Design and Development of Integral Cold Transportation System

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ABSTRACT

A new cold transportation system for a Pulse Tube Refrigerator (PTR) that allows remote placement of the Device to be Cooled (DTC) with respect to PTR is investigated. Such a system has practical applications in on-board space vehicles for cooling of sensors. The system works on the principle of a DC flow circuit and consists of a pressure wave generator, precooler, check valves and heat exchangers at the cold head of the PTR and the load (DTC). The precooler is a regenerator while the other heat exchangers are recuperators. All the heat exchangers are analyzed and designed to match the heat transfer requirement. In the experimental test rig, the cold head is simulated with a liquid nitrogen bath and the load, with a resistance heater. The rig is placed in a cryostat which is maintained at high vacuum condition. Different experiments are performed at both room and cryogenic temperatures. The experiments include pressure testing to observe the AC-DC conversion of flow across the check valves. The temperature variation at different points of the flow circuit is monitored in time to ensure steady state. The load on the cryocooler is estimated by measuring the boil-off rate of LN2 which plays the role of cryocooler in the experimental setup. The boil-off is measured by using a series of PT-100 type temperature sensors placed on a Teflon rod. The sensors are wound with nichrome wire to give a small heat input. The time taken for the rise in temperature of two consecutive sensors is used to calculate the boil-off rate. The results for the quantities of interest are presented.

NOMENCLATURE

C_m – Matrix capacity ratio
FF – Friction factor based on Miyabe
G_h – Mass Velocity by Gedeon (kg/m^2. s)
G_p – Mass Velocity by Miyabe (kg/m^2. s)
kHe – Thermal conductivity of Helium
NS – Number of Screens
NTU_0 – Number of transfer units
Nu – Nusselt number
P - Pressure (bar)
Pr – Prandtl number
Re_h – Reynolds number based on Gedeon
Re_p – Reynolds number based on Miyabe
T_{avg} – Average Temperature (K)
V – Velocity of the gas (m/s)
ΔP – Pressure drop
μ – Dynamic Viscosity of the gas (Pa. s)
ρ – Density of gas (kg/m^3)
ψ – Porosity of the matrix material

Cryocoolers 22, edited by R.G. Ross, Jr., J.R. Raab and S.D. Miller

INTRODUCTION

Cryocoolers are mechanical refrigerators employed to cool various systems to cryogenic temperatures. These stand-alone devices are very useful for cooling sensors such as Infrared (IR) detectors in satellites. Pulse tube refrigerators (PTR) which have no moving parts at cryogenic temperatures are the most durable refrigerators for space applications. The evolution of PTRs is described in [1]. PTRs operating at the frequencies in the range of 30-70 Hz are called as Stirling type cryocoolers and have a refrigeration capacity of 1W at 80K and this can be achieved with a linear motor compressor of input power 50W. This is sufficient for the application of cooling of IR detectors to the operating temperatures. But the difficulty in transferring the cold from PTR to the sensors needs to be addressed. Usually, the sensor is mounted at the cold heat exchanger of PTR. This results in housing the cryocooler very close to the sensor. A cold transportation system like a Cryogenic Heat Pipe (CHP) could be one of the solutions for this. Research and development on CHPs are going on worldwide. Some of the major difficulties involved in the design and operation of CHP are start-up and sustaining of the flow of the working fluid in the heat pipe. The reason for this is that all the working fluids used in CHP are in supercritical state at room temperature. Hence, it is necessary to use a secondary heat pipe system to cool down the working fluid and condense sufficient quantity of the gas to liquid phase before starting the primary heat pipe. Also, the operation of CHP against gravity is another major challenge. The present system which can be described as a passive cold transportation system can overcome the difficulties associated with CHPs and also facilitates vibration isolation of the DTC from the pressure wave generator.

It should be noted that the working principle of the proposed system is different from other cryogenic systems like cryoprobes. For example, in a cryoprobe, a separate transportation system with its own equipment, including the compressor is incorporated. A compressor causes the movement of the working fluid (like helium) in the entire circuit. The cryocooler cools the gas and the cold is transported to the cryoprobe. Hence, the cryoprobe itself can be considered as the cold finger. Moreover, the use of cryoprobe is mostly restricted to NMR technology. A remote cooling loop has been reported earlier by Raab et al. in [2], illustrating the remote cooling by three configurations which include a separate circulator compressor, a single compressor with warm valve and a single compressor with cold valve. In a single compressor configuration, the gas to the remote cooling loop is bled off by the same compressor which drives the PTR, thus affecting the performance of the PTR. The system in the present work uses a twin pressure wave generator (PWG) to drive the PTR and the flow circuit.

