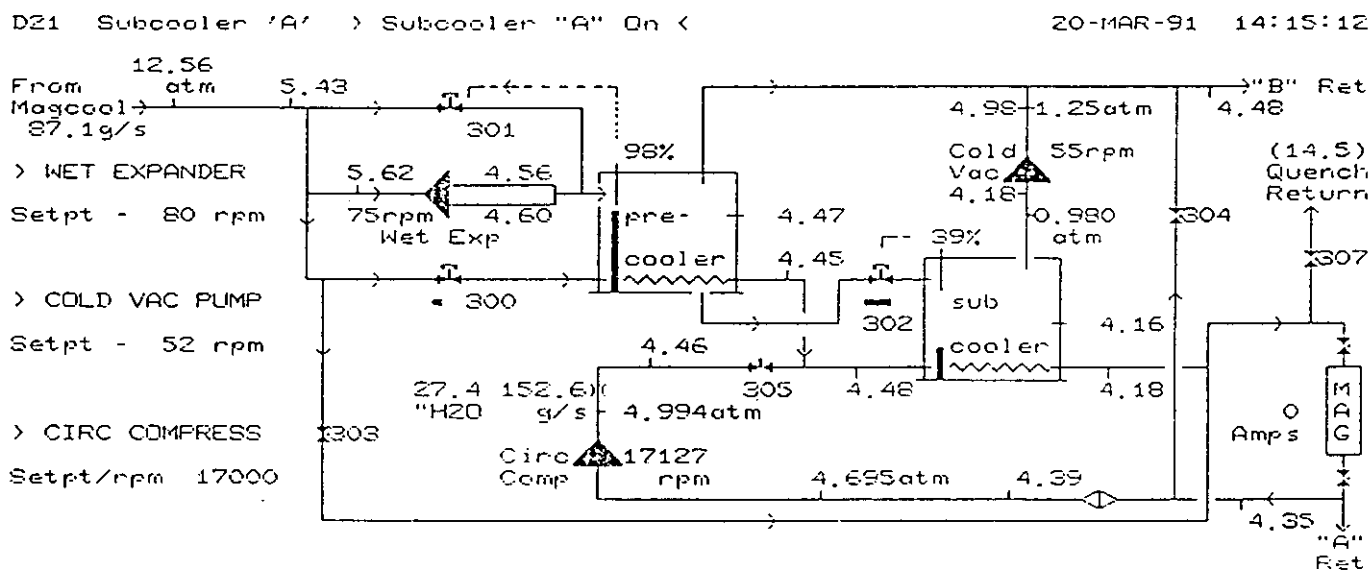

Fig. 3. Typical Operating Condition for Magnet at 4.3 K

Fig.4 shows the condition for the lowest temperature. The circulating loop is maintained at about 5 atm and 3.2 K. The speed of the circulator remains at 17,000 rpm. The flow rate goes up to 164 g/s due to higher helium density at lower temperature. The pressure and temperature inside the subcooler helium pot are 0.243 atm and 2.98 K. The speed of the cold vacuum pump is 370 rpm. The wet expander is operated at 78 rpm.

