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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 the PTR is investigated in this work. Such a system has practical applications in onboard space vehicles for cooling of sensors. The system works on the principle of a DC flow circuit and consists of a pressure wave generator, counter flow heat exchanger, check valves and heat exchangers at the cold head of the PTR and the load (DTC). In the experimental test rig, the cold head is simulated with a liquid nitrogen bath and the load with a resistance heater. Results of the numerical parametric studies and comparison with the experimental results are reported in this paper.

1. Introduction

Cryocoolers are mechanical refrigerators to cool systems to cryogenic temperatures. These stand-alone devices are very useful for cooling sensors viz. Infra-Red (IR) detectors in satellites. Gifford-McMahon (G-M), Split Stirling and Pulse Tube Refrigerators (PTR) are some of the candidates for these types of applications. Of the above, PTRs which have no moving parts at cryogenic temperatures are the most durable refrigerators for space applications. The evolution of PTRs was described in [1]. PTRs are classified into two broad categories: G-M type and Stirling type PTRs. G-M type refrigerators operate at very low frequencies (1-6 Hz) whereas Stirling type coolers operate at frequencies in the range of 30-70 Hz. The G-M type refrigerators require bulk and heavy compressors to operate and are not suitable for space applications. Typically, a Stirling type PTR has a refrigeration capacity of 1 W @ 80 K could be achieved with a linear motor compressor of about 50 W and is sufficient for the application for cooling the IR detectors to the operating temperatures. But the difficulty in transferring the cold from the PTR to the sensor needs to be addressed. Usually, the sensor is mounted at the cold heat exchanger of the PTR. This results in housing the cryocooler very close to the sensor. A cold transportation system viz. Cryogenic Heat Pipe (CHP) could be one of the solutions for this. Research and development on Cryogenic Heat Pipes (CHP) are going on worldwide. Some of the major difficulties involved in the design and operation of CHP are the 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 the 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 fluid to liquid form before starting the primary heat pipe. Also, the operation of the CHP against gravity is another major challenge. Our proposal of a passive cold transportation system can take-up the role of CHP. This also helps in 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 cryogenics systems like cryoprobes. For example, in a cryoprobe, a separate transportation system with its own equipment, including the compressor, is incorporated. In this system, 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 NMR Technology. A remote cooling loop was reported earlier by Raab et al. in [2]. This work shows 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 proposed system uses a twin pressure wave generator (PWG) to drive the PTR and the flow circuit.

2. Design and Working

A twin PWG drives a pulse tube at one end and produces pressure pulses in the circuit at the other end, which push the gas through the one-way valve. The proposed system is illustrated in Figure 1. The cold transportation 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 operation of a passive cold transportation device can handle the transfer of cold from the cold head of the PTR to DTC very easily. When it is successfully developed and implemented, it will 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 needed to operate the system.

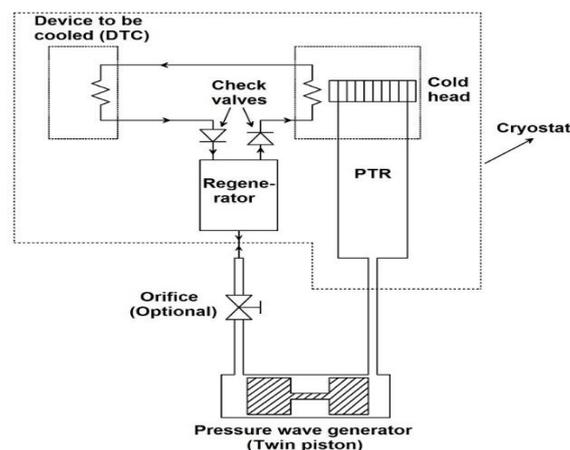


Figure 1: Schematic of the cold transportation system.

The temperature and pressure drop in the flow circuit are to be analyzed at each component and care must be taken care 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 pulse tube refrigerator to be used in the system is of a 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 pre-cooler is carried out to study the influence of the geometrical and operational parameters. For the piping, consideration will be given to materials like metals and plastics considering their strengths to withstand high pressure and low temperature. A metering valve will be used to control the flow in the circuit.

The compact recuperators 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. It is due to the low temperature difference between the hot and cold streams. Various recuperators were designed to meet the requirement of pre-cooling to a temperature of 80.5 K and the length of each exchanger are presented in Table.1. The HX with mesh on both sides 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 opted out. Hence, the regenerator is proposed to be used in place of the counterflow heat exchanger. The flow is converted to DC by the check valves after the regenerator. The following section discusses the design of regenerator.

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

3. Design of regenerator

The properties of the helium gas are calculated from the equation of state and [3], at the average temperature of the regenerator T_{avg} . The correlations for the Nusselt number were taken from Gedeon [4] and the values of G_h and Re_h were calculated accordingly. The pressure drop correlations were given in Miyabe [5] and the Reynolds number and mass velocity used in the equation are considered differently from those by Gedeon and hence are calculated as G_p and Re_p . Barron [6] has taken the Number of Transfer Units, NTU_0 and matrix capacity ratio, C_m equations from the work by Coppage and London [7] and the tabular data from Johnson [8]. It is mentioned earlier that the Coppage and London method is to be used for rotary regenerators and Hausen's [9] method is to be used for fixed bed regenerators. However, Shah [10] has reported that both the methods are related and either of the methods can be employed for the fixed bed calculations.

