Experimental Studies towards Development of a Single Stage High Refrigerating Capacity G-M Type Pulse Tube Refrigerator

Dissertation submitted in partial fulfillment of the requirements of the degree of

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by

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(Roll Number: 614ME1004)

based on research carried out

Under the supervision of

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This is to certify that the work presented in this dissertation entitled "*Experimental Studies towards Development of a Single Stage High Refrigerating Capacity G-M Type Pulse Tube Refrigerator*" by *"K. N. Sai Manoj"*, Roll Number 614ME1004, is a record of original research carried out by him under my supervision and guidance in partial fulfillment of the requirements of the degree of *Master of Technology (Research)* in *Mechanical Engineering*. Neither this dissertation nor any part of it has been submitted for any degree or diploma to any institute or university in India or abroad.

Sunil Kumar Sarangi

Dedicated to my

Parents & Brother

Declaration of Originality

I, *K. N. Sai Manoj*, Roll Number *614ME1004* hereby declare that this dissertation entitled "*Experimental Studies Towards Development of a Single Stage High Refrigerating Capacity G-M Type Pulse Tube Refrigerator"* presents my original work carried out as a postgraduate student of NIT Rourkela and, to the best of my knowledge, contains no material previously published or written by another person, nor any material presented by me for the award of any other degree or diploma of NIT Rourkela or any other institution. Any contribution made to this research by others, with whom I have worked at NIT Rourkela or elsewhere, is explicitly acknowledged in the dissertation. Works of other authors cited in this dissertation have been duly acknowledged under the section "References". I have also submitted my original research records to the scrutiny committee for evaluation of my dissertation.

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Abstract

The absence of moving components at low temperature end gives the pulse tube refrigerator (PTR) a great leverage over other cryocoolers like Stirling and G-M 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 is to design, fabricate and test a single stage G-M type pulse tube refrigerator and study its performances. Experimental studies consists of cooling behavior of the refrigeration system at different cold end temperatures and optimization of orifice and double inlet openings at different pressures.

The developed pulse tube refrigerator consists of compressor, rotary valve, regenerator, pulse tube, hot end heat exchanger, orifice valve and double inlet valve, reservoir or buffer, vacuum chamber and coupling accessories etc. Regenerator and pulse tube have been chosen according to the literature available. Hot end heat exchanger has been designed and fabricated with respect to the regenerator and pulse tube geometry. The assembly of the components has been done in such a way that the set-up can be used as basic pulse tube refrigerator, orifice pulse tube refrigerator or double inlet pulse tube refrigerator as and when required. This has enabled thorough comparison among them.

The effect of operating conditions such as average pressure and pressure ratio of the compressor also has been found out. The optimum operating conditions such as opening of the orifice and double inlet valves have been selected according to the performance i.e. minimum attainable temperature at no load condition. Effect of orifice and double inlet openings at different pressures has been detected by applying the pressure sensors across at various positions in the system. Correspondingly, pressure variations at regenerator inlet, pulse tube and reservoir has been determined.

Keywords: pulse tube refrigerator; double inlet; design; fabrication; testing; optimization; cooling behaviour; pressure variation.

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Abbreviations

BPTR	Basic pulse tube refrigerator
OPTR	Orifice pulse tube refrigerator
DIPTR	Double inlet pulse tube refrigerator
BPT	Basic pulse tube
OPT	Orifice pulse tube
DIPT	Double inlet pulse tube
MOPT	Modified orifice pulse tube
DRPT	Double inlet reversible pulse tube
HPTR	Hybrid pulse tube refrigerator
JT	Joule-Thomson
CE	Cold end
HE	Hot end heat exchanger
PTR	Pulse tube refrigerator
HP	High pressure
LP	Low pressure
PT1	Pulse tube at position 1
PT2	Pulse tube at position 2
PT3	Pulse tube at position 3
RTD	Resistance temperature detector
MRI	Magnetic resonance imaging
SQUID	Super conducting quantum interference device

Chapter 1

Introduction

1.1 Background & Motivation

Cryogenics literally means ‘icy cold’ and is referred to the technology and science of producing low temperatures. However, the term cryogenics generally refers to the entire phenomena occurring at temperatures below 123 K, and processes, techniques and apparatus needed to create or maintain such low temperatures. An increased need for cryogenic temperatures in many areas of science and technology in the last few decades caused a rapid development of cryocoolers. Cryocoolers are refrigerating machines, which are capable of achieving cryogenic temperatures.

Cryocoolers are used in various applications due to high efficiency, high reliability, low cost, low maintenance, low noise level etc. However the presence of moving parts in the cold zone of the most of the cryocoolers makes it difficult to meet all these requirements. A new concept in cryocoolers, pulse tube refrigerator (PTR) has overcome some of these drawbacks. A PTR is a closed cycle mechanical cooler without any moving components, working in the low temperature zone. Conventionally, there exists two types of cooling technologies: open cycle and closed cycle. The open cycle cooling technique, which included the evaporation of stored cryogen and joule-Thomson expansion of pressurized gas, may be relatively low cost and good reliability. But their application is quite limited since they often present logistic problems. The closed cooling system which includes G-M, Stirling and Joule-Thomson cycles are more favourable. The main distinction of cryocoolers from other closed cycle mechanical coolers is that the PTR has no moving parts in the low temperature region and therefore, has a long life and low mechanical and magnetic interferences. The operating principle of the PTR is based on the displacement and the expansion of gas in the pulse tube that results in the reduction of the temperature. Usually helium is used as the working fluid in all closed cycle cryocoolers, including PTR. The working fluid undergoes an oscillating flow due to an oscillating pressure. A typical average pressure in a PTR is 10 to 25 bar. A piston compressor (in case of a Stirling type PTR) or a combination of a compressor and a set of switching valves (G-M type PTR) is used to create pressure oscillation in a PTR. The regenerator of the PTR stores the heat of the gas in its matrix during a half cycle and therefore must have a high heat capacity compared to the heat capacity of the gas.

The concept of pulse tube refrigeration was first introduced by Gifford while working on the compressor in the late 1950's,he noticed that a tube, which branched from high-pressure line and closed by a valve was hotter at the valve than at the branch. He recognized that there was a heat pumping mechanism that resulted from pressure pulses in the line. Thus, in 1963 Gifford together with Longsworth introduced the Pulse tube refrigerator, which is termed as the Basic Pulse Tube (BPT) refrigerator. The cooling principle of the BPT refrigerator is based on the surface heat pumping, which is described as the exchange of heat between the working gas and the pulse tube walls. The major breakthrough in the development of pulse tube refrigerators is with the development of a new type of pulse tube refrigerator called the Orifice Pulse Tube Refrigerator. On the basis of theoretical analysis, a modified version called Double inlet Pulse Tube (DIPT) refrigerator was suggested by Zhou *et al* [11], which has a second inlet valve at the hot end of the pulse tube connected to the pressure wave generator (compressor and rotary valve).

The third most successful type of pulse tube refrigerator is schematically illustrated in Fig.1.1. The pressure wave generator may be either a compressor with a gas distributor (rotary or electromagnetic) or a directly coupled pressure oscillator. Its function is to generate a pressure wave in the system. The regenerator is basically a heat exchanger that helps the gas to reach the low temperature region at high pressure and without carrying heat with it. The regenerator is made of thin walled stainless steel tube filled with stainless steel screens or other porous material with large heat capacities. It does not carry heat in or out of the system but it absorbs heat from the gas during one part of the pressure cycle and returns this heat to the gas during the other part. The high heat capacity of the regenerator matrix with respect to that of the working fluid permits it to store the cooling effect generated in the pulse tube by alternatively cooling down and heating up the gas which flows through it. The pulse tube is considered as the heart of a PTR system and is a thin walled stainless steel tube. The gas inside the pulse tube experiences the cooling effect, if there is a suitable phase shift between the pressure and the gas flow in the tube. The two heat exchangers located in the cold and warm ends of the pulse tube act as flow straighteners. The cold end heat exchanger is the coldest point of the system. Here the PTR absorbs heat from the device to be cooled. The hot end heat exchanger is used to remove the heat carried through the pulse tube section from the cold end. Generally it is an air or water cooled heat exchanger, though other types of cooling are also possible.

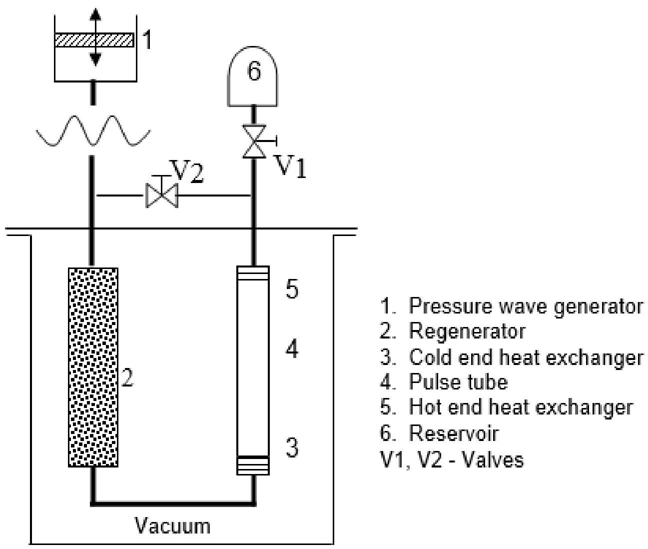


Fig.1.1 Schematic of the Pulse Tube Refrigerator

The orifice and the impedance between the pressure wave generator at the hot end of pulse tube are two adjustable needle valves, V1 and V2. These two valves allow the aforesaid three types of configurations:

Basic Pulse Tube Refrigerator [BPTR], both V1 and V2 closed

Orifice Pulse Tube Refrigerator [OPTR], V1 open and V2 closed

Double Inlet Pulse Tube Refrigerator [DIPTR], both V1 and V2 opened

These cryocoolers as enumerated by Radebaugh (1995), are mainly used for cooling of the infrareds sensors in the missile guided system and satellite based surveillance, as well as in the cooling of superconductors and semiconductors. The cryocoolers can also be used in other applications such as in cryopumps, liquefying natural gases, cooling of radiation shields, SQUID (super conducting quantum interference device), magnetometers, SC Magnets, semiconductor fabrication etc. Although the pulse tube cooler technology has progressed significantly that commercial systems are now available, still there is considerable interest in understanding the fundamental mechanisms of cooling in PTRs. Till today, no one can predict appropriately the working principle responsible for the production of cold effect in the pulse tube. The tube is simple but the occurrences responsible to build the cooling effect are much complicated. It is worth noting that modeling of a complete pulse tube refrigerator is not so easy due to non-linear, unsteady and oscillating flow through different passages particularly regenerator, orifice valve and double inlet valve. At the present scenario the cryocoolers are rapidly increasing based on its applications and usage. Among them the pulse tube cryocoolers are very important for their refrigerating capacity, better performances and no load temperature. The double inlet configuration strikes a good compromise between complexity and performance.

At present, the lowest temperature attained for a single stage system is 22 K [70] in a two stage arrangement.   It is really impossible to reach very low temperature using a single stage pulse tube refrigerator. Many types of Cryocoolers for lower temperature region are carried out as coolers in series which is complex to produce cooling capacity. Because of the simple geometry and the absence of any moving parts, it is possible to attach many stages one after the other. In a single stage: low temperatures have been achieved, but obtaining a cooling capacity in G-M refrigerators of above 50 W is very complex.

Against this background, the main motivation and present research work is undertaken to develop a large refrigerating capacity single stage G-M type pulse tube refrigerator. This is because to generate a liquid nitrogen for storage of live biological materials and tissue engineering products, small scale industrial applications e.g. tool hardening, small natural gas liquefiers and laboratory devices, vacuum pumps and cold traps. In international arena it is very commercial. But in developing countries like India, there is a need for the generation of liquid nitrogen to produce 20-30 liters in a day at high refrigerating capacity to solve the daily requirements as mentioned earlier. A strong potential exists for commercial stuff like G-M type but pulse tube is easier. A G-M refrigerator produce better refrigeration than a pulse tube, but it is far more complex because of cold end moving parts and requires more maintenance. So it is easier to sacrifice some amount of cooling capacity and in terms of electricity. Keeping in view of these facts, an indigenous single stage G-M type pulse tube refrigerator is designed and developed.

Research in the area of pulse tube refrigerators for various applications is the demand of time. Discovery of BPTR, OPTR, DIPTR, four valve and active buffer configurations are just few of them. Intense efforts are going on around the world to make simple and reliable cryocoolers by performing experiments to achieve lowest possible temperatures.

1.2 Objectives

This study aims at broadening the level of understanding of the operations of pulse tube refrigerators. An effort has been made to achieve this by experimental investigations.

The objectives of the research work are

- To conduct an up-to-date survey of literatures on experimental works on single stage and multi stage pulse tube refrigerators.
- To develop an indigenous G-M type single stage pulse tube refrigerator operating at a high cooling capacity of 200 W at 70 K.
- To conduct experimental studies on double inlet configuration of pulse tube refrigerator and study its performances at optimum level.

1.3 Organization of the Thesis

The current thesis consists of six chapters. The basic introduction of cryocoolers and the significance of the present investigation related to pulse tube refrigerator are described in chapter 1 as introduction. Chapter 2 presents a brief review of the literature about the origin and evolution of cryocoolers. This review provides the information regarding current research of experiments undergone and the performance of main configurations of high capacity pulse tube cryocoolers. It consists of effects in cooling capacity and low temperature when subjected to variations by the components. Elaborated briefly about the cryocooler research going across the country. Chapter 3 illustrates and describes the design and fabrication of the components of the pulse tube refrigerator. Chapter 4 consists of assembly of the whole experimental test set-up. It also highlights the instrumentation and procedure of operation. Chapter 5 deals with results and performances of experimental set-up. Chapter 6 sums up the present work with important conclusions and recommendation for future work.

Chapter 2

Review of literature

2.1 Introduction

In this chapter, principle of operation and a brief classification of pulse tube refrigerators are discussed. The various developments took place in the area of PTRs, since its invention in 1964 and the sources of information are presented in a chronological manner.

2.2 Pulse Tube Refrigerator

Cryocoolers, finds wide variety of applications, hence it should be efficient, reliable, durable, economical and less noisy. However, the presence of moving parts in the cold area of most of the cryocoolers makes it difficult to meet all these requirements. The concept of a new cryocooler called the pulse tube refrigerator (PTR) was first introduced by Gifford [1], while working on the compressor in the late 1960's. He noticed that a tube, which branched from high-pressure line was closed by a valve, was hotter at the valve than at the branch. He recognized that there was a heat pumping mechanism that resulted from pressure pulses in the line. Thus, in 1965 Gifford together with his assistant Longsworth introduced the concept of Pulse tube refrigerator, which is currently named as the Basic Pulse Tube (BPT) refrigerator. The cooling principle of the BPT refrigerator is the surface heat pumping, which is based on the exchange of heat between the working gas and the pulse tube walls. The lowest temperature reached by Gifford and Longsworth with the BPT refrigerator, was 124 K with a single stage. Ironically, this is not the basis of the present day pulse tube refrigerators.

Mikulin *et al*. [6] developed a new type of pulse tube refrigerator called, Orifice Pulse Tube (OPT) Refrigerator which has revolutionized the pulse tube technology in the year 1984. This invention resulted in a rapid achievement in the field of cryocoolers and brought an avalanche of new ideas, all with the intention to improve the performance of cryocoolers. The most important types of pulse tube refrigerators are discussed in the following section.

2.2.1 Principle of operation

The operation principles of PTRs are very similar as conventional refrigeration systems. The methods of removing heat from the cold environment to the warm environment are somewhat different. The vapour compression cycle shown in Fig.2.1 operates in a steady flow fashion where heat is transported from the evaporator to the condenser by a constant and steady mass flow rate. The PTR relies on an oscillatory pressure wave in the system for transporting heat from the cold end heat exchanger to hot end heat exchanger.

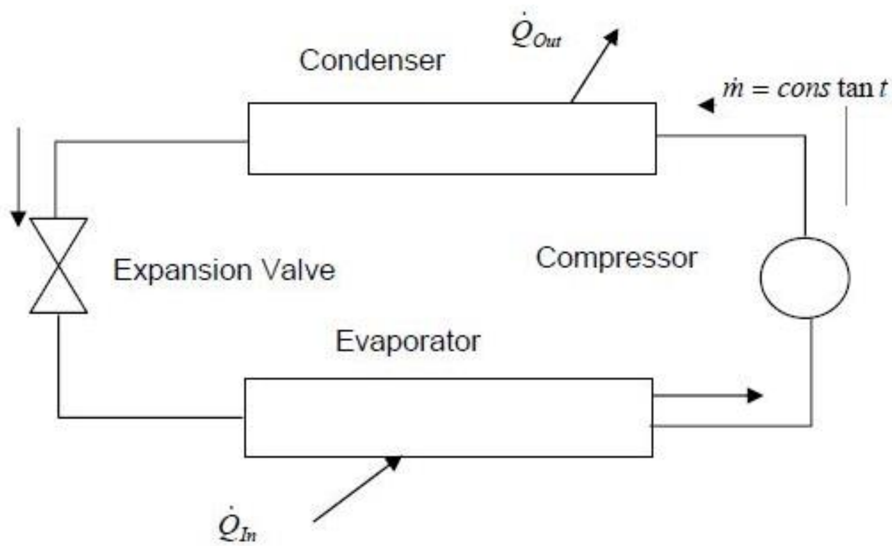


Fig.2.1 Schematic diagram of the simple vapour compression cycle [38]

2.2.2 Advantages of PTR over G-M and Stirling Cryocoolers

- Absence of displacer at cold end.
- Simple construction and reduced cost.
- Higher reliability.
- Reduced vibrations.
- Low mechanical and magnetic interferences

2.2.3 Limitations of PTR over G-M and Stirling Cryocoolers

- Requirement of more gas to pass through pulse tube and reservoir. Hence, viscous losses are increased.
- Difference in density gives rise to convection currents; if the machine is tilted. Thus the performance of the device becomes orientation dependent.

2.2.4 Applications of Pulse Tube Refrigerator

The application area of cryocoolers is very large. Most of the applications require high efficiency and reliability of a cooler as well as its long lifetime and a low cost. Advances in the cryogenic technology and cryocooler design have opened the door for potential applications in cryogenically cooled sensors and devices such as:

- Missile tracking sensors
- Unmanned Aerial Vehicles (UAVs)
- Infrared (IR) search and track sensors
- Satellite tracking systems
- Pollution monitoring sensors
- High Resolution imaging sensors
- Magnetic Resonance Imaging (MRI) and Computer Tomography (CT) for medical diagnosis and treatment.

Studies further indicate that Cryogenic technology has potential applications to Photonic devices, Frequency (RF) sensors, Electro-Optic components and Opto-Electronic devices.

2.3. Classification of Pulse Tube Refrigerators

Even though there are different models of PTR exists, in general, pulse tube cryocoolers are basically classified on the following basis.

- Operating frequency
- Geometry
- Magnitude of Phase shift

2.3.1 Based on Operating Frequency

The most important parameter to achieve cooling capacity and lowest temperature is by varying frequency and can be observed in Stirling and G-M type PTRs. G-M type achieves much lower temperature rather than Stirling one but less efficient.

2.3.1.1 Low frequency /G-M type/Valved PTR

Gifford-Mc-Mahon (G-M) type, is used for lower temperatures (20 K and below) operate at low frequencies (1-5 Hz). At room temperature, the swept volume per cycle can be very high up to one liter and more for these types of refrigerators. Therefore it is more practical to uncouple the compressor from the cooler. The compression heat is removed by cooling water in the compressor. The compressor delivers a constant high pressure (HP) stream corresponding to a given low pressure (LP). A schematic diagram has been given in Fig. 2.2 (b). The varying pressure is obtained through a system of valves, usually of rotary design, which alternately connects the high pressure and low pressure to the hot end of the regenerator. G-M type PTR is less efficient than the Stirling type, since the gas flows through the valves are accompanied by losses, which are absent in the Stirling type.

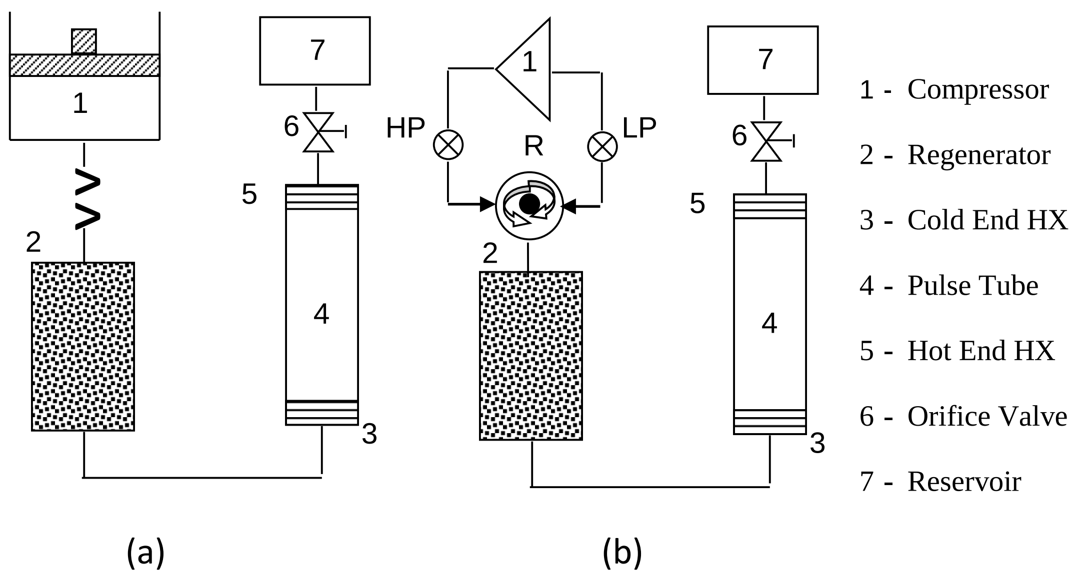


Fig.2.2 (a) Stirling type PTR (b) G-M type PTR

2.3.1.2 High frequency /Stirling type/Valve less PTR

For a Stirling type PTR as shown in Fig. 2.2 (a), a piston-cylinder apparatus is connected to the system so that the piston movement directly generates the pressure fluctuations. The power supplied to the compressor must be removed as heat to the environment by a heat exchanger between the compressor and the entrance of the regenerator commonly known as after cooler.

These types of refrigerators are used for higher temperature ranges of about 80 K and high driving frequency of the range 25-50 Hz. Because of this higher frequency and the absence of valve losses, Stirling PTR systems generally produce higher cooling powers than G-M type PTR. However, the rapid heat exchange required on Stirling type pulse tube refrigerators limits their performance at low temperatures, such as at 10 K and below.

2.3.1.3 Comparisons between Stirling and G- M type Cryocoolers

In general there are two types of pulse tube refrigerators used in practice. The overall comparisons between these two systems are described in Table 2.1.

Table 2.1 Comparisons between Stirling and G- M type Cryocoolers

Stirling type cryocooler	G-M type cryocooler
Works at high frequency (20-120 Hz)	Works at Low frequency (1-5 Hz)
Compressor directly connected to expander	Compressor connected to expander through a valve
Use of dry compressor	Use of oil lubricated compressor
High COP	Low COP
Pressure ratios are low	Pressure ratios are high
Can attain 20 K using two stages of cooler	Can attain below 2 K using two stages of Cooler
Compressors are small (capacity is in few hundred Watts)	Compressors are bulky(capacity is in kW)

2.3.2 Based on Geometrical Arrangement

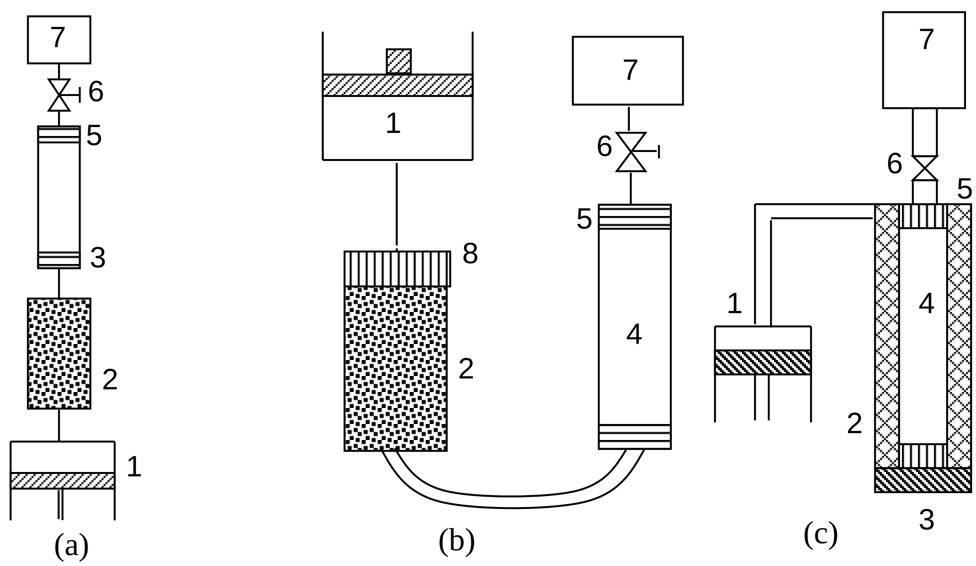
Pulse tube refrigerators are also classified as linear type, U- type and co- axial type, according to their geometry or shape and are briefly described in detail.