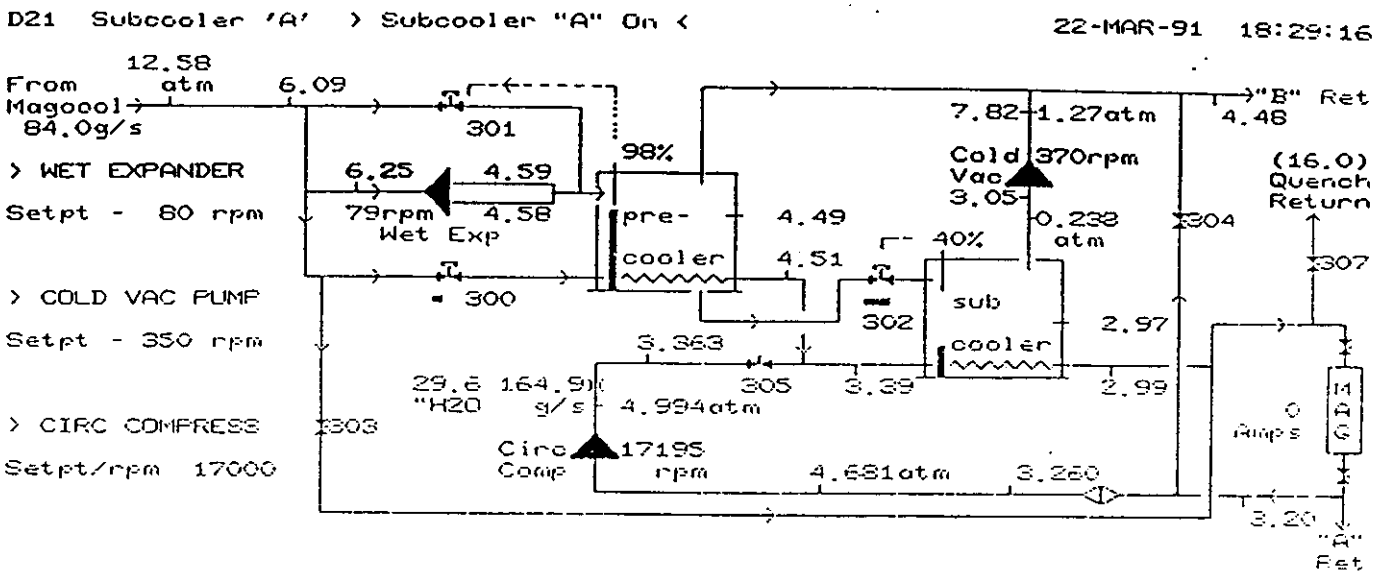

Fig. 4. The Lowest Temperature Performance of the Subcooler Assembly

Nominal operating speeds for the circulating compressor is between 14,000 and 21,000 rpm. For the cold vacuum pump, 50 rpm for 4.3 K, 175 rpm for 3.5 K, 320 rpm for 3.22 K, and 380 rpm for 3.2 K operation. The wet expander is operated at its lowest speed limit due to the fact the existing expander is slightly larger than the process required. But during cooldown and quench recovery, the wet expander could run as fast as 150 rpm and the cold vacuum pump was run at 230 rpm.

PERFORMANCE OF CIRCULATING COMPRESSOR

The circulating compressor showed poor thermal performance for about a year after the subcooler assembly began operation. There were indications of an unusually large "heat leak" between the room temperature and the cryogenic ends of the machine. Through extensive investigation, thermal acoustic induced heat leak, on the order of 150 watts magnitude, was identified and was completely eliminated by increasing the "warm end" volume of the circulator by 0.7 liter.

The test results reported in this section were taken when the circulating compressor reached about 4 K and 5 atm. The loop resistance was adjusted by a manual valve located at the suction side of the circulator. A differential pressure gage and a flow meter were used for head and flow rate measurements. Temperature and pressure sensors, installed at the suction and discharge of the compressor, were used for evaluation of the adiabatic and aerodynamic efficiency.

The design conditions for the circulating compressor are 100, 125 and 150 g/s flow of helium at head rises of 0.23, 0.36 and 0.52 atm. The compressor performance was evaluated first with the manual valve fully open for the condition of least loop resistance. Then the valve was throttled until the design loop resistance was obtained. The head versus capacity curves at several speeds are shown in Fig. 5. with corresponding adiabatic efficiencies given for each point. The measured efficiency at the three design conditions of 100, 125 and 150 g/s are 36%, 47% and 53%. General speaking, efficiencies are higher at higher head since the heat leak is essentially constant.

