NUMERICAL ANALYSIS OF DOUBLE INLET PULSE TUBE REFRIGERATOR

A THESIS SUBMITTED IN PARTIAL
FULFILLMENT OF THE REQUIREMENT FOR
THE DEGREE OF

MASTER OF TECHNOLOGY
In
Mechanical Engineering with Specialization
In "Cryogenic and Vacuum Technology"

By

Samarendra Panda
Roll No.: 213ME5452

Under the Guidance of
Prof. Ranjit Kumar Sahoo

Department of Mechanical Engineering
National Institute of Technology, Rourkela (Odisha)
June, 2015

NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA

CERTIFICATE

This is to certify that the thesis entitled**, "NUMERICAL ANALYSIS OF DOUBLE INLET PULSE TUBE REFRIGERATOR"** submitted by **Mr. Samarendra Panda** in partial fulfilment of the requirements for the award of Master of Technology Degree in Mechanical Engineering with specialization in Cryogenic and Vacuum Technology at the National Institute of Technology, Rourkela is an authentic work carried out by him under my super vision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other university/institute for the award of any degree or diploma.

Date:

Prof. Ranjit Kumar Sahoo
Department of Mechanical Engineering
National Institute of Technology Rourkela
Rourkela– 769008

SELF DECLARATION

I, **Mr. Samarendra Panda, Roll No. - 213ME5452**, student of M. Tech (2013-2015), Cryogenics and Vacuum Technology at Department of Mechanical Engineering, National Institute of Technology, Rourkela do here by declare that I have not adopted any kind of unfair means and carried out the research work reported in this thesis ethically to the best of my knowledge. If adoption of any kind of unfair means is found in this thesis work at a later stage, then appropriate action can be taken against me including withdrawal of this thesis work.

NIT, ROURKELA
DATE:

Samarendra Panda

ACKNOWLEDGEMENT

I would like to thank and express my gratitude towards my supervisor **Prof. Ranjit Kumar Sahoo** for his extensive support throughout this project work. I am greatly indebted to him for giving me the opportunity to work with him and for his belief in me during the hard time in the course of this work. His valuable suggestions and constant encouragement helped me to complete the project work successfully. Working under him has indeed been a great experience and inspiration for me.

I would also like to thank **Mechanical Department** for providing the **CFD Lab** where I completed the maximum part of my project work.

Date: **Samarendra Panda**

Place:

ABSTRACT

Pulse Tube cryocooler or Pulse Tube Refrigerator (PTR) is a compact refrigerator which is capable of attaining a cryogenic temperature below 123 K (-150 K). As there are no moving components at low temperature side of PTR, it attracts the attention of various researchers. The advantages are mainly its simplicity, low vibration at cold end side, low cost, long life span and ease of manufacturing. With the tremendous development in the performance of PTR and due to its compactness, PTR has a wide range of applications like infrared sensors, aerospace, night vision equipments, communication, micro-biological sciences, superconductivity, in medical science and SQUID. In this present work, modeling and numerical simulation is carried out for Double Inlet Pulse Tube Refrigerator (DIPTR).

There are various types of PTRs on the basis of way of their development stages. DIPTR is the modification of Orifice Pulse Tube Refrigerator (OPTR), by simply adding one more valve in between the hot end side of pulse tube and inlet to the regenerator. This arrangement is very much useful to prevent the flow of large amount of gas into pulse tube through the regenerator as in case of OPTR. Here, a two dimensional (2D) model of DIPTR is created using ANSYS Design Modeler and Computational Fluid Dynamics (CFD) solution approach is chosen for numerical simulation purpose. The detailed study on cool down behavior at cold end of DIPTR, pressure variation at inlet and also heat transfer has been performed using CFD package FLUENT software (ANSYS FLUENT 15.0.0). Four numbers of cases have been chosen in which the pressure User Defined Functions (UDFs) are different and all other parameters remain unchanged. Pressure UDFs are applied at inlet as a boundary condition to define the oscillating motion of piston inside the piston-cylinder arrangement. The four pressure UDFs are of different wave forms such as Sinusoidal, Rectangular, Triangular and Trapezoidal. Pulse tube dimension is taken as 15 mm diameter and 250 mm length. The regenerator and other heat exchangers are specified as porous zones with a porosity of 0.6. The operating frequency for all cases is 2 Hz. After simulation using four pressure UDFs, it has been found that pressure UDF generating triangular pressure wave is more efficient than other pressure UDFs and a temperature of 111 K is obtained using triangular wave form of pressure.

Key words: DIPTR, Pressure UDF, CFD

CONTENTS

Description **Page No.**

ABSTRACT i

LIST OF FIGURES iv

LIST OF TABLES vi

NOMENCLATURE vii

CHAPTER I

Introduction

1.1.Classification of Cryo-coolers 1

1.2.Pulse Tube Refrigerator 3

1.3.Working Principle of PTRs 3

1.4.Categorization of PTRs 4

1.5.Main Components of Pulse Tube Refrigerator (PTR) 11

1.6.Basic Theories of PTRs 13

1.7.Applications of PTRs 15

1.8.Objective of Work 16

1.9.Outline of Thesis 16

Chapter II

Literature Review

2.1. Introduction 17

2.2. Basic Pulse Tube Refrigerator (BPTR) 17

2.3. Orifice Pulse Tube Refrigerator (OPTR) 19

2.4. Inertance Tube Pulse Tube Refrigerator (ITPTR) 21

2.5. Double Inlet Pulse Tube Refrigerator (DIPTR) 22

Chapter III

Modeling of Double Inlet Pulse Tube Refrigerator (DIPTR)

3.1. Introduction 26
3.2. Geometry of DIPTR 26
3.3. Technique for Solution 28
3.4. Governing Equations for Pulse Tube Refrigerator 29

Chapter IV

Simulation and Result Discussion

4.1. Defining the Model 32
4.2. Simulation for Different Boundary Conditions 34

Chapter V

Conclusion and Future Work

5.1. Conclusion 45
5.2. Future Work 45

REFERENCES 46

LIST OF FIGURES

Figure No.	Name of Figure	Page No.
Fig 1.1	Classification of Cryo-coolers	2
Fig 1.2	Regenerative type Cryocoolers	2
Fig 1.3	Recuperative type Cryocoolers	3
Fig 1.4	Schematic diagram of (a) Stirling type, (b) G-M type PTRs	4
Fig 1.5	Schematic diagram of BPTR showing various components	5
Fig 1.6	Schematic diagram of OPTR showing various components	6
Fig 1.7	Schematic diagram of DIPTR showing various components	7
Fig 1.8	Schematic diagram of ITPTR	7
Fig 1.9	Stirling type 2-stage double inlet pulse tube refrigerator (DIPTR)	8
Fig 1.10	Three stage arrangement of PTRs	9
Fig 1.11	Schematic diagram of V-M type PTR	9
Fig 1.12	Schematic diagram of (a) Line type PTR, (b) U-type PTR, (c) Co-axial PTR and (d) Annular PTR	11
Fig 1.13	Typical design of Rotary Valve	13
Fig 1.14	Surface heat pumping mechanism	14
Fig 3.1	Physical model of G-M type DIPTR	27
Fig 3.2	Two-Dimensional geometry of DIPTR with Meshing	28
Fig 4.1	Pressure wave generated at inlet of DIPTR for Case – 1	35
Fig 4.2	Temperature decrement behavior of DIPTR for Case – 1	35
Fig 4.3	Phase relationship between mass flow rate and Pressure at inlet of DIPTR	35
Fig 4.4	Temperature contour of DIPTR for Case-1	36
Fig 4.5	Density contour of DIPTR for Case-1	36
Fig 4.6	Pressure wave generated at inlet of DIPTR for Case – 2	37
Fig 4.7	Temperature decrement behavior of DIPTR for Case – 2	38
Fig 4.8	Temperature contour of DIPTR for Case-2	38

Fig 4.9 Density contour of DIPTR for Case-2 39
Fig 4.10 Pressure wave generated at inlet of DIPTR for Case – 3 40
Fig 4.11 Temperature decrement behavior of DIPTR for Case – 3 40
Fig 4.12 Temperature contour of DIPTR for Case – 3 41
Fig 4.13 Density contour of DIPTR for Case -3 41
Fig 4.14 Pressure wave generated at inlet of DIPTR for Case – 4 42
Fig 4.15 Temperature variation in CHX of DIPTR for Case – 4 43
Fig 4.16 Temperature contour of DIPTR for Case – 4 43
Fig 4.17 Density contour of DIPTR for Case – 4 44

LIST OF TABLES

Table No.	Name of Table	Page No.
Table 1.1	Difference between Stirling and G-M type PTRs	4
Table 3.1	Dimensions of various components of DIPTR	27
Table 4.1	Boundary conditions and Initial conditions for various components of DIPTR	33

