Mobile Cryogenic System for Industrial and Laboratory Applications

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ABSTRACT

A previous paper by Sumitomo (SHI) Cryogenics of America Inc. describes a mobile cryogenic system that has been fully commercialized to cool down or warm up a superconducting MRI magnet by circulating helium in a closed-loop system. Sumitomo (SHI) has been exploring other configurations of the system that will enable it to be used to warm up and/or cool down a device to about 25 K, or to cool a device and keep it cold. Such devices might be proton therapy systems or HTS applications. In this paper we present the results of studies including the effect of circulator speed on cooling at the device, capacity with different configurations of the expanders, and flow through interface tubing that will facilitate adapting the system to other applications.

The present system circulates helium which is cooled by single stage Displex—-type Gifford-McMahon (GM) expanders in a cryostat with heat exchangers integrated on the cold ends and warmed by heaters on the heat exchangers. The mobile & compact nature of the system enables users to cool superconducting magnets at an end-user site or manufacturing hub. The benefits are: conservation of helium, the option of shipping a magnet warm, reduced cryogen expense, reduced on-site maintenance, and reduced cryogen handling on-site. There are currently systems in the field at production facilities, manufacturing hubs, and maintenance facilities, allowing customers to be more cost competitive and more responsive to their customer's needs.

INTRODUCTION

The mobile cryogenic refrigeration system described in this paper has been designed to cool down or warm up devices in cryostats by circulating cold or warm gas through transfer lines, bayonets, and the cryostat. The mobility is achieved by designing the system components to be of a size that can be wheeled through corridors in a hospital or laboratory and easily connected together at the point of use. Typically, the device being cooled or warmed is turned off and the power for the device is available to run the mobile refrigerator. Component custom reusable shipping containers are developed and designed ergonomically for ease of handling as shown in Figure 1 and are available for transporting the system, these include one for each of four compressors, one for the main cryostat, and one for the control console. Provisions are made to load & unload system components in/out of the containers.

Cryocoolers 20, edited by S.D. Miller, R.G. Ross, Jr. and J.R. Raab

Figure 1. Refrigerator cryostat cart with custom reusable shipping container

SYSTEM DESCRIPTION

Referring to Figure 1, mounted on the main cart is the refrigerator cryostat that has four GM cycle expanders, a cold rotary fan, and bayonets on the cover plate: a scroll vacuum pump and turbo pump to evacuate the refrigerator cryostat and circulator lines, and piping to supply make-up gas and control the pressure. Compressors are on their own carts and connect to the expanders through flex hoses. The control console is also on a wheeled cart and has wiring connections to the other components. Provision is made to have control signals from the device cryostat interact with the refrigerator controls, such as control of fan speed. More details about the system can be found in a previous paper [1].

Cooling Capacity

Priority has been given in the design of the present system to reach a lower temperature rather than speeding cool down. The model CH110 expanders that are used in this system are designed for minimum temperatures of about 12 K and as a result have more cooling capacity below 40 K than expanders that are optimized to operate near 80 K. The latter expander has more capacity above 40 K and would cool down a device faster but not get it as cold. For the same reason the fan is located in the return line so that the gas going to the device is colder at minimum temperature than if the fan is in the supply line.

Circulating gas to cool the device results in less cooling being available at the device cryostat than is being produced by the expanders. Figure 2 shows the capacity of the expanders, GME, at their cold ends, Tce, and the capacity at the device cryostat, MRI, Tmri, for different rates of helium circulation. The reduction in available refrigeration is due to the temperature change in the gas, heat input from the circulator, and the thermal losses in the piping and transfer lines. At a constant pressure and constant speed the flow rate increases as the gas gets colder.