2.3.2.1 Linear type/Inline cryocooler

If the regenerator and the pulse tube are in line as shown in Fig. 2.3(a) is called as linear type refrigerator. The best arrangement for mounting the PTR in the vacuum chamber is with the hot end of the tube, where heat is released to the environment, connected to the vacuum chamber wall and the cold end of the regenerator inside the vacuum chamber. Thermodynamically, this is the most efficient geometrical arrangement. The only drawback is that the cold end of the pulse tube is difficult to access.

2.3.2.2 U- type cryocooler

The disadvantage of the linear PTR is that the cold region is in the middle of the cooler. U-type PTRs are made by arranging the pulse tube and the regenerator parallel to each other with an interconnecting tube of U-shape, as shown in Fig. 2.3(b).



1- Compressor 2-Regenerator 3-Cold end heat exchanger 4-Pulse Tube
5-Hot end heat exchanger 6-Orifice Valve 7-Reservoir 8- After cooler

Fig.2.3 (a) Linear type (b) U- type (c) Co axial type

For many applications it is preferable that the cooling is produced at the end of the cooler. The hot ends of the pulse tube and regenerator can be mounted on the flange of the vacuum chamber at room temperature. This is the most common shape of pulse tube refrigerators.

2.3.2.3 Co-axial type cryocooler

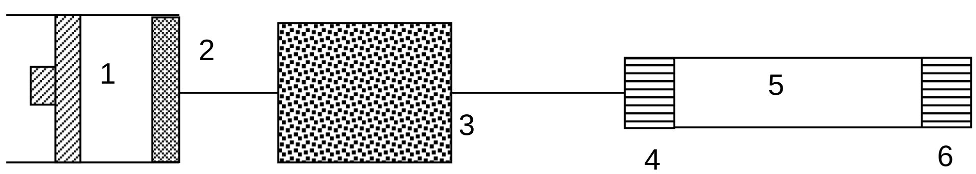
For some applications of PTR it is preferable to have a cylindrical geometry. In that case the PTR can be constructed in a co-axial way so that the regenerator becomes a ring shaped space surrounding the tube shown in Fig. 2.3(c). The major disadvantage of this construction is that there is thermal contact between the tube and the regenerator, which results in a degradation of performance.

2.3.3 Based on Magnitude of Phase Shift

This is the most important classification of PTRs where the phase shift plays a prominent role in achieving better performance rather than above two categories.

2.3.3.1 Basic type (BPTR*)*

The basic pulse tube refrigerator shown in Fig. 2.4 consists of a pressure wave generator (Stirling type compressor or GM type arrangement), regenerator, cold heat exchanger, hot heat exchanger and pulse tube.



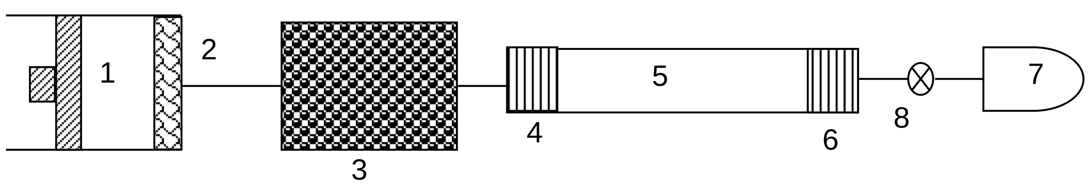
1. Compressor 2. After cooler 3. Regenerator 4. Cold end heat exchanger
5. Pulse tube 6. Hot end heat exchanger

Fig.2.4 Schematic of basic pulse tube refrigerator

During the pressure build up period, the valve admits high pressure gas through the regenerator, where it is cooled to the cold end temperature. There is some gas present in the tube at the beginning of the cycle. The entering gas acts as a gas piston and compresses the gas present in the pulse tube (refer Fig. 2.4). The gas piston pushes the gas to the far end of the tube where a heat exchanger is employed as a heat sink. The temperature of the gas will then cool down to the temperature of the cooling medium of the heat exchanger. After that, the high-pressure gas is allowed to expand during the exhaust phase of the cycle to a very low temperature thus producing refrigeration. Although the heat exchange between the gas and the wall takes place along the length of the pulse tube, it is assumed that only in the region of hot end heat exchanger heat can be rejected from the system. After the expansion takes place adiabatically, the temperature of the gas becomes lower than the wall temperature. So, heat will be transferred from the wall to the gas. However, when the gas enters the cold end heat exchanger, since its temperature is lower than the room temperature, heat is absorbed from the heat exchanger producing cooling power. The net result of this effect is that heat is extracted from the cold end exchanger and rejected at the hot end exchanger. Due to this, the cold end heat exchanger and the regenerator will cool down a bit and the next cycle starts at a slightly lower temperature.

2.3.3.2 Single inlet or Orifice type (OPTR)

The major drawback of BPTR can be overcome by placing an orifice valve and a reservoir after the hot heat exchanger to reduce the phase difference between the pressure and mass flow rate to a value less than 90^o. The reservoir is large enough to be maintained at a nearly constant intermediate pressure during operation. The valve and the reservoir cause the gas to flow through the orifice valve at the points of maximum and minimum pressures. Therefore the reservoir improves the phase relationship between the pressure and gas motion.



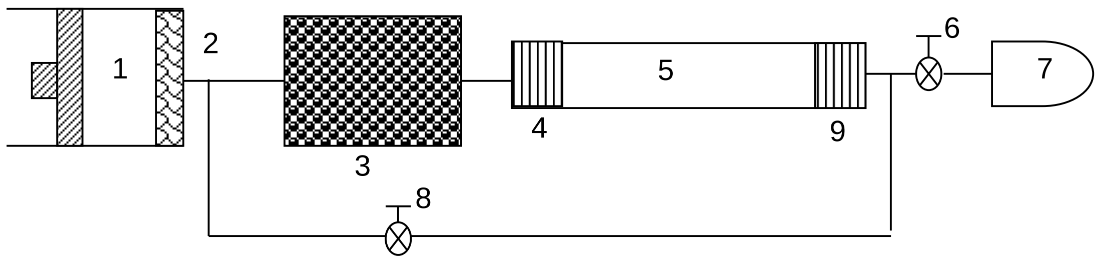
1. Compressor 2. After cooler 3. Regenerator 4. Cold end heat exchanger
5. Pulse tube 6. Hot end heat exchanger 7. Reservoir 8. Orifice

Fig.2.5 Schematic of orifice pulse tube refrigerator

In a BPT refrigerator, the lowest temperature to which the gas can be cooled after compression is the wall temperature of the tube or the temperature of the cooling medium. But in an OPT refrigerator, due to the expansion through orifice, the gas can be cooled to a lower temperature after compression and is shown in Fig. 2.5. Thus during the expansion still lower temperature can be attained.

2.3.3.3 Double inlet type (DIPTR)

In the double-inlet PTR the hot end of the pulse tube is connected to the entrance (hot end) of the regenerator by an orifice adjusted to an optimal value as shown in Fig.2.6. The double inlet valve is a bypass for the regenerator and the pulse tube and hence reduces the cooling power. In addition, it is a dissipative device, which leads to a deterioration of the performance. However, both these disadvantages are overcome by the fact that the double inlet reduces the dissipation in the regenerator. As a result, the performance of the overall system is improved significantly.



1. Compressor 2. After cooler 3. Regenerator 4. Cold end heat exchanger 5. Pulse tube 6. Orifice 7. Reservoir 8. Double inlet valve 9.Hot end heat exchanger

Fig.2.6 Schematic of double inlet pulse tube refrigerator

Double valved double inlet PTR

The double valved double inlet type which is a part of double inlet configuration, two metering valves are used in order to eliminate DC flow loss. This configuration yields better refrigerating capacity, lowest possible temperature and achieves high efficiency rather than single valve operating double inlet configuration. Schematic view of the double valved double inlet type is shown in Fig.2.7.

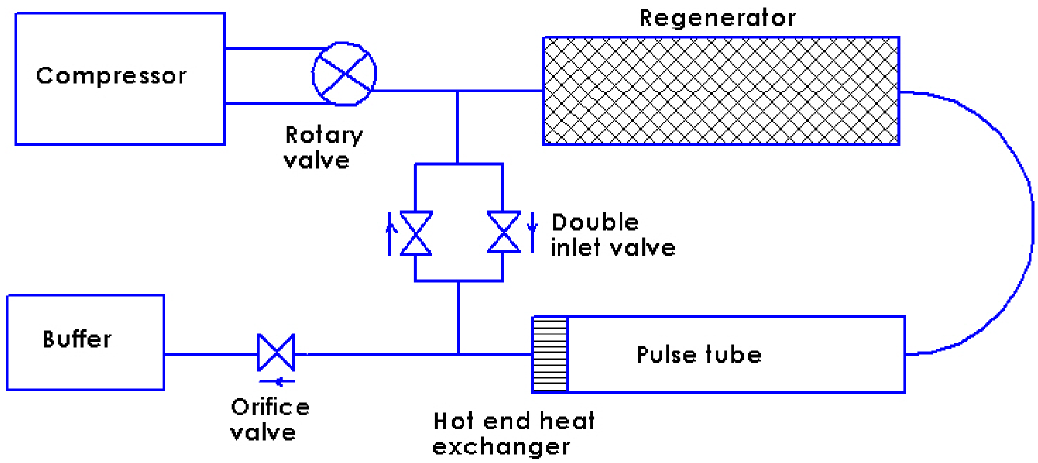


Fig.2.7 Schematic of double valved double inlet pulse tube refrigerator

2.4 Sources of Information

Before the mid 1950's there was no single source of comprehensive fluid or material properties for low temperature applications. Cryogenic data were hard to find and not always in a form convenient for use. To complete a cryogenic system, design engineers relied on multiple books, hand books and compendiums, each with a bit of information needed for material and fluid data. Some of the early hand books commonly found in the engineering library were" 'Hand book of Engineering Fundamentals' and 'Standard hand Book for Mechanical Engineers'. These handbooks contain a wealth of reference tables and charts. In the early 1950s, this has been replaced with the 'Cryogenic Materials Data Handbook' which contains mechanical and thermal property data on different structural alloys and non-metals. In the early 1960s, the entire information was provided in a series entitled 'A Compendium of the Properties of Materials at Low Temperature". From the early 1970s, onwards this scenario has been completely changed and the sources of information are provided in various journals and conferences. The main pillars of information for the rapid growth and research in cryogenics and its related areas are available below.

- Advances in Cryogenic Engineering materials
- International Cryogenic Engineering Conference and International Cryogenic Materials Conference (ICEC-ICMC)
- Cryocoolers Conference
- Journal of Cryogenics

In occasional there are other publications such as ASME, Elsevier and Springer etc. shares the valuable information and developments undergoing in low temperature materials throughout the world.

2.5 Development history of pulse tube refrigerators

Gifford and Longsworth [1] introduced the concept of pulse tube refrigerator, a new method of achieving cryogenic temperature in 1965. Their machine worked by the cyclic compression and expansion of helium gas in a half open tube. They observed that cyclic alternative pressurization and depressurization of a tube from one end of it, while the other end remained closed, could establish a considerable temperature gradient along the tube wall. Despite its mechanical simplicity and high reliability, its performance was very poor. In their first report, a cold end temperature of 150K was achieved. The valuable points in the paper are:

- Pressurization and depressurization of a constant volume system will lead to transfer of heat within the volume and outside the volume.
- Pressurizing and depressurizing a constant volume system due to unsymmetrical transfer of heat may lead to the build-up of large temperature differences within the volume.
- The unsymmetrical transfer of heat in pressurization and depressurization of a constant volume may be used in combination with heat exchangers and a regenerator, which has achieved a temperature as low as 150K.

Gifford and Longsworth [2] developed a relationship for the cold end temperature with zero heat pumping rate in terms of length ratio, hot end temperature and the ratio of specific heats of gas with the help of surface heat pumping (SHP) mechanism.

Colangelo *et al* [3] developed a simplified heat transfer model for the performance analysis of basic pulse tube refrigerators. This model takes into account the heat and mass transfer processes in the regenerator and pulse tube. They assumed that the convective heat transfer between the gas and pulse tube wall or regenerator matrix during flow periods is a controlling mechanism.

Gifford and Kyanka [4] returned to the problem of reversible pulse tube and attempted to compare with that of a valved pulse tube, although it would seem that the experimental comparison was based on limited data. The pressure ratio used in this work was 4.2:1 and a low temperature limit of 165 K was achieved. It was concluded that other factors being equal and the refrigeration capacity of a reversible pulse tube is inferior to that of the valved type. Later, the research on pulse tube cryogenerators was undertaken by Wheatley in the Los Alamos National Laboratory using a thermoacoustic pressure wave generator instead of mechanical one.

Narayankhedhkar and Mane [5] did theoretical and experimental investigations on pulse tube refrigerator. The method for the derivation of cold end temperature with zero heat pumping rates was introduced. Lowest cold end temperature obtained with air as the working fluid was 214.5 K, with a frequency of 50 Hz. Experimental investigations indicated that there exists an optimum speed and hot end length, and this speed decreases with increase in the total length of pulse tube. They verified Longsworth's conclusion about the variation of heat pumping rate with pulse tube length by experiments up to a total length of 550 mm and with air as the working fluid.

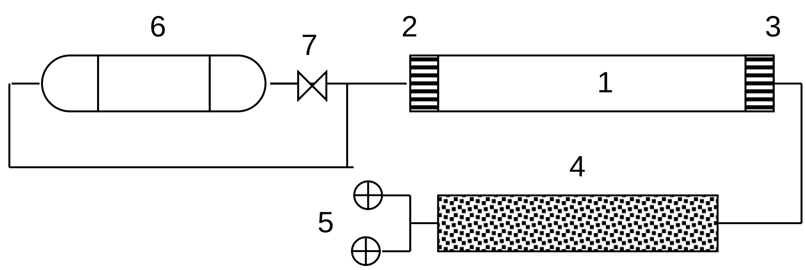
The main achievement when Mikulin *et al.* [6] and his co-workers published their innovative modification of the basic pulse tube refrigerator. They showed that the efficiency of pulse tube refrigerator could be increased by fastening a reservoir to the warm end of the pulse tube, through an orifice instead of being closed. Using air as the working fluid, they achieved a low temperature of nearly 105 K and the net refrigeration capacity at 120 K was ~10 W.

Richardson [7] updated Longworth analysis for BPT refrigerators by considering the maximum value of the gas charging period and he reached the prediction of an optimum pulse rate, which was verified qualitatively by experiments. However this study was mostly experimental and no system modelling performance analysis was done.

Zhou *et al* [8] made an experimental investigation to compare the performance of coiled pulse tubes with those of straight ones having similar cross sections, length and operating conditions. The performance degradation of coiled pulse tube had also been reported when ratio of the axial radius to the radius of the cross section is reduced.

Some new concepts for pulse tube refrigeration has been proposed and investigated by Matsubara and co-workers [9]. In one experiment they replaced the orifice with a moving plug (also at room temperature) and lowered the temperature from 78 K to 73 K. Normally, a mechanical compressor is used to drive the pulse tube, but Matsubara tried a thermally activated pulse tube, where a hot displacer is used to move gas between a heated volume and a room temperature volume to generate a pressure oscillation. The thermally actuated pulse tube refrigerator has been operated at the temperature of about 200 K.

Richardson [10] reviewed the development of valved PTR and explained clearly the heat pumping mechanism inside it. He experimentally optimized the valved pulse tube, which involves the two variables of throttle setting and buffer volume. The schematic of valved PTR is shown in Fig.2.8. It can be seen from the figure that, the valved pulse tube differs from that of the simple design in having a buffer volume linked to the warm end heat exchanger. A throttle valve or a fixed orifice controls the flow of gas between the pulse tube and buffer volume. With the valve fully closed, the device functions as a BPT refrigerator.



1. Pulse tube 2.Warm end heat exchanger 3.Cold end heat exchanger 4.Regenerator
5. Pressure source 6.Buffer volume 7.Throttle valve.

Fig.2.8 Schematic of valved pulse tube refrigerator [10]

Zhou *et al.* [11] achieved a new constructional solution to increase the OPTR refrigeration efficiency. On the basis of theoretical analysis, a modified version called double inlet pulse tube refrigerator (DIPTR) was suggested in Fig.2.9, which had a second inlet at the hot end of the pulse tube connected to the pressure wave generator. Numerical analysis and experimental results confirm that the double inlet pulse tube has improved performance over the OPTR. Numerical analysis and experimental results confirm that the double inlet pulse tube refrigerator can produce higher refrigerating power for unit mass flow rate through the regenerator.

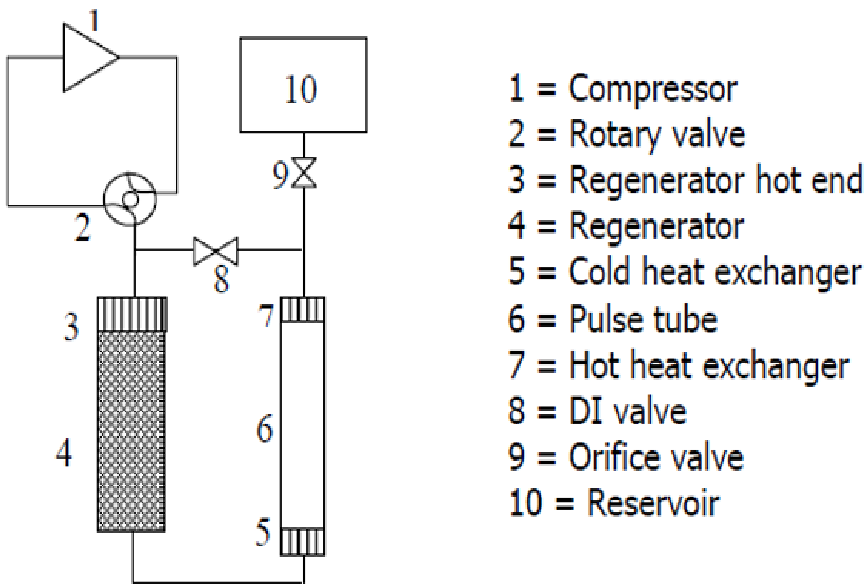
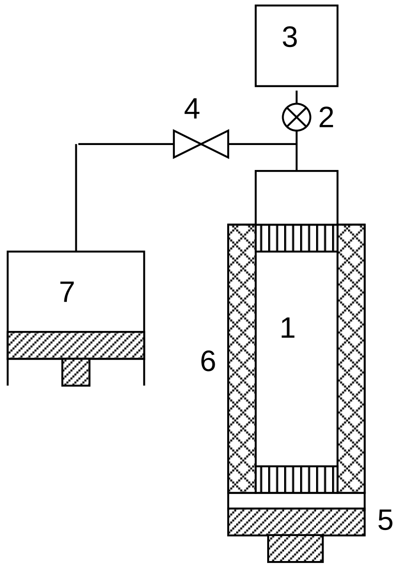


Fig.2.9 Schematic of G-M type double inlet pulse tube refrigerator

Shaowei *et al.* [12] conducted experiments on a single stage DIPT refrigerator. The experimental results shows that a minimum temperature of 42 K was achieved with a single stage DIPT refrigerator with a frequency of 7 Hz and an average pressure of 1.1 MPa, whereas the minimum temperature obtained from a OPT refrigerator of same configuration was 55 K.

Orifice pulse tube refrigerators developed had a U-shape configuration that made it inconvenient for practical applications. To solve this problem Wang *et al* [13] adopted a co-axial configuration of the pulse tube and regenerator to make the system small and compact. Experiments were conducted with this co-axial design and the influence of different parameters on the minimum temperature was investigated. A no load temperature of 62 K was achieved and about 2.5 W of net refrigeration power was attained at 77 K. The main negative aspect of the coaxial type was that the temperature distribution along the pulse tube was different from that along regenerator, which caused heat transfer between the pulse tube and regenerator. Hence the refrigeration capacity was decreased. The schematic of a co-axial PTR is shown in Fig.2.10.



1. Pulse tube
2. Orifice
3. Reservoir
4. Double inlet valve
5. Cold end heat exchanger
6. Regenerator
7. Compressor

Fig.2.10 Schematic of Co- axial Pulse tube refrigerator [13]

Baks *et al.* [14] did an experimental verification of an analytical model developed for orifice pulse tube refrigerator. The cooling power of pulse tube refrigerator was expressed in terms of regenerator loss and average enthalpy flow through the pulse tube. They concluded that the deviation of the experimental results from the theoretical results presented by Radebaugh was due to the thermal contact between the gas in the thermal boundary layer of the pulse tube and the wall of the pulse tube.

Liang *et al.* [15] by improving the regenerator, hot end heat exchanger and the insulation of low temperature sections. A low temperature of 49 K and refrigeration power of 12 W at 77 K achieved experimentally at the cold end and also investigated the relation between the ratio of regenerator volume to pulse tube volume and the minimum temperature of OPTR.

Kasuya *et al.* [16] studied on the role of heat exchange between the gases in the pulse tube and the tube wall in a pulse tube refrigerator. They experimentally investigated a system where the working fluid going through the pulse tube without heat exchange by mounting a piston on the hot end of the pulse tube. Refrigeration power was found to increase as the work flow reaching the hot-end piston increases. On the contrary, the heat flow released into a room temperature environment decreases as the workflow increases. This suggests that the work flow becomes more important as the refrigeration power increases.

Kasuya *et al*. [17] conducted a study to investigate how the phase angle between pressure oscillation and gas displacement affects pulse tube refrigeration performance. For this purpose, a pulse tube refrigerator involving a piston at the hot end of the pulse tube is constructed. It is found that the lowest temperature is 47 K with an operating speed of 1.3 Hz. The improvement achieved with double-inlet pulse-tube refrigerators can be explained by the phase angle versus refrigeration performance relation found in their experiment. At the optimum phase angle, the gas elements near the hot end of the pulse tube move towards the cold end during compression and towards the hot end during expansion.

Marc David *et al.* [18] gave practical methods to calculate the theoretical gross refrigeration power of an ideal OPT or DIPT refrigerator. The difference between the theories of Radebaugh and Marc David is; Radebaugh assumed small sinusoidal oscillations of the gas pressure in the tube instead of gas flow in the tube as time dependent of the pressure oscillation. They could achieve a temperature of 3.2 K with a DIPT refrigerator configuration.

Wang *et al.* [19] developed a modified refrigerator called a double inlet reversible pulse tube (DRPT) refrigerator and the schematic of the same is shown in Fig.2.11. In a DRPT refrigerator, an auxiliary piston is used instead of the orifice and reservoir used for an OPT refrigerator, and the main and auxiliary pistons are arranged in the same axis and driven by the same flywheel. Numerical predictions show that the refrigeration power of the DRPT refrigerator is about three times greater than OPT refrigerator and the efficiency is doubled. Experimental results also show that the performance of a pulse tube is greatly improved by modifying to DRPT type refrigerator.