NOMENCLATURE

Symbols

C_p	Specific heat of gas at constant pressure (J/kg-K)
$\vec{F}_E$	Momentum source term or external body forces
$\vec{g}$	Acceleration due to gravity (m/s^2)
h	Local enthalpy (W)
k	Thermal conductivity of gas (W/m-K)
k_t	Thermal conductivity due to turbulence (W/m-K)
p	Static pressure (N/m^2)
Q	Heat transfer (W)
$\dot{Q}$	Heat rate (W)
$\dot{Q}_0$	Heat Rejected to Atmosphere (W)
S	Source term
S_E	Energy source term
t	Time (sec)
T	Temperature of gas (K)
T_0	Atmospheric Temperature (K)
$\vec{v}$	Velocity of gas (m/sec)
W	Work transfer (W)
$\dot{W}$	Power (W)

Greek Symbols

ρ	Density (kg/m^3)
$\vec{\tau}$	Stress tensor (N/m^2)
ψ_j	Diffusion flux
μ	Fluid viscosity (kg/m-s)

∇	Gradient operator
ω	Angular frequency (rad/sec)

Subscripts

C	Cold end
Comp	Compressor
Exp	Expander
r	Radial co-ordinate
Refrig	Refrigeration
Reject	Rejection
x	Axial co-ordinate

Abbreviations

BPTR	Basic Pulse Tube Refrigerator
CHX	Cold end heat Exchanger
DC	Direct flow
DIPTR	Double Inlet Pulse Tube Refrigerator
DIV	Double Inlet Valve
G-M	Gifford McMohan
HHX	Hot end Heat Exchanger
ITPTR	Inertance Tube Pulse Tube Refrigerator
OPTR	Orifice Pulse Tube Refrigerator
PTR	Pulse Tube Refrigerator
SQUID	Super Conducting Quantum Interference Device
JT	Joule-Thomson

CHAPTER I

INTRODUCTION

Generally cryogenic is the branch of science which deals with study of the physical phenomena occurs at low temperature. Cryogenic is derived from two Greek words: “Kryos” (means low temperature or freezing point temperature) and “Genes” (means to generate or produce). In engineering science, cryogenic can be explained as an application which includes the temperature range from absolute temperature 0K to 120K (-150^{o}C). At a glance, cryogenic includes liquefaction, refrigeration, transport and storage of cryogenic fluids and also the study of various physical properties of material at low temperature.

Cryogenic refrigerators or Cryo-coolers are devices those produce low temperature below 120K and these cryo-coolers are operated with a working cycle. Formerly there are two types of cryo-coolers: Joule Thomson cryocooler and Brayton cryocooler. In 1983, Walker classified cryocoolers as Recuperative type and regenerative type. In the later stage, the cryocoolers include Stirling type cryocooler and Gifford McMohan (G-M) type cryocooler. Now-a-days, the advantages of cryocoolers having less moving parts bring the attention of researchers and scientists. The advantages are mainly reliability, low cost, low mechanical vibration, less maintenance, long life and compactness. In the last few decades, there have been magnificent developments in cryocoolers, especially in the field of Pulse Tube Refrigerators (PTRs). So many developments and researches are going on to improve the performance of PTRs.

1.1. Classification of Cryo-coolers

Considering the gas flow pattern Matsubara [1] divided the cryocooler as: Circulating flow cryocooler and Oscillating flow cryocooler. The Circulating flow cryocooler is comprised of a turbo expander or a low temperature valved expander (Reciprocating in nature) with a counter flow heat exchanger, whereas the Oscillating flow cryocooler comprised of a valve less expander (Oscillating in nature) and a regenerator.

Considering the working cycle, Radebaugh [2] classified the cryocooler as: Open cycle and closed cycle cryocooler. In open cycle cryocooler, makeup gas is supplied in a quantity that balances the liquefaction rate of the system, while in case of closed cycle cryocooler; the working gas undergoes cyclic processes without any inflow and outflow of the medium across the system.

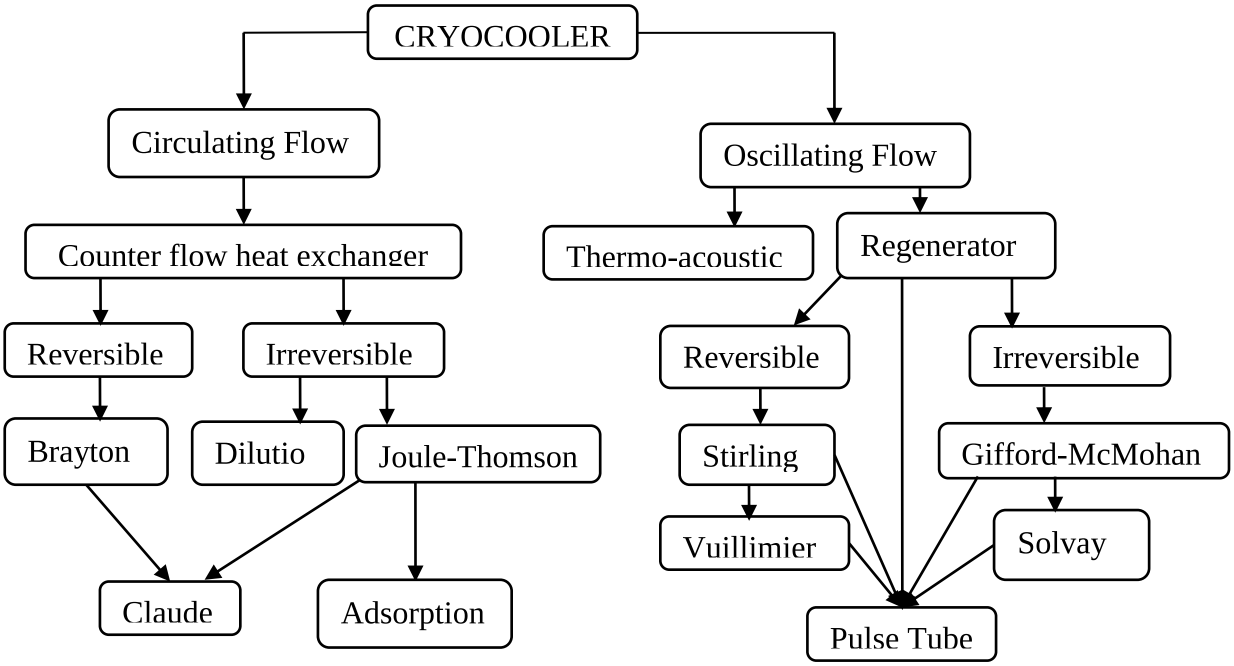
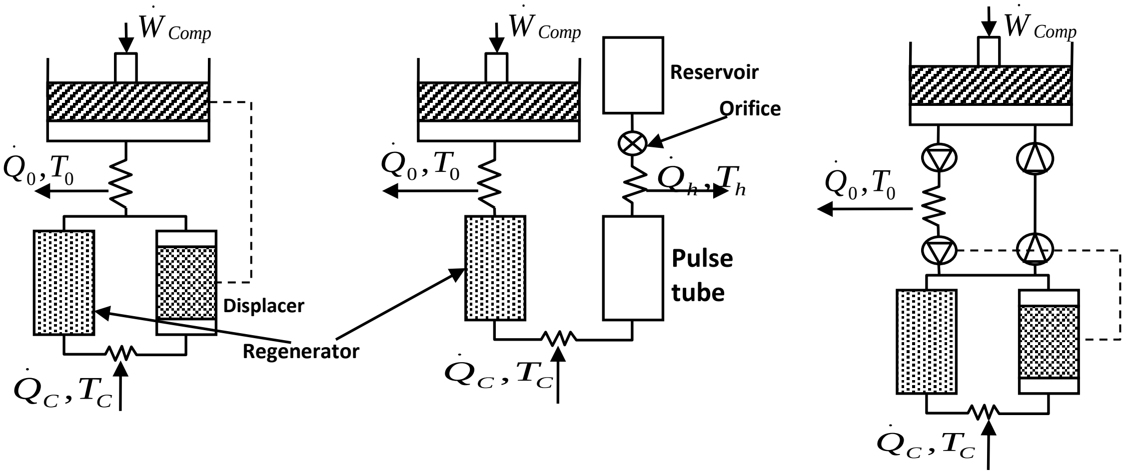