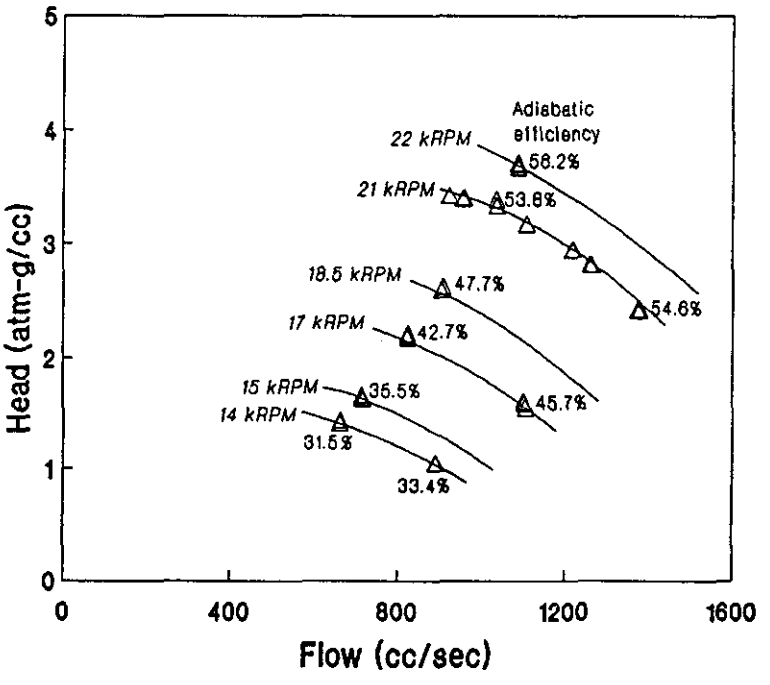

Fig. 5. Circulator Performance Curve

A second test was performed at 21,000 rpm for six resistances from the minimum to the maximum allowable. Beyond the maximum allowable loop resistance, surge phenomena were observed. The adiabatic efficiency and the aerodynamic efficiency, which is the compressor efficiency without the effect of heat leak, are given in Fig. 6. The heat leak is assumed to be 20 watts. The peak aerodynamic efficiency is about 72% at 1300 cc/s.

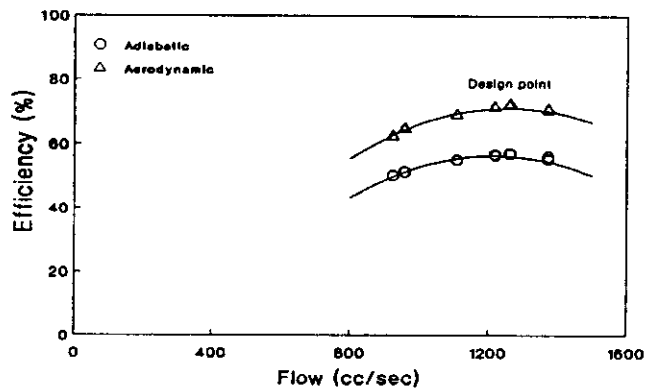

Fig. 6. Circulator Efficiency versus Flow Rate

The curve for head coefficient ψ and flow coefficient φ, defined in equations 1 and 2, are given in Fig. 7.

$$\psi = \Delta P / \rho U^2 \qquad (1)$$
$$\varphi = \dot{m} / \rho U A \qquad (2)$$

where

ΔP is the pressure rise across the compressor
$\dot{m}$ is the mass flow rate
ρ is the fluid density
U is the tip velocity of the impeller
A is the exit area of the impeller

As can be seen, the data for different speeds collapses to a single curve when expressed in this manner.

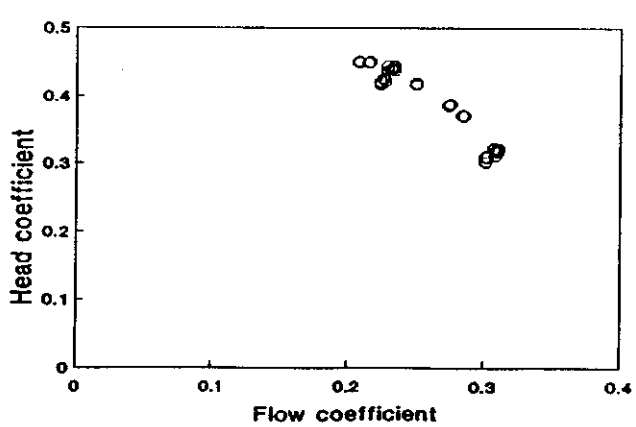

Fig. 7. Head Coefficient versus Flow Coefficient for the Circulator

PERFORMANCE OF COLD VACUUM PUMP

The cold vacuum pump is designed to compress saturated helium vapor and create subatmospheric pressure in the helium pot for temperatures below 4.2 K. The saturation pressure at 4 K, 3.5 K and 3.0 K are 0.8 atm, 0.46 atm and 0.24 atm respectively. With a discharge pressure of 1.26 atm, the operating conditions would require pressure ratios of 1.5, 2.7 and 5. Since the density of helium decreases as pressure decreases, a fixed mass flow would require a larger volume flow rate at lower suction pressure.