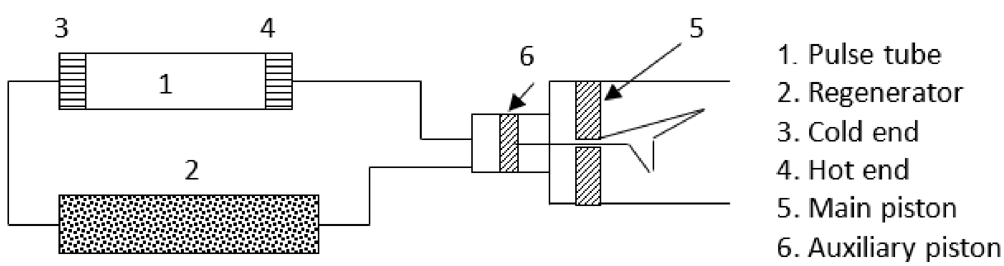
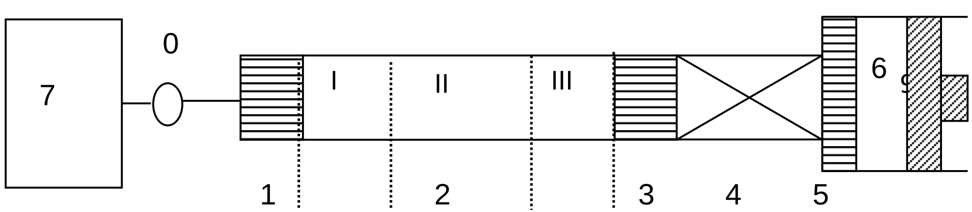


Fig.2.11 Schematic of a double inlet reversible pulse tube refrigerator [19]

Cai *et al.* [20] described the experimental results on the double inlet pulse tube refrigerator. The effects of varying the amplitude and phase difference of the pressure wave and mass flow were discussed. The main contribution of the double inlet is to adjust the phase shift between the pressure wave and the mass flow rate in the pulse tube and to increase their amplitude. There is an optimum matching between double inlet resistance and orifice resistance. The orifice can reduce the phase shift between the pressure wave and mass flow rate in the pulse tube, but the minimum phase difference is 48 degree instead of zero.

Wang *et al.* [21] developed a modified version of OPT refrigerator in which reservoir was eliminated with the objective of reducing the size of OPT refrigerator. Experiments and mathematical simulation were conducted with the so-called Modified Orifice Pulse Tube (MOPT) refrigerator. In MOPT refrigerator crankcase of the compressor was used instead of reservoir to bring the appropriate phase shift between the pressure and flow velocity in the pulse tube. From the comparative study, it was observed that MOPT refrigerator obtained a level of refrigeration power a little larger than OPT refrigerator. Also, a slightly more work is needed for MOPT refrigerator and has same efficiency as that of the OPT refrigerator.

Zhu *et al.* [22] in applied an isothermal model for simulating the pulse tube refrigerator. They considered the pulse tube as split type Stirling refrigerator and the gas inside the pulse tube was divided into three parts; the cold part which flows from the regenerator and expands to deliver work, the hot part which flows from the orifice and absorbs work and the middle part which never flows out of the pulse tube and is similar to a displacer in Stirling refrigerator. The schematic of the model is shown in Fig.2.12.



0. Orifice 1.Hot end heat exchanger 2. Pulse tube 3. Cold end heat exchanger
4. Regenerator 5. After cooler 6. Compressor 7.Reservoir

Fig.2.12 Layout of the numerical model of an orifice pulse tube refrigerator [22]

Liang *et al.* [23] idealized the pulse tube refrigeration process by simplifying the practical conditions without losing the main characteristics of pulse tube refrigeration. Based on the idealization, the thermodynamic non-symmetry effect of gas element working at cold end of the pulse tube has been described. The gas element enters the cold end of the pulse tube at wall temperature of cold end heat exchanger, but return to the cold end of pulse at much lower temperature. They termed it as thermodynamic non-symmetry in entering and leaving the pulse tube during one cycle. This effect had been conveniently used to explain the refrigeration mechanism of basic, orifice, and double inlet pulse tube refrigerator.

Liang *et al.* [24] developed the theoretical model was compared and validated with the experimental results. The influence of the important parameters, such as opening of the orifice and double inlet valves, frequency, average pressure, pressure oscillation amplitude in the pulse tube, diameter of the pulse tube on the refrigerator were investigated. The first series of experiments were focused on the influence of principal parameters on the cold end temperature. The optimum frequency was found to increase with decrease in pulse tube diameter, other parameters being constant. It was higher when the pulse tube works at higher temperature regions under the same pressure amplitude. The cold end temperature decreases as the average pressure decreases.

Thummes *et al.* [25] noticed that the use of double-inlet mode in the pulse tube cooler opens up a possibility of DC gas flow circulating around the regenerator and pulse tube. Numerical analysis shows that effects of DC flow in a single-stage pulse tube cooler are different in some aspects from that in a 4 K pulse tube cooler.

Xu *et al.* [26] analyzed the behaviour of the various gas elements that enter the tube of a pulse tube refrigerator from its cold end using the method of characteristics. They found that in an orifice pulse tube refrigerator, the gas elements can be divided into three parts. The specific cooling capacity produced by the second part of the gas element will be maximum. If the total mass is fixed, in order to improve the overall cooling capacity of an orifice pulse tube refrigerator, the ratio of the gas elements in the second part should be increased, while those in the first part and the third part should be decreased.

Tward *et al.* [27] tested the performance and flight qualification of miniature pulse tube cooler designed specifically for use on small satellites. They reported that the miniature pulse tube cooler is intended for greater than 10 year long-life space application and incorporates a non-wearing flexure bearing compressor vibrationally balanced by a motor controlled balancer and a completely passive pulse tube cold head.

Huang *et al*. [28] carried out an experimental study to derive a correlation for the design of an OPT refrigerator. Seven OPT refrigerators with different dimensions of pulse tube were tested and their performances were evaluated up to the cold end temperature for zero cooling capacity. It was shown experimentally that, there exists an optimum frequency, which increases with decrease in pulse tube volume. The experimental results were used to derive a correlation for the performance of an OPT refrigerator.

Kasthurirengan *et al*. in their technical report [29] detailed design parameters and experimental results have been presented for single stage G-M type DIPTR. Karunanithi *et al*. [30] have designed and developed a single stage G-M type double inlet pulse tube refrigerator. They have used a rotary valve for pressure wave generation.

Kasthurirengan *et al*. [31] tested a single stage pulse tube cooler of 7 W at 77 K. The pulse tube refrigerator can be performed in basic, orifice and double inlet type and examined their performances and variations of all three types. They found that the pressure wave form is in between the rectangular and sinusoidal shape. They finally concluded that double inlet type yields the best performance and refrigeration capacity.

Von *et al.* [32] described the cooling performance of a pulse tube extending to room temperature which is precooled by a single stage Refrigerator. They found that this system is possible to reach liquid helium temperatures without using rare earth compounds as regenerator material. Neveu *et al*. [33] developed both ideal and dynamic models for better understand the energy and entropy flows occurring in the pulse tube coolers. Ideal modelling is sufficient to quantify the maximum performance, which could be reached, but dynamic modelling is required to perform a good design.

Chen *et al*. [34] introduced a modified Brayton cycle predicting the thermodynamic performance of pulse tube refrigeration with a binary mixture refrigerant. They established theoretical expressions of cooling power, thermodynamic efficiency and required work of a refrigeration cycle.

Huang *et al*. [35] carried out an experimental steady on the design of a single stage orifice pulse tube refrigerator (OPTR). It was shown experimentally that there exists an optimum operating frequency, which increases with decreasing pulse tube volume. For a fixed pulse tube volume, increasing the pulse tube diameter will improve the performance. The experimental results are used to derive a correlation for the performance of OPTR, which correlates the net cooling capacity with the operating conditions and the dimensions of the OPTR.

Ju *et al.* [36] measured the flow resistance and flow inductance of inertance tubes at high acoustic amplitudes for different inner diameters at various tube lengths at different frequencies. Lu *et al*. [37] carried out numerical and experimental study on a single stage double inlet G-M pulse tube refrigerator, where the oscillating amplitude of physical quantities are large and oscillating frequencies are low in the system. They have measured the temperature distribution on the surface of the regenerator and the pulse tube, as well as the refrigeration capacities at different refrigeration temperatures under optimum operating conditions. A transient one-dimensional numerical simulator has been developed to verify experimental data and to study the nonlinear characteristics in the double inlet pulse tube.

Roy *et al*. [38] developed a single stage G-M type pulse tube refrigerator and carried out experimental studies that consists of cooling behaviour of the refrigeration system, cooling capacity at different cold end temperatures and optimization of orifice and double inlet openings. An indigenous rotary valve has been designed and developed to produce pressure pulsation and finally investigated the opening effects of orifice and double inlet by applying a differential pressure transducer across the orifice valve.

Baik [39] developed a reliable and scalable design tool for active valve G-M (Gifford-McMahon) type pulse tube refrigerators The design tool begins focuses on the limitations imposed by the reciprocating-type compressor commonly used for G-M type pulse tube refrigerators and maximizes the ideal pulse tube cooling power that can be produced from a compressor of fixed capacity. The setup of a single stage G-M type 5-valve pulse tube refrigerator yields 30 W at 30 K driven by a 5.5 kW compressor. They developed an analysis that provides an improved prediction of DC flow and shuttle heat loss and an enhanced ability to scale the design of G-M type pulse tube refrigerators. They have also investigated the results between the single stage G-M type double inlet and active 5 valve PTR and concluded that active 5 valve PTR yields better performance and refrigeration capacity.

Dang *et al*. [40] have designed and tested a set of Stirling-type non-magnetic and non-metallic co-axial pulse tube cryocoolers, intended to achieve portable cryogen-free systems with very low interference for high-Tc SQUIDs operation. Yong *et al*. [41] have examined individual loss associated with the regenerator and combined these effects to investigate size effects on the performance of Stirling cycle cryocoolers. For the fixed cycle parameters and given regenerator length scale, it was found that only for a specific range of the hydrodynamic diameter can produce net refrigeration and there is an optimum hydraulic diameter at which the maximum net refrigeration is achieved. Tanaeva *et al*. [42] developed a new three-stage pulse tube refrigerator (PTR) by scaling down a working model PTR by 50%. With He^3 as a working fluid a no-load temperature of 1.73 K is reached and a cooling power of 124 mW at 4.2 K is realized.

Masuyama *et al*. [43] has experimentally investigated a Stirling type pulse tube refrigerator with an active phase control. A phase shifter, which controls the phase angle between the mass flow and the pressure inside a pulse tube, plays a key role in the performance of pulse tube refrigerators. In this study, an electrically driven and mechanically damped linear compressor, which is directly connected at the warm end of the pulse tube using a connecting tube, is used as the active phase controller (APC).

Wang *et al*. [44] have constructed single stage four-valve pulse tube refrigerator (FVPTR) with a ‘L’ type pulse tube structure and two orifice valves at the hot end of pulse tube in order to simplify the structure of the cold end of the pulse tube refrigerator (PTR) and have a better utilization of the cold energy of the system. Verification by experiments shows that a two-orifice valve structure gives different adjustments to the gas flow rate of the hot end of the pulse tube than that of the one-orifice valve structure.

Kwanwoo Nam *et al*. [45] presented the experimental results and correlations on the friction factor of screen regenerators, being focused on the effect of cryogenic temperature. In their second paper [46] they described development of novel regenerator geometry for cryocoolers. They developed a parallel wire type which is a wire bundle stacked in parallel with the flow in the housing, which is similar to a conventional parallel plate or tube. They performed hydrodynamic and thermal experiments to demonstrate the feasibility of the parallel wire regenerator. The pressure drop characteristic of the parallel wire regenerator is compared to that of the screen mesh regenerator. De Waele *et al*. [47] studied the performance of pulse tube at very low temperatures. They found that the cooling power of pulse tube coolers is zero when the thermal expansion coefficient is zero.

Wang *et al*. [48] proposed a new type of copper foaming metal with high heat transfer area and low flow resistance in the heat exchanger instead of the copper screens. The heat transfer performances of the copper screens and the copper foaming metal are compared by theoretical calculation.

Qiu *et al*. [49] have optimized a three-layer regenerator, which consists of woven wire screen, lead sphere and Er_3Ni to enhance the cooling performance and explore the lowest attainable refrigeration temperature for a single-stage PTC. The efforts focus on the temperature range of 80–300 K, where woven wire screens are used. They have carried out theoretical and experimental studies to study the metal material and the mesh size effect of woven wire screens on the performance of the single stage G-M type pulse tube cryocooler.

Chen *et al*. [50] have analyzed heat transfer characteristics of compressible oscillating flow in two kinds of simple regenerators filled with circular tubes or parallel plates under assumption of small perturbation. They have applied linear thermoacoustic theory for analysis. They have derived exact expressions of Nusselt number in complex notation based on the cross-sectional oscillating velocity and temperature distributions.

Koettig *et al*. [51] have experientially investigated the direction and the quantity of transferred heat within a pulse tube refrigerator (PTR) in coaxial configuration. They located the pulse tube inside the regenerator matrix in axial direction. They found that an internal thermal contact between these two main components of the cold finger occurs. Results showed that intermediate cooling of the regenerator by the corresponding part of its own pulse tube can improve the cooling performance of a PTR. Therefore, a well-adapted geometrical arrangement between the pulse tube and the regenerator is essential.

2.6 High capacity single stage pulse tube refrigerators

Gan and Thummes [52] were the first to achieve 100 W at 80 K with a double valved double inlet configuration. They introduced a new phenomenon in form of a temperature hysteresis in a single stage G-M type orifice pulse tube refrigerator that depends on the heat load and on the adjustment of the needle valve connecting the pulse tube warm end with buffer volume. Also proposed that compared to the instability in some double inlet pulse tube refrigerators due to the DC flow in the cooling system, the single OPTR is usually considered to operate stably. This is the highest refrigerating capacity achieved by any of the single stage G-M type PTRs.

Zhu *et al*. [53] introduced a new type of waiting time effect of a G-M type PTR and examined that there is an optimum waiting time for the no load temperature, cooling capacity and efficiency. A no load temperature of 45.1 K and the cooling capacity of 45 W at 80 K achieved with 1^0 waiting time. They also proved experimentally that pressure difference across high pressures valve and low pressure valve are decreased by long waiting times and thus the cooling and efficiency are increased.

Deasi *et al*. [54] developed a thermodynamic model of a G-M type double inlet PTR and obtained 37 W at 80 K and investigated the effect of orifice valve opening, double inlet opening, and frequency on the performance of cryocooler in terms of net refrigeration power and no-load temperature. The basic fundamentals of cryogenics and low temperature properties of matter [55-56] can be easily referred and regained. The knowledge on the principle and working of various cryocoolers, liquefaction of various cycles can be thoroughly understandable.

Yanlong *et al*. [57] evaluated experimentally by the influence of DC flow induced by the introduction of double inlet on the refrigeration performance of the pulse tube cooler. Also investigated that the double inlet configuration was the best to reduce DC flows successfully instead if a conventional single valved one under different operation modes and achieved a cooling capacity of 35 W at 80 K.

Koh *et al*. [58] developed a design technology of PTR and acquired an output of 23 W at 80 K. A lowest temperature of 28 K was achieved with a single double inlet configuration. Also investigated the refrigeration performance of the basic, orifice and double inlet PTRs and examined the cool down rates will be higher for double inlet rather than orifice and basic pulse tube refrigerators. They also observed that the refrigeration capacity increases with the operating frequency in OPTR and has a maximum at the operating frequency of 2.5 Hz in the double inlet PTR.

Ravex and Rolland [59] developed characterized the pulse tube refrigerator by taking into considerations of wall heat pumping, enthalpy flow and regenerator efficiency and experimentally succeeded by achieving a refrigerating capacity of 20 W at 80 K with a single stage double inlet pulse tube refrigerator at no load temperature of 28 K. Further investigated the behaviour of orifice valve, double inlet valve opening and net refrigeration capacity.

Liang *et al*. [60] developed a new type of orifice pulse tube refrigerator, which could reach much lower temperature compared to that achieved by earlier designs. The relation between the ratio of regenerator volume to the pulse tube volume and minimum temperature of the orifice pulse tube was experimentally investigated. They also experimentally investigated the influence of the dimensions and the matrix materials of the regenerators on the performance of the orifice pulse tube refrigerator by obtaining 12 W at 77 K.

A G-M type single stage pulse tube cooler has been designed, fabricated and operated by Kasthurirengan *et al*. [61] with an indigenous helium cooler of 2 kW and performances are compared with 3 kW imported helium compressor, concluded that with increase in pressure ratio of the indigenous compressor, 50% more cooling power seen in case of indigenous helium compressor by achieving 7 W at 77 K. They also concluded that higher the pressure ratio and more trapezoidal the pressure wave form, higher is the cooling power of the pulse tube refrigerator with indigenous helium compressor.

A minimum temperature of 22.4 K and cooling power of 5.65 W at 80 K was obtained when 2 kW input power was inputted by Gan *et al*. [62]. Also investigated that the DC flows in the pulse tube refrigerators will be controlled well with double valved configuration.

Lu *et al*. [63] proposed a new phenomenon of dynamic pressure of various compressible flow oscillating at different locations in a G-M type PTR oscillating at cycle steady states with a cooling capacity of 2 W at 80 K. They examined the oscillating amplitude of the pressure was the largest at the hot end of regenerator while the cycle averaged pressure was the largest in the reservoir. In addition the effect of cycle averaged pressure on the refrigeration performance is discussed with proper asymmetric charging and discharging periods that has a better performance than a symmetric one in a G-M type PTR.

Radebaugh *et al.* [64] compared various pulse tube refrigerators and Stirling refrigerators using enthalpy flow model. He suggested that the displacer or expansion piston of the Stirling refrigerators, which was used to cause a phase shift between the mass flow rate and pressure, was replaced with irreversible heat transfer or irreversible expansion through an orifice to bring the necessary phase shift.

Storch *et al*. [65] developed an analytical model of the OPT refrigerator. However, the magnitudes predicted by the model were three to five times higher than experimental values because of over simplified assumptions used in the model. The enthalpy flow model developed by Storch and Radebaugh is not only subjected to large error (three to five times higher than the experimental values) in the performance prediction of OPT refrigerators, but also not applicable to the analysis of basic pulse tube refrigerators due to different working principles.

Radebaugh *et al*. [66] conducted experiments to determine the minimum temperature and maximum refrigeration power available with an OPT refrigerator driven by a compressor that yields a net refrigeration power of 2 W at 80 K was obtained. Three different pulse tube volumes were tried and the lowest temperature achieved was 67 K for the pulse tube having a volume of 7.9 cm^3. They also concluded that stainless steel was much better than phosphor bronze as screen material for regenerator because of reduced axial conduction.

Radebaugh [67] studied about the overall system performance with various sizes of compressor and did analytical modelling of pulse tube behaviour. The analogy between the pulse tube and AC electrical systems was first introduced. In this model, a low temperature of 26 K was achieved in two stages and experimentally carried out.

Marc David *et al*. [68] conducted research to achieve the efficiency of a Gifford Mc-Mahon cryocooler with a pulse tube refrigerator. Experiments were done on a newly introduced Hybrid pulse tube refrigerator (HPTR), OPTR and on G-M cryocooler. They obtained a 57 K limit temperature with a single stage and a net refrigeration power of 12 W at 72 K. The schematic of a Hybrid pulse tube refrigerator is shown in Fig.2.13.

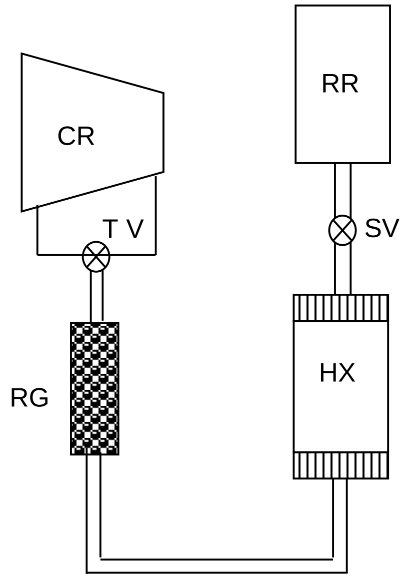


Fig.2.13 Schematic of Hybrid pulse tube refrigerator [68]

2.7 Multi stage pulse tube refrigerators

Wang *et al.* [69] developed a two-stage pulse tube refrigerator with a rotary valve and a valved compressor. A minimum temperature of 11.5 K and a cooling capacity of 1.3 W at 20 K were obtained with this arrangement.

Wang *et al.* [70] designed and constructed a two-stage double inlet pulse tube refrigerator for cooling below 4 K by the aid of numerical analysis. The hot end of the two stage pulse tube is connected to the phase shifting assembly at room temperature without the use of a regenerator tube. They used a three-layer second stage regenerator filled with ErNi0.9, CO0.1, Er3Ni and lead spheres and obtained a lowest temperature of 2.23 K and cooling power of 370 mW at 4.2 K and 700 mW at 5 K. Wild *et al*. [71] developed a two-stage pulse tube refrigerator, driven by 6 kW compressor equipped with magnetic valves. The first stage of their refrigerator when operated separately as double inlet system achieved more than 30 W at 80 K, the no-load minimum temperature was 26 K. The first stage refrigeration is limited to about 30 W to 50 W because most of the focus is on the three stage pulse tube cryocooler. Up to second stage pulse tube cryocooler, it achieves a very low temperature. Overwhelming the majority of the design provide the cause for a 200 W in the first stage, however if the pulse tube is going to be used to produce a liquid nitrogen at a capacity of 100 W or 200 W. It is considered as a high refrigerating capacity.

Wang [72] developed a computer program for numerical simulation of 4 K pulse tube coolers, which takes into account non-ideal gas properties of the magnetic regenerative material, and the heat transfers in the heat exchangers and regenerator. The suggested model was very efficient for visualizing physical process in 4 K pulse tube coolers. The numerical predictions were compared with the performance of existing pulse tube coolers for liquid helium temperature and were found to be in reasonable agreement. In their second part of the work [73], the processes and performance of 4 K pulse tube coolers and G-M coolers were analyzed by numerical simulation. Several features, such as flat temperature region in the low temperature regenerator and increased mass flow rate at the cold end of the regenerator were discussed in their analysis.

The important conclusions of the work were;

- The behavior of the 4 K regenerator, such as an extended region of constant temperature near 4 K and comparatively large mass flow rate at the cold end are completely different from the behavior of regenerators working at higher temperatures.
- A configuration of the pulse tube cooler where the phase shifter located at room temperature is not capable for an efficient phase shifting of the moving liquid helium at the cold end.
- Double inlet operation significantly improves the performance of 4 K pulse tube coolers by reducing mass flow rate and losses in regenerator. A DC flow through the double inlet tube was discovered in the simulation.

On the basis of results obtained from the pulse tube refrigerator that was able to produce a net cooling power of ~ 0.37 W at 4.2 K, a modified version has been built to use for small scale ^{4}He liquefaction by Wang *et al*. [74].

Xu *et al*. [75] investigated on the lambda transition of ^{4}He at low temperature. This acts as a barrier for reaching temperature below 2 K. Theoretical analysis in this paper shows that, using ^{3}He, the temperature limit is below 2 K, and the efficiency of a 4 K pulse tube refrigerator can be improved significantly. They constructed a three-stage pulse tube refrigerator and with ^{4}He they reached a minimum temperature of 2.19 K. Using ^{3}He, at the same valve setting and operating parameters, the minimum average temperature goes down to 1.87 K and the cooling power at 4.2 K is enhanced by 60%. After fine-tuning of the valves a minimum average temperature of 1.78 K was obtained.

Qui *et al*. [76] investigated the valve timing effects on cooling performance of a two stage 4 K pulse tube cooler. Their experimental results shows that optimization of valve timing can considerably improve the cooling performance of both stages.

Regenerative cryocoolers that employ ^{4}He as working fluids can only reach a lowest temperature of about 4 K. This limitation can be overcome by the use of ^{3}He as the working fluid and was experimentally proved by Jiang *et al*. [77]. For this, they analyzed the performance of a two-stage pulse tube cooler that consists of two parallel stages with independent gas circuits. This feature makes it possible to run only the second stage with either ^{4}He or expensive ^{3}He as the working fluid. With ^{3}He as the working fluid, the two stage reaches low temperatures between 1.27 K and 1.38 K, depending on the operating and optimization conditions. This is the lowest temperature achieved by any of the mechanical refrigerators. As compared to the operation with ^{4}He, the cooling power and cooling efficiency with ^{3}He are enhanced by 30-50% at 4.2 K. They also suggest that even lower temperatures than 1.27 K might be possible by replacing the $HoCu_2$ in the coldest regenerator by one of the new ceramic regenerator materials like GAP (GdAlO3) or GOS (Gd_2O_2S).

