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Theoretical and Experimental Studies on a Two Stagepulse Tube Refrigerator

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Abstract—***Pulse Tube Refrigerators (PTRs) are devices with no moving parts at cryogenic temperatures and produce the needed refrigeration power at the specific cryogenic temperature. Staging of the Pulse Tubes leads to lower temperatures and hence two stage PTRs have been used for a variety of applications such as cooling of superconducting magnets, re-condensation of liquid helium, cryosorption pumps etc. Although two stage PTRs are available commercially and significant developments have occurred in this field, the processes responsible for the production of refrigeration in these systems are not fully understood. Different models such as enthalpy flow theory, models based on thermoacoustic theory, isothermal model etc. have been proposed over the years for predicting the performances of the PTRs.***

A two stage PTR operated in the double inlet configuration has been developed in our laboratory which produces a no load temperature of ~ 3K and ~65 K in its 2nd stage and second stage cold ends respectively. We present here the theoretical and experimental studies carried out on this system. The numerical analysis of this system has been carried out assuming an adiabatic gas flow and sinusoidal volume variations of the compressor. Optimized performance of PTR has been arrived at byvarying the orifice and double inlet valves for obtaining the maximum cooling powers at the respective stages. The experimentally measured cooling powers at the 2nd stage cold end of the PTR are found to be in reasonably good agreement with theoretical predictions by the adiabatic numerical model. The above studies of two stage PTR are presented in detail in this work.

Keywords— Two stage Pulse Tube Refrigerator, Numerical model, Regenerator, orifice, double inlet

I. INTRODUCTION

The cryocoolers (also known as cryo refrigerators) are used to obtain a known amount of refrigeration power at a specific low temperature of the several types of coolers such as GM, Sterling, pulse tube and JT. Pulse tube refrigerators(PTRs) have gained significant importance over the last two decades in view of the absence of moving parts at cryogenic temperature range. Further PTRs are suitable for long term performance with improved mechanical stability. Multi-staging of pulse tube cryocoolers is utilized to obtain lower and lower temperatures, which are useful for applications such as re-condensation, cooling of superconducting magnets, cryopumps etc. Although commercial PTRs may be procured from different sources at a high cost , the fundamental processes responsible for producing the refrigeration in pulse tube system are not fully understood. Several theoretical models such as enthalpy flow theory, thermo-acoustic theory etc. have been proposed for predicting the performance of pulse tubes. However, all the experimental features are not explained by any single theory. In view of the above we have attempted a numerical model based on adiabatic gas flow through the pulse tube.

We present here the design and development of a 2 stage PTR which achieves a no load temperature of 3.3K in the 2nd stage cold end. Further the numerical modelling based on adiabatic flow through the pulse tube has been attempted to predict the performance of the PTR. The refrigeration powers predicted by the model are compared with the measured experimental data. The details of the above are presented in the manuscript.

II. DETAILS OF EXPERIMENTAL SETUP

The schematic of the two stage PTR and the fabricated setup are shown in Figures 1 (a) and (b) respectively. The sizes of the pulse tube and regenerator are given in Table 1. Both the pulse tube and regenerator were made of thin walled stainless steel tubes with end flanges. The 1st stage regenerator was prepared using meshes of stainless steel (size 200), while the second stage regenerator was fabricated with lead powder (grain size 250mm) and Er3Ni (grain size 0.3 to 0.5mm). The room temperature seals were made by o-rings, while the low temperature seals were made by Indium. The heat exchangers were fabricated from electrolytic grade copper with slit type arrangement. The heat removal at the warm end was performed by utilizing copper heat exchanger with external fins. Both the pulsetube and regenerator were insulated with aluminized Mylar to take care of radiation heattransfer. These PTR is housed in a vacuum jacket to minimize a heat transfer by gas conduction. The 1ststage refrigeration power is used as a radiation shield for the second stage.