In Figure 2, system1, system2 curves show the cooling that is available down to 100 K on the present system for initial flow rates of 3 g/s and 2 g/s. The benefit of operating at an initial flow rate above 3 g/s is small while the loss for operating below 2 g/s becomes large when compared against 15 g/s, 5 g/s and 1.5 g/s as shown in Figure 2. In the SHI capacity test cryostat, the fan, running at 18,000 rpm with helium at 290 K, 220 kPa, can flow 3.5 g/s at a head of 270 m (0.85 kPa) of which less than 35 % of the total pressure drop is lost in the refrigerator piping and transfer lines. With the device at 300 K and 3 g/s flow rate the colder expander is operating at about 190 K as shown by GME in Figure 2 and 1,400 W of cooling is available at the device as shown by System 1 in Figure 2. As shown in Figure 2, with a flow rate of 2 g/s the expanders are at about 170 K and 1,300 W is available, and for 1.5 g/s the expanders are at about 140 K and 1,180 W is available. It is recommended that the interface tooling and circulation pressure be designed with 2 g/s as a minimum goal. At this flow rate the pressure drop within the refrigerator circuit and transfer lines is small and most of the head produced by the fan is available to move gas through the tooling.

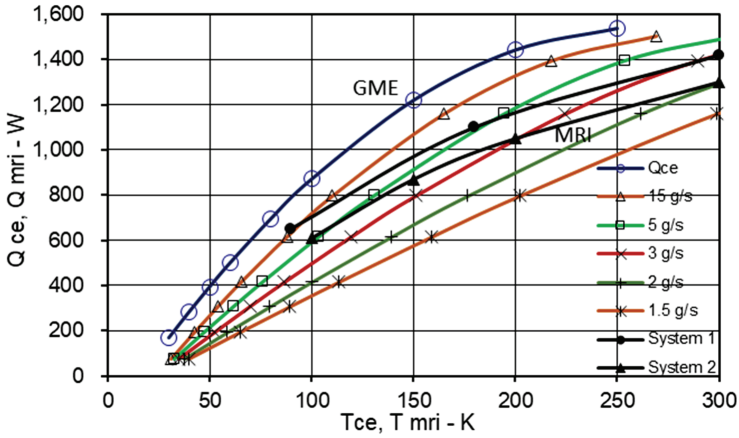

Figure 2. Capacity at the expander, GME, and capacity at the MRI cryostat for different circulation flow rates. Capacity with 2 and 3 g/s initial flow rates at constant circulator speed are superimposed.

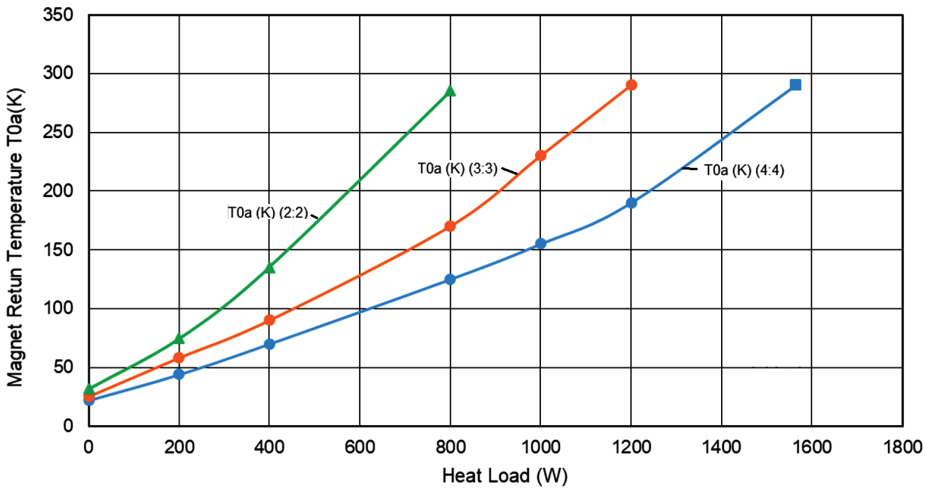

Figure 3. System capacity with two, three and four expanders operating

Capacity tests have been run in a test cryostat with only two and three expanders operating on the present system [1]. The non-operating expander(s) have a very small heat loss and the temperature change in the gas between the operating and non-operating expander is smaller at a given flow rate. With four expanders operating the minimum temperature is 22 K, three expanders is 25 K, and two expanders is 32 K. Fewer than four expanders can be used to cool down smaller devices in a reasonable time, or larger devices if more time is allowed. System capacity from 300K to minimum temperature is shown in Figure 3 when operating two, three and four expanders.