DESIGN AND WORKING

A twin PWG drives a PTR at one end and produces pressure pulses in the circuit at the other end, which push the gas through the one-way valve. A schematic of the cold transportation system is shown in Fig. 1. The system consists of a precooler, a couple of non-return valves, a heat exchanger at the cold head of the PTR, a heat exchanger at the DTC, and an optional valve to match the cooling capacity to the load. The alternating pulse generated from the PWG is passed through a check valve and is converted to a DC flow. The gas is then cooled by the precooler and then passed through the cold head of PTR to attain its temperature. The gas then passes along the line to the DTC at the remote location. The gas exiting the load at remote location is at lower temperature than the gas coming from PWG and hence can be used to cool the oncoming gas. The system can automatically transfer the cold from the refrigerator to the sensor whenever the PTR is in operation. There is no necessity to manage the cold transportation device separately as it does not involve any critical adjustment/control. There is no need for any additional instrumentation to operate the system.

The temperature and pressure drop in the flow circuit are to be analyzed at each component and care must be taken to match its performance with pulse tube. Also, the maintenance of the system in high vacuum condition to prevent heat ingress poses an important challenge in the design. The PTR to be used in the system is of few watts of refrigeration capacity. Thus, the flow in the circuit can be expected to be in the oscillatory laminar or transitional regime. A parametric analysis of the precooler is carried out to study the influence of the geometrical and operational parameters. For the piping, consideration can be given to materials like metals and plastics considering their strengths to withstand high pressure and low temperature. A metering valve can be used to control the flow in the circuit.

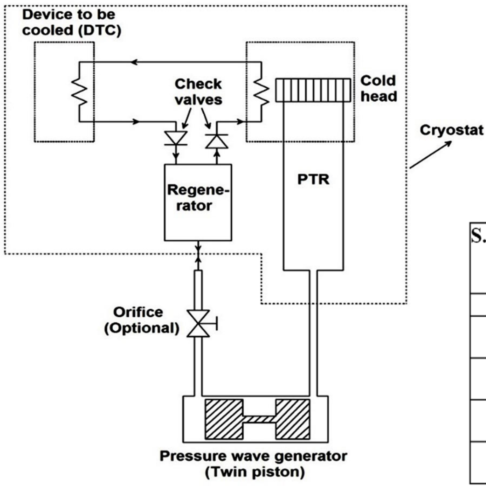


Figure 1. Schematic of the cold transportation system.

Table 1. Comparison of different recuperators for the required cooling.

S.No.	Type of HX	Length of HX (in m) (approx.)
1.	Tube-in Tube Heat Exchanger	350
2.	Mesh type HX (Mesh in the annular side)	210
3.	Helical Heat Exchanger (G-H Type)	189
4.	Spiral fin Heat Exchanger (Koch Helium Plant)	22.5
5.	Mesh type with mesh in both tube and annulus	8.5

The compact recuperators as studied in [3-7] are found to be not feasible for the design of the DC loop as their length is of the order of few tens of meters. This is due to the low temperature difference between the hot and cold streams. Various recuperators were designed to meet the requirement of precooling to a temperature of about 80.5K and the length of each heat exchanger is presented in Table. 1. The HX with mesh on both tube and annular side is found to be compact when compared to the other designs, but the pressure drop on the tube side is of the order of few tens of bars and thus is opted out. Hence, a regenerator is used in the place of counter flow heat exchanger. The flow is converted to DC by the check valves after the regenerator. The following section discusses the design of the regenerator.