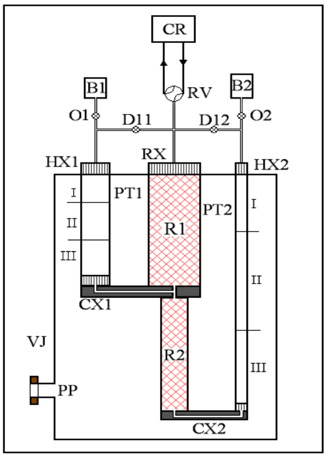

Fig. 1 (a) Schematic of 2 stage PTR.   Fig. 1 (b) Assembled setup of 2 stage PTR.

The temperatures at different locations above 40K were measured by Platinum resistance thermometer Pt500. The cool end temperature at the second stage was measured with a pre-adjusted silicon diode (DT470/DT410) sensor. The cooling powers produced at the second stage cold end have been measured as follows. The 2nd stage cold end is fitted with a heater of low wattage. This can be energized by applying a known voltage which allows the current flow in accordance with the resistance of the heater. Initially the pulse tube system is allowed to reach the no-load temperature. Subsequently a known amount of voltage is applied to the heater mounted on the 2nd stage cold end. This allows a current flow through the heater wire inversely proportional to its resistance. The heat load applied through the heater is equal to product of the applied voltage and current. For different heat loads applied to the cold end the steady state temperatures achieved on the 2nd stage cold end are measured. A plot of the applied heat load as a function of the cold end temperature gives the cooling power produced by the cryocooler.

The system uses Swagelok M-series needle valves as orifice and double inlet valves. The pressure wave was generated using an indigenous rotary valve with a 6kW water-cooled helium compressor. The pressure ratio obtained from the above was 22.5/6.8 at an operating frequency of ≅1.8Hz. High purity helium gas(99.999% pure) is used as the working fluid in the pulse tube system. The data acquisition for the measurements of temperatures and cooling powers were carried out using a LABVIEW based program. The cool down behaviour and the cooling power characteristics of the 2 stage PTR are discussed in the Results and Discussion section. In the following, we present the numerical modelling based on adiabatic behaviour of gas flow through the pulse tube system.

TABLE I: DETAILS OF EXPERIMENTAL PULSE TUBE CONFIGURATIONS.

All dimensions in mm	**Pulse Tube**		**Regenerator**		**Buffer volume**	**Regenerator materials**
	Diameter	**Length**	**Diameter**	**Length**		**SS - Stainless Steel meshes (size 200mm)**
First stage	Ø19	270	Ø 25	200	0.5 liters	SS + Lead (85%+15%)
Second stage	Ø14	190	Ø 19	390	0.5 liters	HoCu2+Lead+SS (30%+40%+30%)

III. NUMERICAL ANALYSIS

A numerical model based on the adiabatic gas flow behaviour in the pulse tube has been developed for the 2 stage PTR. Some of the assumptions used in this model are as follows. A) Ideal gas behaviour is assumed for the working fluid (in the present case helium is the working fluid). B) The flow in the system is assumed to be one dimensional. C) There are no pressure drops in the entire system. D) The effectiveness of the regenerator and the heat exchangers is 100%. E) The orifice and double inlet valve flows are assumed to be adiabatic. The methodology of the numerical modelling is as follows.

Since helium is treated as an ideal gas we can write $PV = mRT$. This is used to obtain the time rate variations of pressure, mass and volume for the pulse tube, the compressor and the reservoir. Initially the mass flow rate through the regenerator is obtained through Ergun’s equation. When the mass flow rate through the double inlet valve is added to this, we obtain the compressor mass flow rate. The mass flow rate through the orifice and the double inlet valve are obtained using nozzle equations. The sum of hot end mass flow rate and the double inlet mass flow rate gives the mass flow rate through the orifice. The detailed expressions for the Ergun's equations, nozzle equations, time rate equations for pressure, mass and volume are given in reference [3] and hence not presented here.