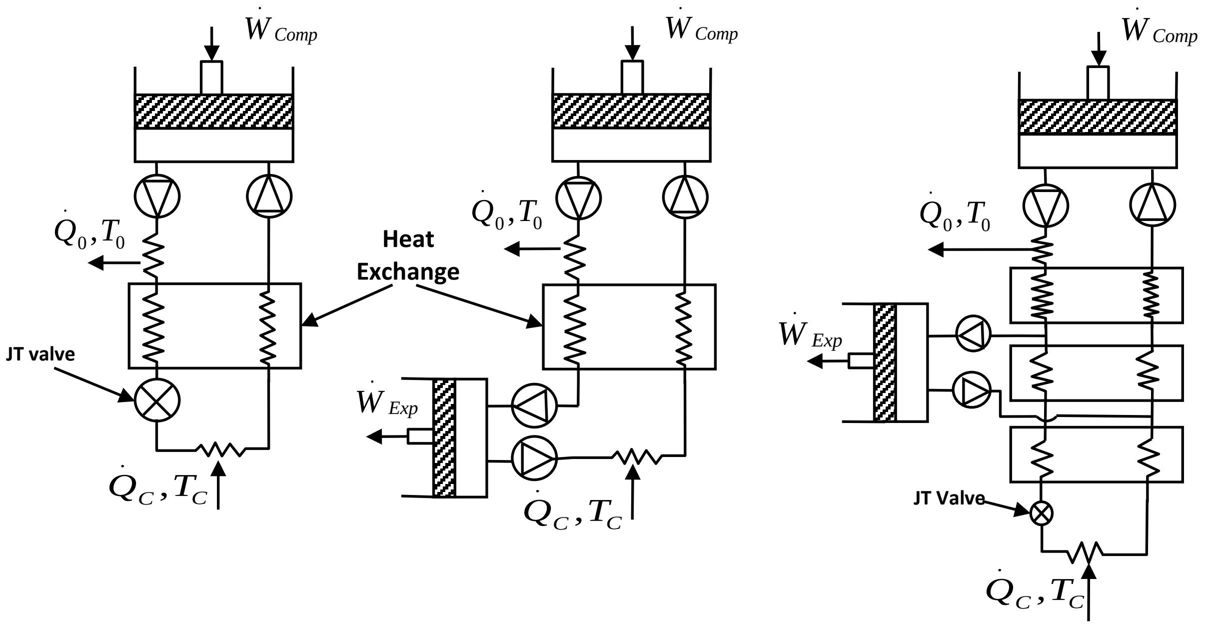
Fig 1.1. Classification of Cryo-coolers.

According to the type of heat exchanger used, Radebaugh [2] classified the cryocoolers as Regenerative type (Fig 1.2) and Recuperative type (Fig 1.3).



(a) Stirling **(b) Pulse tube** **(c) Gifford-McMohan**

Fig 1.2. Regenerative type Cryocoolers.



(a) Joule-Thomson **(b) Brayton** **(c) Claude**

Fig. 1.3. Recuperative type cryocoolers.

1.2. Pulse Tube Refrigerator

The most impressive feature of pulse tube refrigerators (PTRs) is absence of the moving parts at cold end. This feature of PTR makes it long life, reliable, low vibration and more efficient. The concept of PTR was 1st proposed by Gifford and Longsworth [3] in 1963. They constructed a device that consists of a hollow cylinder with one end closed and the other end open. The open end was connected to a pressure wave generator (compressor) through a regenerator, causing open end to cool, while the closed end was connected to an atmospheric temperature heat exchanger. This device was named as Basic Pulse Tube Refrigerator (BPTR).

1.3. Working Principle of PTRs

Pulse Tube Refrigerators (PTRs) can produce temperatures below 123K. PTRs are different from refrigerators those work on the principle of vapour compression cycle. PTRs are based on the theory of oscillating expansion and compression of gas (the working fluid) within a closed volume to acquire the desired refrigeration. Being oscillatory motion, PTRs are non steady systems and require time dependent solution. But, PTRs attain quasi-static steady periodic state like the other periodic systems. This oscillating compression and expansion of the gas occurs due to the oscillating pressure wave generated by a piston-cylinder arrangement. This pressure wave is created at one end of the system which generates an oscillating flow of gas in rest part of the system. This oscillating gas flow takes away heat from cold end heat exchanger i.e. low temperature section of pulse tube to the other side i.e. hot end heat exchanger. The refrigeration capacity of PTR depends upon its size and the power required for driving it.

1.4. Categorization of PTRs

2. *According to the nature of pressure wave generator*
    a. Stirling type PTRs
    b. Gifford McMohan (G-M) type PTRs

Stirling type PTR	**G-M type PTR**
Operating frequency is high (10Hz to 120Hz)	Operating frequency is low (1Hz to 5Hz)
Compressor is directly connected to expander without any valve in between.	Compressor is connected to expander with a valve in between (generally rotary or solenoid valves are used).
Dry compressor is used and capacity is low (in few hundred of Watts).	Bulky and oil lubricated compressor is used and capacity is more (in kW).
Coefficient of performance (COP) is high.	COP is low.
Pressure ratio is low.	Pressure ratio is high.
A temperature of 20K can be achieved.	A temperature of 2K can be achieved.

Table 1.1. Difference between Stirling and G-M type PTRs

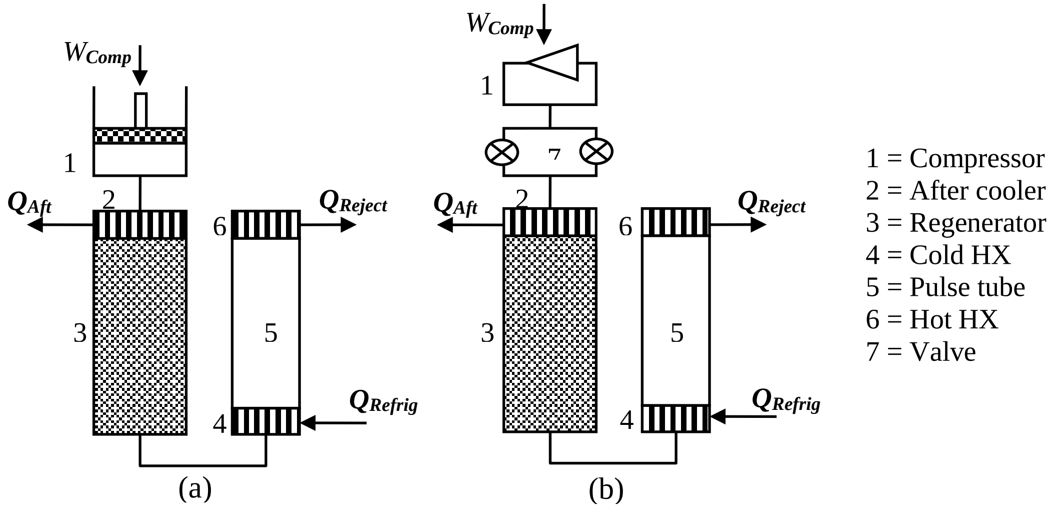


Fig 1.4. Schematic diagram of (a) Stirling type, (b) G-M type PTRs.

3. *Based on the way of development*
    a. Basic pulse tube refrigerator (BPTR)
    b. Orifice pulse tube refrigerator (OPTR)
    c. Double inlet pulse tube refrigerator (DIPTR)
    d. Inertance tube pulse tube refrigerator (ITPTR)
    e. Multi stage pulse tube refrigerator
    f. V-M type pulse tube refrigerator

BPTR is the 1st pulse tube refrigerator (PTR), which was proposed by Gifford and Longsworth [3] in the year 1963. Fig 1.5 shows the main elements of BPTR.

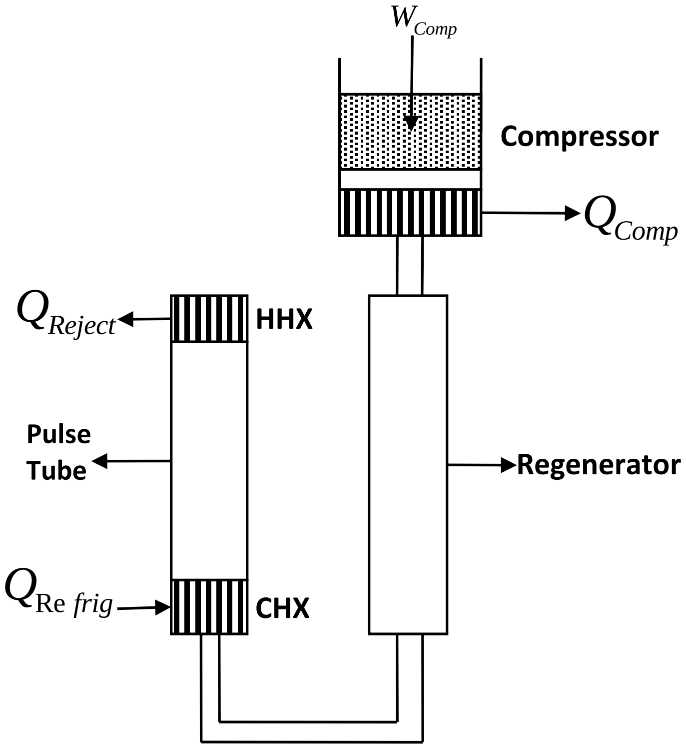


Fig 1.5. Schematic diagram of BPTR showing various components.

The demerit of BPTR is 90^0 phase difference between the mass flow rate and the pressure wave due to which the temperature obtained at cold end limits to 123K. This limitation can be understood as, when the pressure is maximum then mass flow rate is zero at the hot end side of the pulse tube. This limitation can be overcome by providing an orifice valve and a surge volume or reservoir after the hot end of the pulse tube. The addition of valve and reservoir decreases the phase angle below 90^0. This arrangement is named as Orifice Pulse Tube Refrigerator (OPTR) and a temperature of 60K can be achieved by using OPTR. Fig 1.6 shows the various components of OPTR.