Qiu *et al*. [78] developed a simpler and more reliable pulse tube cooler driven by a thermoacoustic engine that can completely eliminate mechanical moving parts. A Stirling thermoacoustic heat engine has been constructed and tested. The heat engine can generate a maximum pressure ratio of 1.19, which makes it possible to drive a pulse tube cooler and get good performance. Frequency is one of the key operating parameters, not only for the heat engine but also for the pulse tube cooler. In order to adapt to the relatively low design frequency of the PTR, the operating frequency of the thermoacoustic heat engine was regulated by varying the length of the resonance tube. Driven by the thermoacoustic engine, a single stage double-inlet pulse tube cooler obtained the lowest refrigeration temperature of 80.9 K with an operating frequency of 45 Hz, which is a new record for the reported thermoacoustically driven refrigerators.

The lowest temperature reached by Gifford and Longsworth with the BPT refrigerator was 124 K with a single stage PTR and 79 K with a two stage PTR. The major important discovery happened when Mikulin *et al*. developed an OPT refrigerator and achieved a low temperature of 105 K using air as working medium. Later, Radebaugh reached 60 K with a similar device using Helium. Since then the development of PTR progressed much fast. In 1990, Zhu *et al.* connected the warm end of the pulse tube with the main gas inlet by a tube, containing second orifice and named the configuration as DIPT refrigerator. Matsubara used this configuration to reach a temperature of as low as 3.6 K with a three stage PTR. The same year low temperature group of Eindhoven University of Technology began the research work of PTR. In 1999, with a three stage DIPTR a temperature of 1.78 K was reached. In 2003, professors of Giessen University have developed a double circuit ^{3}He/ ^{4}He PTR that has reached 1.27 K. This is the lowest temperature achieved by any of the mechanical refrigerators.

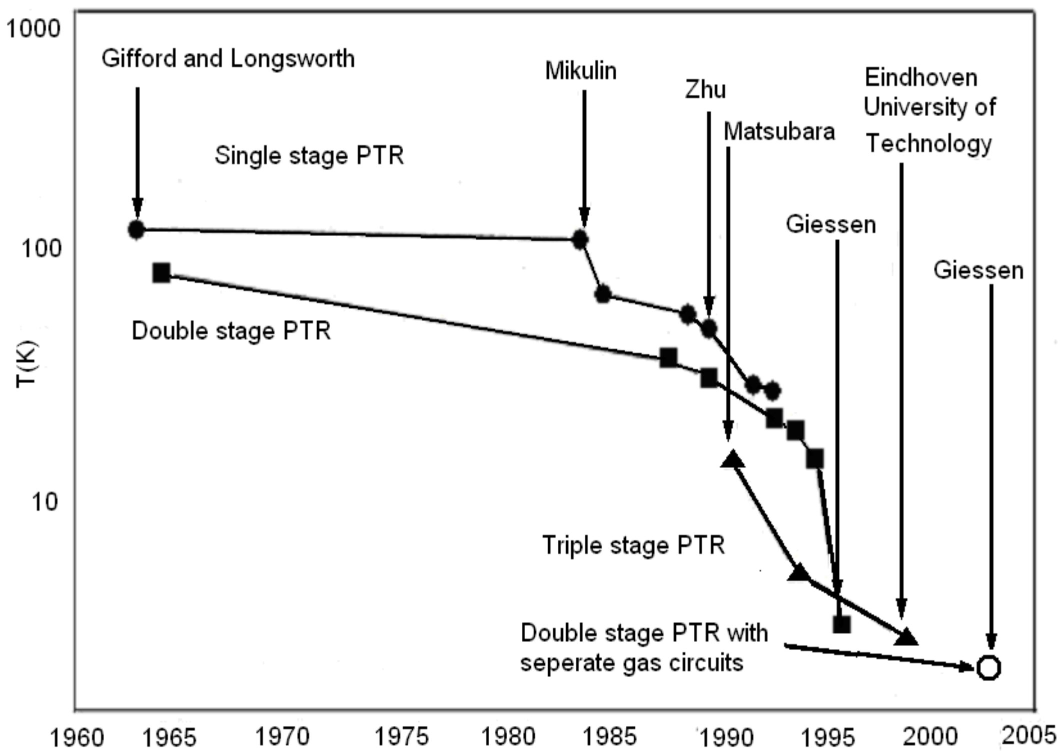


Fig.2.14 History of Pulse tube refrigerators [42]

2.8 Effect of cooling effect and low temperature

The cooling effect and low temperature have a major impact on the compressor, rotary valve, hot end heat exchanger and regenerator.

Compressor

The cooling effect of the pulse tube refrigerator depends mainly on the capacity and type of compressor. Normally, reciprocating helium compressors of high capacity ranging 1 kW to

6 kW are used for G-M type pulse tube refrigerators and low capacity compressors for Stirling type refrigerators. The cooling system for compressors is either water cooled or air cooled, but water cooled is most effective one for high capacity compressors. For pulse tube refrigerators of multistage, four valve type, five valve type and active buffer type pulse tube refrigerators use compressors of reciprocating with very high power capacity. A high cooling effect of 100 W at 80 K was achieved by university of Giessen [52] with a reciprocating helium compressor of 6 kW. Till now, G-M type PTRs have been commissioned successfully by 6 kW reciprocating compressors. At IIT Bombay [54] a cooling effect of 37 W has been achieved with compressor capacity of 6 kW. At IISc Bangalore [61] cooling capacity of 7 W has been obtained with a 3 kW compressor. Cryomech, Leybold, Sumitomo and Kirloskar etc. are the reciprocating helium compressor companies usually preferred.

Rotary valve

It is the main key component of the G-M pulse tube cooler that acts as a junction between compressor system and pulse tube system. It operates with a frequency between 1 Hz to 5 Hz. The actual construction of rotary valve depends on the suction, discharge ports of the compressor and regenerator inlet of pulse tube system. Detailed survey regarding functioning, construction is not available in open literature. Even proper mathematical modeling and CFD analysis does not exist for a rotary valve. Kasthurirengan *et al*. [79] developed an indigenous rotary valve for the applications of single stage and two stage G-M type pulse tube cryocoolers. The rotary valve has been designed to produce pressure waveforms in the frequency range from 1 Hz to 3 Hz. They have conducted the experiment with different configurations of rulon part in the rotary valve. The shapes of the gas flow passage in the Rulon part of the rotary valve plays an important role in determining the cold end temperature, since it determines the times during which the high and low pressures are applied to the pulse tube cryocooler. With the best configuration, a no load temperature of 62.3 K in first stage and 3.5 K in two stage has been achieved.

Hot end heat exchanger

The next most important component of pulse tube refrigerator. Normally, heat exchangers of water or air cooled type are being used. Out of those slit type, shell and tube type are most commonly used. For low capacity and high capacity pulse tube refrigerators, slit types and shell and tube heat exchangers are generally preferred. A slit type heat exchanger [61] has been used for both single and two stage pulse tube refrigerators and achieved no load temperatures of 67 K and 3 K. A low of temperature of 28 K and a cooling capacity of 37 W at 80 K [54] has been achieved with the application of shell and tube heat exchanger. A five way valve G-M pulse tube cooler [39] with a lowest temperature of 30 K is achieved by using a shell and tube heat exchanger with no baffles.

Regenerator

It is the key component that effects the cooling capacity and no load temperature by varying the mesh material and mesh sizes. Usually, stainless steel (SS) mesh of size ranging from 50 to 250 is commonly used along with other mesh materials copper, lead, Er_3Ni are used for achieving lowest temperatures. Major part of the matrix is occupied by stainless steel mesh and minor part by above aforesaid meshes. Kasthurirengan *et al*. [61] achieved a lowest temperature of 3 K by two stage G-M type pulse tube refrigerator with $HoCu_2$+Er_3Ni+Lead+SS meshes and 67 K with SS+ lead. They have tested with different configurations of mesh materials and sizes and finally concluded with the best suitable mesh material. Desai *et al.* [54] obtained a low temperature of 28 K with SS mesh of size 250 by G-M type pulse tube cryocooler. Radebaugh *et al*. [66] conducted experiments to determine the minimum temperature and a net refrigeration power of 2 W at 80 K was obtained. Three different pulse tube volumes were tried and the lowest temperature achieved was 67 K with the application of SS mesh of 300 and lead balls. Wang *et al*. [69] experimentally investigated a two stage OPTR. A lower temperature of 31 K was achieved in the second stage with maximum and minimum pressure levels of 0.95 MPa and 0.6 MPa respectively at a frequency of 5 Hz by SS+lead+Er_3Ni. Wang *et al.* [13] adopted a co-axial configuration of the pulse tube and regenerator to make the system small and compact. Experiments were conducted with this co-axial design and the influence of different parameters on the minimum temperature was investigated. A no load temperature of 62 K was achieved and about 2.5 W of net refrigeration power was attained at 77 K. Matrix material used is $HoCu_2$+Lead+SS. Tward *et al*. [27] developed a single stage and a two-stage pulse tube test cooler by varying mesh sizes and concluded that Er_3Ni+SS is suitable matrix material for regenerator. They claimed that, the unoptimized two-stage cooler has reached 26 K while rejecting heat above 300 K.

2.9 Cryocooler research in India

Basically, the cryocooler research in our country carried out in three major places and a major contribution in the field of low temperature area.

Indian Institute of Technology Bombay (IIT)

Mainly the macro research is concentrated on the Stirling cryocoolers which are widely used in space applications. Minor research focuses on other areas such as G-M and Pulse Tube Cryocoolers, Mixed Gas Refrigerants for Joule-Thomson Cryocoolers, Integration of Cryocoolers for Cooling Superconducting Magnets, Linear compressors driven cryocoolers. A first Stirling type pulse tube cryocooler of 15 W capacity at 77 K has been developed. The technology finds applications in re-condensation of nitrogen gas for MRI shield cooling, liquefaction of hydrogen and oxygen for space applications, and helium liquefaction for SQUID. The cooler has a modular compressor design, and produces cryogenic temperatures without the use of displacers. At IIT Bombay, various pulse tube cryocoolers have been designed and developed; the technology has been transferred to users and attempts are made to reach down to lower and lower temperatures using multi-staged cryocoolers with minimum power input to the compressor. Their work has been diverted to pulse tube cryocoolers where massive application can be seen in low temperature application. In the year 2007-2010 a mixed refrigerant J-T cooler has been developed using simple Air Conditioning compressor to reach down to low temperature. They have used a mixture of 6 gases. With the mixture of Neon-11 in the J-T cooler, a cooling capacity 6.1 W at 80 K / 21 W at 100 K by Compressor Power of 868 / 1031 W. A lowest Temperature of 65 K is obtained. In the year 2006-2009 a two stage pulse tube cooler with a low temperature of 25 K was achieved. Different configurations such as inline type, U-type and co-axial type are developed with the two stage pulse tube cooler and compared the efficiency of each of them. With the Inline type and U-type lowest temperatures of 50 K and 70 K and cooling effect of 6 W at 80 K, 1 W at 80 K have been obtained. With the co-axial configuration a lowest temperature of 89 K was achieved. Present research is carrying out in Investigation on Cryocooler based helium recondensing cryostat, thermodynamic and engineering investigations of 4.5 K helium cryogenic systems focusing on cold compressors, J-T cryocoolers using sorption compressor, multistage pulse tube cryocooler [80].

Indian Institute of Science Bangalore (IISc)

It focuses mainly on both Stirling and G-M type refrigerators as well as pulse tube type. They have developed a single stage G-M type pulse tube refrigerator with a cooling effect of 7 W at 77 K and a low temperature of 40 K has been achieved by an air-cooled compressor of 1.6 kW. An indigenous rotary valve has been successfully developed and used for single stage and two stage GM-type pulse tube refrigerator is highly appreciated. They have commissioned the single stage pulse tube refrigerator with one indigenously developed a water cooled reciprocating helium compressor of 3 kW capacity and a imported helium compressor. They have achieved 7 W at 77 K and a no load temperature of 37.5 K with developed compressor and 6 W at 80 K with an imported one. A two stage G-M type pulse tube cooler has been developed and achieved a lowest temperature of 2.5 K. They have tested with different mesh materials of varying sizes like that of 14 configurations. This is lowest temperature ever achieved with pulse tube cooler in Indian history. A research on Numerical simulation of pulse tube refrigerator with inertance tube is carried out by employing One-dimensional form of the time-dependent equations. Equations are discretized using second-order upwind differencing for the convective terms. The formulation consists of other algebraic equations for quantities such as the properties of the gas and matrix, friction factor and the Nusselt number. Projects have been carried out in development of space pulse tube cryocooler, zero-helium loss magnet cryostat using hybrid cryocoolers and pulse tube based liquid helium re-condenser system. Present research is focused on development of helium recondensation system based on two stage pulse tube coolers and thermoacoustic pulse tube refrigerator operating down to 100 K [81].

National Institute of Technology Surat (NIT)

It works on G-M pulse tube coolers and thermoacoustic refrigerators in collaboration with IIT Bombay. A lowest temperature of 30 K and cooling effect of 37 W at 80 K [54] with a single stage G-M type pulse tube cryocooler. Present research is focused on design, development & experimental investigations on two stage G-M type pulse tube cryocooler at 10 K applications, development and investigations on thermoacoustic refrigerator.

The dimensions of regenerator, pulse tube, buffer and other important parameters for the present experimental set-up are chosen from the table 2.2 according to the desired output i.e. on the cooling capacity and low temperature. Most of the various configurations of pulse tube cryocoolers of high and low capacity are not widely available in the market which are commercial.

Table 2.2 Data obtained from literature review

Ref.	Year	Input (kW)	Avg. press (bar)	Freq (Hz)	Regen. dia. (mm)	Regen. length (mm)	Pulse tube dia. (mm)	Pulse tube length (mm)	Buffer (lit)	Output W @ K
52	2004	6	17.5	1.4	48	200	41	200	2.5	100@80
53	1998	3.3	17.65	2	55	94	50	202	4.4	45@80
54	2008	6	16	1.7	38	175	25.4	250	1	37@80
57	2004	-	16.5	-	32.35	129	28	155	-	35@80
58	1997	2.2	14.7	1.5-2.5	21	203	35	100	1	23@80
59	1992	-	12	3	18	-	14	200	-	20@80
60	1990	-	10.6	8.3	36	200	19	420	1	12@77
61	2004	3	15.5	2.3	19	210	14	250	0.5	7@77
62	2004	2	14	2	20	210	14	220	0.5	5.65@80
63	2002	3	18.3	2	18.3	150	12.7	180	0.5	2@80

Chapter 3

Design and Fabrication of Pulse Tube Refrigerator

3.1 Introduction

The main components of the pulse tube refrigerator such as regenerator, pulse tube, hot end heat exchanger and reservoir have been designed and fabricated. The present pulse tube cryocooler is of single stage double inlet configuration. It has been designed for a cooling capacity close to 100W to 200 W. Detailed drawings of the components are available in appendix.

3.2 Regenerator

Regenerator is a thermal energy storage device. The thermal energy is stored in porous matrix of high heat capacity material and used to heat and cool a fluid flowing through the matrix. The matrix cools the incoming fluid stream to working temperature and warms the exhaust stream to ambient. Another way a matrix is cooled by the exhaust stream and warmed by the incoming stream. It maintains a constant temperature gradient over the inlet and outlet at steady operating condition. The regenerator used in the experiments is stainless steel tube of external diameter Φ51 mm, 180 mm in length with 1 mm thickness is shown in Fig.3.3.

Regenerator Materials

Regenerator materials and geometries are to be selected based on the temperature range over which they are most commonly used. The most commonly used woven wire screen used for the regenerator is stainless steel because it is easy to weave in to the screen. It is used over temperature range from 30 to 300 K, where it provide the following advantages.

- Low pressure drop
- High heat transfer area
- Low axial conduction
- High heat capacity

Wire mesh screen

The woven wire of stainless steel mesh screen is most commonly used regenerator material. It is readily available in useful mesh sizes from 50 mesh to over 250 mesh. It is available in different materials and relatively inexpensive to use. The small diameter and high thermal conductivity of the wire used to weave the screen provides full utilization of the thermal capacity of the material. In the present case, stainless steel mesh screens of size 250 and copper mesh screen size of 40 have been taken.

The stainless steel wire mesh is first cut in to roughly square pieces and stacked one over another till a long stack is obtained. Then this stack is machined on a conventional lathe to get the circular stack of meshes to be fitted in to the tube. This is done to obtain a tight packing inside the regenerator tube and to minimize occurrences of air spaces, to increase its heat capacity and hence its effectiveness. For every tenth layer of stainless steel mesh, copper meshes have been inserted in order to maintain the temperature uniformity.

Optimization of the regenerator is one of the main problems associated with the development of a pulse tube refrigerator. For example, by increasing the filling factor, the pressure drop becomes higher. Another difficulty is regarding the fixation of the regenerator material inside the regenerator. In the experimentation of pulse tube cryocooler, the regenerator material is very standard and the mesh type commonly used is stainless steel of mesh size 250. Copper is also used along with stainless steel for temperature uniformity. When it achieves low temperatures, the specific heat of stainless steel will become very small and then the preferred regenerator material are lead balls, Er_3Ni etc. However, the optimization of the regenerator material and fixation are highly complicated in the experimentation.

Fig.3.1 (a) Stainless steel mesh

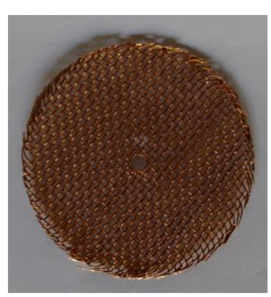


Fig.3.1 (b) Copper mesh

Fig.3.2 (a) Top flange of Regenerator

Fig.3.2 (b) Bottom flange of Regenerator

Fig.3.3 Photographic view of Regenerator

3.3 Pulse tube

The pulse tube is most critical component of the whole refrigeration system. This is the component where main functioning works. But geometrically, as well as from the fabrication point of view this is the simplest component of the system. Only a thin walled stainless steel tube is used to reduce the axial heat transfer over the large temperature gradient between the cold and hot end heat exchangers. The main objective of the pulse tube is to carry the heat from the cold end to the warm end by an enthalpy flow. The pulse tube used in the present case is stainless steel tube of external diameter Φ45 mm, 250 mm in length of 1 mm thickness with end flanges is shown in Fig.3.5.

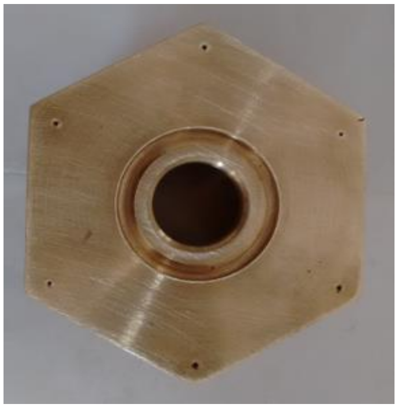


Fig.3.4 (a) Top flange of Pulse tube

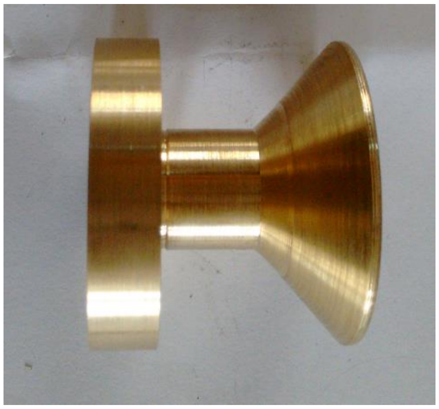


Fig.3.4 (b) Bottom flange of Pulse tube

Fig.3.5 Photographic view of Pulse tube

3.4 Hot end heat exchanger

Hot end exchanger is where the gas rejects heat of compression in every periodic cycle of operation. Upon receiving the enthalpy flow from the pulse tube, the heat load at a higher temperature is rejected to the environment. A shell and tube type heat exchanger has been designed and fabricated to extract heat out of helium gas at the hot end of pulse tube [39]. Helium gas flow through a total of 55, 4 mm outer diameter with 0.5 mm thickness capillary copper tubes that are cooled by a continues flow of 15^0 C cold water from the chiller. The outer shell of heat exchanger is made of Φ55 mm outer diameter, thickness of 5 mm and length 30 mm. Holes of 4 mm have been drilled equally on two circular plates and baffles of 45 mm in diameter with each 3 mm thickness. The bottom flange of hot end heat exchanger is fixed to top flange of vacuum chamber with O-ring seal and nut-bolt arrangement. The top end is made convergent for proper flow distribution which is connected to stainless steel tube of diameter of 6 mm. The convergent section has been fixed carefully with the shell by thread arrangement and with araldite the gap has been blocked by ensuring leak proof. Two holes of 6 mm diameter have been drilled on the shell to make proper flow distribution for water inlet and outlet from the chiller. Schematic view of hot end heat exchanger is shown in Fig. 3.6. Detailed fabrication views of hot end heat exchanger are shown in Figures (3.7 to 3.10).

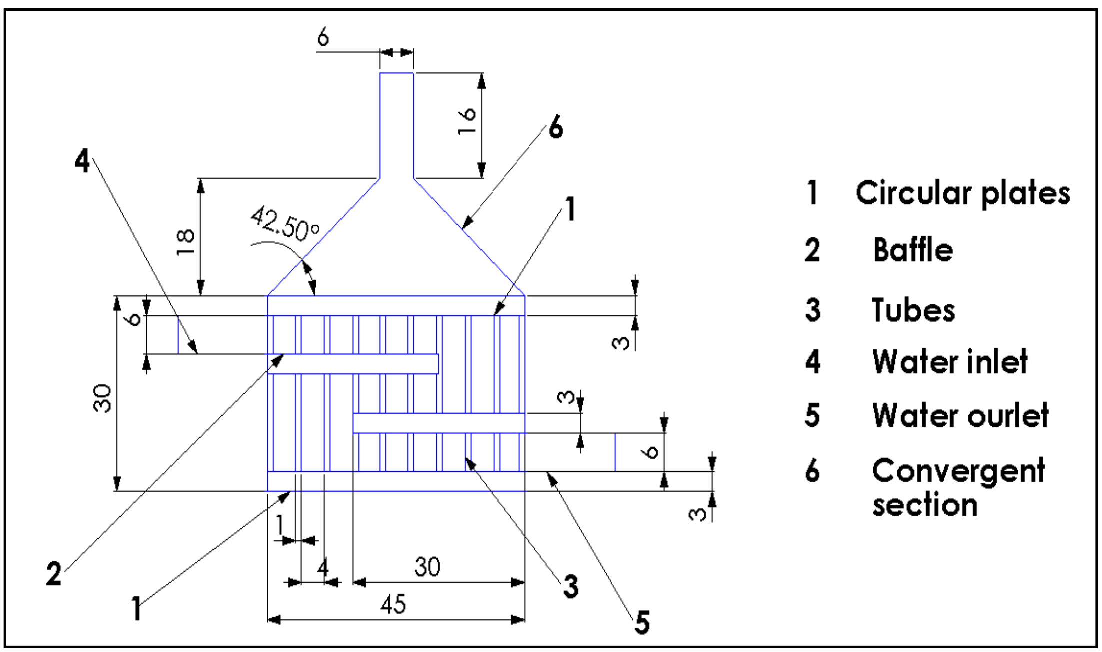


Fig.3.6 Schematic view of Hot end heat exchanger

Fig.3.7 Shell of Hot end heat exchanger with a flange

Fig.3.8 Circular plate

Fig.3.9 Convergent section Hot end heat exchanger

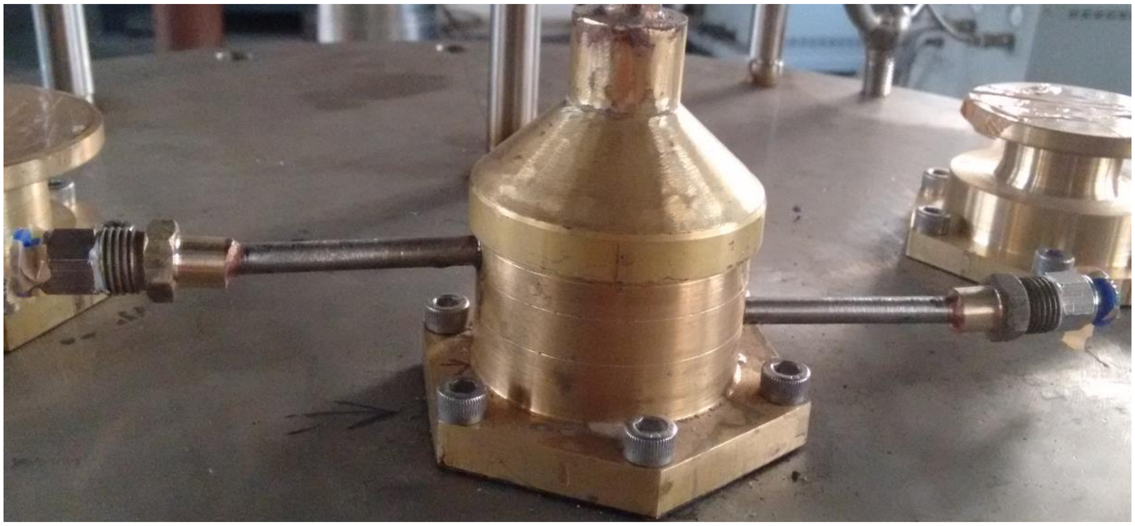


Fig.3.10 Photographic view of Hot end heat exchanger

3.5 Reservoir or Buffer

The reservoir or buffer is mainly used to stabilize pressure oscillations of the system where gas comes out from the orifice. It helps to keep the gas pressure more or less constant. The reservoir is made of stainless steel with a volume of 3 liters. A schematic view of reservoir is shown in Fig.3.11.

