A

Project Report

On

THEORETICAL AND EXPERIMENTAL STUDIES ON PULSE TUBE REFRIGERATOR

Submitted by:

Mr. Vivek Kumar Rawat

Roll No. 10503068

Guided by:

Prof. R. K. Sahoo

Mechanical Engineering

NIT Rourkela

Mechanical Engineering Department

National Institute of Technology Rourkela

Rourkela, Orissa - 769008

CERTIFICATE

This is to certify that the thesis entitled "Theoretical and Experimental studies on Pulse Tube Refrigerator", being submitted by Mr. Vivek Kumar Rawat, in the partial fulfillment of the requirement for the award of the degree of B. Tech in Mechanical Engineering, is a record of bona fide research carried out by him at the Department of Mechanical Engineering, National Institute of Technology Rourkela, under our guidance and supervision.

Prof. R. K. Sahoo

Mechanical Engineering

NIT Rourkela

Date:

ACKNOWLEDGEMENT

I would like to express my sincere gratitude to my guide **Prof. R. K. Sahoo** for his invaluable guidance and steadfast support during the course of this project work. Fruitful and rewarding discussions with him on numerous occasions have made this work possible. It has been a great pleasure for me to work under his guidance.

I am also indebt to **Mr. Bishwanath Mukharjee** of cryogenics laboratory for his technical support during the experiments which has made this work possible.

I would like to express my sincere thanks to all the faculty members of Mechanical Engineering Department for their kind co-operation.

I would like to acknowledge the assistance of all my friends in the process of completing this work.

Finally, I acknowledge my sincere gratitude to my family members for their constant encouragement and support.

Vivek Kumar Rawat

ABSTRACT

The absence of moving components at low temperature end gives the pulse tube refrigerator (PTR) a great leverage over other cryo-coolers like Stirling and GM refrigerators that are conventionally in use for several decades. PTR has greater reliability; no electric motors to cause electromagnetic interference, no sources of mechanical vibration in the cold head and no clearance seal between piston and cylinder. Moreover, it is a relatively low cost device with a simple yet compact design.

The objectives of the present work are to 1) understand the basic phenomena responsible for the production of cold effect with the help of simple theoretical models based on ideal behavior of gases and to 2) test a single stage GM type pulse tube refrigerator present in the cryogenics lab of Mechanical Engineering Department of NIT Rourkela. Experimental studies consist of cooling behavior of the refrigeration system and suggesting modifications to improve the performance of the PTR.

CONTENTS

CHAPTER 1

INTRODUCTION

1.1 CRYOCOOLERS

Cryocoolers are small refrigerators that can reach cryogenic temperatures and provide refrigeration in the temperature range 10 K to 120 K. The use of cryocoolers has been propelled by many necessities of modern day applications such as adequate refrigeration at specified temperature with low power input, long lifetime, reliable and maintenance free operation with minimum vibration and noise, compactness, and lightweight. The requirements imposed in each of these applications have been difficult to meet and have been the impetus for considerable research in the field of cryocoolers for the past forty years.

Typical applications of cryocoolers are:

- Liquefaction of gases such as nitrogen, oxygen, hydrogen, helium, natural gas
- Cooling of super-conducting magnets
- Cooling of infra-red sensors for missile guidance
- Cryo vacuum pumps
- SQUID magnetometers
- Gamma ray sensors for monitoring nuclear activity
- Cooling of high temperature superconductors and semiconductors
- Cryosurgery
- Preservation of biological materials, blood, biological specimens etc.

1.2 CLASSIFICATION OF CRYOCOOLERS

Cryocoolers are classified into two types based on the type of heat exchanger used:

(1) Recuperative cryocoolers and (2) Regenerative cryocoolers.

Recuperative types, as shown in figure 1.1, utilize continuous flow of the refrigerant in one direction, analogous to a DC electrical system. As a result, the compressor and expander must have inlet and outlet valves to control the flow direction, unless rotary or turbine compressor or expanders are used. The recuperative heat exchangers have two or more separate flow channels. The performance of the recuperative cryocoolers is dependent on the properties of the working fluids used. Also, the maximum exergy loss in most cryocoolers occurs in the compressor. The main advantage of DC cryocoolers, however, is that they can be scaled to any size (up to few MW of refrigeration).

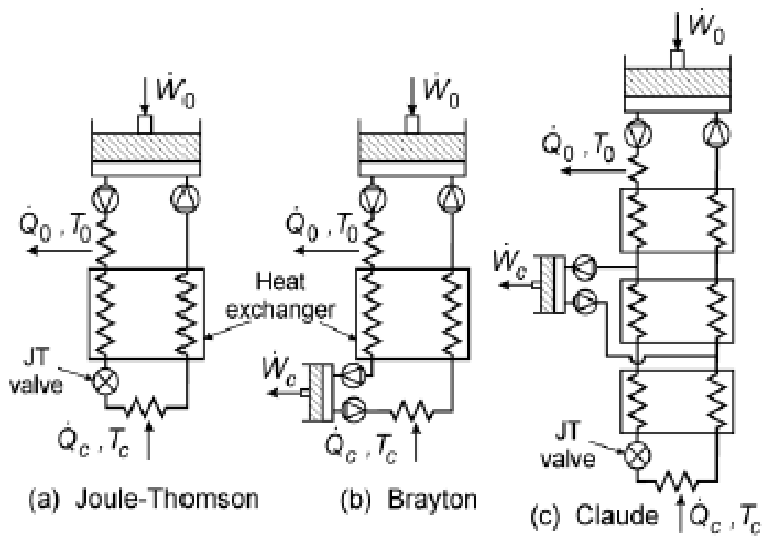

Fig. 1.1: Schematic of Recuperative Cryocoolers

In regenerative types, as shown in figure 1.2, the refrigerant undergoes an oscillating flow or an oscillating pressure analogous to an AC electrical system. The compressor and the pressure oscillator for the regenerative cycles need no inlet or outlet valves. The regenerator has only one flow channel, and the heat is stored for a half cycle in the regenerator matrix, which must have a high heat capacity. The performance of the regenerative type cryocoolers is dependent on the phase difference between the pressure and mass flow rate phasors. Helium is the refrigerant of choice for most regenerative type cryocoolers. The main disadvantage with regenerative type cryocoolers is that they cannot be scaled to large sizes like the recuperative cryocoolers.