Effect of Fan Speed

Tests were run in a test cryostat with constant heat input while changing the fan speed. Results are shown in Figure 4. High fan speeds circulate gas at a higher rate and add more heat of compression to the gas than low speeds thus there is a minimum temperature for a given load at the optimum fan speed. It is seen that there is a wide range of speeds for which there is a small change in temperature. The present system has a default program that holds the fan speed at

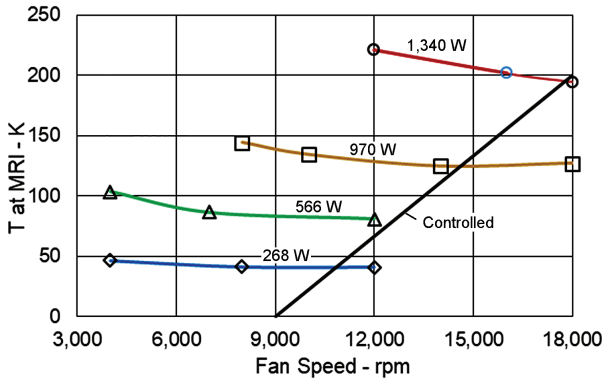

Figure 4. Tests at constant heat loads, variable fan speeds, 220 kPa absolute pressure

18,000 rpm until the return temperature drops to 200 K then reduces the speed on the line labeled "controlled". The fan speed program can be user adjustable based on the application requirements.

Interface to Device Cryostat

Special tooling is needed to have the circulating helium flow from the supply transfer line to the bottom of the device cryostat then return warmer gas from the top of the cryostat through the return transfer line. Vacuum jacketed sleeves are needed on both the supply and return bayonets and the termination of the return bayonet needs to be sufficiently above the supply bayonet to avoid direct recirculation of the gas. The return bayonet is at a higher temperature than the supply bayonet and preferably has a larger ID. While the original system was designed with some latitude on the size of the interface tubing, the question now being asked is whether or not sufficient flow can be brought through the ports on existing cryostats.

Interface Simulation Test Setup

In order to provide some test data that can be used as a reference for calculating flow rates, a test was set up with a tubing size that might be close to the minimum size that is practical to use. A 2 m length of 14.1 mm ID tubing was coiled in 260 mm loops, wrapped with super insulation, and connected to the bayonets in the same cryostat that had been used to measure system capacity. A second 14.1 mm ID coil, 0.5 m long was also tested. A manometer was constructed to measure the pressure drop across the coil using a low vapor pressure oil. Figure 5 is a photo of the test setup. The tubing became slightly flattened as it was coiled and the equivalent diameter was determined to be 13.9 mm.

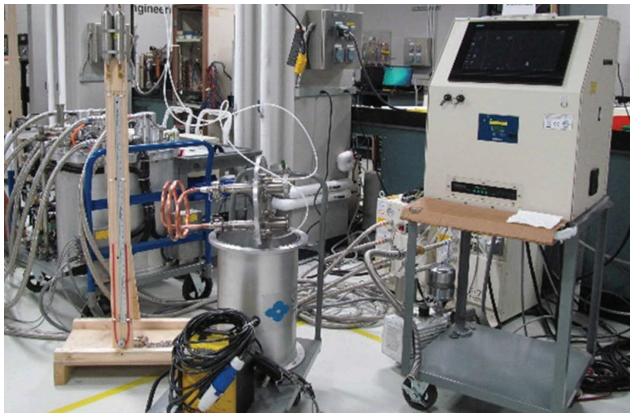

Figure 5. Photo of test setup. The oil filled manometer is to the left of the 2 m long test coil