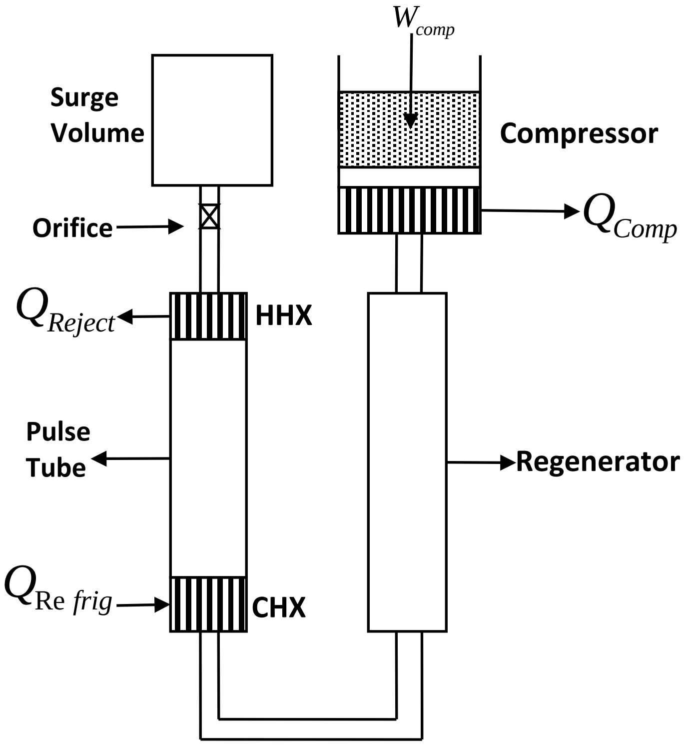


Fig 1.6. Schematic diagram of OPTR showing various components.

Also few limitations were there with OPTR. In case of optimal phase angle, the mass flow rate should lag the pressure at the inlet to the expansion space. But for OPTR, this optimal phase angle cannot be achieved as the mass flow rate always leads the pressure at limiting case. Moreover, a large volume of gas flows inside the pulse tube through regenerator without cooling. To overcome these limitations, one more valve is attached in between the hot end of pulse tube to the inlet of regenerator. This arrangement is called Double inlet Pulse Tube Refrigerator (DIPTR) and it enhances the refrigeration power and cooling per unit mass flow rate. Fig 1.7 shows the various components of DIPTR and arrangement of the components of DIPTR. A temperature of 41K can be achieved with the help of DIPTR and also the rate of temperature fall is more as compared to OPTR.

The drawback associated with DIPTR is the fluctuation of cold end temperature. This fluctuation of temperature occurs due to the circulating flow of gas through the pulse tube and regenerator, which is known as DC gas flow. To overcome this drawback, a long thin tube known as inertance tube is added in place of orifice valve of OPTR. Like the inductance in an electric circuit, inertance tube provides impendance which allows an adjustable phase relationship between pressure and mass flow rate in the pulse tube and reservoir. Thus, the phase relationship between mass flow rate and pressure can be adjusted to maximum and a higher cooling efficiency can be achieved. Fig 1.8 shows the various components of Inertance tube Pulse Tube refrigerator (ITPTR).

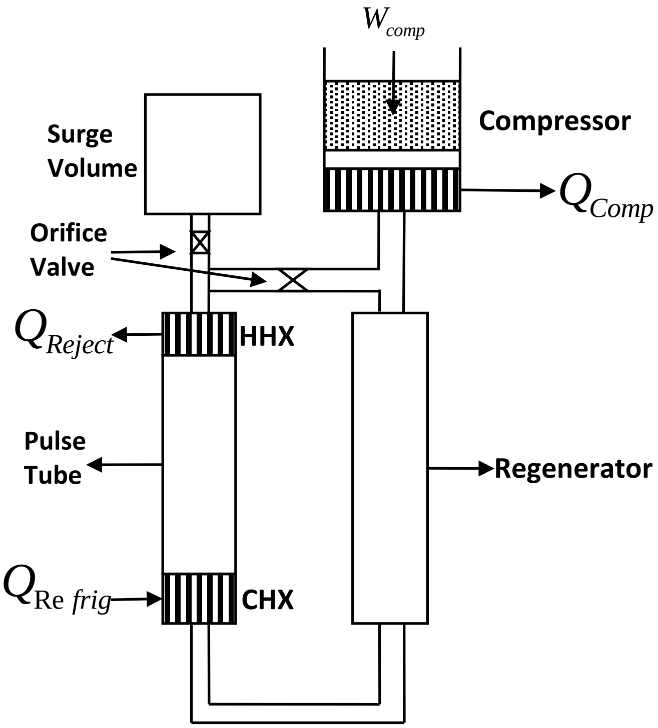


Fig 1.7. Schematic diagram of DIPTR showing various components.

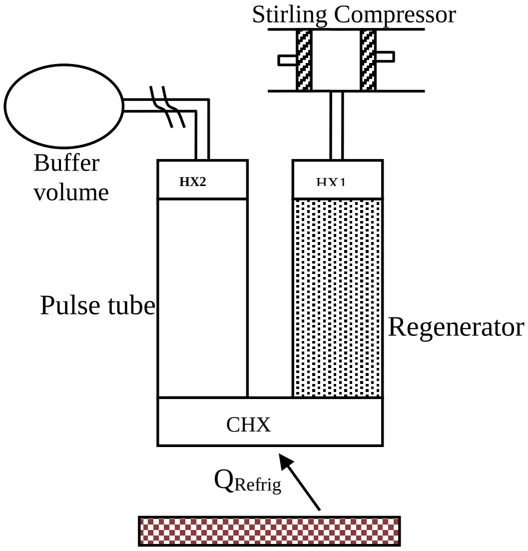
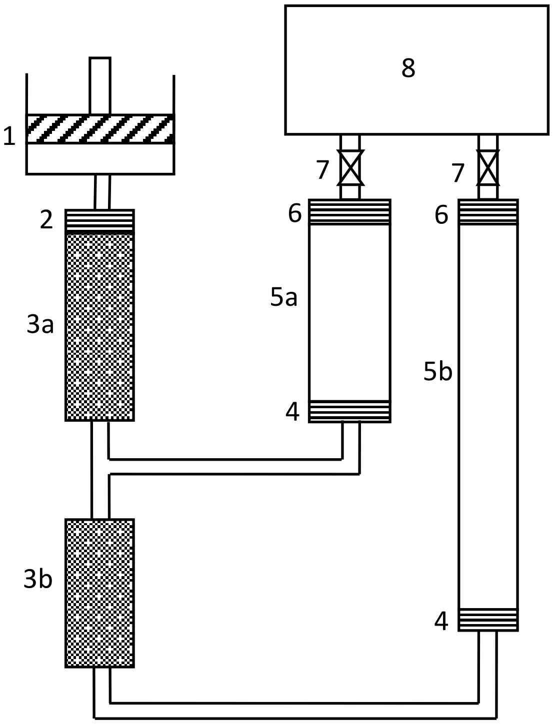


Fig 1.8. Schematic diagram of ITPTR.

Multistaging of pulse tube refrigerator (PTR) is done to achieve much lower temperature below 30K. Multistaging of PTR is done by simply arranging the PTRs in which the hot end of pulse tube in 2nd stage is connected to room temperature instead of connecting to the cold end of 1st stage. By 3-stage arrangement of PTRs, a temperature of 1.78K can be achieved by taking ^{3}He as working fluid. Fig 1.9 shows the arrangement of a Stirling type 2-stage DIPTR and Fig 1.10 shows a three stage arrangement of PTRs.



1- Compressor
2- After cooler
3a- Regenerator (1st stage)
3b- Regenerator (2nd stage)
4- Cold end HX
5a- Pulse tube (1st stage)
5b- Pulse tube (2nd stage)
6- Hot end HX
7- Orifice valve
8- Reservoir

Fig 1.9. Stirling type 2-stage double inlet pulse tube refrigerator (DIPTR).

The advantage of V-M type pulse tube refrigerator is that, it uses thermal compressor instead of mechanical compressor. In case of mechanical compressor, the oscillatory motion of piston causes the pressure oscillation, but in case of thermal compressor pressure oscillation is generated by temperature difference. Fig 1.11 shows the various components of V-M type pulse tube refrigerator. The main components of V-M type PTR are displacer, expander, work transfer tube, regenerator, pulse tube, heat exchanger. The heat exchangers are immersed in liquid nitrogen bath.