DESIGN OF THE REGENERATOR

The properties of the helium gas such as density ñ, viscosity ì, and thermal conductivity k_{He} were calculated from the ideal gas equation and the relations reported in [8], at the average temperature of the regenerator T_{avg} and pressure P. the correlations of Gedeon [9] were used for finding the Nusselt number, the values of G_h and Re_h were calculated accordingly. The correlations of Miyabe [10] were used for pressure drop calculations. It may be noted that the Reynolds number Re_p and mass velocity G_p in these correlations are different from those in the Gedeon's correlations. Barron [11] has taken the number of Transfer Units, NTU_0 and matrix capacity ratio, C_m equations from the work by Coppage and London [12] and the tabular data from Johnson [13] to determine the dimensions of the regenerator. It is mentioned earlier that the Coppage and London method is to be used for rotary regenerators and Hausen's method [14] is to be used for fixed bed regenerators. However, Shah [15] has reported that both the methods are related and either of the methods can be employed for the fixed bed calculations.

The Nusselt number correlation from [9] is

$$Nu = (1 + 0.99Pe^{0.66})\psi^{1.79} \tag{1}$$

The friction factor and pressure drop correlations from [10] are

$$FF = 0.337 + 33.6/Re \tag{2}$$

and

$$\Delta P = (NS * \rho * FF * V * V/2 \tag{3}$$

where, Pe is the Peclet number which is the product of Reynolds number Re and Prandtl number Pr; ø is the porosity of the matrix; FF is the Darcy's friction factor; NS is the number of screens; V is the velocity of fluid in the matrix; ñ is the density of the fluid.

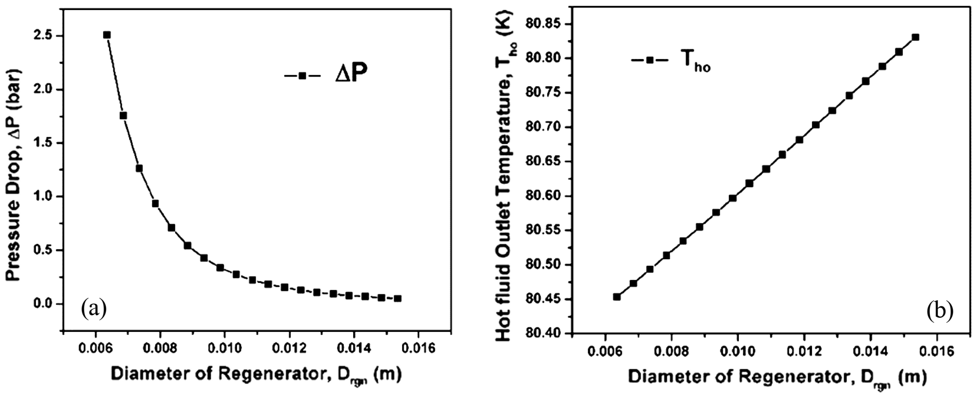


Figure 2. Effect of regenerator diameter (D_{rgn}) on (a) Pressure drop (P); and (b) hot fluid outlet temperature (T_{ho}).

A computer program was written to calculate the dimensions of the regenerator. Coppage and London [12] mentioned that for values of C_m above 10, the ratio can be considered as ∞. Hence, the calculations were initiated with Cm = 10 and are performed iteratively to get the required NTU. The length and pressure drop for a set of Cm values for each diameter of regenerator were determined. A value of Cm = 180 was selected for sample calculations and to observe the trends of temperature and pressure drop and the results are plotted.

The plots showing the variation of pressure drop and hot fluid outlet temperature (here the cold side temperature of regenerator) with diameter of the regenerator are shown in Fig. 2. It can be observed that an increase in diameter decreases the length of the regenerator for a fixed matrix capacity ratio and hence the pressure drop decreases with increasing diameter. However, the difference in the pressure drop for diameters above 0.012 m is very minimal and thus can be chosen for fabrication. The outlet temperature of the hot fluid increases with decrease in length. The required temperature can be obtained by increasing the matrix capacity ratio which increases the length for a particular diameter.

EXPERIMENTATION

Experiment with Small Compressor

Figure 3 shows the experimental test rig of the DC Loop for the cold transportation. The main aim of this experiment is the validation of the computer results and the operation of check valves at cryogenic temperatures. An existing regenerator and a small compressor are coupled to the DC loop which is passed through the LN2 bath for cooling. The heat ingress is considered as the thermal load. The length of the