As shown in Fig. 2, a pressure transducer and a temperature sensor are located on the suction and discharge sides of the cold vacuum pump. In addition there is a temperature sensor mounted inside the subcooler helium pot. The pressure and temperature in the helium pot have been found to agree well with the published saturation curve. There is no flow meter installed on the pump so mass flow is evaluated indirectly from an energy balance on the subcooler helium pot and the circulating loop. Operating experience shows all pressure and temperature readings are stable except the pump discharge temperature which is periodically effected by liquid helium overflow from the precooler helium pot. Therefore it is difficult to evaluate the efficiency for the cold vacuum pump.

The results given in the report consist of three groups of data with the cold vacuum pump operating between 50 and 380 rpm. The first group was taken when the subcooler was connected to a calorimeter. The cold vacuum pump was operated at 310 rpm and the calorimeter was set between 0 and 300 watts. The second group of data were taken for an SSC magnet. Both the first and second group were taken with the thermal acoustic heat leak in the circulating compressor was present. The third group were taken during subcooling of a SSC magnet after this heat leak was eliminated. The second and third group represent constant heat loads and mass flow rates conditions.

The mass flow rate as a function of pressure ratio is given in Fig. 8. As one can see from the first data group, the mass flow rate decreases with the pressure ratio at constant pump speed. For constant mass flow rate, higher speeds correspond higher pressure ratios. The difference between the second and the third data group represent the 150 watts heat load from thermal acoustic oscillation of the circulator.

The volumetric efficiency, defined as the ratio of actual flow divided by the pump displacement due to cylinder clearance, and is plotted against pressure ratio as shown in Fig. 9. At the lowest operating pressure ratio of 1.2, the volumetric efficiency is about 90% which is very close to the 97% calculated from the built in clearance volume. This shows the adequacy of the present measurements. As expected the volumetric efficiency decreases with pressure ratio. The efficiencies are 0.77 and 0.5 at pressure ratio of 2.2 and 5.2. In the low pressure regime, the pump needs to be operated at high speeds to compensate for both lower density and volumetric efficiency. Higher pressure ratio or lower suction pressure can be reached if the heat load of the system is smaller.