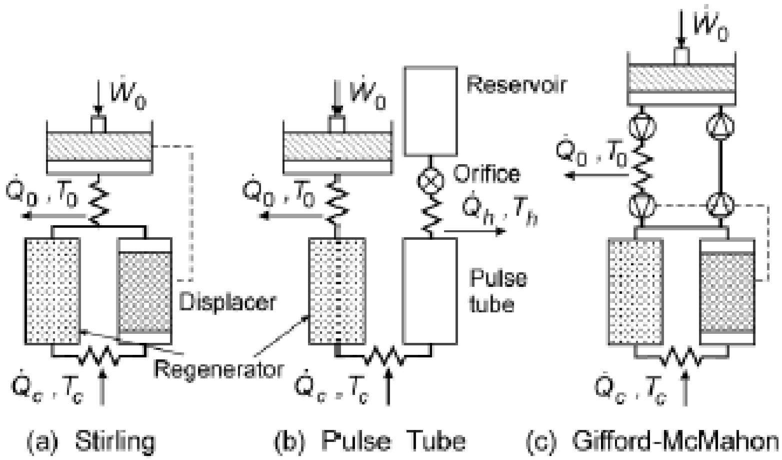

Fig.1.2: Schematic of Regenerative Cryocoolers

1.3 PULSE TUBE REFRIGERATORS (PTRs)

The moving displacer in the Stirling and Gifford-McMahon refrigerators has several disadvantages. It is a source of vibration, has a short lifetime, and contributes to axial heat conduction as well as to a shuttle heat loss. In the pulse tube refrigerator, shown in Figure 1.2b, the displacer is eliminated. The proper gas motion in phase with the pressure is achieved by the use of an orifice and a reservoir volume to store the gas during a half cycle. The reservoir volume is large enough that negligible pressure oscillation occurs in it during the oscillating flow. The oscillating flow through the orifice separates the heating and cooling effects just as the displacer does for the Stirling and Gifford-McMahon refrigerators. The orifice pulse tube refrigerator (OPTR) operates ideally with adiabatic compression and expansion in the pulse tube.

The four steps in the cycle are as follows.

1. The piston moves down to compress the gas (Helium) in the pulse tube.
2. Because this heated, compressed gas is at a higher pressure than the average in the reservoir, it flows through the orifice into the reservoir and exchanges heat with the

ambient through the heat exchanger at the warm end of the pulse tube. The flow stops when the pressure in the pulse tube is reduced to the average pressure.

3. The piston moves up and expands the gas adiabatically in the pulse tube.
4. This cold, low-pressure gas in the pulse tube is forced toward the cold end by the gas flow from the reservoir into the pulse tube through the orifice. As the cold gas flows through the heat exchanger at the cold end of the pulse tube it picks up heat from the object being cooled. The flow stops when the pressure in the pulse tube increases to the average pressure. The cycle then repeats.

The function of the regenerator is the same as in the Stirling and Gifford-McMahon refrigerators in that it precools the incoming high-pressure gas before it reaches the cold end. The function of the pulse tube is to insulate the processes at its two ends. That is, it must be large enough that gas flowing from the warm end traverses only part way through the pulse tube before flow is reversed. Likewise, flow in from the cold end never reaches the warm end. Gas in the middle portion of the pulse tube never leaves the pulse tube and forms a temperature gradient that insulates the two ends. Roughly speaking, the gas in the pulse tube is divided into three segments, with the middle segment acting like a displacer but consisting of gas rather than a solid material. For this gas plug to effectively insulate the two ends of the pulse tube, turbulence in the pulse tube must be minimized. Thus, flow straightening at the two ends is crucial to the successful operation of the pulse tube refrigerator.

1.4 ANALYSIS OF PULSE TUBE REFRIGERATORS

1.4.1 Enthalpy and Entropy Flow Model

The refrigeration power of the PTR is derived using the First and Second Laws of Thermodynamics for an open system. Because of the oscillating flow the expressions are simplified if averages over one cycle are made. Even though the time-averaged mass flow rate is zero, other time-averaged quantities, such as enthalpy flow, entropy flow, etc., will have nonzero values in general. We define positive flow to be in the direction from the compressor to the orifice. The First Law balance for the cold section is shown in Figure 1.3. No work is extracted from the cold end, so the heat absorbed under steady state conditions at the cold end is given by

$$Q_c = \langle H \rangle - \langle H_r \rangle \tag{1.1}$$

where <H> is the time-averaged enthalpy flow in the pulse tube, and $\langle H_r \rangle$ is the time-averaged enthalpy flow in the regenerator, which is zero for a perfect regenerator and an ideal gas. The maximum, or gross, refrigeration power is simply the enthalpy flow in the pulse tube, with the enthalpy flow in the regenerator being considered a loss. Combining the First and Second Laws for a steady-state oscillating system gives the time-averaged enthalpy flow at any location as

$$\langle H \rangle = \langle P_d V \rangle + T_o \langle S \rangle \tag{1.2}$$

where P_d is the dynamic pressure, V is the volume flow rate, T_o is the average temperature of the gas at the location of interest, and is the time-averaged entropy flow. The first term on the right hand side of Eq. (1.2) represents the potential of the gas to do reversible work in reference to the average pressure P_o if an isothermal expansion process occurred at T_o in the gas at that location. Since it is not an actual thermodynamic work term, it is sometimes referred to as the hydrodynamic workflow, hydrodynamic power, or acoustic power. Equation (1.2) shows that the acoustic power can be expressed as an availability or exergy flow with the reference state being P_o and T_o. The specific availability or exergy is given as h- T_os.
Processes within the pulse tube in the ideal case are adiabatic and reversible. In this case entropy remains constant throughout the cycle, which gives

$$\langle S \rangle = 0 \tag{1.3}$$

Equations (1.1) and (1.2) are very general and apply to any oscillating thermodynamic system, even if the flow and pressure are not sinusoidal functions of time. If they are sinusoidal, the acoustic power can be written as

$$\langle P dV \rangle = (1/2) P_1 V_1 \cos\theta = (1/2) R T_0 m_1 (P_1 / P_0) \cos\theta \qquad (1.4)$$

where P_1 is the amplitude of the sinusoidal pressure oscillation, V_1 is the amplitude of the sinusoidal volume flow rate, θ is the phase angle between the flow and the pressure, R is the gas constant per unit mass, and m_1 is the amplitude of the sinusoidal mass flow rate. Equations (1.1-1.3) can be combined to give the maximum or gross refrigeration power in terms of acoustic power as

$$Q_{\max} = \langle P dV \rangle \qquad (1.5)$$

This simple expression is a very general expression and applies to the Stirling and Gifford-McMahon refrigerators as well. In those two refrigerators the acoustic work is converted to actual expansion work by the moving displacer. That work is easily measured by finding the area of the PV diagram. In the case of the pulse tube refrigerator there is no moving displacer to extract the work or to measure a PV diagram. Thus, the volume or mass flow rate must be measured by some flow meter to determine the acoustic power. Such measurements are difficult to perform inside the pulse tube without disturbing the flow and, hence, the refrigeration power. Because there is no heat exchange to the outside along a well-insulated pulse tube, the First Law shows that the time-averaged enthalpy flow through the pulse tube is constant from one end to the other. Then, according to Eq. (1.2) the acoustic power remains constant as long as there are no losses along the pulse tube to generate entropy. The instantaneous flow rate through the orifice is easily determined by measuring the small pressure oscillation in the reservoir and using the ideal-gas law to find the instantaneous mass flow rate. The instantaneous pressure is easily measured in the warm end of the pulse tube, and the product of it and the volume flow is integrated according to Eq. (1.5) to find the acoustic power.