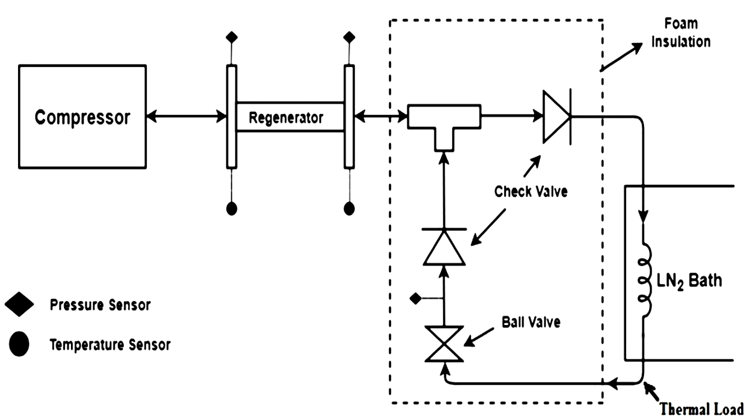


Figure 3. Experimental test rig.

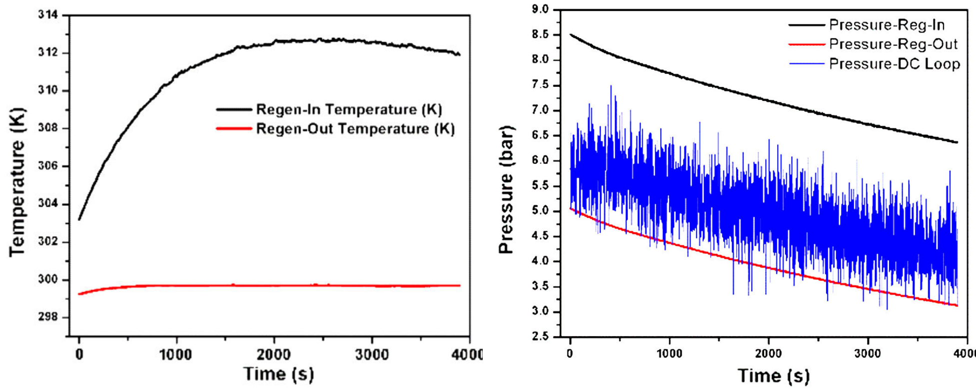


Figure 4. Experimental Results: (a) Time vs. Temperature of regenerator; (b) Time vs. Pressure of regenerator and loop.

regenerator is 64 mm, and the diameter is 9.5 mm; the porosity of the mesh used is 0.661; the charging pressure is 8 bar.

Fig 4(a) shows the trend of temperature at the entry and exit of the regenerator with time. It is observed that there is an initial rise in temperature at the entry of the regenerator. It is due to the heat of compression, and the system reached a steady state after some time. The exit temperature, however, remained almost constant throughout. Hence, it is concluded that the pressure wave from the compressor is not travelling through the DC loop to get cooled, due to the insufficient stroke of the compressor. The reason for this could be that the regenerator used in the circuit needs a larger capacity compressor. The pressure variation at the regenerator inlet, outlet, and at the DC loop were recorded, and a steady decrease in the mean pressure over the entire circuit is observed as shown in Fig. 4(b). The pressure fluctuations in the DC loop are due to the gas entering the DC loop and not exiting the loop. Thus, the experiment confirms the successful operation of check valves at the cryogenic temperatures.

Experiment with larger compressor

As the stroke of the small compressor is insufficient to drive the fluid, a larger compressor was used to drive the circuit. Figs. 5(a) and 5(b) show a schematic and a photograph of the experimental setup, respectively, with the large compressor. The regenerator and the LN2 bath were removed, as the prime objective of this experiment was to observe the trends in pressure in the flow circuit. The system was charged with helium gas to a pressure of 14 bar, and the compressor was operated with 500 W input. Two pressure sensors were placed, one at the exit of the compressor (Pressure Sensor 1), and another in the DC loop in between the ball valve and check valve (Pressure Sensor 2). These are diaphragm type