Fig.3.11 Photographic view of Reservoir

3.6 U-tube

It is a passage connecting regenerator and pulse tube where helium flows smoothly between them. The U-tube that connects the regenerator and pulse tube is of soft copper tube with external diameter of 12 mm and thickness of 1 mm is shown in Fig.3.12.

Fig.3.12 Photographic view of U-tube

Chapter 4

Construction of Experimental Test-rig

4.1 Introduction

Though the theoretical and analytical investigations on pulse tube refrigerator have been carried out from early stage of its invention and continue till date, the most of work is primarily experimental investigation; which plays a very prominent role in its development. The double inlet pulse tube refrigerator system consists of a compressor, regenerator, pulse tube, reservoir, heat exchanger and the valve system. This chapter gives a detailed construction and other components required for the experimental setup.

4.2 Experimental Technique

Experimental studies on the three common types (BPTR, OPTR and DIPTR) of the pulse tube refrigerator have been investigated in the present work. The present objective of the study is to develop the design technology of pulse tube refrigerator and find out its optimum operating condition and its performance with respect to various operating conditions. As a preliminary test, the refrigeration performance of the basic, orifice and double inlet pulse tube refrigerators has been investigated according to their cooling behavior and minimum attainable temperature at no load condition. The important studies made are listed below along with their objectives.

Cool down behaviour of the system

It is required to investigate the cool down behaviour of the system to know the required time for reaching the equilibrium state.

Effect of pressure ratio

Performance of pulse tube cooler is strongly dependent on the pressure ratio i.e. the ratio of highest to lowest pressure because this parameter determines the range of compression and expansion work and fluid characteristics inside the system.

Effect of flow resistance

Performance variation of the different pulse tube refrigerators are mainly due to phase relationship between the pressure and mass flow in the system. This is achieved with the help of flow resistance devices i.e. orifice and double inlet valves and controlling their opening. To determine the optimum operating condition, data has been taken at different opening of the valves.

Performance comparison

Comparison among the different types of pulse tube refrigerators (BPTR, OPTR and DIPTR) has been conducted at various pressures, valve openings and at different cold end temperatures.

The indigenously developed pulse tube refrigerator test rig consists of several sub systems such as compressor, rotary valve, regenerator, pulse tube, hot end heat exchanger, flow resistance valves, reservoir or buffer and a U-tube. A vacuum pumping system has been used to provide thermal insulation outside the pulse tube system. The schematic of the experimental set up has been shown in Fig.4.1. All the accessories have been discussed separately along with their specifications, design criteria and fabrication. The experimental test rig has been made in such a way that it can be operated as basic, orifice as well as double inlet type to facilitate the requirement of comparative study among them.

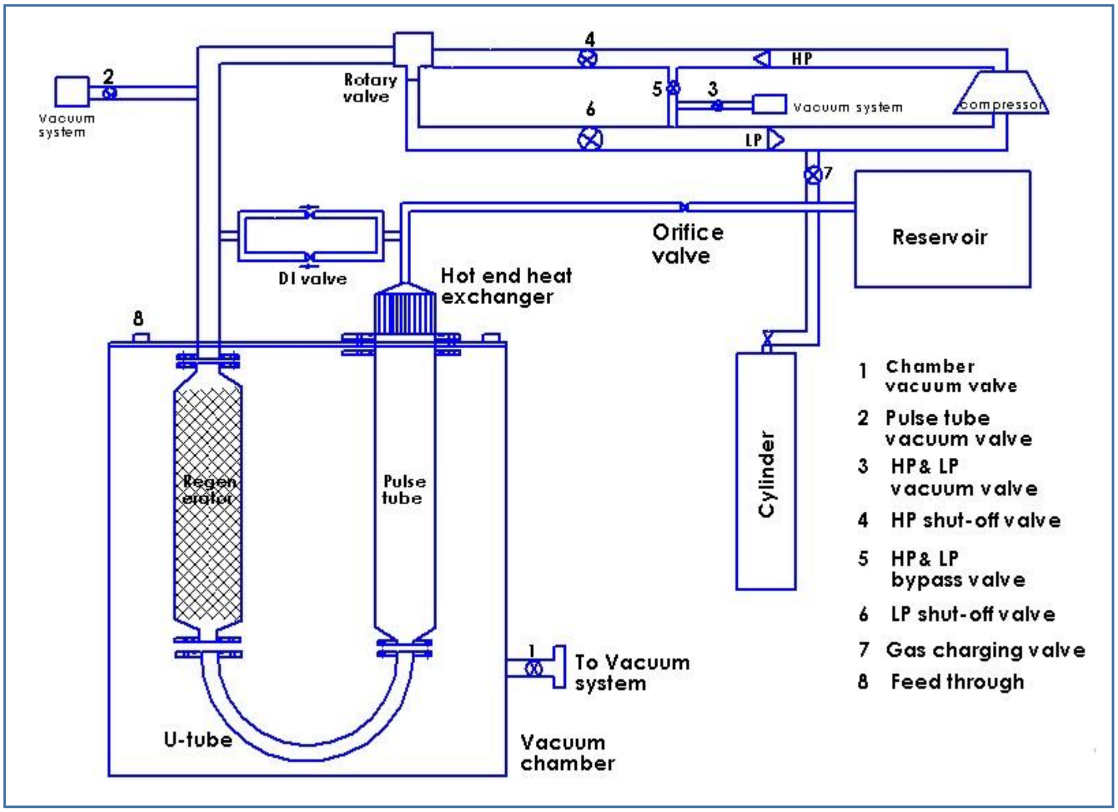


Fig.4.1 Schematic view of Experimental set-up

The test set-up has been designed and developed for a scroll compressor of power 8 kW. But it has been tested successfully with a reciprocating helium compressor of power 1.5 kW. The performances and its effects have been discussed in chapter 5.

Fig.4.2 Experimental test-rig of Pulse Tube Refrigerator

4.3 Compressor

The main role of the compressor in a pulse tube refrigerator is to deliver high pressure working fluid to the regenerator and pulse tube. A 1.5 kW reciprocating helium compressor (Cryomech made model CP04) has been used for experimental purpose is shown in Fig.4.3. The high pressure (HP) port and low pressure (LP) port of the compressor are directly connected to the rotary valve with help of suitable connecting mechanism, which alternately connects the HP, LP line and the regenerator hot end. This is operated at a high pressure of 15 bar and low pressure of 4 bar.

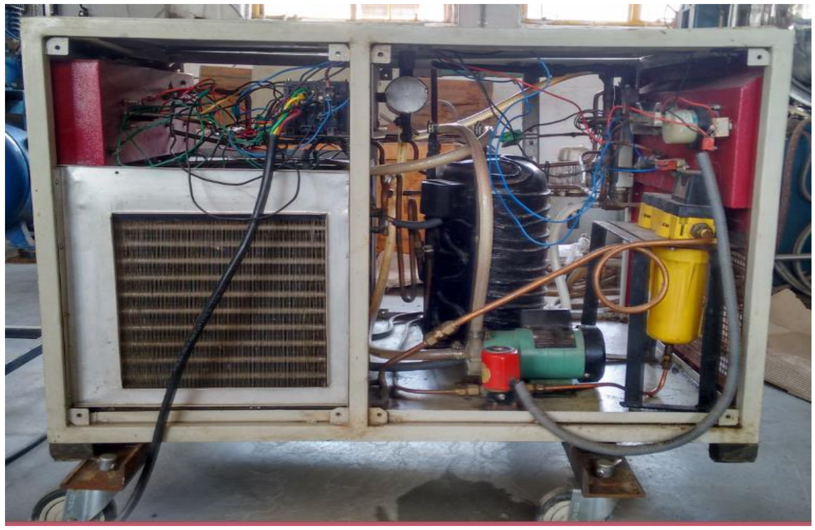


Fig.4.3 Photographic view of reciprocating helium Compressor

4.4 Metering Valves

Most of the cases adjustable needle or metering valves are used as flow resistance devices. Orifice and Double inlet valve are most common flow resistance in pulse tube refrigeration system. Minor orifice and multi bypass valves are also used in more advanced stage of research. These are the most vital components of a pulse tube refrigerator that helps to get minimum temperature as well as high cooling capacity providing better phase shift between mass flow and pressure flow oscillation of the system. These valves are highly accurate and has large number of turns to facilitate various opening positions. Three metering valves (Swagelok, SS-6MG-MM-MH) of 6mm size have been used for this purpose and is shown in Fig.4.4 (a) and Fig.4.4 (b). The end connections are ferrule type and are suitable for a 6 mm OD stainless steel tubes.

Fig.4.4 (a) Metering valve

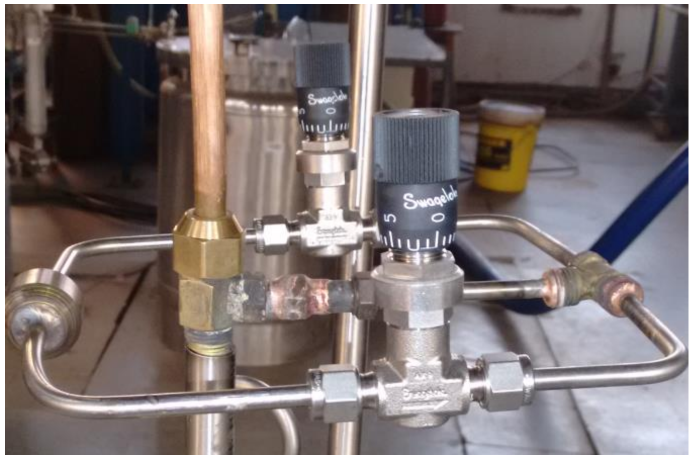


Fig.4.4 (b) Double inlet configuration

4.5 Rotary valve

The rotary valve is one of the critical components of most cryocoolers such as Gifford-Mc-Mohan and pulse tube. It is used to switch high and low pressures from a helium compressor to the required system. The commercially available cryocoolers have the pressure alternating arrangements already built into them. It is a valve which interfaces the helium compressor and pulse tube refrigerator system. The rotary valve has a rulon which is made to rotate with the help of a synchronous motor against an aluminum block with predefined passages connecting the high and low pressures to the pulse tube side. By the rotation of the Rulon part, the pulse tube side is alternately connected to high and low pressure side of the helium compressor. The frequency of the rotary valve is 2 Hz. Detailed description of the working of the rotary valve is available in reference [38]. The rotary valve that has been used for the experimental purpose is shown in Fig.4.5.

Fig.4.5 Photographic view of Rotary valve

4.6 Vacuum pumping system

The pulse tube & regenerator assembly are kept inside the vacuum chamber to prevent direct contact with the atmosphere and to facilitate the accurate reading of the refrigeration power produced i.e. to maintain proper thermal insulation between the system and surrounding. So the regenerator, pulse tube and U-tube (cold ends of regenerator and pulse tube) are placed in a cylindrical vessel made of stainless steel of length of 700 mm, diameter 44 mm and 3 mm wall thickness with proper top & bottom flanges is shown in Fig.4.7. This has a vacuum port of size KF-32 type, which can be used for evacuation. The chamber has been coupled to rotary pump to maintain a low pressure of 10^{-2} mbar which is shown in Fig.4.6.

Fig.4.6 Vacuum pumping system

Fig.4.7 Vacuum chamber

4.7 Valve manifold

The valve manifold is required for evacuation of lines, charging of pulse tube and helium compressor is shown in Fig.4.8. In the experimental set-up, total 7 valves has been used and out of which three valves (valves 1, 2 and 3) are used for evacuation of vacuum chamber, pulse tube system and HP, LP lines of compressor. Two shut-off valves (Valve 4 and valve 6) have been used in order to isolate rotary valve and compressor. Valve 5 acts as bypass for HP and LP lines of compressor .Valve 7 is used for the accomplishment of gas charging purpose to the pulse tube system.

Process of purging and operation of valves

- The whole pulse tube system must be charged with Helium gas from the cylinder with valve 7 through LP line of compressor by operating valve 6. Valves 4, 5 are also opened and other valves are closed. The compressor is in motion at the time of charging.
- Then evacuation has been carried out in order to avoid air pockets by valve 2 and valve 3. All the remaining valves are in operational. This process of charging and evacuation continues for 2-3 minutes until the whole system has been charged with Helium gas.
- During running of the experiment, only valves (4, 6) are in operational and other valves (2, 3, 5, and 7) remained closed.
- Valve 1 is continuously operational for supply of vacuum to the vacuum chamber. This valve isolates vacuum chamber and vacuum pump.

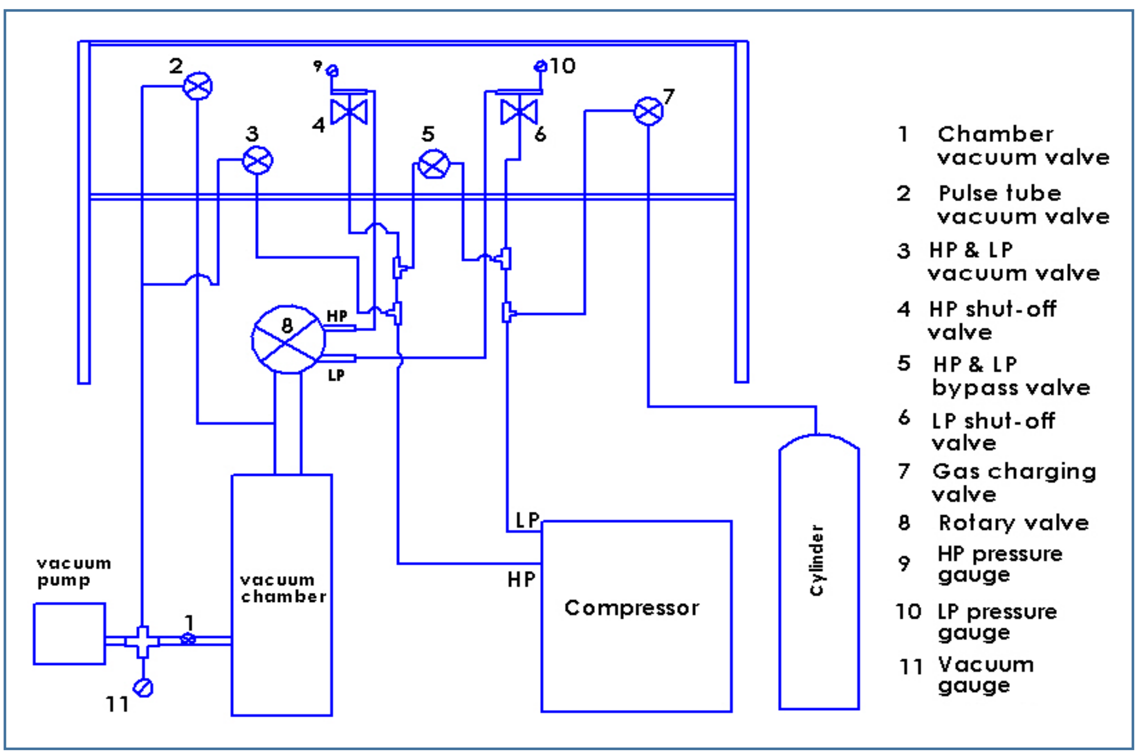


Fig.4.8 Schematic view of Valve manifold

4.8 Connecting tubes

The connecting tubes that have been used are copper of 3/8 inch, 1/4 inch in diameter and stainless steel tube of 6 mm in diameter. They have been cut to desired lengths before fitting them onto the apparatus.

4.9 Pulse tube Assembly

One side of the pulse tube and the regenerator with flanges are connected to the flanges of U-tube and similarly other ends have been connected to the top flange of vacuum chamber by O-ring seal and nut-bolt arrangement such that leakage should be minimum. Leak check has been tested for the entire pulse tube system and connecting tubes. A complete pulse tube assembly has been depicted below in Fig. 4.9.

Fig.4.9 Typical assembly of pulse tube refrigerator before and after incorporation of pressure and temperature sensors.

4.10 Procedure of Operation

After the purging process, the pulse tube refrigeration system is ready to operate. The detailed operation procedure is as follows:

- All the instruments are turned ON to monitor the pressure and the temperature of the system.
- The vacuum pump is connected to the vacuum chamber. The vacuum chamber is evacuated to a pressure level less than 10^{-2} mbar.
- Ensure the cold water supply from the chiller unit to the hot end heat exchanger.
- The initial pressurization is started by opening the regulating valve slowly of the high-pressure helium gas cylinder through LP line of compressor. The refrigeration system and the compressor are charged with helium gas to 12bar pressure.
- After the pressurization, the regulator valve, valves (7, 5) are closed and the gas is allowed to enter the system by opening the respective valves (4, 6).
- Switch ON the compressor and wait for one or two minutes such that the pressure is stabilized in the system.
- Pressure and the temperatures are monitored as a function of time until the steady state conditions are obtained.
- The orifice and double inlet valves can be adjusted during the operation.
- The no load temperature is measured and the corresponding pressure waves are recorded on the oscilloscope.

4.11 Instrumentation

The measurements taken with the various types of refrigeration systems provide a description of the effect of regenerator and pulse tube geometry on the minimum cold end temperature. The information is provided in the form of pressure and temperature measurement which are linked with digital storage oscilloscope, 16 channel RTD scanner and PC based data acquisition system.

4.11.1 Pressure Sensors

The measurement of dynamic pressure at the hot end of the regenerator, cold end of pulse tube, before the orifice valve and buffer are more important parameters for the PTR system as the whole cooling mechanism is fully responsible for pressurization and depressurization of gas inside the system. Piezoresistive transducers (Endevco, 8510B-500) as pressure sensors (P1, P2, P3, P4) have been employed for monitoring the dynamic pressure. Photographic view of the pressure sensor and their positions have been shown in Fig. 4.10 and Fig. 4.12.

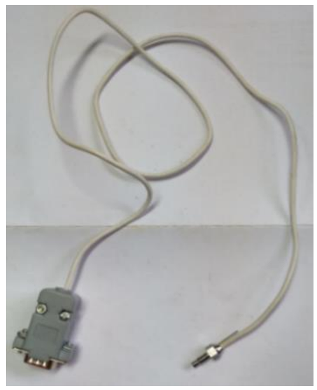


Fig.4.10 Photographic view of Pressure sensor

Table 4.1 Specifications of Piezoresistive pressure transducer

Operating pressure	0-500 psig
Resonance frequency	500 kHz
Sensitivity	0.6 ± 0.2 mV/psi
Excitation	10-18 Vdc

4.11.2 Temperature Sensors

These are the most vital parameter in any refrigeration system. Wire wound and thin film type are the two types of temperature sensors. For the present, PT-100 resistance thermometers (T1 to T10) of thin film type have been used to measure temperature at different points of pulse tube system is shown in Fig. 4.12. The temperature range of the sensors are in range of -200^o C to 200^o C and the response time is of 0.25 seconds. All the points have been connected to an output line connected to a data acquisition system to PC and 16 channel RTD scanner. Platinum Resistance thermometers (PRT) have been employed to measure temperatures of the different longitudinal position of the pulse tube wall and regenerator wall. The resistances of individual platinum sensor have been measured at 0°C and room temperature. All of them showed identical temperature at the respective temperature levels within an accuracy of 0.2° C. A thin film type PRT is shown in Fig. 4.11.

Fig.4.11 A thin film type PT-100 sensor

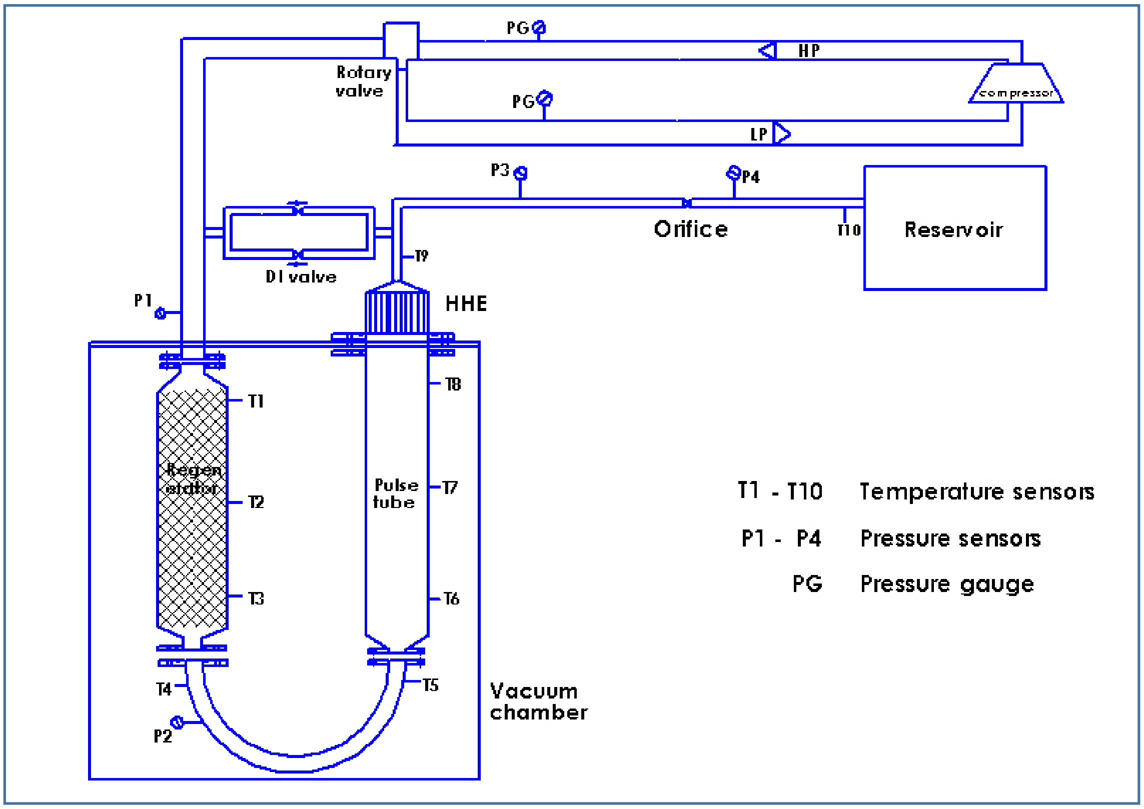


Fig.4.12 Schematic diagram of Pulse tube Refrigerator indicating Pressure and Temperature sensors

Feedthrough

Two feedthrough are fixed above the top flange of vacuum chamber in such a way that pressure and temperature sensors can be measured and monitored.