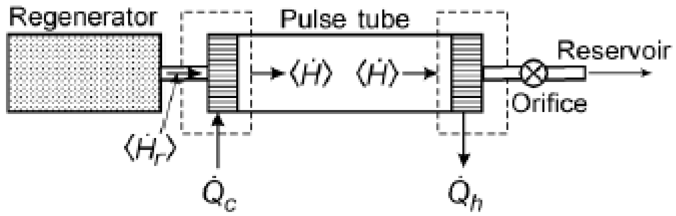

Fig.1.3: First Law Energy balance in PTR

1.4.2 Pulse Tube Losses and Figure of Merit

In an actual pulse tube refrigerator there will be losses in both the regenerator and in the pulse tube. These losses can be subtracted from the gross refrigeration power to find the net refrigeration power. The regenerator loss caused by $\langle H_r \rangle$ is usually the largest loss, and it can be calculated accurately only by complex numerical analysis programs, such as REGEN3.1. The other significant loss is that associated with generation of entropy inside the pulse tube from such effects as (a) instantaneous heat transfer between the gas and the tube wall, (b) mixing of the hot and cold gas segments because of turbulence, (c) acoustic streaming or circulation of the gas within the pulse tube brought about by the oscillating pressure and gas interactions with the wall, and (d) end-effect losses associated with a transition from an adiabatic volume to an isothermal volume. The time-averaged entropy flows associated with (b), (c), and (d) are always negative, that is, flow from pulse tube to compressor. The entropy flow associated with (a) is negative at cryogenic temperatures, where the critical temperature gradient has been exceeded, but is positive at higher temperatures, where the temperature gradient is less than the critical value. For an ideal PTR the figure of merit is defined as

$$FOM = \langle H \rangle / \langle P dV \rangle$$

(1.6)

1.4.3 Effect of Phase between Flow and Pressure

Equation (1.4) shows that for a given pressure amplitude and acoustic power, the mass flow amplitude is minimized for $\theta = 0$. Such a phase occurs at the orifice, that is, the flow is in phase with the pressure. However, because of the volume associated with the pulse tube, the flow at the cold end of the pulse tube then leads the pressure by approximately 30° in a correctly sized pulse tube. The gas volume in the regenerator will cause the flow at the warm end of the regenerator to lead the pressure even further, for example, by 50 to 60°. With this large phase difference the amplitude of mass flow at the warm end of the regenerator must be quite large to transmit a given acoustic power through the regenerator. This large amplitude of mass flow leads to large pressure drops as well as to poor heat exchange in the regenerator. These losses are minimized when the amplitude averaged throughout the regenerator is minimized. This occurs when the flow at the cold end lags the pressure and flow at the warm end leads the pressure.

1.4.4 Intrinsic PTR efficiency

In an ideal PTR the only loss is the irreversible expansion through the orifice. The irreversible entropy generation there is a result of lost work that otherwise could have been recovered and used to help with the compression. All other components are assumed to be perfect, and the working fluid is assumed to be an ideal gas. The *COP* for this ideal PTR is given by

$$COP_{carnot} = Q_c / W_o = T_c / (T_h - T_c) \qquad (1.7)$$

$$COP_{ideal} = Q_c / W_o = \langle P_d V_c \rangle / \langle P_d V_h \rangle = T_c / T_h \qquad (1.8)$$

where Eq. (1.5) was used to relate the refrigeration power to the acoustic power at the cold end. The acoustic power at the hot end of the regenerator is simply the PV power of the compressor. Because the regenerator is assumed to be perfect, the acoustic power varies along its length in accordance with the specific volume, which is proportional to temperature for an ideal gas. The maximum *COP* from Eq. (1.8) is 1.0, but only when the cold temperature becomes equal to the hot temperature. A comparison of the *COP* from Eq. (1.8) with the Carnot *COP* from Eq. (1.7) shows that the only difference is the presence of the T_c term in the denominator of Eq. (1.7). That term represented the work reversibly recovered at the low temperature and used to help in the compression. The Carnot efficiency of the ideal PTR is given by

$$\eta_{ideal} = COP_{ideal} / COP_{carnot} = (T_h - T_c) / T_h \tag{1.9}$$

For T_h = 300 K and Tc = 75 K, ηideal = 0.75. Since practical pulse tube refrigerators have efficiencies less than about 20% of Carnot, the intrinsic loss is dominated by other practical losses when operating at this low temperature. However, for T_c = 250 K, ηideal = 0.17. In that case the lost power at the orifice is a much larger fraction of the total input power. Thus, the PTR cannot compete with the vapor-compression refrigerator for near-ambient operation. It is useful only for much lower temperatures, especially cryogenic temperatures, unless the acoustic power flow at the warm end of the pulse tube is recovered.

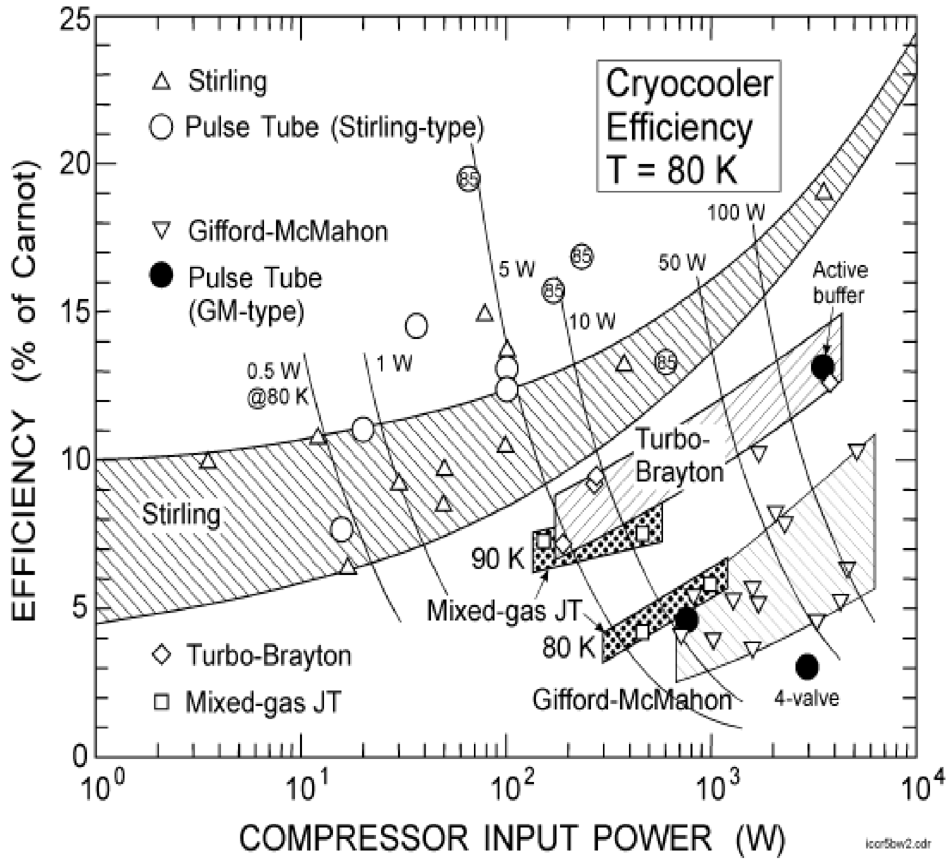

Fig.1.4: Carnot Efficiency of various types of cryocoolers

1.5 OBJECTIVE OF THIS WORK

The objectives of this work is to test and optimize a Pulse Tube Refrigerator and study its performance by varying different parameters like frequency and valve openings and suggest modifications in the present design.