Starting with an initial values of pressure of the compressor, pulse tube and reservoir at time t=0, 4th order Runge-Kutta method is used to obtain the pressure values at the next time interval. The process is continued till one complete cycle is completed. Several such cycles are run to obtain consistent values through a given cycle. The refrigeration power produced by the cryocooler is predicted by the model as

$$Q = \frac{1}{\tau}\int_0^\tau [T_c * C_p * \dot{M}_c]\, dt \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

In the above equation, Tc is the temperature of the cold end and $\dot{M}_c$ is the mass flow rate through the cold end. Cp is the specific heat of the working fluid. This can be compared with the experimentally measured cooling powers at the second stage cold end. In the following we present the typical results obtained by the numerical model.

IV. Results and Discussions

Both the experimental outcomes and results of numerical analysis are presented in this part along with the comparison between them wherever possible. Initially we discuss typical characteristic plots obtained from the numerical model for the variations of pressure, mass flow rate and volumes. Subsequently we discuss the cool down behaviour of the pulse tube followed by a discussion on the cooling powers produced by the cryocooler.

Typical characteristic plots from Numerical analysis:

A. Typical variation of the pressures:

Figure 2 (a) shows the pressure variation in the compressor, pulse tube and reservoir as a function of time. Depending on the frequency of oscillations one observes a specific number of cycles of these pressures. The compressor pressure amplitude is found to be higher when compared to that of the pulse tube. The reservoir pressure variations are very small when compared to those of the compressor and pulse tube. The reservoir pressure variations will be of smaller amplitude when the reservoir volume is increased.

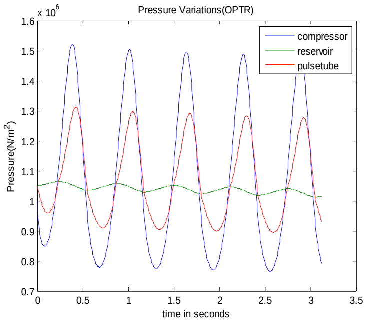

Fig. 2 (a) Plot of pressure variations

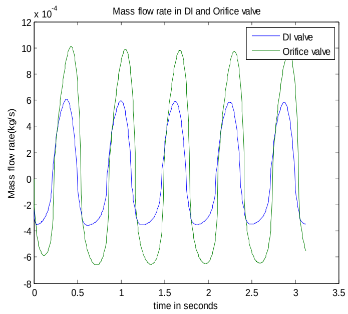

Fig. 2 (b) Plot of mass flow rate variations

B. Typical variation of the mass flow rates

Figure 2 (b) plots the typical mass flow rates through Double Inlet and Orifice. As can be seen from the figure Double Inlet mass flow rate is much smaller compared to that of the Orifice. Although the observed mass flow rates are both positive and negative (indicating flow on either direction). The net mass flow rate is calculated based on absolute values.

C. Typical variation of volumes with respect to pressure

Figure 3 plots the P-V diagram of the Compressor and the 2nd stage cold end. The area within the curves represents the work requirement in each case. P-V curve of compressor refer to compressor work. While P-V diagram of 2nd stage cold end will be proportional to the 2nd stage cooling power. Obviously the area under the P-V curve of 2nd stage cold end is lesser than that of the compressor work.

Figure 4 shows the Cool down behaviour of both the 1st stage and the 2nd stage. Figure 5 shows how cooling power varies when the temperature of 2nd stage cold end is varied. From Figure 4.4 it observed that the steady state no load temperatures accomplished is ~3.3 K and ~65 K at the second and first stage cold end respectively. In Figure 4.5 we see that the no –load temperature reached is 3.3K and the system produces a cooling power of ≅4.7W at 35K.

D. Comparison of experimental and the theoretical cooling powers

The experimentally measured 2nd stage cooling powers are compared with those predicted by the numerical model in Figure 6. The figure shows the reasonably good agreement between the experimental and the theoretical cooling powers. It is observed that the theoretical predictions are somewhat higher than the experimental values. It should however be noted that the numerical model does not consider the various losses of cooling power occurring in the pulse tube such as shuttle losses, axial conduction, radiation etc. If these losses are taken into account perhaps there may be very good agreement between the theory and the experimental data.