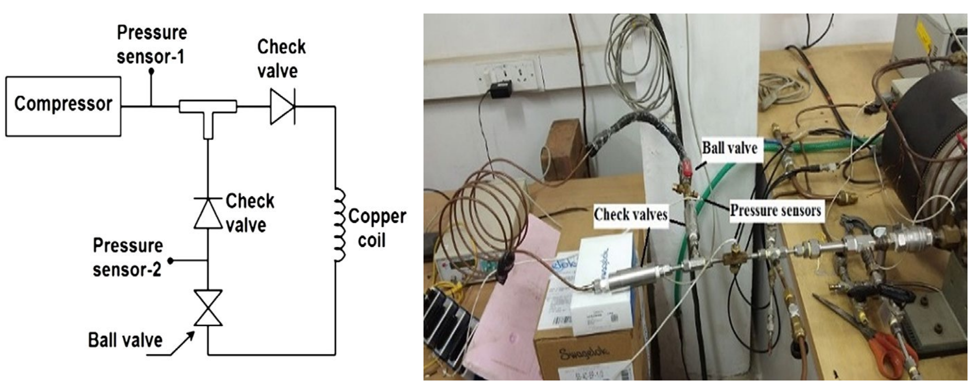


Figure 5. (a) Schematic of the experimental setup with larger compressor; (b) Photograph

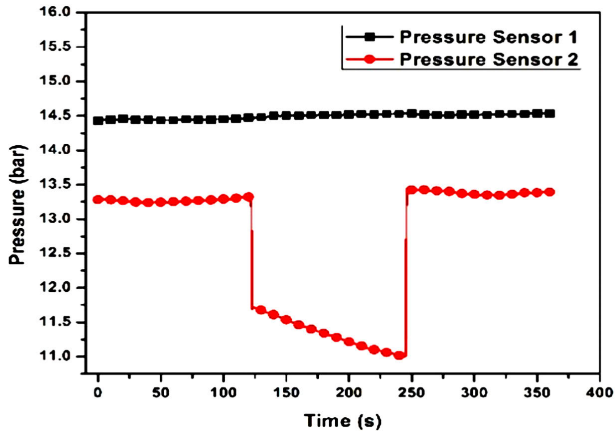


Figure 6. Experimental results with larger compressor, Time vs. Pressure.

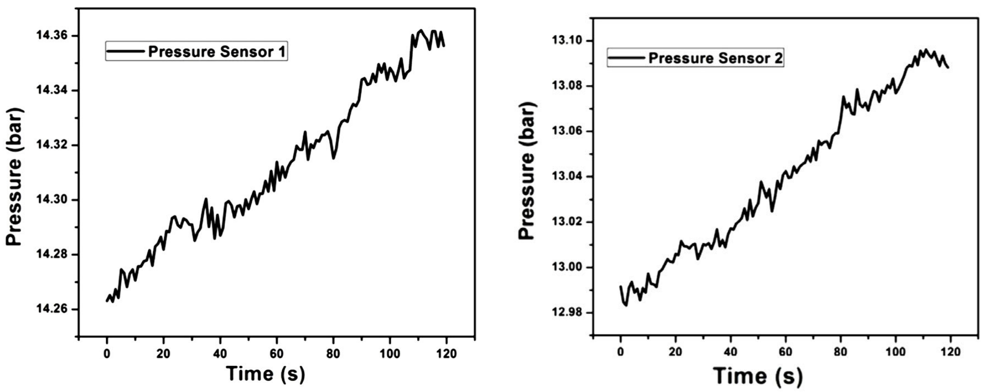


Figure 7. Changes in pressure with time (a) before the check valves and (b) in between the check valves.

pressure sensors. The experiment was run for 360 seconds by keeping the ball valve open, closed, and open, respectively, for a duration of 120 seconds each. The acquired data are plotted in Fig. 6.

It can be observed that as the ball valve is opened, the flow is uniform throughout the circuit, and there is a uniform pressure difference at the two points. But as the ball valve is closed, the gas in between the ball valve and the check valve gets rarefied, and there is an abrupt drop in the pressure at that region. From these readings, it is concluded that the stroke of the compressor is adequate to establish the flow in the DC circuit also. The acquired data from the pressure sensors were plotted individually against time and a steady increase in pressure is observed as shown in Figs. 7(a) and 7(b). This increase in pressure is due to the heat of compression added to the gas. As the compressor is operating at a higher frequency and there is no provision to remove the heat, the gas gets heated and the pressure rises accordingly.

The plotted data do not show a sinusoidal wave form because of the very high operating frequency, and hence an oscilloscope was connected in parallel to the pressure sensors. The wave forms from both sensors are captured and are shown in Fig. 8.