CHAPTER 2

LITERATURE REVIEW

The first pulse tube refrigerator was discovered accidentally at Syracuse University by Gifford and Longsworth in the mid-1960s as they were developing the Gifford McMahon refrigerator. They noticed that the closed end of a pipe became very hot when there was a pressure oscillation inside, whereas the open end toward the compressor was cool. After further studies and optimization of the geometry, they were able to achieve a low temperature of 124 K at one end when the closed end was cooled with water. In their arrangement they used a Gifford-McMahon compressor to drive the system, but there was no orifice or separate reservoir. There was a small reservoir associated with the heat exchanger at the warm end of the pulse tube. Pulse tube diameters were about 20 to 25 mm and operating frequencies were about 1 Hz. This pulse tube arrangement without an orifice is now referred to as the basic pulse tube.

The operating principle of the basic pulse tube refrigerator is entirely different from the orifice type discussed earlier. In the basic pulse tube refrigerator the compression and expansion process inside the pulse tube occurs about halfway between adiabatic and isothermal. Expressed more rigorously, the thermal penetration depth in the gas is comparable to the tube radius. This large boundary layer pumps heat from the cold to the hot end of the pulse tube by a shuttle action of all the gas parcels. A parcel of gas is compressed and moved toward the closed end. At the plateau of the sinusoidal motion there is time for the hot parcel of gas to transfer heat to the adjacent tube wall. Next the parcel of gas is expanded and moved away from the closed end. Near the end of its motion this cooled parcel picks up heat from the adjacent tube wall. That heat is then carried back toward the closed end as the cycle repeats. Other parcels contribute to the heat pumping action all the way from the cold to the hot end. Because heat transfer with the wall is involved, there is a critical temperature gradient where the heat pumping effect goes to zero. At that point the temperature profile of the gas during its movement exactly matches that in the tube wall. If a steeper temperature gradient were imposed on the wall, the gas would shuttle heat from the hot to the cold end. This critical temperature gradient then prevents such a system from reaching cryogenic temperatures. Efficiencies of this arrangement were very poor, and, as a result, little work was done with the basic pulse tube refrigerator after about 1970.

In the early 1980s Wheatley and coworkers at Los Alamos National Laboratory began to investigate heat-pumping effects at frequencies much higher than that used in the basic pulse tube refrigerator. At frequencies of 500 to 1000 Hz resonance would occur in short tubes and lead to a standing wave. At such high frequencies the thermal penetration depth in helium is on the order of 0.1 mm. Proper heat transfer was achieved by the use of closely spaced plates inside the tube. This resonant pulse tube, better known as the thermoacoustic refrigerator also has a low-temperature limit set by the critical temperature gradient because it, also, relies on heat transfer with the solid structure. A low temperature of about 195 K has been achieved with this type of refrigerator. Further research and development on the thermoacoustic refrigerator is continuing for use in near-ambient refrigeration and air conditioning. At the Moscow Bauman Technical Institute in 1984 Mikulin *et al*. introduced an orifice inside the pulse tube near the warm end and achieved a low temperature of 105 K.

In 1985 Radebaugh *et al*. at NIST/Boulder placed the orifice outside the pulse tube, as shown in Figure 2c, to allow the warm heat exchanger to act as a flow straightener. The orifice was a needle valve, which allowed an easy optimization of the flow impedance. A temperature of 60 K was then achieved. Frequencies of 5 to 10 Hz were used in those early studies and were limited by the available valveless compressor. Fundamental studies of the orifice pulse tube refrigerator were carried out at NIST over the next several years to better understand the operating principles of this device. Such studies showed that the orifice pulse tube refrigerator did not rely on heat transfer with the tube wall. In fact, such heat transfer degraded the performance. A simple harmonic model was developed to calculate the time-averaged enthalpy flow in the pulse tube and the resultant refrigeration effect. The model assumed adiabatic conditions inside the pulse tube. In 1990 temperatures below 40K were achieved at NIST and other laboratories.

One of the first pulse tube refrigerators developed for space applications was the miniature double-inlet system developed by Chan *et al*. and discussed previously. It produces 0.5 W at 80 K with 17 W of input electrical power. It is being considered for many different space applications and about 30 have been made to date. Figure 2.1 shows a photograph of this refrigerator. It uses an inline arrangement of the regenerator and pulse tube, which is the most efficient because it reduces the dead volume at the cold end and minimizes turbulence from changing flow directions.

Figure 2.2 shows a recent pulse tube refrigerator developed at NIST for NASA to be used in the laboratory to study the process of liquefying oxygen on Mars. The flight program scheduled for the year 2007 would chemically convert the carbon dioxide atmosphere of Mars into oxygen, after which it is to be liquefied and stored. After about 500 days enough liquid oxygen should be collected to fire rockets for lifting off from Mars and returning to Earth with rock samples. The pulse tube liquefier shown in Figure 2.2 is a coaxial geometry with the pulse tube located inside the annular regenerator. In use, the cold tip points down to eliminate gravitational-induced convection in the pulse tube. The dual-opposed compressor uses flexure bearings and moving coils. It produces 19 W of refrigeration at 90 K with 222 W of input PV power. Though this compressor is only 63% efficient, the system would have a Carnot efficiency of 17% with an 85% efficient compressor.

Recent advances in linear compressor technology at the University of Oxford have led to reduced volumes and masses for a compressor of a given PV power. A recent pulse tube refrigerator using this compressor technology and developed for NASA to cool infrared focal plane arrays to 55 K with 0.5 W of refrigeration power is shown in Figure 2.3. The compressor uses 35 W of input power and has a mass of only 3.6 kg.