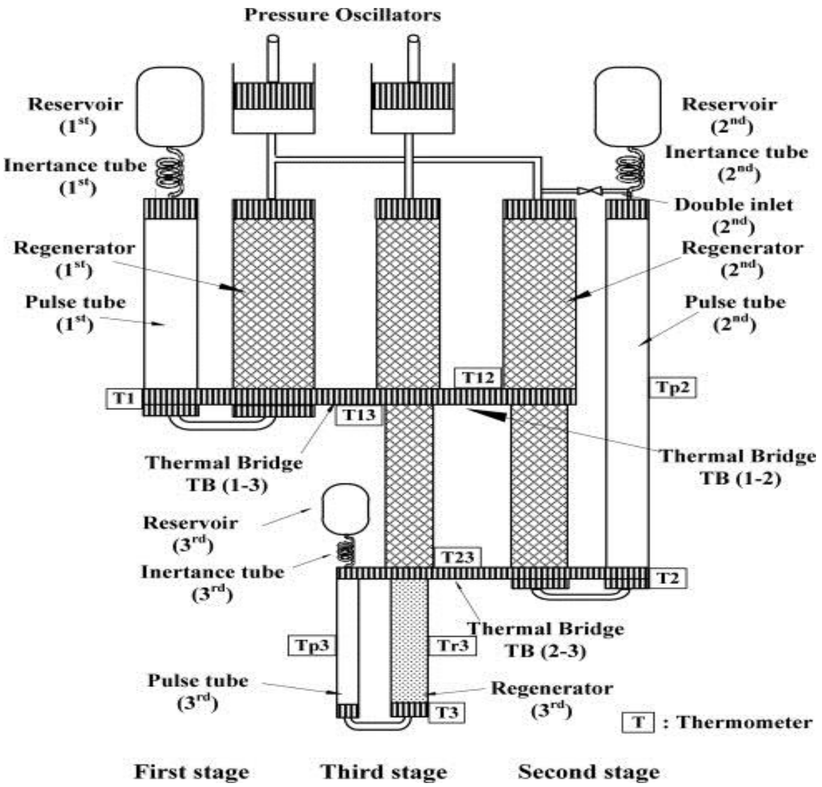


Fig 1.10. Three stage arrangement of PTRs.

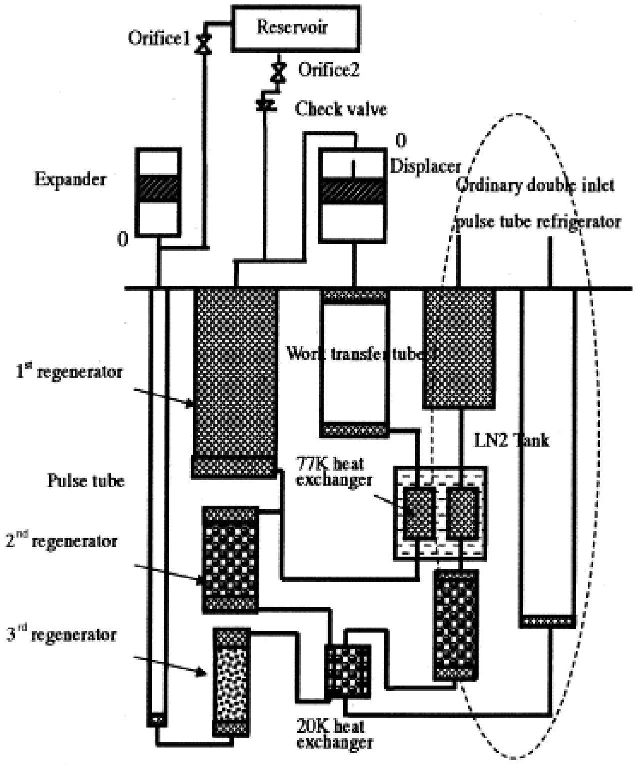


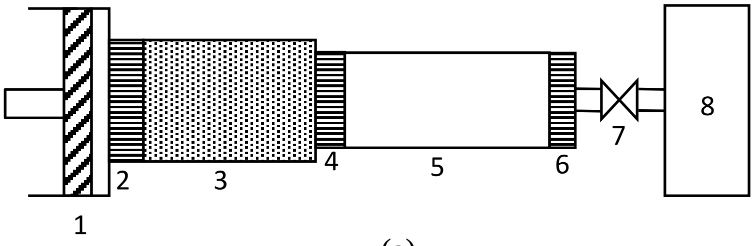
Fig 1.11. Schematic diagram of V-M type PTR.

4. *Considering Shape and Geometry*
    a. Line type pulse tube refrigerator
    b. Co-axial type pulse tube refrigerator
    c. U type pulse tube refrigerator
    d. Annular type pulse tube refrigerator

In Line type pulse tube refrigerator (PTR), all the components of PTR are arranged in a straight line starting from compressor to reservoir. As the loss due to the conductance of pipe is minimum in Line type of PTRs, these PTRs can be used where high performance is required. But, the disadvantage of Line type PTRs is the access of cold end as it is placed in middle of the arrangement. Also these PTRs required more space. Fig 1.12(a) shows the configuration of Line type PTR.

The configuration of U-type PTR is compact and it allows easy access to cold end. But, there are some losses due to pipe conductance and also due to the bend of pipe. Fig 1.12(b) shows the configuration of U-type PTR. Co-axial arrangement of PTRs is very much compact. The regenerator of these type PTRs are constructed as a ring shape. This ring shape regenerator surrounds the pulse tube as shown in Fig 1.12(c). As there is a physical contact between regenerator and pulse tube, there is a large temperature difference and hence heat transfer between pulse tube and regenerator. The remedy for this difficulty is to provide a thin layer of insulation between pulse tube and regenerator, which increases the overall size of refrigerator.

Fig 1.12(d) shows arrangement of annular type PTR. In this arrangement, the regenerator is placed inside the pulse tube, whereas in case of co-axial PTR pulse tube is placed inside the regenerator.



(a)

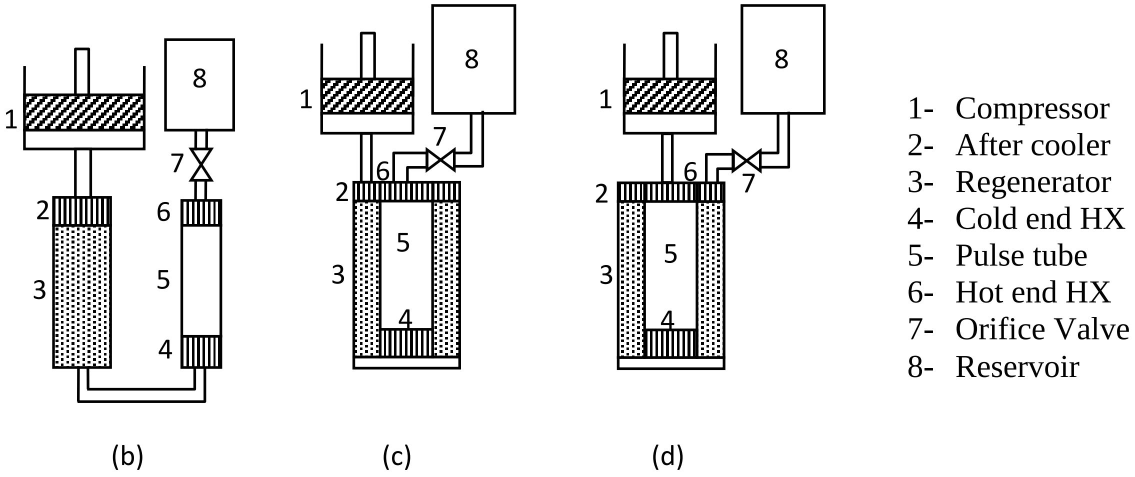


Fig 1.12. Schematic diagram of (a) Line type PTR, (b) U-type PTR, (c) Co-axial PTR and (d) Annular PTR

1.5. Main Components of Pulse Tube Refrigerator (PTR)

The main components of a PTR include a compressor, an after cooler, a regenerator, a cold end heat exchanger, a pulse tube, hot end heat exchanger, an orifice valve for Orifice Pulse Tube Refrigerator (OPTR) and an inertance tube for Inertance Tube Pulse Tube Refrigerator (ITPTR) and at last a reservoir.

- **Compressor**

  The compressor is a device which is used to generate high and low pressure oscillation in PTR. Hence, gas pressurization and depressurization takes place inside a closed volume. Input to the compressor is electric power and it converts the electric power to equivalent mechanical power that oscillates the piston inside the compressor. Generally, reciprocating compressors are used in Stirling type PTRs and for high frequency moving coil type linear motor driven compressors are preferred.

- **After Cooler**

  It is a device which is placed in between compressor and regenerator in the arrangement of PTR. The gas is slightly heated while compressing in compressor. The main purpose of after cooler is to remove almost all the heat from the gas coming from compressor and to supply cold gas to the system.