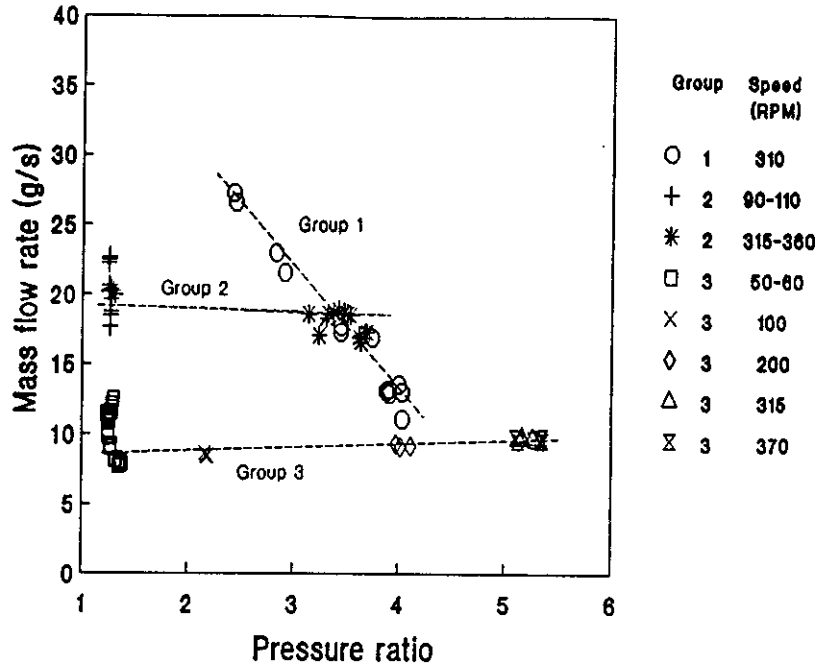

Fig. 8. Mass FLow Rate Versus Pressure Ratio for CCI Cold Vacuum Pump

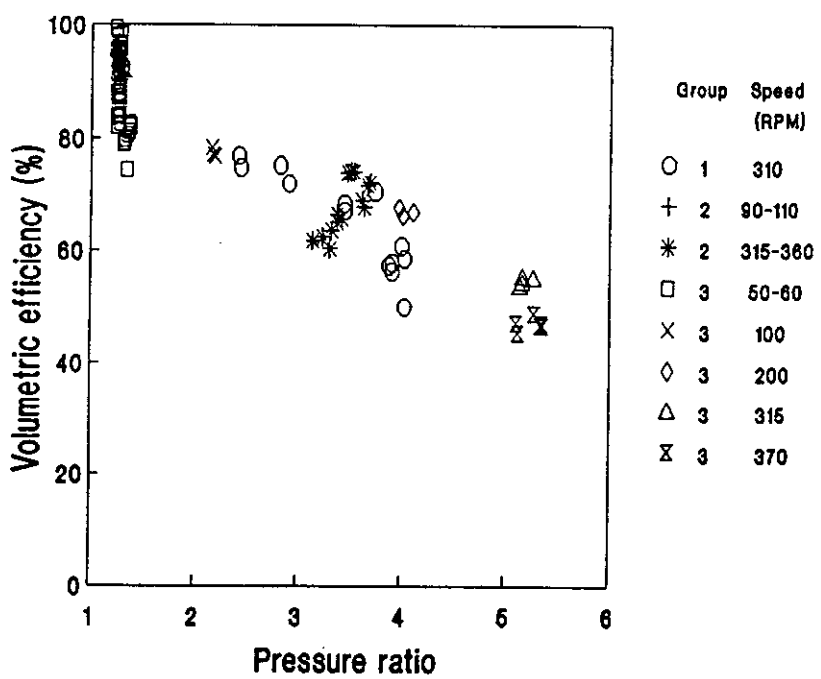

Fig. 9. Volumetric Efficiency Versus Pressure Ratio for CCI Cold Vacuum Pump

For evaluation of the adiabatic efficiency, analysis has been performed to discard invalid data due to the effect of liquid helium at the pump discharge. The heat leak for the cold vacuum pump is estimated at 35 watts. We calculate the pump efficiency without the heat leak and retain those data for which this efficiency is less than 80%. The results are given in Fig. 10 and Fig. 11. The adiabatic efficiency varies between 38% and 65% for pressure ratios from 2.5 to 5.5. The pump efficiencies, without heat leak, vary between 42% and 80 %.

The uncertainties associated with results given in Fig. 10 and 11 are large and the efficiencies are probably on the optimistic side. We plane to reevaluate the pump efficiency in the future. However some conclusions can be made. The efficiency curve seems rather flat in the region of investigation. In both the very low and very high pressure ratio regime, the adiabatic efficiency will be low since the compression work becomes less than the 35 watts heat leak. The pump efficiency at pressure ratio of 2.5 agree with that given in reference 1. But at large pressure ratios, the efficiency does not decrease as fast as reference 1 suggests. At pressure ratio of 5.3, the pump efficiency is still quite adequate for the present application.