Nusselt number correlation from [4] is $Nu = (1 + 0.99Pe^{0.66})\psi^{1.79}$

Friction factor and pressure drop correlations from [5] are $FF = 0.337 + \left(\frac{33.6}{Re_p}\right)$ and $\Delta P = \frac{(NS * \rho * FF * V^2)}{2}$

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

Coppage and London [5] mentioned that, for values of C_m above 10, the ratio can be considered as ∞ . Hence the calculations were initiated with $C_m = 10$ and are performed iteratively to get the required NTU. The length and pressure drop for a set of C_m values for each diameter of regenerator are determined. A value of $C_m = 180$ was selected as a sample for calculation and to observe the trends of temperature and pressure drop and the results are plotted.

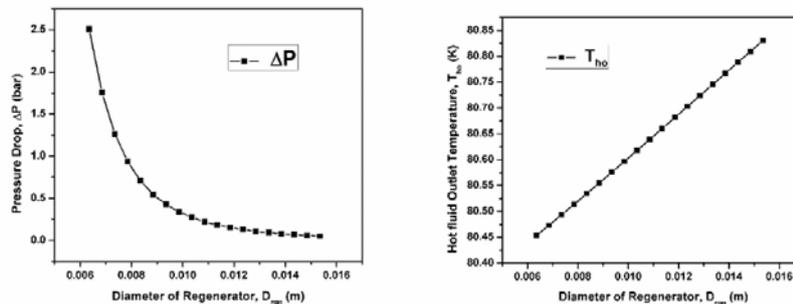


Figure 2: Effect of Regenerator diameter (D_{rgn}) on Pressure drop (P) and hot fluid outlet temperature (T_{ho}).

The plots showing the variation of pressure drop and hot fluid outlet temperature with diameter of the regenerator were shown in Figure 2. It can be observed that the 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 the 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.

4. Experimentation

Figure 3 shows the experimental test rig of the DC Loop for the cold transportation. The main aim of this experiment is to test the validation of the computer program and the operation of check valves at cryogenic temperatures. An existing regenerator and a small compressor were coupled to the DC loop which is passed through the LN_2 bath for cooling. The heat ingress is considered as the thermal load. The dimensions of the regenerator were fed to the program which yielded the result. The length of the regenerator is 64 mm and the diameter is 9.5mm. the porosity of the mesh used is 0.661. The charging pressure is 8 bar.

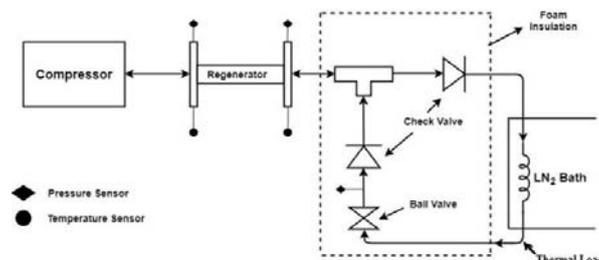


Figure 3: Experimental Test Rig.

Figure 4(a) shows the trend of temperature at the entry and exit of regenerator with time. It is observed that there is an initial rise in temperature at the entry of the regenerator. The initial rise 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. There is neither increase nor a decrease in the temperature. 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 type of heat exchanger used in the circuit needs a larger capacity compressor. The pressure change at regenerator inlet, outlet and at the DC loop were acquired and a steady decrease in the mean pressure over the entire circuit is observed as shown in Figure 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 the check valves at the cryogenic temperatures.

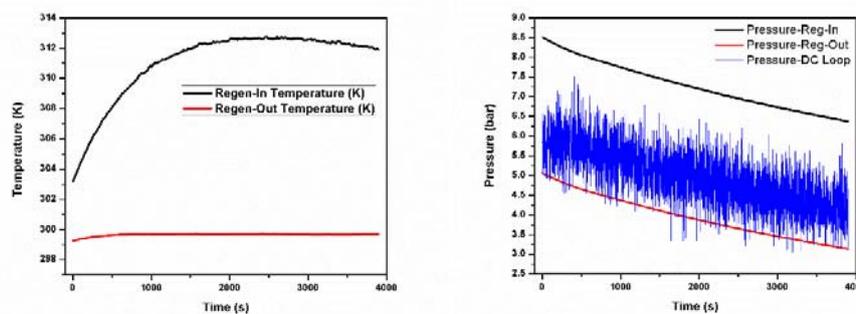


Figure 4: Experimental Results. (a) Time Vs Temperature of regenerator, (b) Time Vs Pressure of regenerator and loop.

5. Conclusion

The design had shown that the standard compact recuperators cannot be used as a precooler in this system due to very low LMTD. The experiments conclude that the check valves can withstand the cryogenic temperatures and thus, this system can be employed for the application of remote cooling. Efforts are underway to employ a higher capacity compressor.

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