- **Regenerator**

  Regenerator is the heart of PTRs. The main function of regenerator is, it absorbs heat from gas during compression stroke and gets heated. Then subsequently release the same heat to cold gas during return stroke and gets cooled. The regenerator is made up of porous matrix. Stainless steel wire screens are generally used for constructing porous matrix. To get maximum enthalpy flow in pulse tube, regenerator should be ideal i.e. there should be no pressure drop and effectiveness should be 100%. But, it is very much difficult to maintain effectiveness of regenerator 100% at cryogenic temperature. So, stainless steel is used as regenerator packing material as they provide better heat transfer area, high specific heat value, low pressure drop and low thermal conductivity.

- **Cold end Heat Exchanger (CHX)**

  As compared to the evaporator in a Vapour Compression Refrigeration cycle, there is CHX in case of PTRs. The function of CHX is same as that of evaporator. The system absorbs refrigeration load at CHX. It is placed in between regenerator and pulse tube. Copper wire screens are used for exchanging heat between CHX and housing wall.

- **Pulse Tube**

  Pulse Tube is a hollow cylindrical tube made up of stainless steel. The main objective is to carry heat load from cold end HX to the hot end HX with the help of enthalpy flow. The mechanism behind heat carrying from cold end to hot end of the pulse tube is phase shifting mechanism. The thickness of the pulse tube wall is kept optimum to promote surface heat pumping.

- **Hot end Heat Exchanger (HHX)**

  HHX is used to remove heat of compression from gas in every stroke of cycle. The heat is released to atmosphere. Generally, water cooling or air cooling systems are used to take heat away.

- **Phase shifter**

  Orifice valve for OPTR and inertance tube for ITPTR are usually known as phase shifter. These are placed in between the hot end of pulse tube and reservoir. The optimum phase relationship can be achieved by adjusting orifice valve’s diameter in OPTR and by adjusting length and diameter of inertance tube in ITPTR. Physically, inertance tube is a long thin tube and orifice valve is a needle valve type.

- **Surge volume**

  Surge volume or reservoir is placed after the phase shifter. It is a closed volume. Its volume can be adjusted in such a way that pressure variation and mass flow variation is negligible inside it.

- **Rotary Valve**

  For G-M type pulse tube refrigerator, rotary valve is used to generate pressure wave. Generally pressure oscillation frequency generated by rotary valve is 1Hz to 3Hz. Fig 1.13 shows the design of rotary valve.

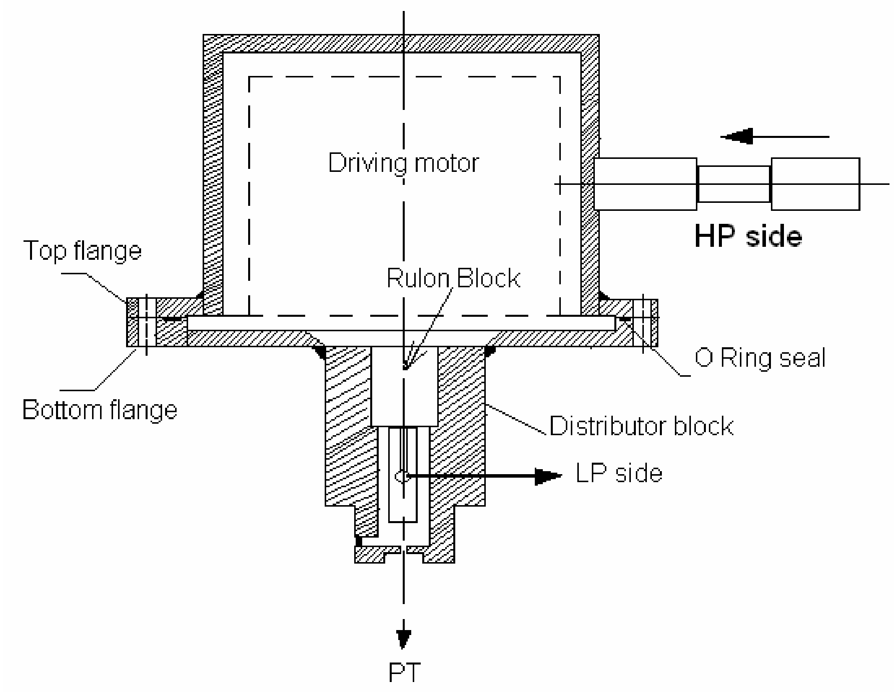


Fig 1.13. Typical design of Rotary Valve.

1.6. Basic Theories of PTRs

- **Surface Heat Pumping Theory**

  Surface heat pumping mechanism was explained by Gifford and Longsworth [4]. According to this mechanism, as there is temperature gradient in oscillating flow, there is heat transfer between surface wall and oscillating gas. This theory is the basic theory of BPTR.

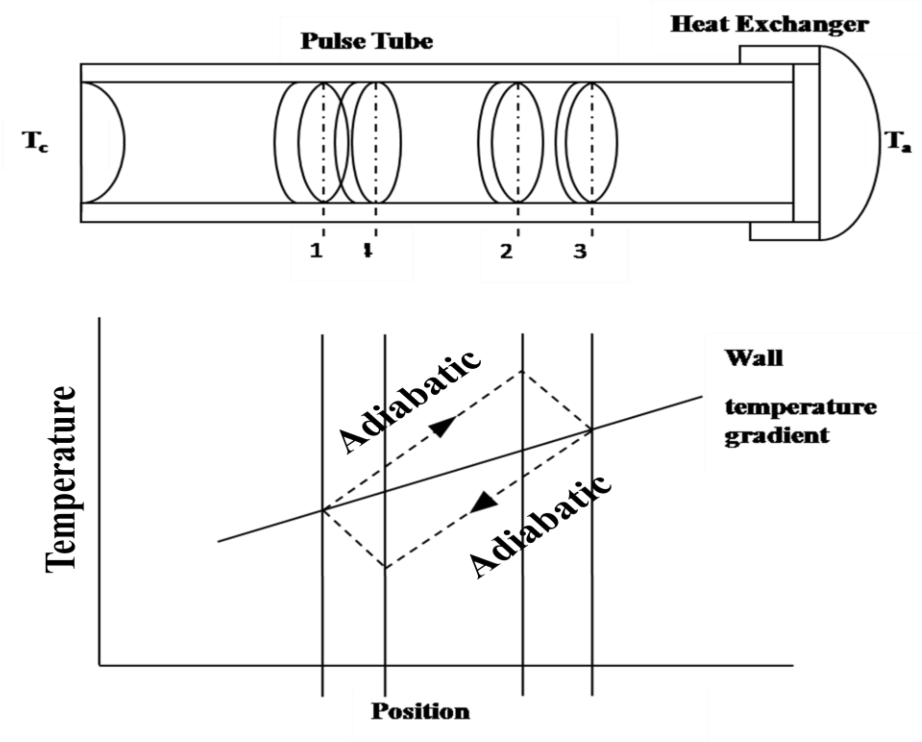


Fig 1.14. Surface heat pumping mechanism.

- **Enthalpy flow model**

  Enthalpy flow theory is applied to various components individually in a PTR. The cyclic average of heat flow can be calculated by simply integrating the governing differential equations. Using this theory time-dependent analysis is carried out to find heat flow and temperature variation at different sections of PTR. Based on this theory, phasor analysis is developed to know the dependence of various parameters on the performance of PTRs.

- **Thermo-acoustic Theory**

  In a thermo-acoustic device, heat flow, work flow and mutual conversion of them is possible. The thermo-acoustic device generally contains two media; one is solid and other one fluid. The solid is the matrix of regenerator or the wall of pulse tube and fluid is generally oscillating in nature. If a large temperature gradient exists inside a tube which is closed at one end, the gas inside that tube starts oscillating. Similarly, the oscillating gas inside the closed tube generates temperature gradient across the ends.

1.7. Application of PTRs

2. *Defence*
   a. Infrared sensors used for guidance of missile, surveillance and night vision.
   b. Monitoring the nuclear activity by Gamma-ray sensors.
3. *Environmental*
   a. Infrared sensors used for monitoring of pollution and also for studying atmosphere.
4. *Commercial purpose*
   a. Cryo-pumps used for fabrication of semiconductors.
   b. Superconductors used for high speed communication and electric transmission with minimum losses.
   c. Semiconductors used for high speed computers.
   d. Liquefaction of industrial gases like Nitrogen, Helium, Oxygen.
5. *Medical science*
   a. Used for cooling of superconducting magnets for Magnetic Resonance Imaging (MRI).
   b. To study heart and brain (SQUID magnetometers).
   c. Cryosurgery and cryo-ablation catheters.
6. *Transportation*
   a. Liquefied Nitrogen gas used for vehicles.
   b. Superconducting magnets used for Maglev trains.
7. *Security*
   a. Infrared sensors used for rescue and for security purpose in night.
8. *Biology and Agriculture*
   a. For storing specimens, biological cells and tissues.
   b. High pressure cooling for preservation of macromolecular crystals.