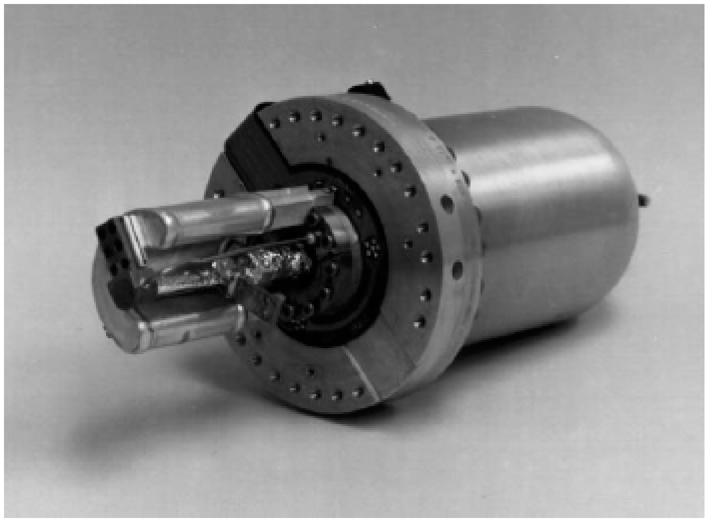

Figure 2.1: Mini pulse tube refrigerator for space applications

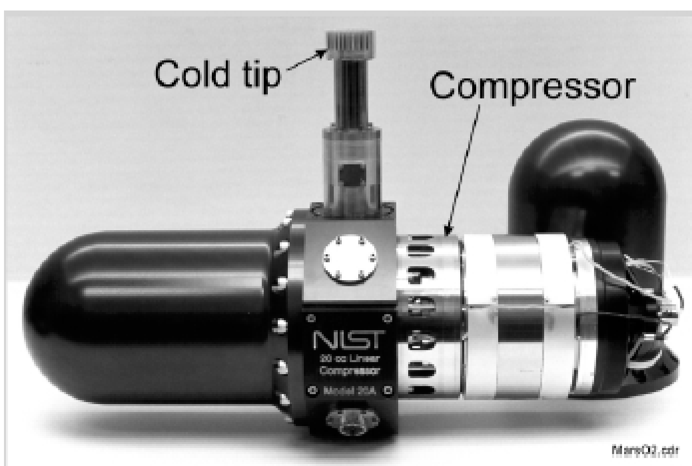

Figure 2.2: Pulse tube refrigerator for studies of oxygen liquefaction on Mars

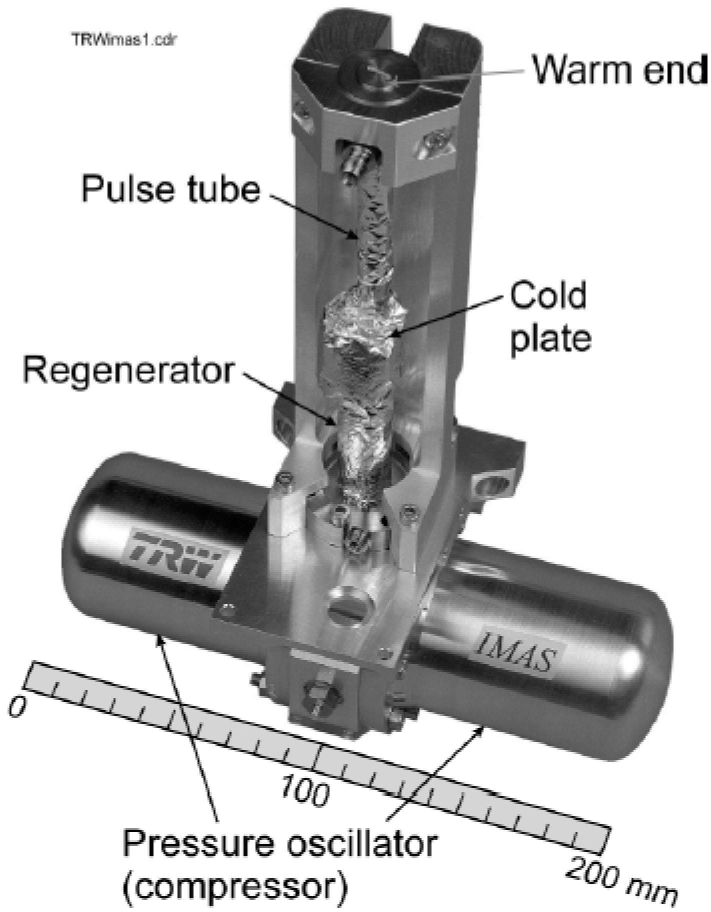

Figure 2.3: Pulse tube refrigerator for Integrated Multispectral Atmospheric Sounder

CONCLUSIONS

In a time span of about 15 years the pulse tube refrigerator and its variations have become the most efficient of all cryocoolers for a given size, even exceeding that of Stirling refrigerators in some cases. Efficiencies above 17% of Carnot have been achieved. They have no moving parts at the cold end, and for large systems can be driven with thermoacoustic drivers that also have no moving parts. The lack of moving parts in the cold end gives them the advantage of less vibration, higher reliability, and lower cost than all other cryocoolers, except for Joule-Thomson refrigerators, which also have no cold moving parts. However, the Joule-Thomson refrigerators currently have lower efficiencies than pulse tube refrigerators, at least for temperatures below about 100 K. Commercial and industrial applications of pulse tube refrigerators are slower to develop because of the need to reduce cost while maintaining high reliability. Nevertheless, at least three companies now sell pulse tube refrigerators for commercial applications. In all three cases the compressors are mostly Gifford-McMahon type compressors, and rotary valves are used to switch between the high- and low-pressure lines. Thus, these commercial systems do not have the high efficiency of the space systems where valveless compressors are used. In a few cases Stirling-type pulse tube refrigerators are being sold commercially for high efficiency applications. So far most of the development of pulse tube refrigerators has been for rather small systems with less than a few watts of cooling at 80 K or lower. Recently there has been much more interest in pulse tube refrigerators for industrial applications in gas liquefaction and in power applications of superconductors. In many of these cases refrigeration powers of kilowatts or even tens of kilowatts are required. These are intermediate-size applications and are smaller than the large air and gas liquefaction plants where megawatts of refrigeration power are needed.

At present there is not a clear upper limit to the useful size of pulse tube refrigerators.

The many advances in pulse tube refrigerators in such a short period of time have brought these refrigerators to the point where they are beginning to replace other types of cryocoolers in several applications. With further improvements, especially reduced costs, many other applications are beginning to develop, particularly in the area of superconductivity. Improved cryocoolers are an enabling technology for many cryogenic and superconductor applications. Pulse tube refrigerators now have the potential to be used in many of these applications.

CHAPTER 3

EXPERIMENTAL SET-UP AND PROCEDURE

3.1 DESCRIPTION OF THE EXPERIMENTAL SET-UP

The schematic of the experimental set-up is as shown in the figure 3.1. The whole experimental set-up can be divided into four units namely the compressor unit, the pressure wave generating unit, the cold box unit and the data acquisition system. The compressor unit consists of the compressor, the after cooler and the oil filters. The low-pressure working fluid is compressed to a high pressure in the compressor. The after cooler removes the heat of compression and brings the working fluid to near-ambient temperature. The working fluid then passes through the oil-filters where the oil and other fine impurities are removed. The suction and discharge ends are connected to a rotary valve that consists of a synchronous motor that is actuated by an electrical frequency varying unit.

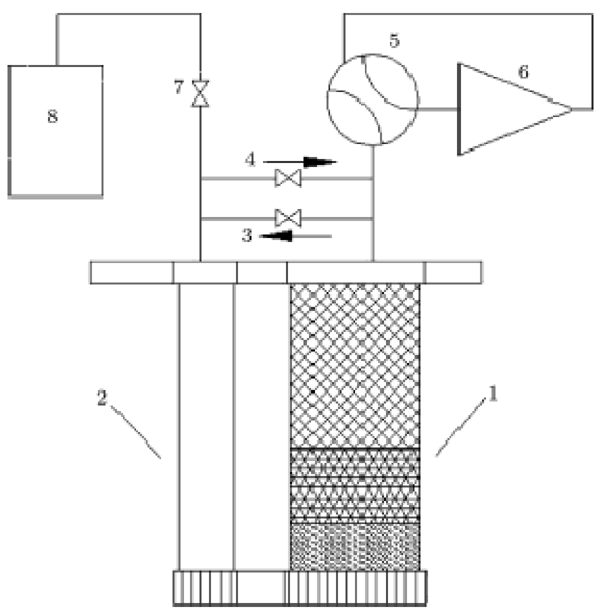

Fig 3.1: Schematic of the experimental set-up

The regenerator and pulse tube are connected to these valves as shown in figure 3.1. The regenerator, pulse tube and the heat exchangers are placed in an evacuated vessel, normally known as the cryostat or the cold box. The cold end of the pulse tube has provisions for fixing a heater wire and a Platinum resistance thermometer (PT 100). In the pulse tube heat is intermittently transferred from the cold end to the warm end heat exchanger Cold water, cooled in a water bath, cools the warm end heat exchanger. A buffer volume is attached to the warm end of the pulse tube through a metering valve. The cryostat is maintained at a pressure of 10^{-4} mbar so as to minimize the heat leakage from the ambient. Multilayer insulation is used to reduce the heat leak from ambient. A number of pressure transducers and temperature sensors have been located at critical positions in the set-up. All electronic sensors are connected to the data acquisition system, which helps to digitally monitor and store data.