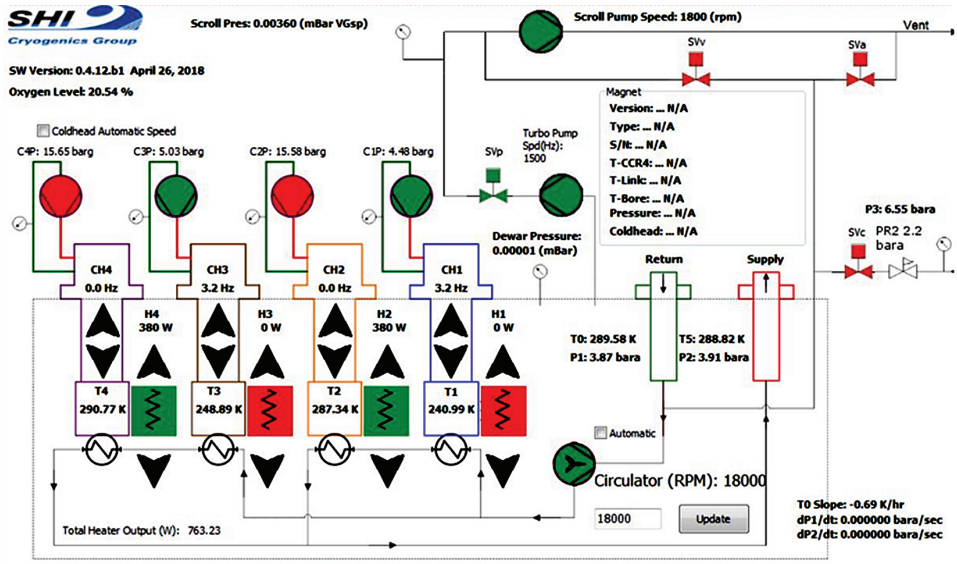

Figure 6. System user interface on the control console during the test

The circulation circuit is designed for a maximum pressure of 1.5 MPa but the relief valves and control settings on the present unit are set for pressures below 0.2 MPa. For the tests with the coil the settings were adjusted for circulating pressures within the range of 110 to 410 kPa absolute. All tests were run at the maximum fan speed of 18,000 rpm and at temperatures from room temperature down to 100 K. The flow through the four expanders is split into two circuits, each having a first expander and a second expander in series. Flow rate was measured by running the first expanders to cool the circulating gas then, with the second expanders not operating, the heaters were used to warm the gas to the temperature that flows through the test coil as shown in system user interface on the control console in Figure 6. The mass flow rate is then calculated based on the temperature difference across each cold end and the refrigeration produced at the cold ends of the first expanders, and the heat input at the second expanders. The total flow rate is taken as half of the sum of the four individual flow rates.

TEST RESULTS

Figure 6 is a screen shot of the control console while holding the coil at a stable temperature of 289 K with helium flowing into the circulator at an absolute pressure of 3.87 bar (387 kPa absolute). The first expanders cool the gas close to their operating temperature of 241 K and 249 K where they are producing over 380 W of refrigeration each. The gas is then heated back to 289 K by putting 380 W of heat into each of the second expander heat stations with the expanders turned off. A summary of test results with the 2 m and 0.5 m coils is presented in Table 1.

The volume flow rate is calculated from the measured flow rate and the density of the gas at the fan. It is interesting to note that the volume flow rate of about 15 m^3/hr was nearly constant for all of the tests with the 2 m long coil. This indicates that the velocity of the gas leaving the fan and the effective flow area are nearly constant for the range of temperatures and pressures of the tests. The head produced by the fan at 18,000 rpm and 15 m^3/hr is 319 m. The circulator head is shown in Table 1 in kPa absolute based on the density of the gas at the fan. The pressure drop measured with the oil manometer across the test coil is slightly less than the pressure rise in the circulator indicating that the pressure drop in the refrigerator circuit is small relative to that in the test coil. Calculated values of pressure drop, assuming the length of the tubing is straight, are listed in Table 1. The difference between the calculated pressure drops and the measured pressure drops is due to head losses in the entrance, exit, and bends. The last line of table 1 shows the