1.8. Objective of Work

- Developing the geometry of DIPTR and meshing of DIPTR.
- Writing pressure UDFs of different wave forms. The different wave forms are Sinusoidal, Rectangular, Triangular and Trapezoidal.
- Simulation of DIPTR using ANSYS FLUENT 15.0.0 software to know, for which pressure UDF, minimum temperature is obtained at cold end of DIPTR.

1.9. Outline of Thesis

- Chapter I contains basic introduction to PTRs, developing stages of PTRs, working principle of PTRs, Basic theories employing for PTRs and applications of PTRs on various fields.
- Chapter II contains literatures review on PTRs.
- Chapter III contains the modeling procedure for DIPTR, geometry and also meshing and also the governing equations for DIPTR.
- Chapter IV contains the simulation procedure, pressure UDFs, boundary conditions specified for DIPTR, cool down behavior of temperature at cold end and also the contours.
- Chapter V contains the conclusion and future scope of this work.

CHAPTER II

LITERATURE REVIEW

2.1. Introduction

This chapter contains literature survey those are published on the way of development of Pulse Tube Refrigerators (PTRs). This chapter focuses on theoretical investigations on PTRs, experimental and numerical investigations on PTRs and the efforts of various researchers to improve the regenerator performance and also for improving the overall performance of PTRs.

2.2. Basic Pulse Tube Refrigerators (BPTR)

Two scientists from Syracus University, Gifford and Longsworth introduced a new method to achieve low temperature in 1963, Pulse tube refrigerator. They published their 1st paper in 1964 [3] which described that “temperature gradient is generated in a closed volume by simply pressurizing and depressurizing the gas inside that closed volume from a point on its periphery”. The temperature gradient hence generated depends upon operating conditions and the size and geometry of closed volume.

There 1st design comprised of a hollow cylindrical tube with one end closed and the other end open. The closed end was exposed to the atmospheric heat exchanger and the open end is the clod end. Due to the oscillatory motion of the piston, an oscillatory pressure wave is created inside the tube which causes the cooling of open end. This refrigeration system is named as Basic Pulse Tube Refrigerator (BPTR).

Gifford and Longsworth [4] predicted the relation between cooling temperature at cold end and zero heat pumping concerning length ratio, hot end temperature and the ratio of specific heats of gas by using the mechanism of surface heat pumping. They observed that the surface heat pumping occurs due to the abnormal interaction between the fluid displacement and exchange of energy in the surface and fluid. Gifford and Longsworth [5] examined the BPTR between the critical pressure ratios and got the useful refrigeration.

Gifford and Longsworth [6] compared the operation of BPTR with the valved pulse tube to overcome the problems reported by them for BPTR. Various improvements were done by de

Boer [7] considering the gas motion during cooling and heating steps and a thermodynamic model of BPTR was developed. He got more accurate temperature profiles. An important improvement in BPTR was done by de Boer [8] by adding a heat exchanger and regenerator at both the ends of pulse tube. He studied the heat flow in regenerator, heat exchangers and pulse tube by using control volume analysis. Jeong et al. [9] carried out an analytical study on the secondary flow of BPTR. They studied the secondary flow of gas within the pulse tube considering the effect of axial temperature gradient.

Longsworth [10] experimentally investigated the heat pumping rates of PTR and also furnished an empirical relation for correlating experimental data with the empirical solution. He observed that the heat pumping rate is directly proportional to the tube length by keeping all other parameters unchanged. Longsworth performed an experiment on heat pumping rates of pulse tube refrigeration and also an empirical formula has been given to relate with the experimental data. He concluded that the heat pumping rate varies directly with the tube length when all the remaining parameters are kept constant.

Barrett et al. [11] developed a model of a pulse tube cryocooler for studying oscillating flow inside the cryocooler by using a commercial Computational Fluid Dynamics (CFD) software package. They developed a 2D axis-symmetric model and demonstrated the time varying velocity and temperature fields in the pulse tube and also the heat fluxes at the hot and cold end heat exchangers. Jiao et al. [12] experimentally and numerically studied the heat transfer behavior of a cryogenic fluid (Helium gas) flowing through a miniature tube. They used thermo-physical properties those depend on temperature for their analysis. They concluded that the heat flow characteristics of cryogenic fluid (Helium gas) considering temperature dependent thermo-physical properties are deviated from those with constant thermo-physical properties at ambient condition.

Ashwin et al. [13] performed the numerical simulation of high frequency miniature PTRs. They simulated the model by considering different length-to-diameter ratios (L/D ratios) of the pulse tube by using CFD package FLUENT software. For modeling the porous zones, the local thermal non-equilibrium of gas and porous matrix were considered. They examined the dynamic characteristics of gas flow and mechanism of heat transfer in the tube. They found that, considering thermal non-equilibrium of gas and matrix a much lower temperature at cold end side of tube as compared to that of thermal equilibrium conditions. They concluded that for pulse tube diameter 4 mm and L/D ratios between 10 and 20 is beneficial.

2.3. Orifice Pulse Tube Refrigerator (OPTR)

Orifice Pulse Tube Refrigerator (OPTR) was 1st invented by Mikulin et al in 1984.

Wu et al. [14] analysed an OPTR analytically with valveless compressor and they explained the processes occurring inside the pulse tube of OPTR. Richardson [15] developed valved pulse tube refrigerator, known as orifice pulse tube refrigerator. He observed that these refrigerators reach much lower temperature than valveless PTRs. Lee et al. [16] studied the OPTR by using surface heat pumping mechanism. They considered the effect of velocity of gas on surface heat pumping. Kasuya et al. [17] studied the optimum phase angle, for which temperature obtained is minima at cold end, between pressure and gas displacement oscillation in PTR.

Wang et al. [18] numerically studied the detailed performance and characteristics of an OPTR. They used an improved numerical modeling technique which considers the friction of flow, heat transfer characteristics in regenerator and heat exchanger and also the properties of material. Ames research center of NASA organized a research program on PTR in 1992 which was led by Kittel [19]. This program mainly focused on the optimization of compressor i.e. pressure wave generator and thermal regenerator to improve the performance of PTR.

Storch et al. [20] has developed an analytical model of OPTR by considering phasor analysis. Zhu et al. [21] developed an isothermal model for OPTR and compared the obtained results with the nodal analysis. They concluded that the values obtained for input power and gross refrigeration power are nearly 20% lower than that of the former nodal analysis. And also the obtained values for average mass flow rate and pressure ratio are nearly 5% lower using this nodal analysis model. A new technique was developed for instantaneous measurements of temperature and mass flow rate for OPTR in actual operations by Rawlins et al. [22].

Kittel et al. [23] described the fundamental behavior of PTR using the concept of phasor analysis. They developed a simple one dimensional model and studied the concept of enthalpy flow and Gibb’s free energy flow to identify the loss mechanism and to find the importance of different losses. Huang and Chuang [24] experimentally designed a linear flow network model of an OPTR. This analysis considered the pressure as electric voltage and the mass flow as electric current. De waele [25] studied the dynamic behavior of temperature profiles in the cold end and hot end of pulse tube and also for the regenerator of OPTR and double inlet pulse tube refrigerator (DIPTR) analytically.

Gerster et al. [26] furnished a loss at hot end by taking enthalpy flow model into account. They concluded that the loss at hot end occurs due to regenerative effect of the heat exchanger at hot end. De Boer [27] presented the optimization results on OPTR. He has taken the orifice volume very large and the regenerator volume zero. He concluded that all the dimensional quantities of importance depend upon the dimensionless frequency and the ratio between orifice conductance and the regenerator conductance. Roach [28] described a model for evaluation of oscillating pressure, enthalpy flows and mass flows in the main components (like pulse tube, regenerator, cold end heat exchanger) of OPTR. He also presented the phasor analysis for pressure and mass flow rate at the cold end for OPTR.

To increase the efficiency of the PTRs Mikulin et al. [29] have designed and constructed an OPTR by joining a buffer reservoir to warm end side of pulse tube. Here, the reservoir is a buffer of quasi-steady pressure. They concluded that, optimum rate of flow of gas entering to the reservoir and also the optimum phase shift between the reflected and incident pressure waves can be obtained by changing the flow area. Radebaugh et al. [30] have been experimentally developed a new model for OPTR and temperature of 60 K was achieved using single stage of pulse tube.

Liang et al. [31] developed a set up to conduct an experiment on OPTR which can produce a temperature below 49K. They studied the relationship between ratio of volumes between regenerator and pulse tube and the minimum temperature obtained at the cold end of OPTR and also the influence of matrix materials and dimensions on the performance of OPTR. They concluded that in OPTR the amount of gas passes through regenerator is slightly large and load on the regenerator is also large. Zhang et al. [32] simulated 2D axis-symmetric model of GM-type orifice pulse tube cryocooler using CFD (Computational Fluid Dynamics) package fluent software. They demonstrated the modeling process and results like phase relationship between velocity and pressure at the cold end, the temperature profiles along the wall of OPTR and also the oscillations of temperature at cold end considering various heat load conditions.