3.2 COMPONENT DESCRIPTION

3.2.1 Compressor Unit

The compressor used is single-stage oil lubricated hermetically sealed reciprocating compressor. A thermal overload protector, attached to the compressor prevents overloading by switching off upon overheating. The compressor is water cooled to remove the large amount of heat produced during compression of Helium. The oil used in the compressor may get carried over to the cold box where it can freeze and foul the pipes, pulse tube and the regenerator. Superior quality coalescing filters followed by activated charcoal bed have been used. The oil filters are of Domnick Hunter, UK make. Oil, collected in the oil filters, is periodically sent back through a solenoid valve. Solenoid operated automatic by-pass valve has also been provided.

Table 3.1: Specifications of compressor

Make	Kirloskar Copeland Limited
Suction Pressure	6 bar
Discharge Pressure	28 bar
Flow rate	8 m^3/hr
Voltage rating	400 V – 3 Phase – 50Hz, 3kW
Size	710mm x 585mm x 510mm
Weight	80 kg
Gas	Commercial Helium

3.2.2 Rotary Valves

The rotary valve is one of the critical components of most cryocoolers such as Gifford c-Mahon and pulse tube. It is used to switch between high and low pressures from a helium compressor to the required system. In most commonly used valves the rotor is pressed tightly against the stator and large driving torques are needed.

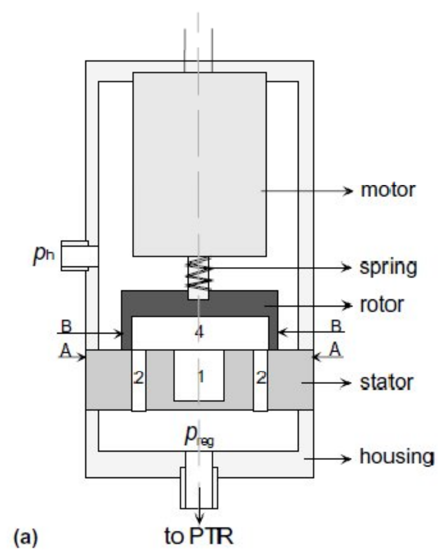

Fig. 3.2 Rotary valve

3.2.3 Electrical Frequency Varying Unit

The electrical frequency varying unit serves the purpose of varying the frequency of the electrical supply to the synchronous motor attached to the rotary valve. Hence it controls the speed of the motor which varies the frequency of the pressure wave fed to the pulse tube. Synchronous speed is given by,

$$N_s = 120*f/p$$

F – frequency of the electrical signal

P – no of poles of motor

It consists of many components like integrator, potentiometers, FET switches, microcontroller etc. integrated on a printed circuit board.

3.2.4 Regenerator

Regenerator was made of stainless steel tube filled with 200 mesh size stainless steel meshes. The tube is of length 200mm and of inner diameter 17.6mm. The tube was machined to reduce its thickness to 0.5mm to reduce axial heat conduction. The meshes were machined by turning process so that they fit exactly into the tube. The meshes were closely packed into the tube. Stainless steel meshes were used because they have high heat capacity and provide a large surface area for heat transfer.

3.2.5 Pulse tube

Pulse tube was made of stainless steel tube of length 300mm and of inner diameter 12.7mm. It was also machined to reduce thickness to 0.5mm. The pulse tube is bounded by a warm-end heat exchanger and a cold-end heat exchanger.

3.2.6 Cold-end Heat Exchanger

It is brazed to the cold end of the pulse tube. Made out of a copper rod it has provisions for Pt 100s and a heater wire. It is packed with Copper meshes (mesh size 200) for enhanced heat transfer between the working fluid and the load.

3.2.7 Warm-end Heat Exchanger

It is connected to the warm end of the pulse tube. It is a wire mesh heat exchanger. It is made of two co-axial stainless steel tubes filled with stainless steel wire meshes. The working fluid flows in the inner tube. Cold water from a water bath is pumped through the shell side of the heat exchanger.

3.2.8 Buffer Volume

It is connected to the warm-end heat exchanger through a metering valve. It is a two liter stainless steel cylinder. The buffer volume was chosen to be greater than ten times the volume of the pulse tube.

3.2.9 Cryostat

The regenerator, pulse tube and the heat exchangers are enclosed in a cryostat and maintained at a vacuum of approximately 0.01 mbar. This almost fully eliminates the infiltration of heat by convection. No part of the cold end should be in contact with the cryostat. This helps avoid heat infiltration.

Table 3.2: Details of Cryostat

Material	SS304
Leak Rate	< 10-6 Torr-lit/sec
Length	484 mm
Diameter	300 mm
Outlets from the flange	4 nos (3/4 ” dia 100 mm long)

3.2.10 RTD Scanner System

The temperature sensor connected to the RTD scanner system, which helps to digitally monitor and store the data. Temperature of the cold end is measured using Platinum resistance thermometer (Pt 100s) for accurate measurements. It was calibrated and used in the four-wire system for improved accuracy. A 24 V DC supply is used as the power source for the transducers. An electrical feed-through has been used to preserve vacuum while making the electrical connections from inside the cryostat to the data-logger. An aluminium transition piece has been fabricated and used for fixing the electrical feed-through to the cryostat.

Fig 3.3: Pictorial view of the Data Acquisition System

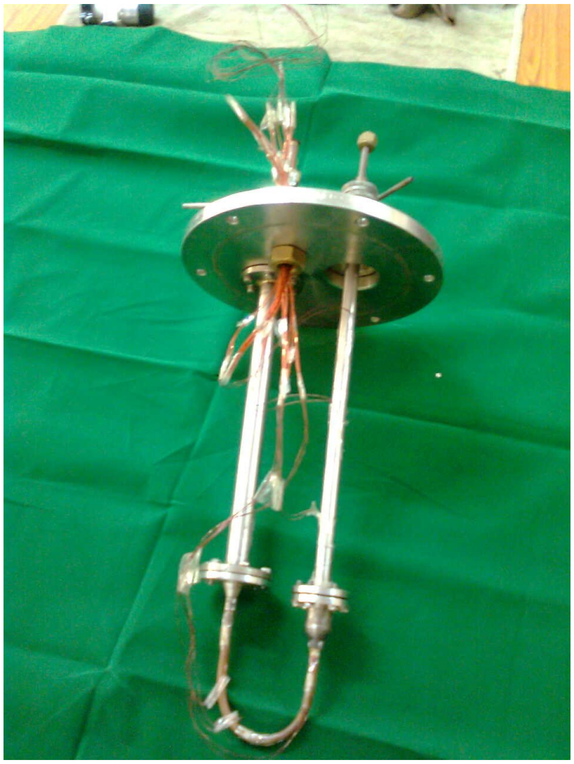

Fig 3.4: Pictorial view of regenerator and pulse tube

Fig 3.5: Pictorial view of electrical unit

Fig 3.6: Pictorial view of rotary valve

Fig 3.7: Pictorial view of compressor

3.3 LEAK TEST

First the system was charged with nitrogen and checked for leaks using soap solution. Later it was filled with small quantity of helium and a helium leak detector was used. Leaks were found in the compressor unit especially in the pressure gauges and in the bypass valve. These components were replaced and all other leaks were plugged so that the leak was minimized to be within the allowable limits.