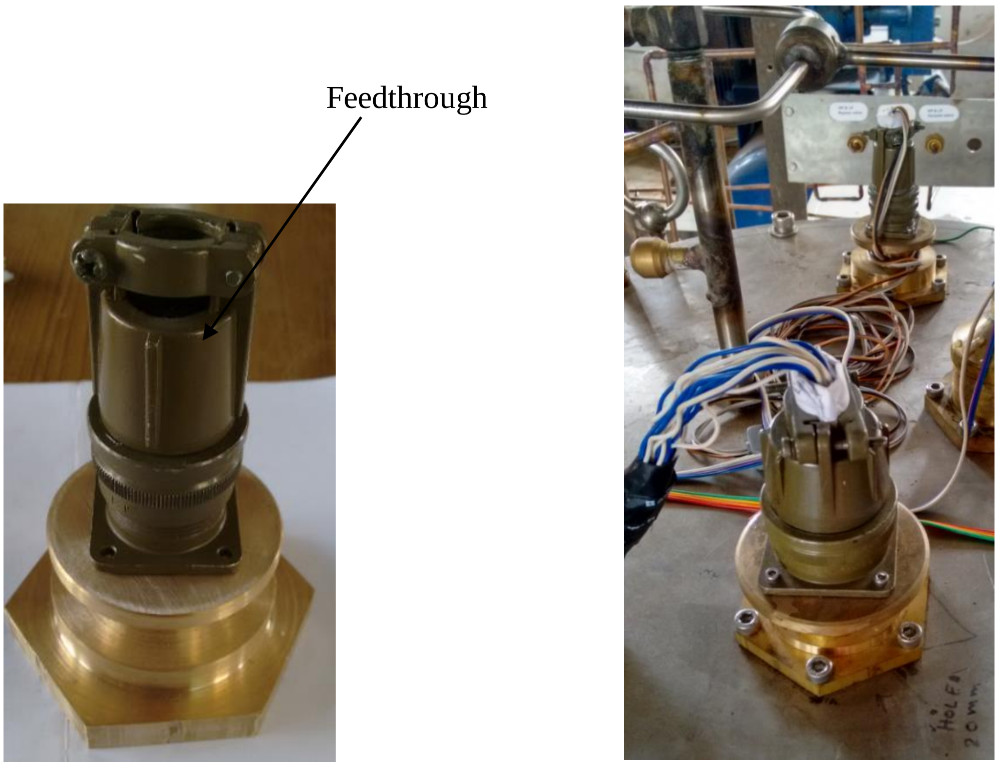


Fig.4.13 Feed through for temperature and pressure sensors

4.11.3 Data Acquisition system

The data acquisition system are the most vital accessory in any experimental set-up. The necessary equipment have been used to monitor the data and are briefly elaborated in temperature and pressure measurements.

4.11.3.1 Temperature measurements

A 16 channel RTD scanner and ADAM module have been used as data acquisition system for the measurement of temperature. The temperature sensors have been connected to a PC through ADAM 4000 series data acquisition module. The ADAM 4000 series is a set of intelligent sensor to computer interface modules containing built-in microprocessor. They are remotely controlled through a simple set of commands issued in ASCII format and transmitted in RS-485 protocol. The host computer has been connected to the RS-485 network with one of its COM ports through the ADAM RS-232/RS-485 converter which has been transformed the host signals to the correct RS-485 protocol. One 6-channel data acquisition module (4015) has been to acquire all the signals. The output voltages obtained from different sensors varies from millivolt to volt range. Schematic arrangement ADAM module and the data acquisition system are shown in Fig.4.14 and Fig.4.15.

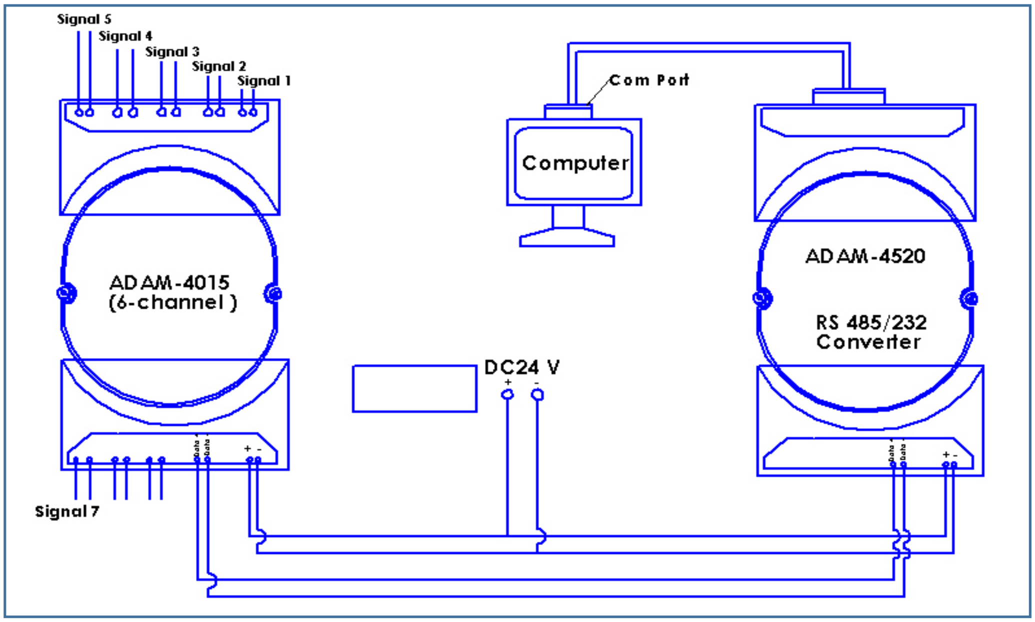


Fig.4.14 Schematic view of arrangement of ADAM module

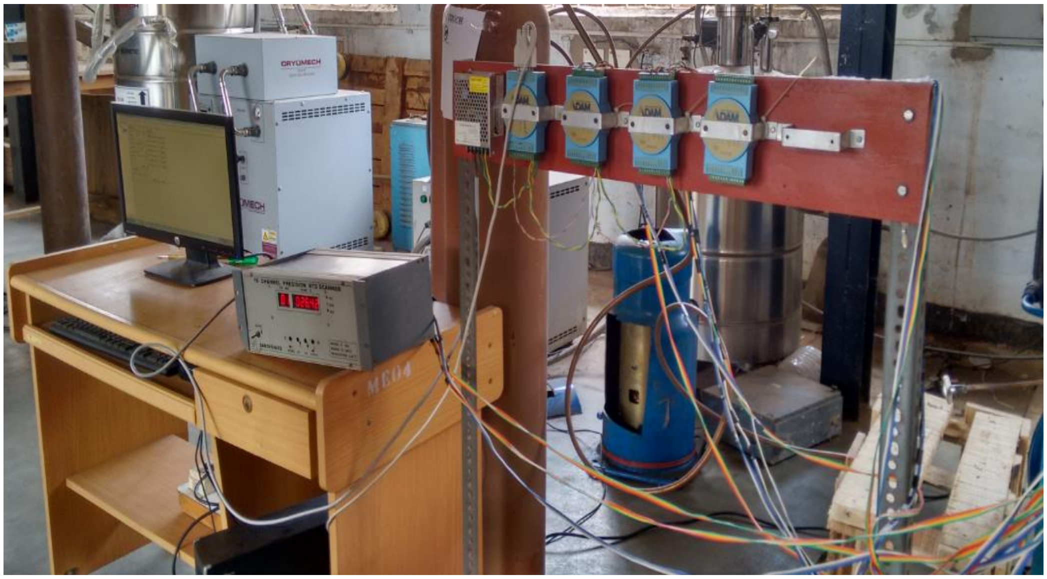


Fig.4.15 Data acquisition system for temperature measurements

4.11.3.2 Pressure measurements

The electrical input for the pressure sensors is given by the differential voltage amplifier (Endevco). The output of the sensors are connected to digital storage oscilloscope where the results can be displayed and stored and is shown in Fig. 4.16. The amplifier acts as a junction between the pressure sensors and digital storage oscilloscope.

Table 4.2 Specifications of differential voltage amplifier

Input	0 to ±10 Vdc
Linear output	10 V
Accuracy	±0.5% of full scale.
Power requirement	90-264 VAC, 50 to 60 Hz
Channel	3

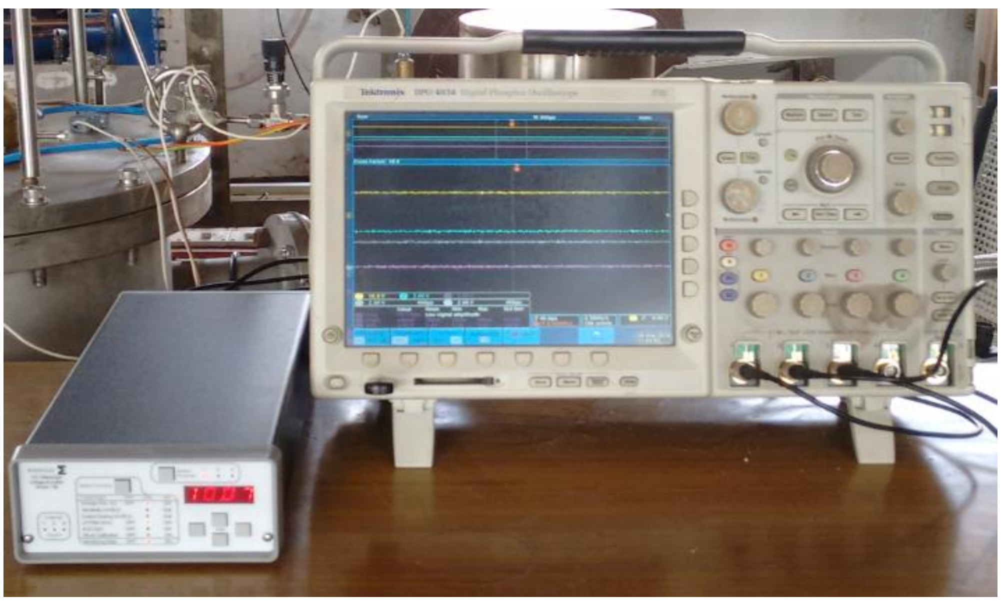


Fig.4.16 Data acquisition system for pressure measurements

Chapter 5

Experimental Results and Discussions

Experimentation has been carried out on the pulse tube refrigerator test rig by varying the different inputs such as charging pressure, double inlet valve opening and orifice valve opening. The set-up has been operated as BPTR, OPTR and DIPTR to study its performances and effects at cold end temperature.

5.1 Cooling behaviour

OPTR and DIPTR have been shown better cool down characteristics compared to BPTR. Since the compressor is of small capacity of 1.5 kW, the steady state obtained is slow. By trail run it has been found that pulse tube refrigerator comes in steady operation after 3600 seconds (approx.). Figures of (5.1 to 5.6) have shown the cool down behaviour when operated at basic, orifice and double inlet type respectively at their particular operating condition. It has been found that higher pressure gives the minimum cold end temperature i.e. better performance. It has been seen that higher orifice opening gives lower cooling but comes steady state quickly compared to smaller opening. The cold end temperature decreases with the increase of pressure due to higher compression and expansion of the gas inside the tube.

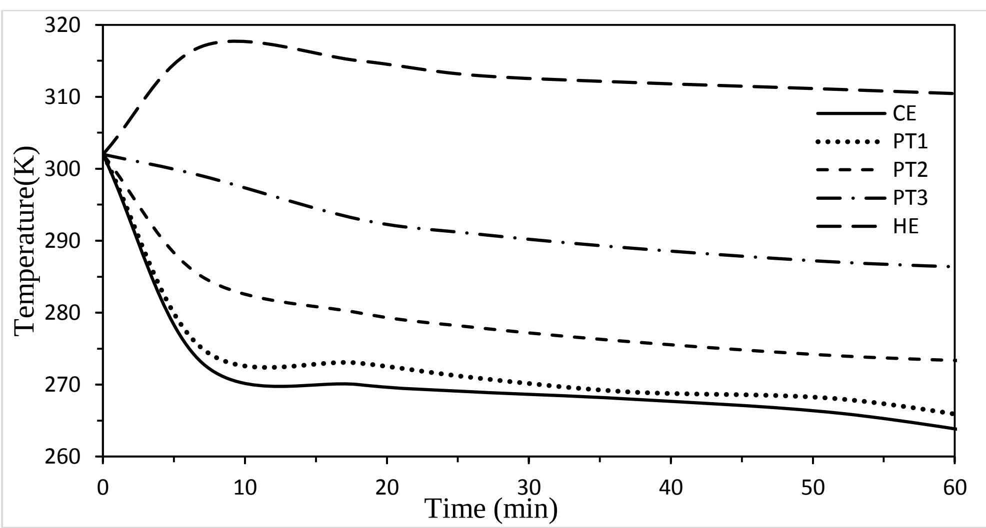


Fig.5.1 Cool down behaviour at optimum opening of orifice valve at HP =10 bar and LP=8 bar at no load as OPTR.

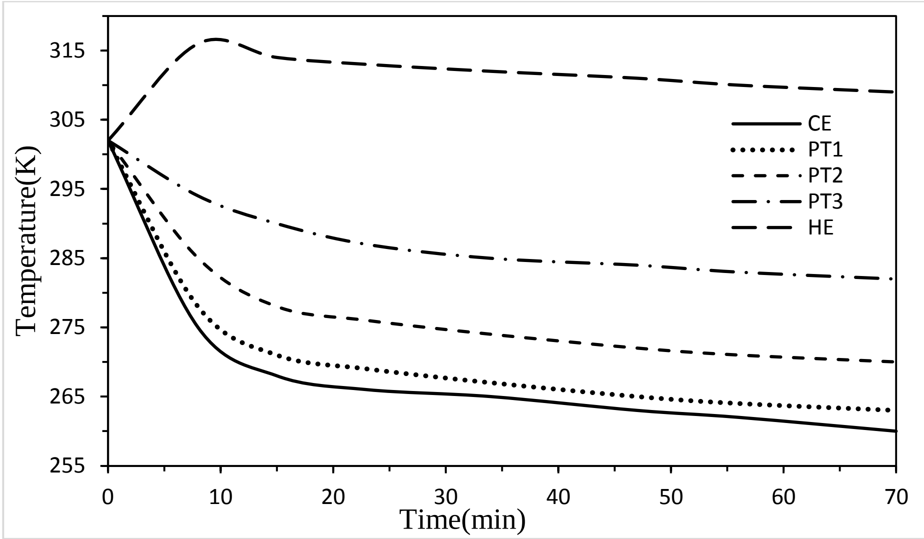


Fig.5.2 Cool down behaviour at optimum opening of double inlet valve at HP =10 bar and LP=8 bar at no load as DIPTR.

A temperature of 260 K has been observed at high pressure of 10 bar and low pressure of 8 bar when operated at optimum opening of double inlet valve at 0.197 inches and of orifice at 0.152 inches. At an optimum orifice opening of 0.158 inches a temperature of 262 K at cold end has been achieved at same pressure when operated as orifice type.

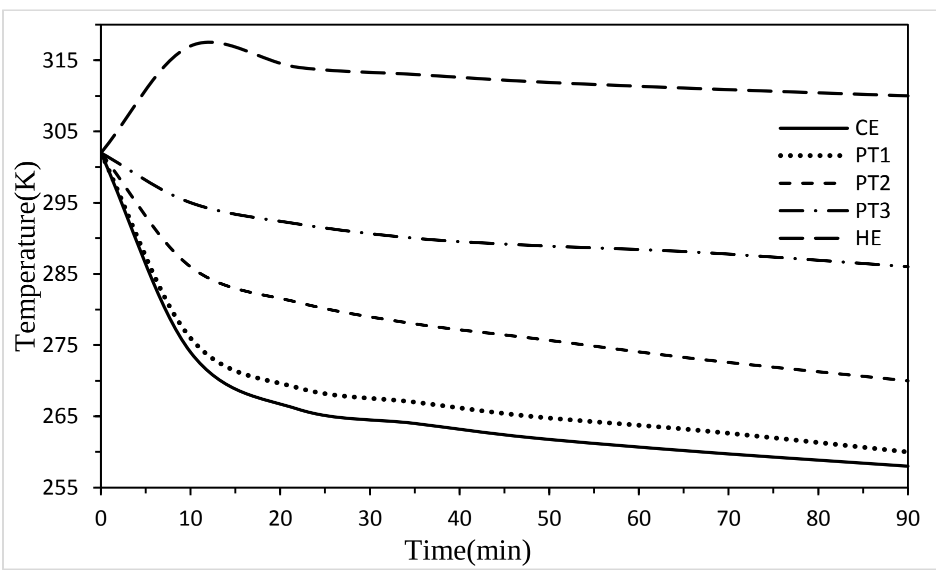


Fig.5.3 Cool down behaviour at optimum opening of double inlet valve at HP =14 bar and LP=10 bar at no load as DIPTR.

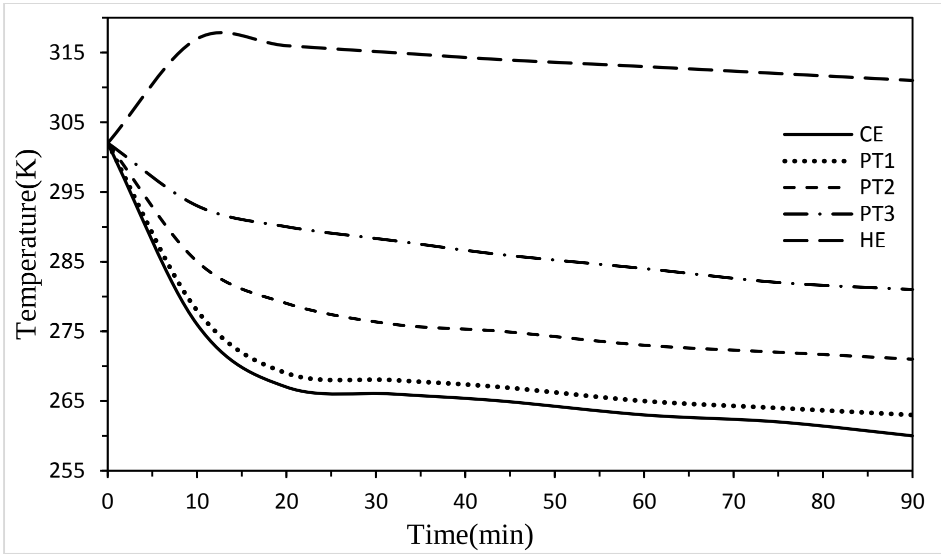


Fig.5.4 Cool down behaviour at optimum opening of orifice valve at HP =14 bar and LP=10 bar at no load as OPTR.

The experimental set-up when operated as a double inlet type achieved a lowest temperature of 258 K at an optimum opening of double inlet valve at 0.197 inches and of orifice at 0.152 inches at a high pressure of 14 bar and low pressure of 10 bar. At same pressure when operated as an orifice type a lowest temperature of 259 K has been obtained at an optimum opening of orifice at 0.158 inches.

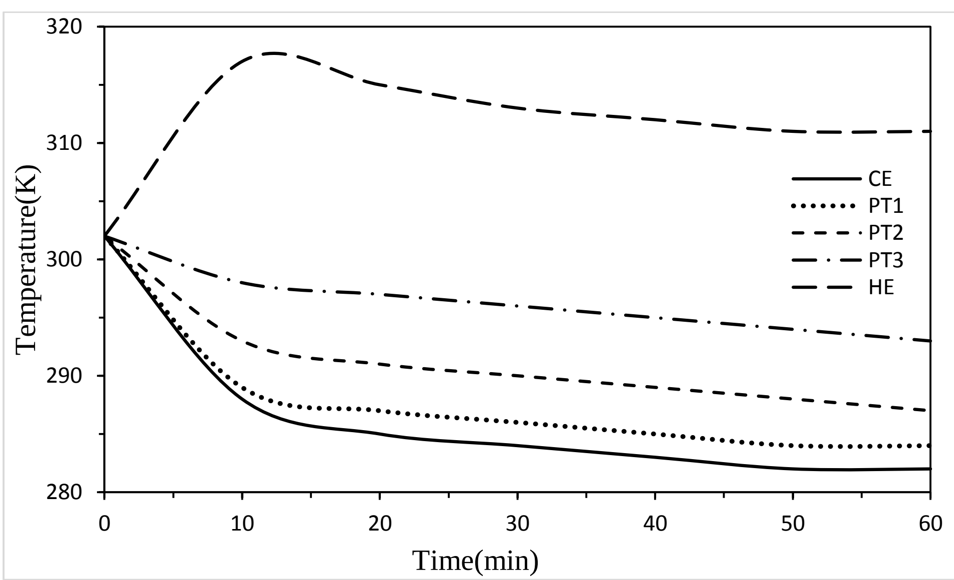


Fig.5.5 Cool down behaviour of BPTR at HP =10 bar and LP=8 bar at no load.

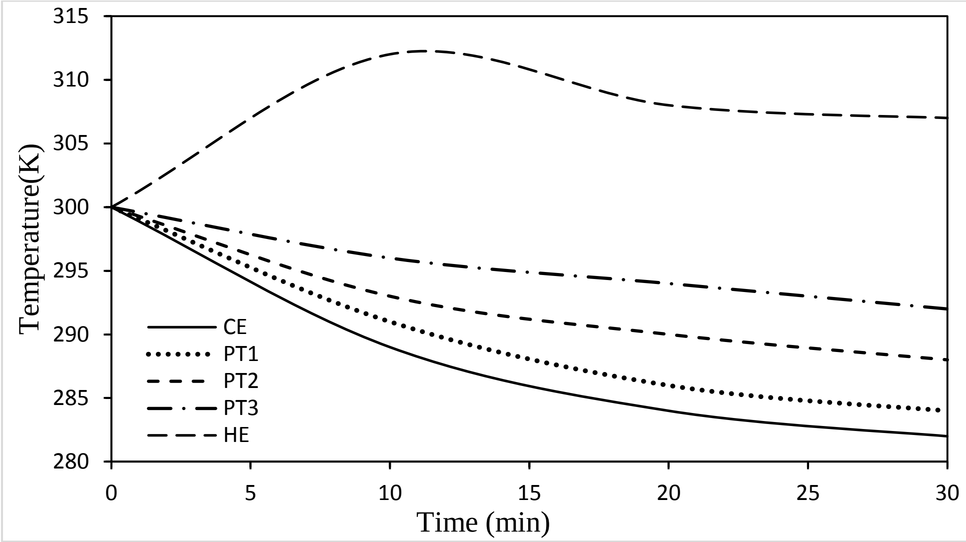


Fig.5.6 Cool down behaviour of BPTR at HP =10 bar and LP=5 bar at no load

Form the figures (5.5 and 5.6), it has been observed that BPTR is inefficient and less effective in terms of cool down behaviour irrespective of operating pressure and valve openings.

5.2 Valve optimization

The pulse tube refrigerator test rig has been optimized with respect to minimum attainable temperature at no load by varying both double inlet and orifice valve opening. Figures of (5.7 to 5.10) have been shown the minimum attainable temperature of OPTR and DIPTR at their different valve openings. It has been observed that at lesser and higher opening it is not affected much due to improper phase relation between pressure and mass flow rate. Minimum attainable temperature at cold end has been achieved at optimum opening of double inlet valve at 0.197 inches and orifice at 0.152 inches as double inlet type. As orifice type, minimum attainable temperature at cold end has been obtained at an optimum opening of orifice valve at 0.158 inches. This is the optimum opening in the developed set-up with a frequency of 2Hz.