Table 1. Summary of test results with 13.9 mm ID coils

Coil length - m	2.0	2.0	2.0	2.0	2.0	2.0	0.5	0.5	0.5	0.5
P - Bara	3.91	2.05	2.38	3.91	3.85	2.3	3.44	2.52	3.6	2.1
T0 - K	290	285	202	198	107	104	301	299	102	93
Flow - g/s	2.7	1.5	2.4	4.0	7.3	4.4	4.0	2.8	9.5	6.3
Flow - m^3/hr	14.8	15.0	15.2	15.0	15.1	14.6	25.8	24.4	19.9	20.5
Head - m	319	319	319	319	319	319	297	301	311	310
Head, fan curve - kPa	2.1	1.1	1.8	3.1	5.6	3.5	1.6	1.2	5.3	3.4
dP coil, manometer - kPa	1.9	1.0	1.6	2.9	5.0	3.1	1.4	1.0	4.6	3.5
dP straight, calc -kPa	1.0	0.7	0.9	1.3	1.9	1.2	0.6	0.4	0.8	0.6
dPstraight/dPmanometer	0.52	0.66	0.53	0.43	0.36	0.38	0.41	0.41	0.16	0.15

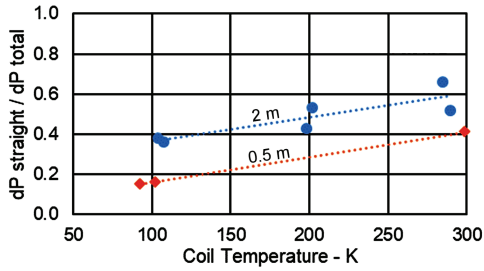

Figure 7. Fraction of pressure drop in straight section of 13.9 mm ID, 2.0 m and 0.5 m long test coils.

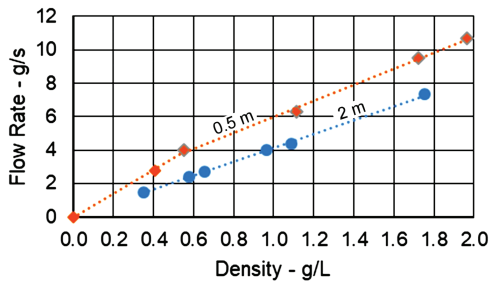

Figure 8. Measured flow rate vs gas density at the fan for 2.0 m and 0.5 m long, 13.9 mm ID tubes.

fraction of calculated pressure drop for the assumed straight length of tubing relative to the total pressure drop. This fraction is plotted it Figure 7 as a function of the temperature of the gas at the fan. It is seen that the pressure drop calculated for the "straight" length decreases as the temperature drops, and the shorter coil has more form factor losses.

The relation between the density of the gas at the fan and the mass flow rate for the tests summarized in Table 1 is shown in Figure 8. For a constant fan speed and pressure, the flow rate increases proportional to the initial temperature divided by the return temperature from the device cryostat, during cool down. From Figure 8 it is seen that the 2 m long test coil has to have an initial density at the fan of 0.5 g/L to have a flow rate of 2 g/s. This requires a circulating pressure of 300 kPa absolute at 290 K in the test coil. The data for the tests with the 0.5 m long coil are puzzling because they show a break in the flow rate curve at a density of about 5.6 g/L. The break is also reflected in the drop in the volumetric flow rate. A density of about .3 g/L at the fan is needed to have an initial flow rate of 2 g/s. An initial pressure of 185 kPa absolute at 290 K will have a flow rate of 2 g/s. Consideration of the fact that for a flow rate of 2 g/s the temperature of the gas flowing through the supply tooling drops from around 290 K to less than 200 K results in the flow rate increasing by about 25 % if the return tooling has a low pressure drop.