3.4 EXPERIMENTAL PROCEDURE

The cryostat is connected to the vacuum pump and the vacuum pump is switched on until the vacuum within the cryostat is of the order of 1×10^{-4} mbar. The pulse tube refrigerator is evacuated and then filled with the working fluid to appropriate pressure. The water bath is switched to cool the water. The electrical switching unit is adjusted to produce the required pressure wave. The metering valves are also adjusted according to requirements.

The setup is started by switching on the compressor and the electrical switching unit. The pressure ratio is controlled by adjusting the bypass valve. The pump circulates the water through the warm-end heat exchanger. The setup is run at no load until steady state is achieved. The temperature values are continuously monitored by the data acquisition system.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 GENERATION OF PRESSURE PULSE

The primary task was to create a sinusoidal pressure wave in the regenerator and pulse tube. This was achieved using rotary valves controlled by electrical unit.

Experiments were conducted with optimum pressure range of 5 bar – 12 bar. Frequencies of 1 Hz and 2 Hz were used during experiments

4.2 EFFECT OF VALVE OPENING

The performance of the pulse tube refrigerator primarily depends upon the phase relationship between pressure and mass flow. This phase relationship is dependent on many parameters like pulse tube volume, orifice valve opening, bypass valve opening and frequency of pressure pulse. Several experiments were done to reach an optimum configuration with respect to the no load temperature. Figures 4.1 and 4.2, show the effect of valve opening on the performance of pulse tube refrigerator.

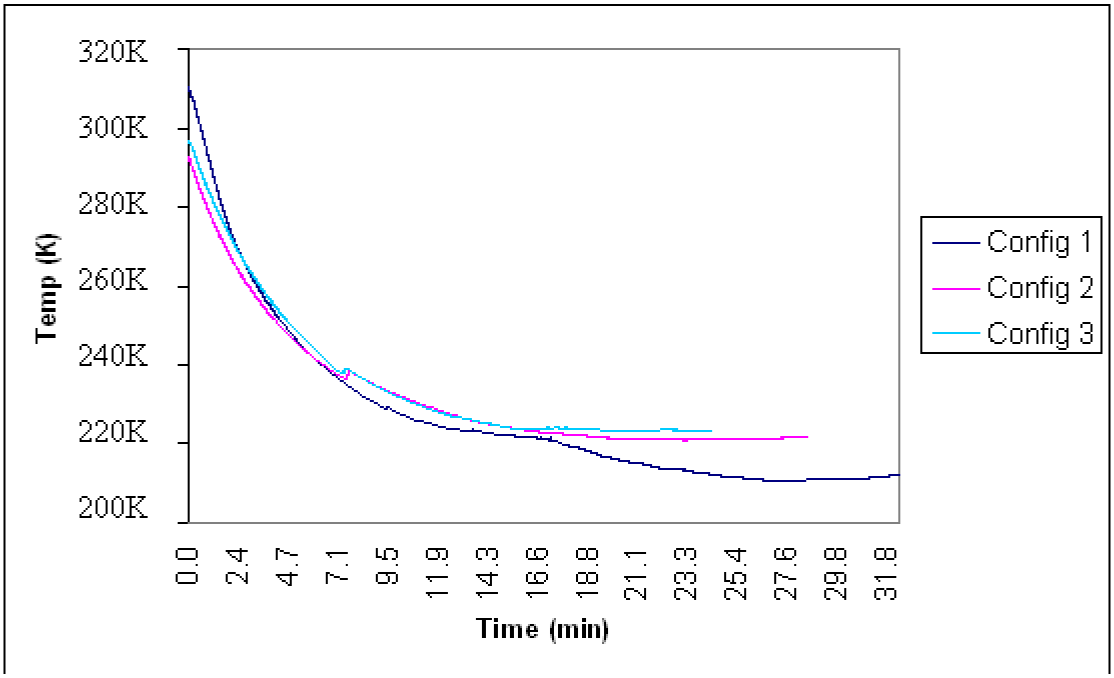

Fig 4.1: Cool down characteristics for different orifice valve openings

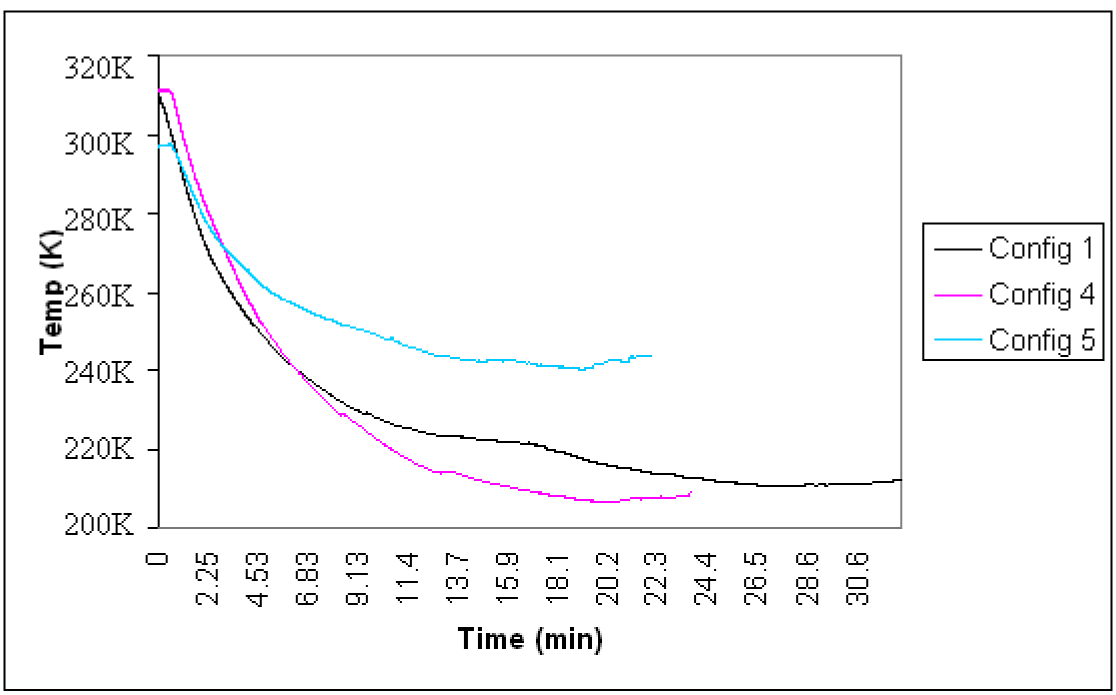

Fig 4.2: Cool down characteristics for different bypass valve openings

Table 4.1: Configuration description

	Orifice valve opening (as read on the metering scale)	**Bypass valve opening (as read on the metering scale)**
Configuration 1	7	0
Configuration 2	14	0
Configuration 3	21	0
Configuration 4	7	10
Configuration 5	7	20