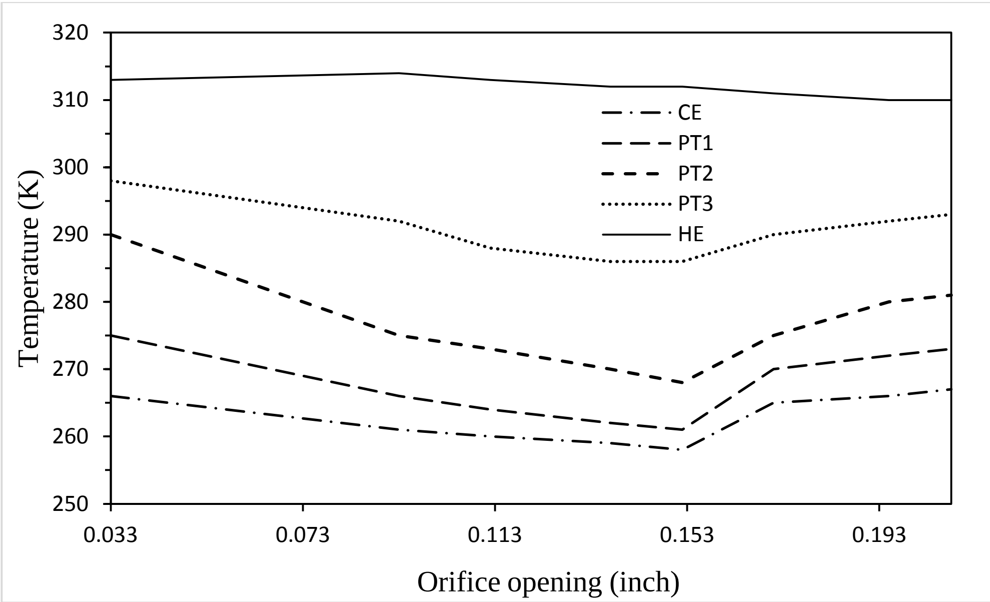


Fig.5.7 Effect of orifice valve opening on minimum attainable temperature of double inlet valve in optimum condition at no load at HP=14 bar and LP=10 bar as DIPTR

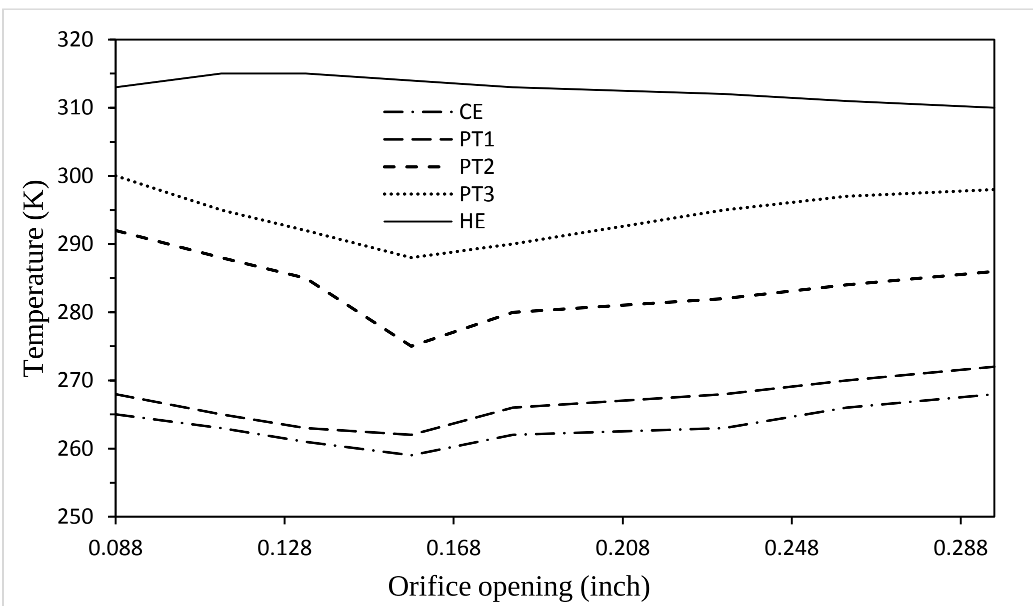


Fig.5.8 Effect of orifice valve opening on minimum attainable temperature at no load at HP=14 bar and LP=10 bar as OPTR

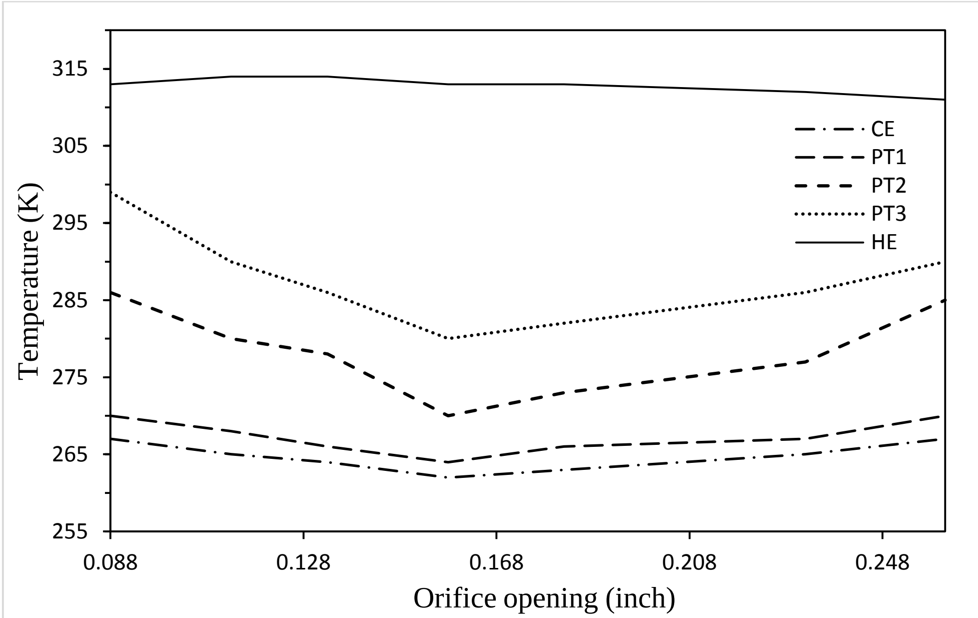


Fig.5.9 Effect of orifice valve opening on minimum attainable temperature at no load at HP=10 bar and LP=8 bar as OPTR

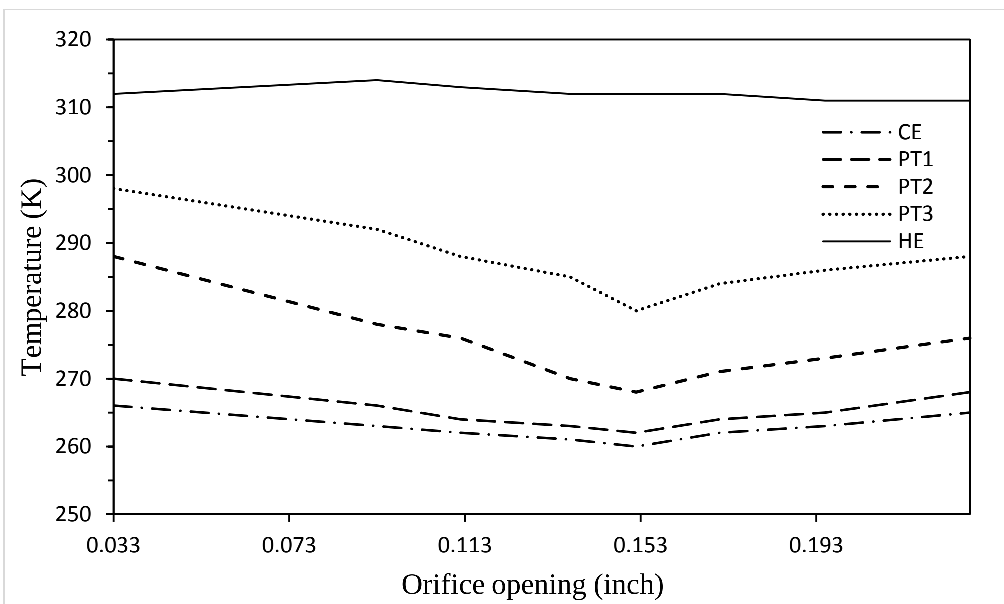


Fig.5.10 Effect of orifice valve opening on minimum attainable temperature of double inlet valve in optimum condition at no load at HP=10 bar and LP=8 bar as DIPTR

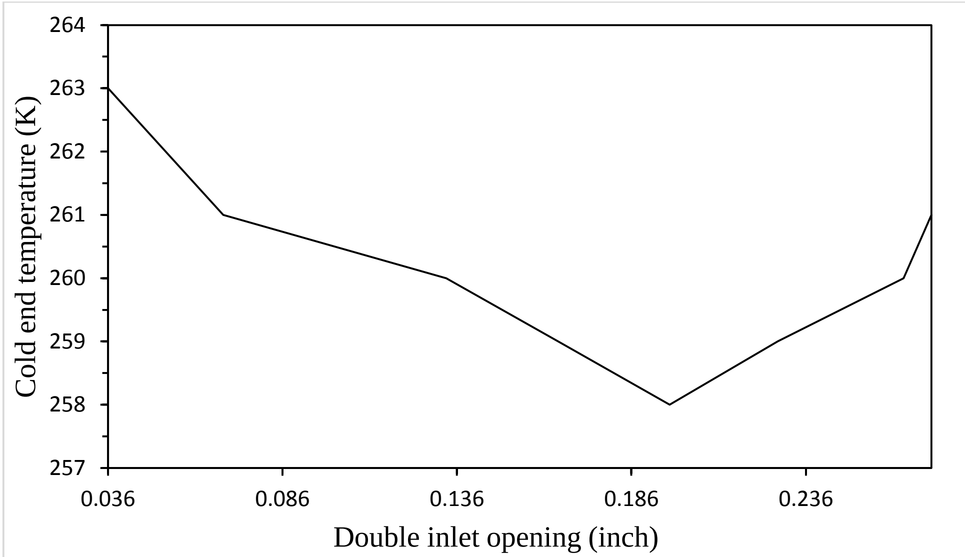


Fig. 5.11 Effect of change in double inlet opening on minimum attainable temperature at no load at HP=14 bar and LP=10 bar as DIPTR

From the fig. 5.11, it can be seen that at higher opening of double inlet valve minimum temperature is achieved very rapidly rather than smaller opening.

5.3 Pressure variation

The pressure variations at regenerator inlet, pulse tube and reservoir at different pressures have been shown in figures. It has been observed that reservoir pressure is almost constant irrespective of valve opening and the type it is being operated. The best performance of the pulse tube refrigerator is observed when the pressure variation established for the regenerator and pulse tube is in between sinusoidal and trapezoidal form.

From the figures (5.12 to 5.19), it can be seen that even with the reciprocating compressor of low capacity and depending on pressure ratio, there exists a pressure wave more or less sinusoidal for both regenerator and pulse tube at a frequency of 2 Hz which resembles that the set-up has functioned well. If the test set-up will be commissioned by the high capacity compressor, at high pressure ratio and by varying frequency, then definitely a pressure wave form between sinusoidal and trapezoidal form will be achieved. It also shows that though there have some quantitatively difference between them, but a phase relationship among the pressures are always there.

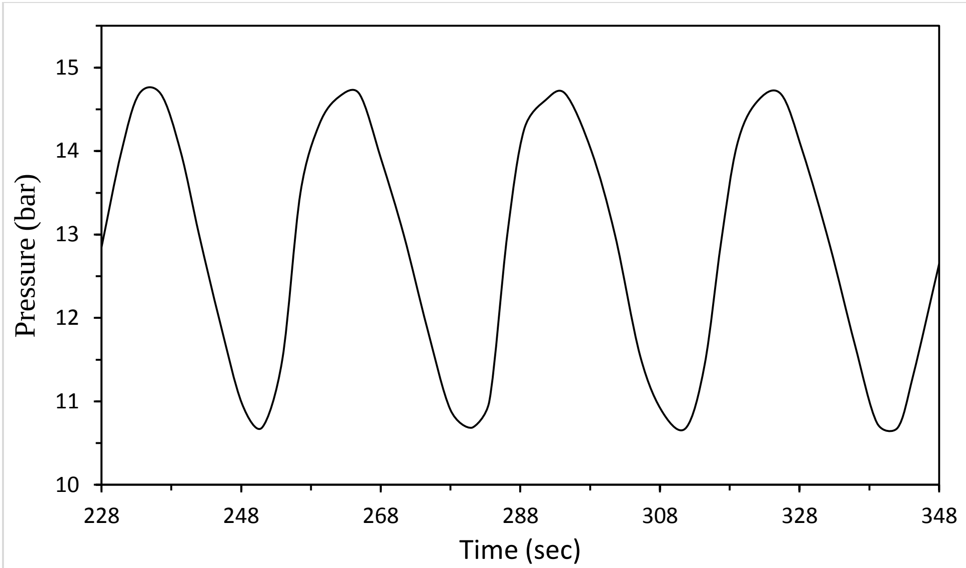


Fig 5.12 Pressure variation at regenerator inlet at an optimum opening of double inlet valve at HP=14 bar and LP=10 bar

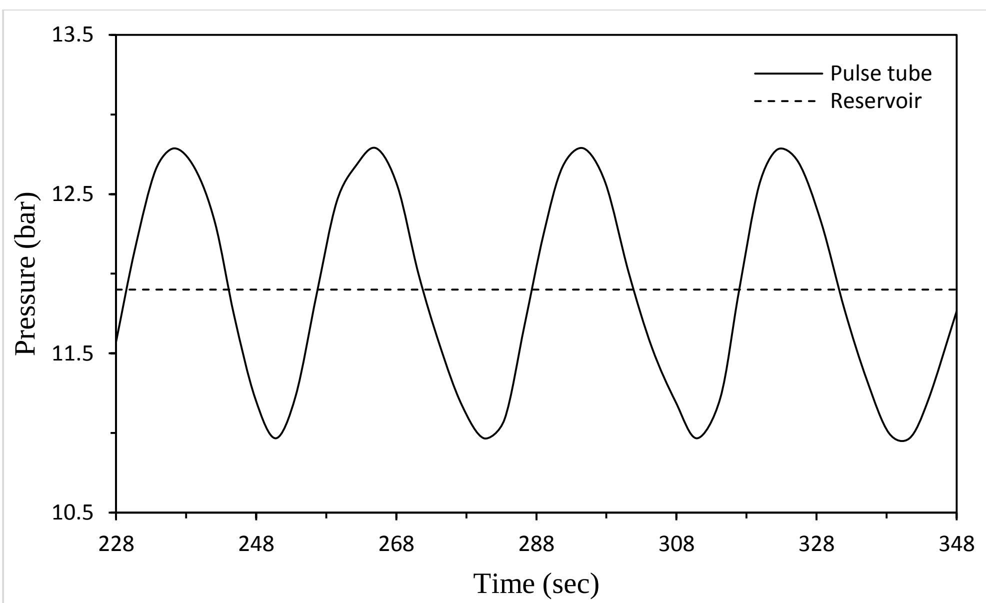


Fig 5.13 Pressure variation at pulse tube and reservoir at an optimum opening of double inlet valve at HP=14 bar and LP=10 bar

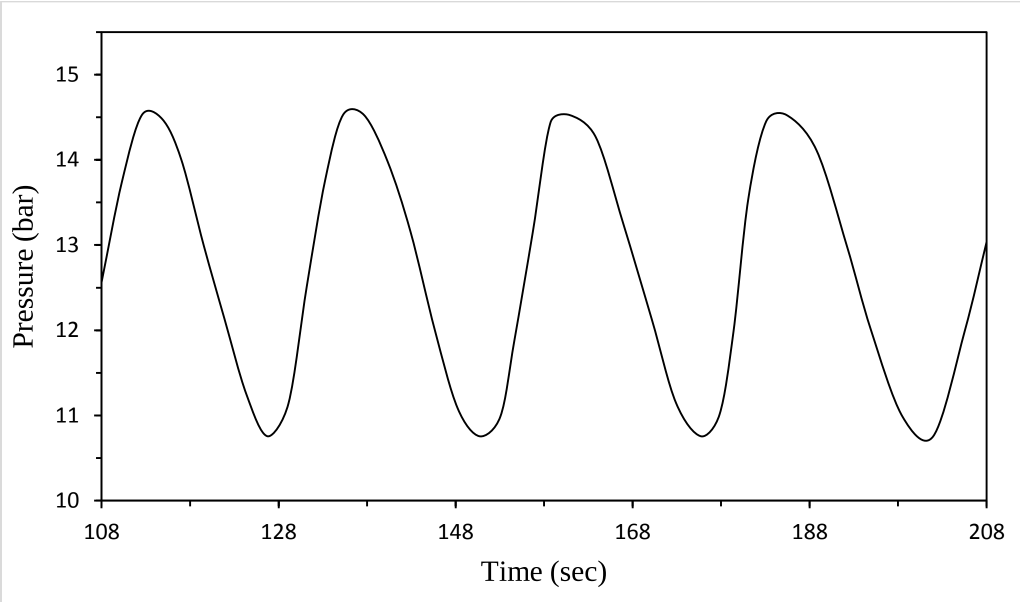


Fig.5.14 Pressure variation at regenerator inlet at an optimum opening of orifice valve at HP=14 bar and LP=10 bar

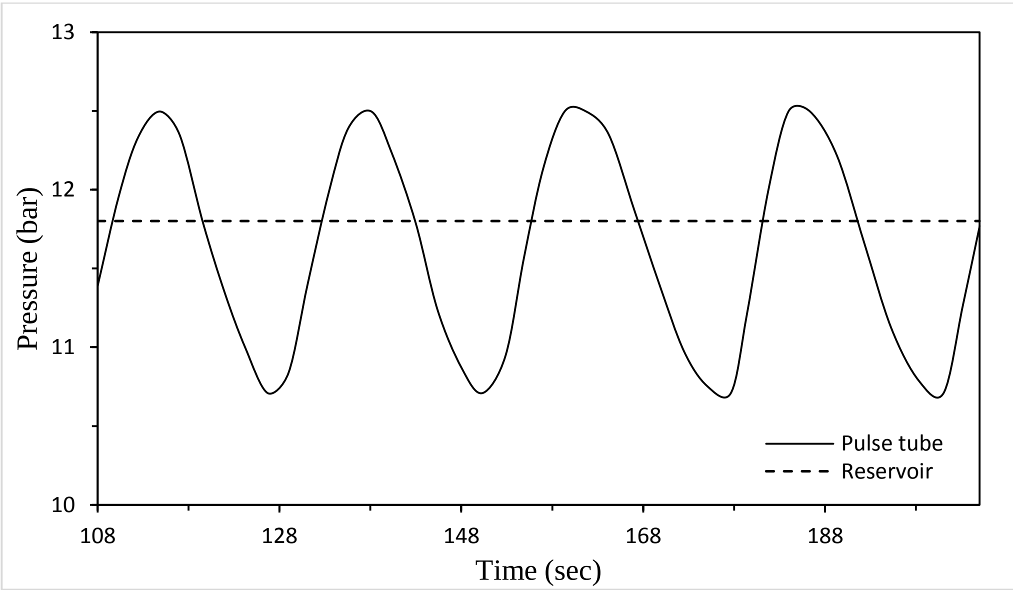


Fig. 5.15 Pressure variation at pulse tube and reservoir at an optimum opening of orifice valve at HP=14 bar and LP=10 bar

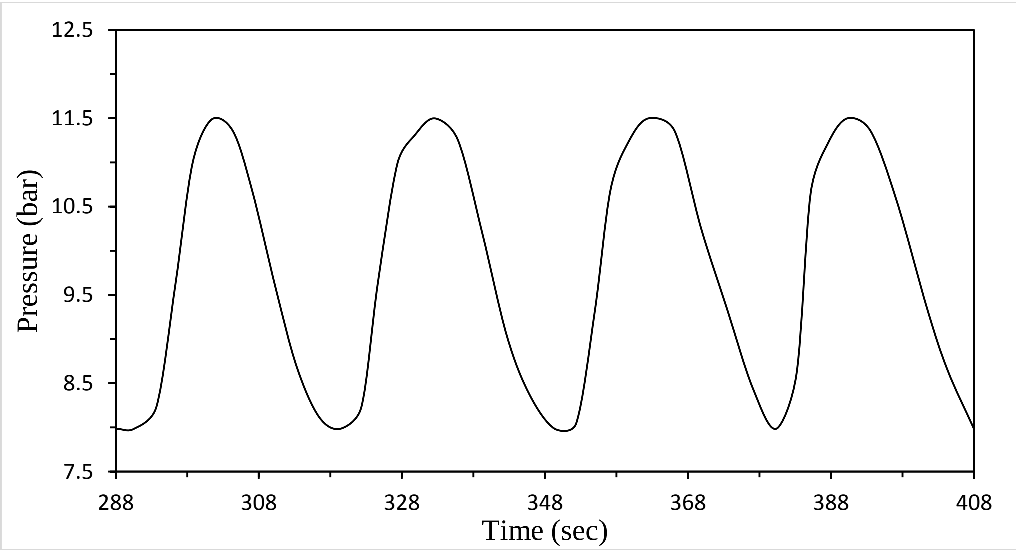


Fig.5.16 Pressure variation at regenerator inlet at an optimum opening of double inlet valve at HP=10 bar and LP=8 bar

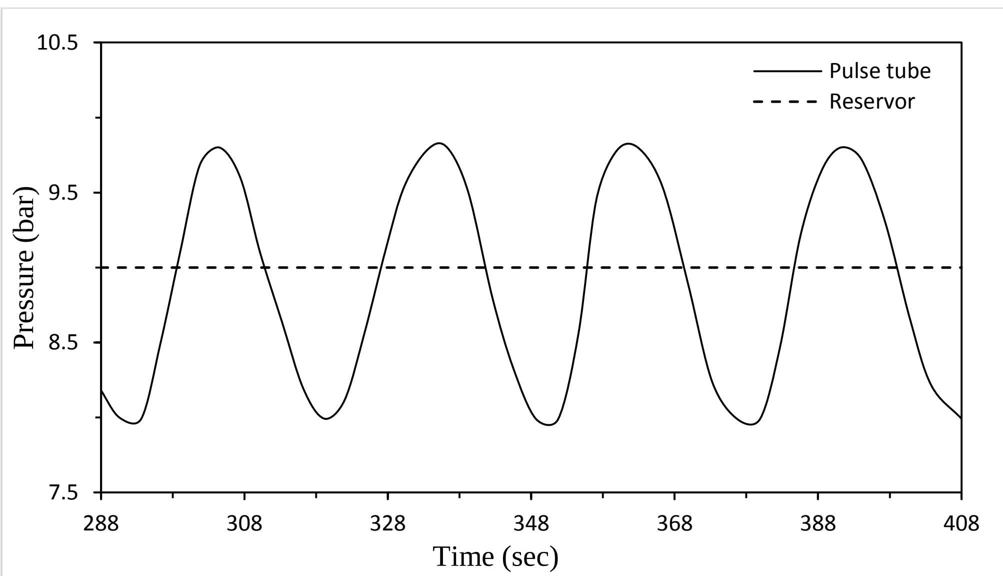


Fig.5.17 Pressure variation at pulse tube and reservoir at an optimum opening of double inlet valve at HP=10 bar and LP=8 bar

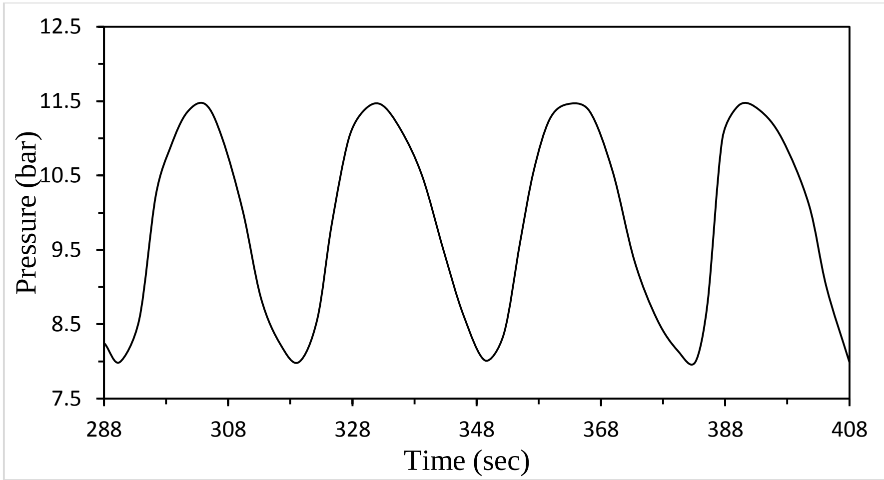


Fig.5.18 Pressure variation at regenerator inlet at an optimum opening of orifice valve at HP=10 bar and LP=8 bar

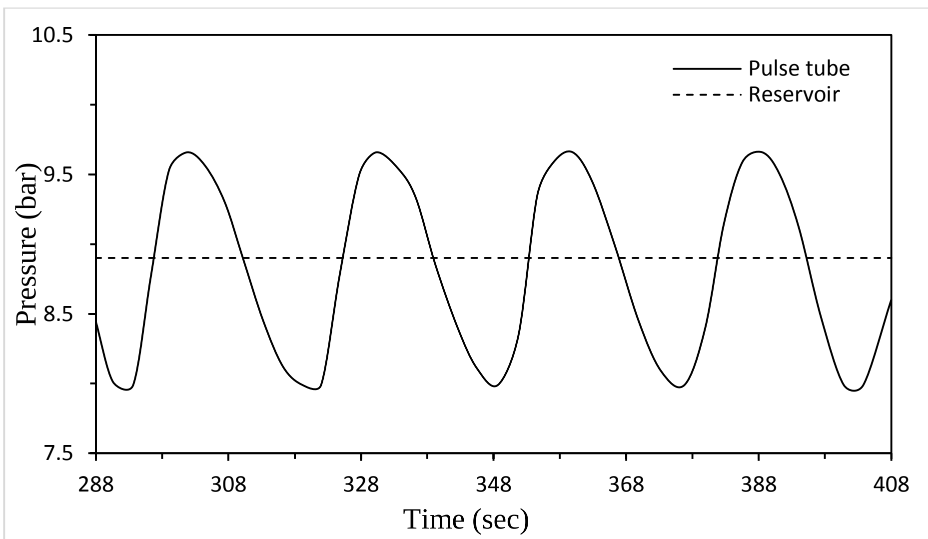


Fig.5.19 Pressure variation at pulse tube and reservoir at an optimum opening of orifice valve at HP=10 bar and LP=8 bar

The pressure is not constant and is fluctuating in the pulse tube. As the power consumption of the compressor is very low, it is difficult to determine the optimum opening of orifice valve and is possible at cooling loads and very low temperatures. For the present case, the optimum opening of orifice valve is at 0152 inches for HP of 14 bar, LP of 10 bar and also for HP of 10 bar, LP of 8 bar. It is highly impossible to obtain any refrigerating capacity and cop at such a low capacity compressor of 1.5 kW and that too high capacity is uncertainty and which will be achieved further.