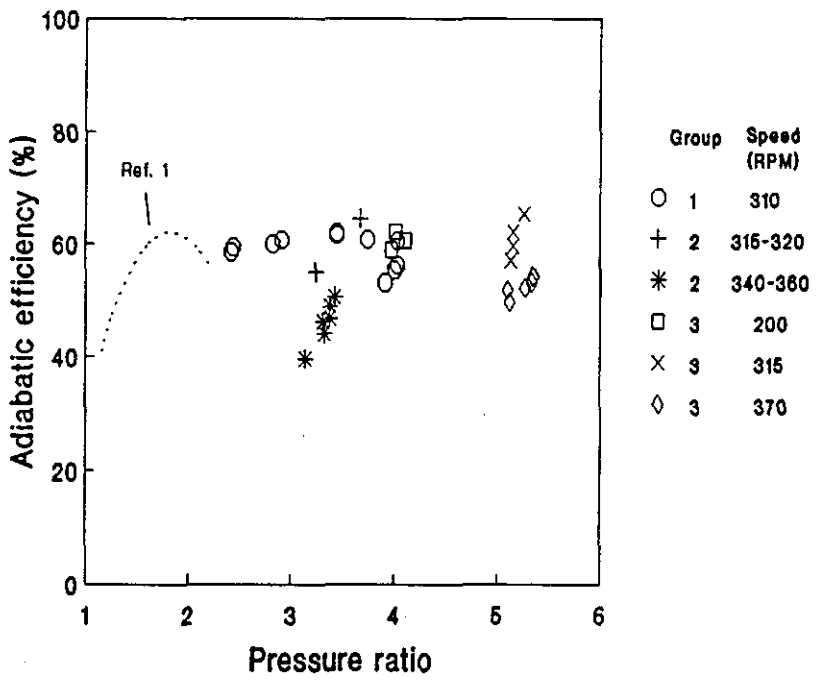

Fig. 10. Adiabatic Efficiency as a Function of Pressure Ratio for Cold Vacuum Pump

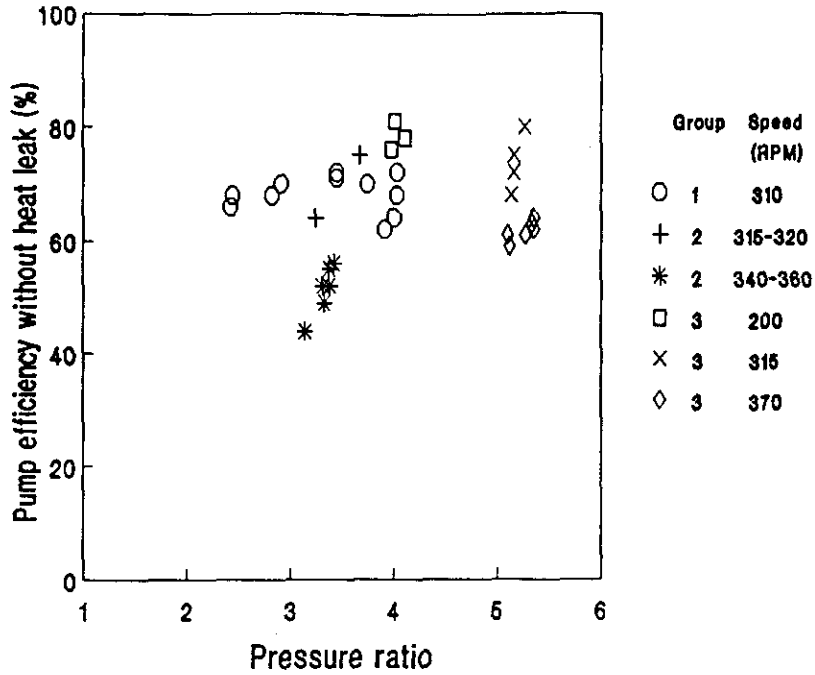

Fig. 11. Efficiency of Cold Vacuum Pump in the Absence of Heat Leak

CONCLUSION

The subcooler assembly has been used for testing SSC dipoles with great success. Equipment is well selected to provide performance and flexibility for the process. An SSC dipole can be tested any where between 4.35 K and 3.2 K with ample subcooler capacity for cooling and quench recovery. From performance data, we believe the existing subcooler can handle a large number of, perhaps 10, magnets in series at 3.5 K. With more than 5000 hours of operation, we conclude the system is well designed and constructed, very reliable, versatile and easy to operate.

ACKNOWLEDGEMENT

Numerous personnel at BNL who contributed to the success of construction and operation of this system are highly appreciated. In particular the authors want to thank H. Hildebrand's group for installation and operation, W. Kollmer's group for electronics and instrumentation, and R. Picinich for MAGCOOL operation.

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