Table 4.2: No load temperature attained

	No load temperature (K)
Configuration 1	245.21
Configuration 2	250.00
Configuration 3	241.55
Configuration 4	213.28
Configuration 5	226.11

The results show that no load temperature obtained is strongly dependent on opening of the orifice and bypass valves. It is clear that there is an optimum value for the valve openings. When the orifice valve is opened there is a drop in its resistance and thus the performance of PTR changes. It is observed that performance of orifice pulse tube refrigerator can be improved by introducing a bypass valve. Introduction of the bypass increases the amplitude of the pressure fluctuation on the cold side of the pulse tube and reduces the phase angle, with clear effects on the performance. The results are seen to be quite sensitive to the bypass resistance. Large opening of the bypass valve has negative effect on the performance. Overall the results reveal a complex behavior with respect to the orifice and bypass adjustments, which can be attributed to the fact that instantaneous division of flow between the two parallel paths depends upon the instantaneous impedances of the two paths.

4.3 EFFECT OF FREQUENCY

Experiments were conducted with frequencies of 1 Hz and 2 Hz with Nitrogen as working fluid. The results are shown in figure 4.3.

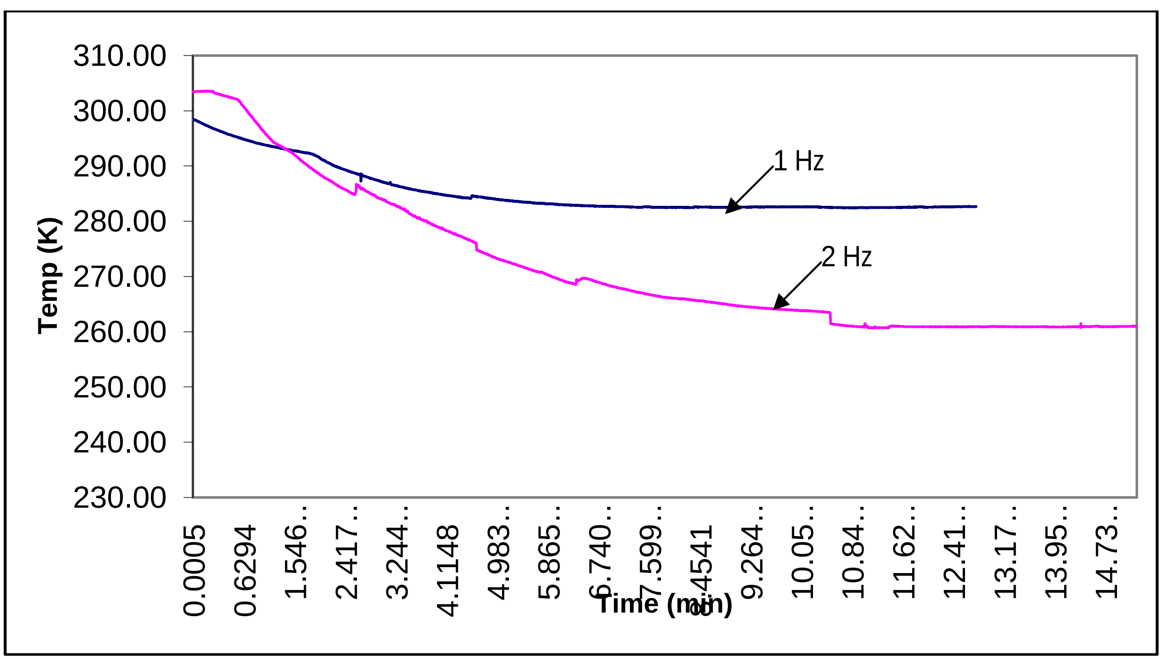

Fig 4.3: Cool down characteristics for different frequencies

Pulse tube refrigerator works at low frequencies. Frequency defines the diffusion depth in the working fluid and the regenerator material. When frequency is increased diffusion depth decreases and the heat storage in the regenerator degrades. High operating frequency means a big pressure drop in the regenerator, which leads to a poor performance. Hence low frequencies (1 Hz and 2 Hz) were used. It was observed that with a frequency of 2 Hz lower no load temperature could be achieved. This can be attributed to the fact that higher frequency increases time averaged enthalpy flow.

4.4 CONCLUSIONS

1. A pulse tube refrigerator has been built and tested with different valve openings, frequencies and working fluid. The proof of concept has been established.
2. The results show that in the presence of relatively high pressure gradients in the regenerator, the bypass improves performance by reducing these gradients and by improving the phase relationship between pressure and mass flow.
3. Lowest no load temperature obtained is **213 K**.

4.5 MODIFICATIONS SUGGESTED

1. Optimization of pulse tube and regenerator dimensions as well as arrangement of regenerator material to achieve lowest possible temperature.
2. Incorporation of an inertance tube to enhance the performance of pulse tube refrigerator.
3. Vortex tube arrangement can be used inside the pulse tube to avoid mixing of cold and hot gas.

REFERENCES

1. **Ray Radebaugh**, *Development of the Pulse Tube refrigerator as an Efficient and Reliable Cryocooler*, Proc. Institute of Refrigeration (London) 1999-2000.

2. **Ray Radebaugh,** *Pulse Tube cryocoolers for cooling Infrared Sensors*, Proceedings of SPIE, The International society for Optical Engineering, Infrared Technology and Applications XXVI, Vol.4130, pp. 363-379 (2000).

3. **Ray Radebaugh,** Advances *in cryocoolers*, Proc. ICEC16/ICMC, Japan, 1966, pp. 33-44, Elsevier Science, Oxford, 1997.

4. **G. Walker**, *Cryocoolers*, Plenum Press, New York, 1983.

5. **B. J. Huang** and **G. J. Yu**, *Experimental study on the design of orifice pulse tube refrigerator*, International Journal of Refrigeration, vol. 24 (2001) pp. 400-408.

6. **S. Zhu, P. Wu,** and **Z. Chen**, *Double inlet pulse tube refrigerators: an important improvement*, Cryogenics **30**, 514 (1990).

7. **E. D. Marquardt** and **R. Radebaugh**, *Pulse tube oxygen liquefier*, Adv. in Cryogenic Engineering, vol. 45, Plenum Press, NY (2000) in press.

8. **E. I. Mikulin**, **A. A. Tarasov**, and **M. P. Shkrebyonock**, *Low temperature expansion pulse tubes*, Adv. in Cryogenic Engineering, vol. 29, Plenum Press, New York (1984) pp. 629-637.

9. **Shaowei Zhu, Yasuhiro Kakimi** and **Yoichi Matsubara**, *Waiting Time effect of a GM type orifice pulse tube refrigerator*, Cryogenics **38** (1998) 619–624.