Chapter 6

Conclusion

Experimental studies have been made on pulse tube refrigeration system. Previous chapters contain the details of the investigation. The salient results and features have been highlighted in the present chapter.

6.1 Summary

- A pulse tube refrigerator along with the test-rig has been designed and fabricated indigenously. Elaborate studies have been carried out to optimize the developed system.
- Cooling behavior of the pulse tube refrigerators has been studied at different average pressures and at different openings of the flow resistance valves. Some distinct features of OPTR and DIPTR compared to BPTR have been discussed.
- Optimum opening of the flow resistance valves (orifice and double inlet valve) has been determined according to minimum attainable cold end temperature at no load condition.
- Instead of single valve double inlet type, a double valve double inlet configuration has been developed.
- The lowest temperature at the cold end has been obtained in this case. It has found that at 0.197 inches opening of double inlet valve and orifice at 0.152 inches are optimum opening for double inlet type.
- Observed that 0.158 inches is the optimum opening for orifice type. Optimization of the valves opening has been carried out at different average pressures of the system.
- Pressure variations of the pulse tube system have been determined at different orifice and double inlet opening. Pressure variations at various positions such as regenerator inlet, pulse tube and at reservoir in the experimental set-up have been shown.
- The pulse tube refrigerator plant has been successfully commissioned.

6.2 Scope of future work

This chapter does not mark the end of our venture; rather we can say that it is the beginning of a major endeavor that has been initiated. Naturally, there are lots of activities left behind. In spite of these studies, there are several possible issues considered for future research work. Some recommendations of these includes

- In the present, studies could not be made at different frequencies of the rotary valve. However, this is a very important parameter. Extensive studies are needed to identify the optimum frequency.
- Well planned strategy of the experimental studies showed can be taken to optimize the geometry of pulse tube.
- Scope of improvement also exists in the design of regenerator.
- Present studies are mainly focused on the effect of cold end temperature and can be extended further to study on the effect of cooling capacity.
- The developed facility set-up can be extended to study the performance of the inertance tube and minor orifice type pulse tube.
- Scope of commissioning the test set-up by a high kW compressor for high refrigerating capacity and for the study of better performances.

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APPENDIX- A

Drawings of PTR components

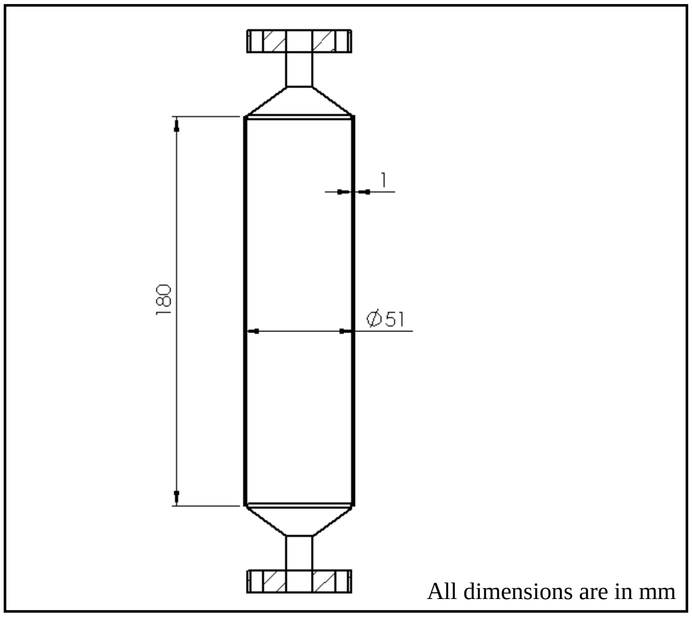


Fig.A.1 Schematic view of Regenerator

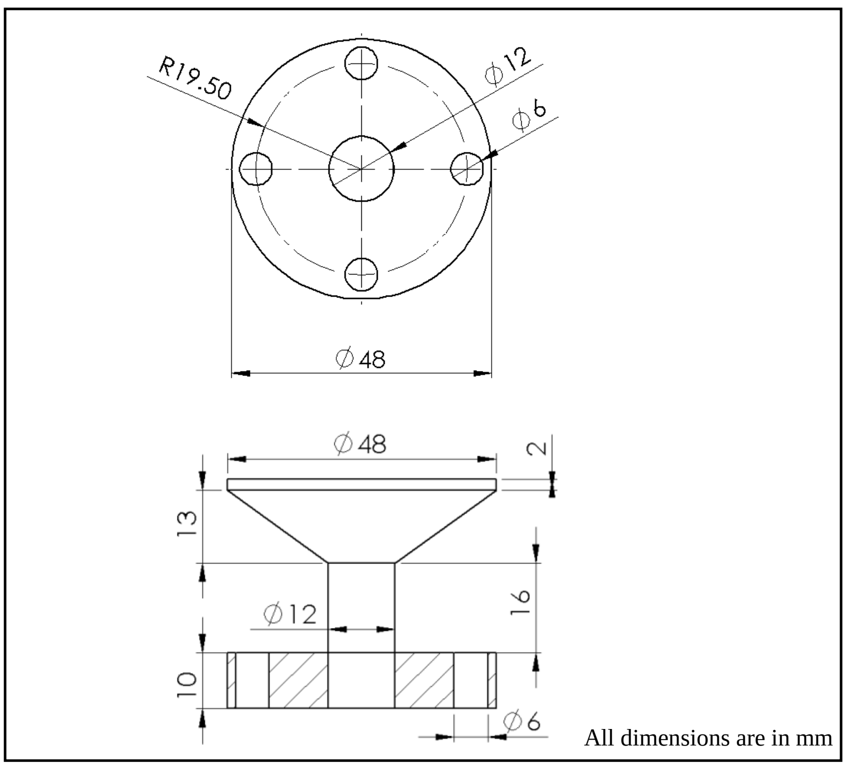


Fig.A.2 Top and bottom flanges of Regenerator

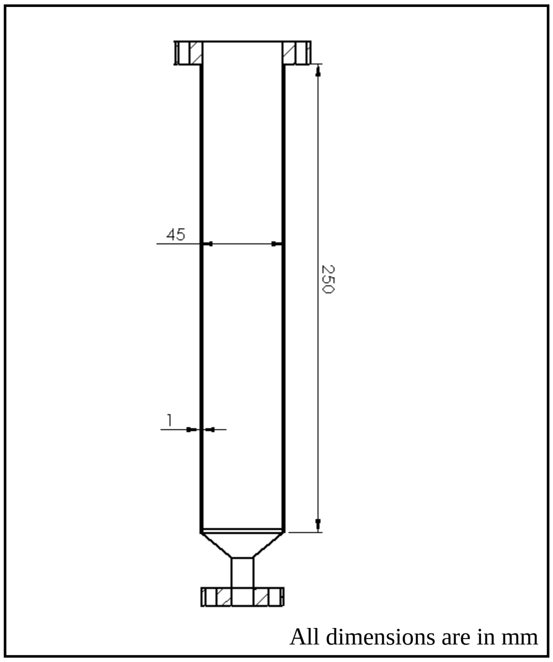


Fig.A.3 Schematic view of Pulse tube

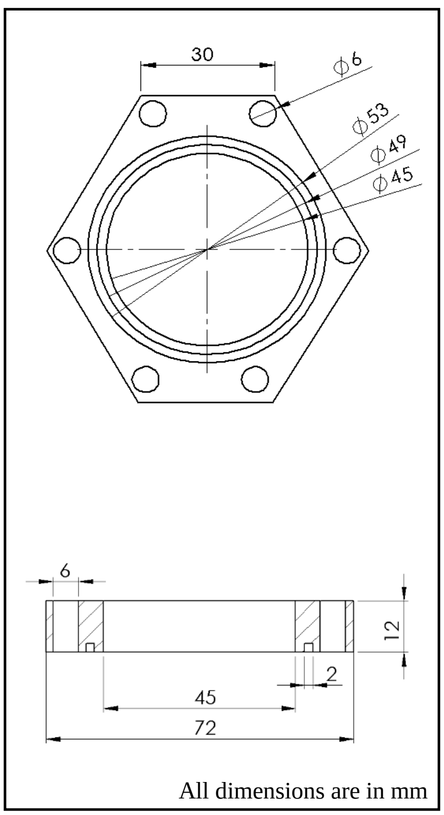


Fig.A.4 (a) Top flange of pulse tube

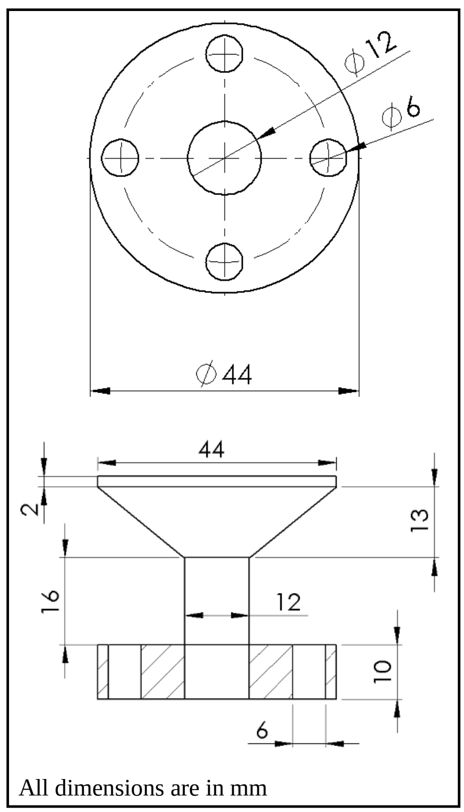


Fig.A.4 (b) Bottom flange pulse tube

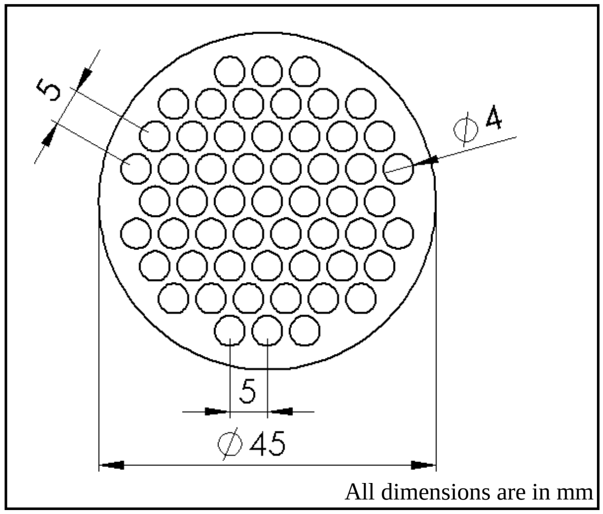


Fig.A.5 Circular plate of hot end heat exchanger

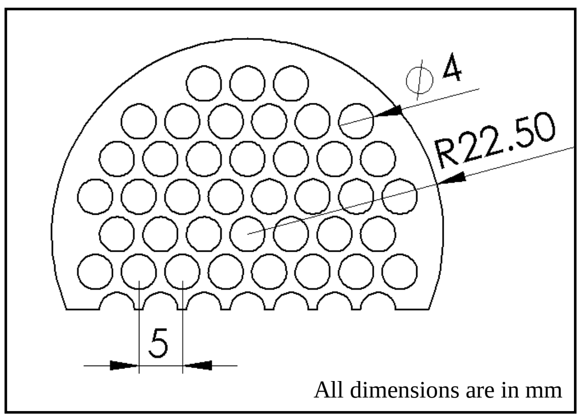


Fig.A.6 Baffle of hot end heat exchanger

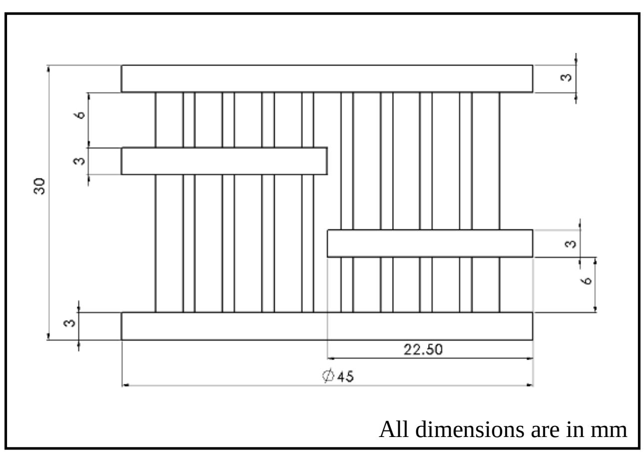


Fig.A.7 Interior part of hot end heat exchanger

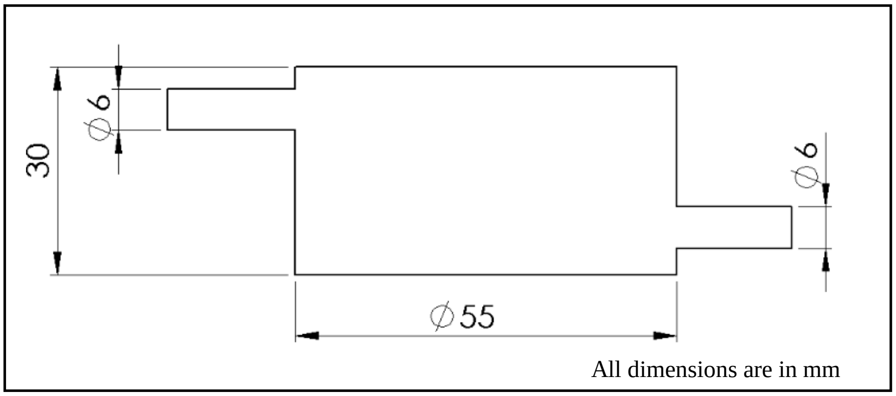


Fig.A.8 Shell of hot end heat exchanger heat exchanger

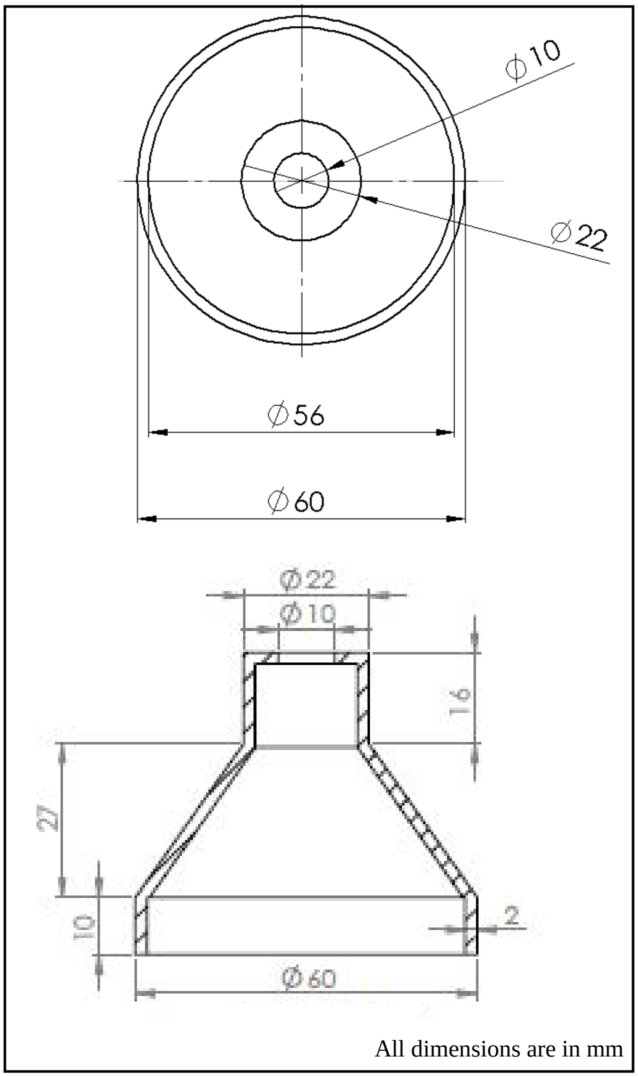


Fig.A.9 Convergent section of hot end heat exchanger

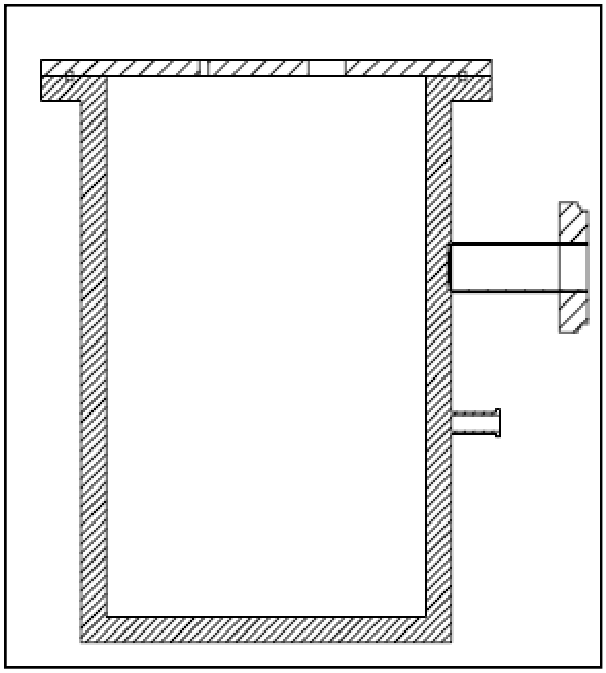


Fig.A.10 Schematic view of Vacuum chamber

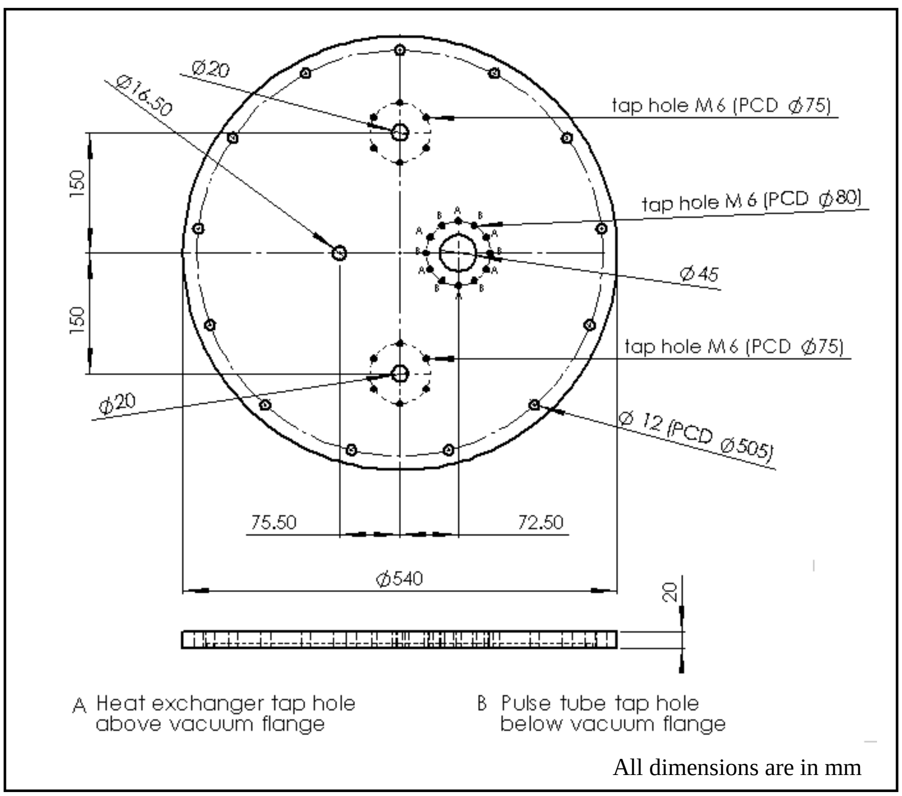
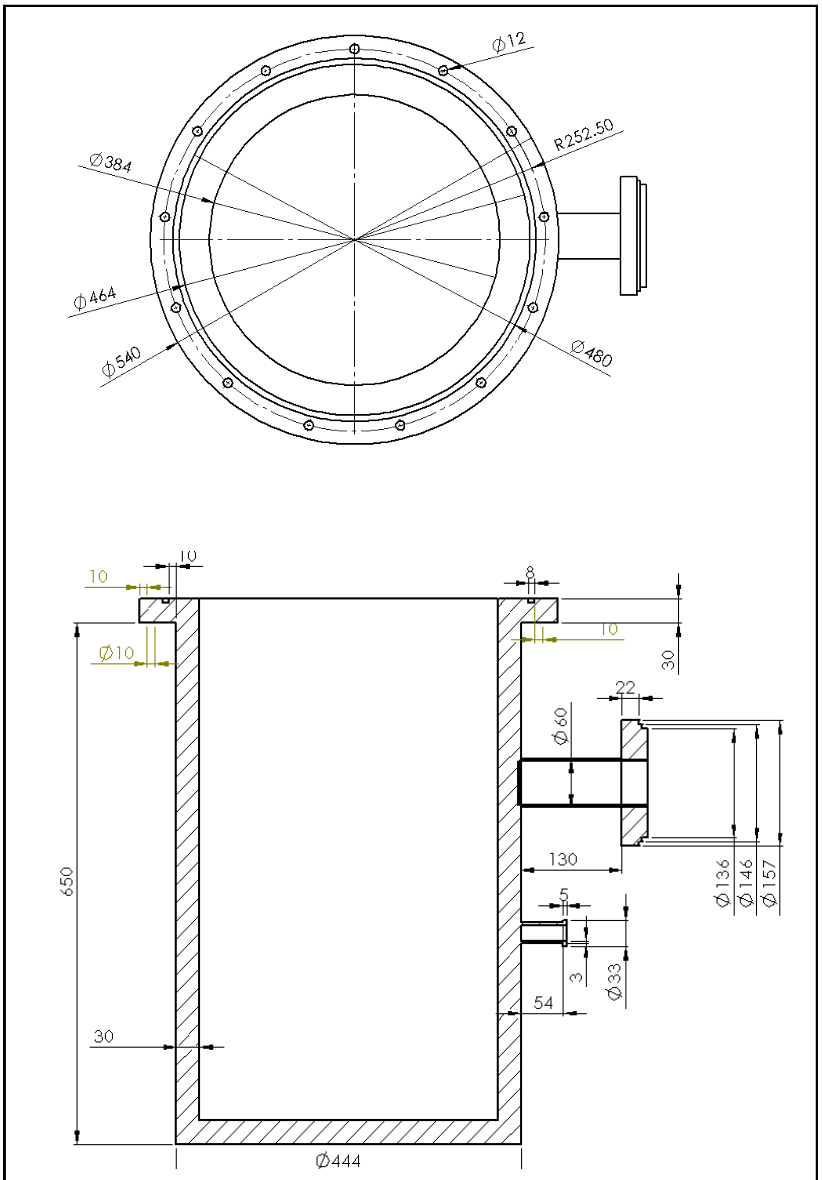


Fig.A.11 Top flange of Vacuum chamber



All dimensions are in mm

Fig.A.12 Bottom flange of Vacuum chamber

Dissemination

International conferences

1. K.N.S. Manoj**,** S. Panda, D. Panda, R.K. Sahoo, S.K. Sarangi, "Design and Fabrication of a High Cooling Capacity G-M Type Pulse Tube Refrigerator", *Proc. of International Cryogenic Engineering Conference-International Cryogenic Materials Conference* (ICEC26-ICMC 2016), March 2016, New Delhi, India.

2. D. Panda, K.N.S. Manoj, R.K. Sahoo, S.K. Sarangi, "A Mathematical Model and Design Software for Pulse Tube Refrigerator", *Proc. of International Cryogenic Engineering Conference-International Cryogenic Materials Conference* (ICEC26-ICMC 2016), March 2016, New Delhi, India.