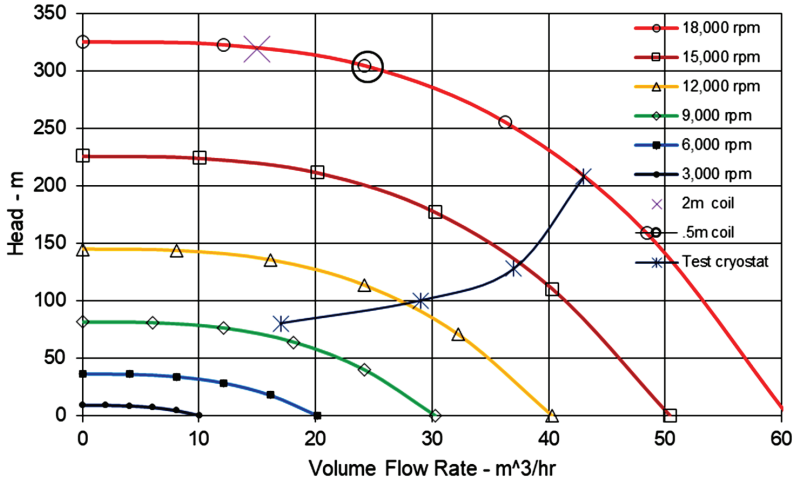

Figure 9. Head vs Volume flow rate at constant rpm with test results superimposed

Correlations with Circulator

Figure 9 is a plot of the circulator head vs. volume flow rate at constant fan speeds. The data of the tests shown in Table 1 cluster around the symbols X for the 2 m long coil and O for the 0.5 m coil which have been superimposed on the plot. Also superimposed is the data from the fan speed test shown in Figure 4 for the "controlled" fan speeds. The coil tests have relatively high flow impedances while the SHI heat load test cryostat has a relatively low impedance. Fan speed is typically reduced during cool down to minimize cooldown time, but it may also be further reduced to control the rate of cooling and the effect of thermal stresses.

CONCLUSIONS

The mobile & modular nature of this refrigeration system enables it to be transported in its custom reusable shipping containers, and provides the option to use fewer compressors and expanders if the full capacity of the system is not needed. The system is designed with the equivalent of 30 mm ID piping and a cold circulating fan having a maximum speed of 18,000 rpm. Tests at different fan speeds show that cool down time in the SHI capacity test cryostat at 220 kPa absolute is minimized by running the fan at 18,000 rpm down to 200 K then reducing the speed to 9,000 rpm at 0 K proportional to temperature. A wide range of speeds resulted in small changes in the rate of cooling. A minimum flow rate of 2 g/s at the start of cool down is recommended for effective use of the system.

Tests were run with 13.9 mm ID coils of tubing having lengths of 2.0 and 0.5 m, and pressures from 2 to 4 Bara, in order to provide some reference data for the design of interface tooling that can be used for adapting the present system to existing device cryostats that have been designed with helium fill ports. The pressure drop across the test coils as measured by a manometer is slightly less than the pressure rise produced by the circulator fan indicating that the pressure drop in the refrigerator and transfer lines is small relative to that of the test coils. It is also observed that for the 2 m long test coil the flow rate is directly proportional to the density of the gas at the fan at 18,000 rpm. Near 300 K the pressure drop in the "straight" tubing was about 60 % of the total for the 2.0 m coil and about 40 % for the 0.5 m coil. The pressure drop due to entrance, exit, and bend effects is significant and increases as the temperature decreases. Tests with the 2 m long coil showed that the volume flow rate at 18,000 is nearly constant over a wide range of temperatures and pressures. The 0.5 m long coil shifted to a lower volumetric flow rate above a density of about .3 g/L. When plotted on the fan curve, it is suggested that the intersection of the volume flow rate and head at 18,000 rpm can be used to characterize the system.

The present design has the fan in the return gas line in order to minimize the minimum temperature, if however the fan is put in the colder supply line the available pressure head at 2 g/s is increased about 50 %. This would increase the minimum temperature about 1 K. Consideration is being given to doing this if it is needed.

REFERENCES

1. Gandla S K, Longsworth R C, "Mobile refrigeration system for precool and warm up of superconducting magnets," *Advances in Cryogenic Engineering: Proceedings of the Cryogenic Engineering Conference (CEC) 2017*, Vol. 278, 012179.