Another set of experiments to study the temperature trends of the DC circuit were carried out with a layout similar to that shown earlier in Fig. 3. The experimental setup consisted of three temperature sensors at the ends of the regenerator and at a second check valve. A pressure sensor was placed before the regenerator to measure the pressure in the system. The DC circuit was covered with foam insulation to avoid excessive heat ingress. The temperatures at the mentioned points were acquired and are shown in Fig. 9. It can be seen that the temperature at the regenerator inlet remains almost constant at room temperature (300 K) as the heat of compression is removed by the cold water coil running at the inlet of the

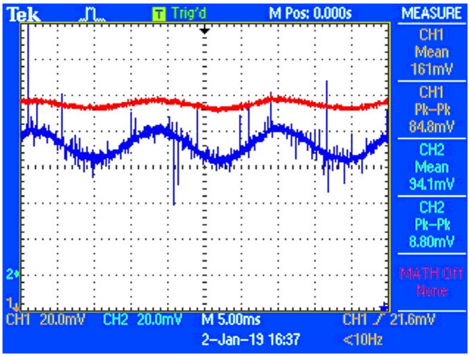


Figure 8. Oscilloscope image showing the sinusoidal wave forms of the gas flow.

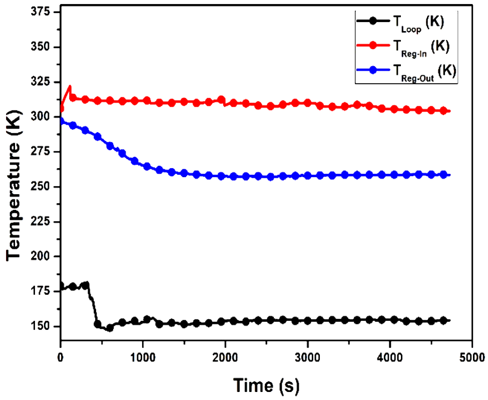
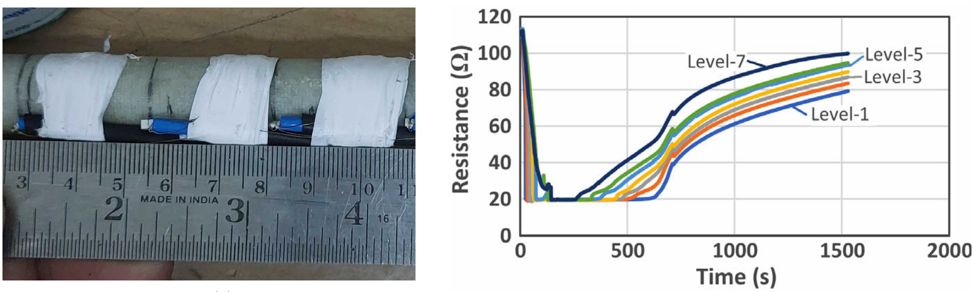


Figure 9. Change in temperature at different point with time.



(a)

(b)

Figure 10. (a) RTDs used to measure the boil-off rate; (b) RTD resistance vs. time for different levels.

regenerator. The temperature at the regenerator outlet is seen to drop gradually from room temperature to a value near 250 K and attained a steady state thereby establishing a gradient between the ends of the regenerator. The temperature in between the check valves dropped to 150 K and remained steady thereafter. Further drop in these temperatures could not be achieved due to the limitation of the insulation provided. The performance of the system can be increased by maintaining the system in a vacuum environment.

The boil off rate of Liquid nitrogen gives the load on the cold head and hence the boiloff is measured by placing RTDs at different locations. A Nichrome wire is wound around each sensor and a minimum heat is given to prevent the formation of ice around the sensors. The heat load provided should be accounted in parasitic losses. In Fig. 10, the initial part of the graph shows the filling of the SS tank with liquid nitrogen, The values of the resistance fell gradually from 100 ohms to near 20 ohms when the sensor is submerged in LN2. As the liquid boils off, the sensor will come into contact with the ambient and thus the resistance increases gradually. The graph clearly shows the rise in resistance of each sensor at different times.

CONCLUSION

The design shows that the standard compact recuperators cannot be used as a precooler in this system due to very low LMTD. The experiments revealed that the check valves can withstand the cryogenic temperatures and the working of the DC loop is satisfactory. The pressure drop is found to be more in regenerator and at the check valves showing that the effect of the transfer line on pressure drop is minimal. Thus, the system can be employed for the application of remote cooling. Further experiments in high vacuum condition are in progress.

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