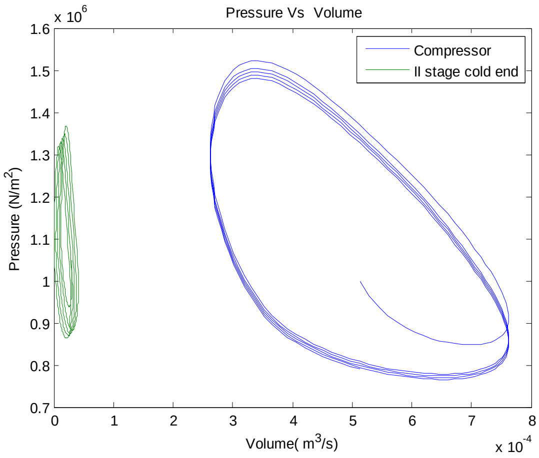

Fig. 3 Plot of pressure vs volume for compressor and cold end. Cool down behaviour and Experimental Cooling powers:

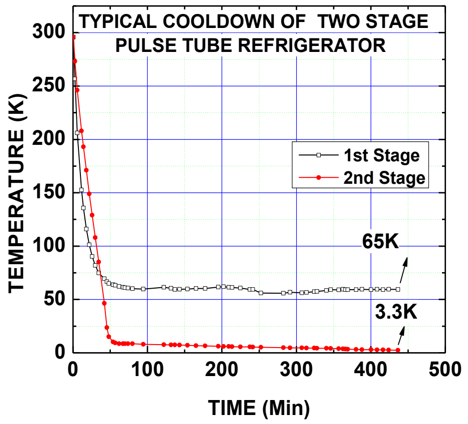

Fig. 4 Typical Cool down of 2 stage PTR

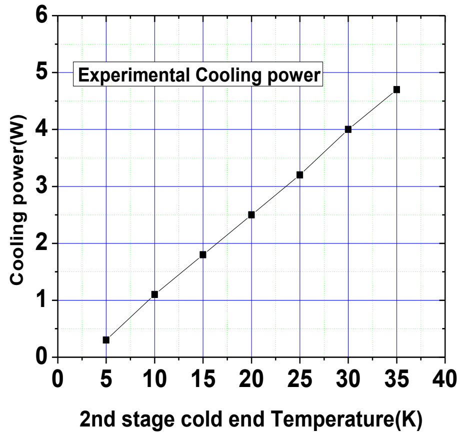

Fig. 5 Plot of Experimental Cooling power

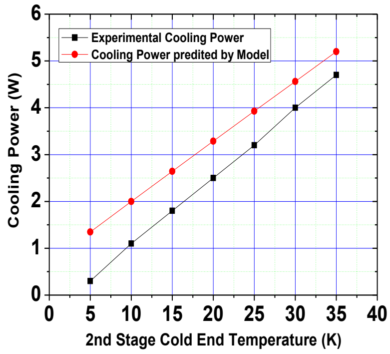

Fig. 6 Cooling powers comparison

V. CONCLUSION

In this work, the development of a two stage PTR which produces a no load temperature of ~ 3.3K and ~65 K in its 2^{nd} stage and the first stage cold end respectively is presented. PTR optimization has been carried out by varying the orifice and double inlet valves for obtaining the maximum cooling powers at the respective stages. The cool down behavior and the cooling powers of the experimental system has been studied.

The numerical analysis of this system has been carried out assuming an adiabatic gas flow. An appropriate MATLAB based program has been developed for solving the different equations in the analysis. The

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experimentally measured cooling powers at the 2nd stage cold end of the PTR are found to be in reasonably good agreement with theoretical predictions by the adiabatic numerical model. The present numerical study has given a better understanding of the processes occurring in the pulse tube system. A more detailed analysis including the losses will however be necessary to have exact predictions of the performances of two stage PTR.

VI. ACKNOWLEDGEMENTS

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