Antao and Farouk [33] have performed a numerical study on OPTR to investigate fundamental fluid flow process and heat transfer process. The OPTR was driven by cyclically oscillating piston and Helium was used as working fluid. Regenerator and all the heat exchangers were defined as porous regions and non-equilibrium models of thermodynamics were applied in these regions. They found that at an optimum frequency of 34 Hz, a bi-cellular structure is obtained which intensifies the performance of OPTR by generating a dead zone (buffer region) that effectively isolates the cold end and hot end of the pulse tube.

2.4. Inertance Tube Pulse Tube Refrigerator (ITPTR)

Gardner and Swift [34] replaced the orifice valve with the help of an inertance tube. They calculated phase relationship between oscillating pressure and oscillating velocity experimentally and concluded that the cooling power is more in Inertance Tube Pulse Tube Refrigerator (ITPTR) than that of the OPTR. An experimental study was performed by Roach et al. [35]. They found that a long thin tube (inertance tube) introduces an additional phase shift between the pressure and flow of mass in the pulse tube segment, which causes more cooling than OPTR.

Zhu et al. [36] performed the numerical simulation of ITPTR. They used nodal analysis for simulation of ITPTR. Dai et al. [37] performed theoretical computation for inertance tube in ITPTR without reservoir volume and concluded that the phase-leading effect is large with these devices. They analysed the relation of phase-leading requirement with the pulse tube geometry with the help of phasor diagram and concluded that larger the void volume of pulse tube, larger will be the phase-leading effect. Cha et al. [38] designed two ITPTR systems and numerically simulated the both using Computational Fluid Dynamics (CFD) package FLUENT software. They simulated the two ITPTRs with different boundary conditions and examined the flow in multi-dimensions and heat transfer effects. They showed that, the 1D analysis is feasible only when length to diameter ratio of all the components of ITPTR is large.

Banjare et al. developed 2D axis-symmetric models for orifice pulse tube refrigerator (OPTR) [39] and ITPTR [40] and simulated them using CFD software at different frequencies. They used dual opposed piston compressor in their simulation. They observed that at higher frequency, due to turbulence and recirculation of fluid, deterioration occurs in the overall performance of the system. Simulation result showed that there is an optimum frequency exists for each PTR model at which maximum refrigeration occurs.

To optimize a single stage ITPTR, Rout et al. [41] have performed a numerical solution by using CFD package FLUENT software. They have taken length of pulse tube as varying parameter and operating frequency of 34 Hz. They found that at a pulse tube length of 125 mm, a minimum temperature of 58K is achieved while keeping all other parameters unchanged. They compared the simulation result with Cha model. Rout et al. [42] numerically studied the impact of porosity on the performance of a single stage co-axial ITPTR. They used CFD package FLUENT software for simulation and studied the cooling behavior, heat transfer inside the pulse tube and variation of pressure inside the whole system. They fixed the dimension of pulse tube at 5 mm diameter and 125 mm length and the operating frequency was 34 Hz. They found that at a porosity of 0.6, a better cooling at cold end of ITPTR is produced. They validated the simulation result with Cha model.

2.5. Double Inlet Pulse Tube Refrigerator (DIPTR)

Zhu et al. [43] have done an important improvement in OPTR and developed Double Inlet Pulse Tube Refrigerator (DIPTR). They have constructed a DIPTR by simply modifying the OPTR that they have added one more valve in between the hot end of pulse tube and inlet to the regenerator. Theoretically gas flowing from cold end to the pulse tube can do maximum work and also rate of mass flow into pulse tube through the regenerator is reduced. They studied numerically and experimentally and concluded that the DIPTR has better performance than OPTR. A numerical simulation model has been developed for OPTR and DIPTR by Ju et al. [44]. They studied the occurrence of physical phenomena in pulse tube and also studied the impact of orifice valve and double inlet (DI) valve on the refrigeration power and efficiency of PTR. A numerical analysis and also a numerical model have been developed for DIPTR by Wang et al. [45]. The equations of continuity, momentum and energy were being solved and also predicted that, refrigeration power is higher and P-V work done is lower for DIPTR than in case of the OPTR.

Herold et al. [46] performed an analytical simulation on DIPTR by using a stepped piston compressor for generating the pressure wave. Zhu et al. [47] performed an analytical study on work loss for DIPTR. They developed analytical equations for calculating the mass flow rate passing through double inlet valve of DIPTR by analyzing pressure drop in the regenerator. Kirkconnell [48] has performed numerically analyze the mass flow rate and thermal behavior in the pulse tube for high frequency pulse tube refrigerators. Zhu and Chen [49] developed integration formula for calculating the flow rate of enthalpy in the pulse tube of a pulse tube refrigerator. They performed the calculation by assuming that, mass flow rate and fluctuation of pressure are sinusoidal. They derived the formula for both ideal OPTR and ideal DIPTR by taking helium as working fluid. They evaluated the volume of pulse tube for OPTR and DIPTR in ideal conditions using Langrange method.

Thummes et al. [50] designed and constructed a two phase 4K G-M type ^{4}He liquefaction system. This GM-cooler was operated with GM-rotary (Gifford-McMohan rotary) valve and a GM type compressor which has an input power of 6 kW. They got the temperature of 4.2K and liquefaction rate of 0.13 l/hr at steady state. Yuan and Pfotenhauer [51] have been developed an analytical model to study the thermodynamics behind the working of an active-valve pulse tube refrigerator. They concluded that the refrigeration capacity of a given pulse tube is achieved by simply controlling the flow of gas to and from both the ends of pulse tube and also the optimized performance depends upon the two intermediate pressures.

Xu et al. [52] have performed a theoretical study on pulse tube refrigerator. They have examined that, using ^{3}He as working fluid, a temperature below 2K can be achieved and the efficiency of 4 K PTR can be improved. De Boer [53] theoretically studied the performance for a DIPTR. The assumptions taken by him are, regenerator was theoretically perfect and the regenerator temperature profile was linear. Chokhawala et al. [54] performed the phasor analysis for double inlet pulse tube cryocooler. They studied that the DIPTR improves refrigeration power and coefficient of performance (COP). Banjare et al. [55] developed a model for numerical simulation of a single stage DIPTR and explained isothermal and adiabatic behavior of gas for pulse tube and compressor.

Zhang et al. [56] performed a numerical analysis for the effect of reservoir volume on the performance of various components of simple model of OPTR and double inlet pulse tube cryocooler. They studied the entropy generation for various components of pulse tube refrigerator and shown that the volume of reservoir has a significant effect on entropy generation in the different components of PTR when the ratio of reservoir volume to pulse tube volume is less than about to 5.

A thermodynamic model based on flow of energy through pulse tube refrigerators has been developed by Razani et al. [57]. They proposed that an exertic efficiency parameter represents the losses inside the pulse tube. They also studied the effect of phase-shifting mechanism for both exergy and energy flow in PTRs. Ju et al. [58] studied the thermodynamic losses of the rotary valve and coefficient of performance of G-M type pulse tube refrigerator (PTR). He used both 1st and 2nd laws of thermodynamics for his study.

Cai et al. [59] developed an experimental set up for DIPTR. They have been discussed the effect of variation of amplitude and phase difference of the mass flow rate and pressure wave. The main function of Double inlet valve in DIPTR is to adjust phase-shift between the mass flow rate and pressure wave in the pulse tube and also to increase the amplitude. Wang et al. [60] theoretically and experimentally did some modifications on DIPTR. They used an auxiliary piston in place of orifice valve of OPTR and reservoir was there as in case of OPTR. They verified the experimental data for both DIPTR and OPTR and concluded that this arrangement significantly improves the refrigeration effect and overall performance of PTRs.

Yang et al. [61] introduced a mechanism of double inlet to decrease the cold end temperature of PTR by analyzing phase relationship between pressure and flow characteristics of double inlet and also considering multi-bypass technique. After conducting the experiment, they found that, a temperature of 77 K is achieved at cold end by using two double inlet valves and 50 K at cold end by using one double inlet valve. Gan et al. [62] have carried out an experimental study on two component multi-phase Nitrogen and Helium mixtures in a single-stage PTR. They concluded that both cooling power and COP (coefficient of performance) can be improved at a temperature above 77K when the fraction of Nitrogen in the mixture is not more than 25%.

A rotary valve has been developed by Kasthurirengan et al. [63] for cryocooler application indigenously. They developed a 6 watt single stage PTR (pulse tube refrigerator) operating at a temperature of 77K in their 2nd paper [64]. In their final technical report [65], the design parameters, experimental set up and experimental results have been demonstrated for a single stage GM type DIPTR. Roy et al. [66] theoretically and experimentally investigated the PTR. They developed a numerical model which could be used for approximate design of PTR (pulse tube refrigerator).

Lu et al. [67] have performed a numerical and experimental investigation on single-stage double inlet GM type PTR, in which the fluctuating amplitudes of physical quantities were taken as large and fluctuating frequencies were taken as low for the system. They measured the temperature distribution on the surface of pulse tube and regenerator and also the cooling capacities under various refrigeration temperatures. Furthermore, they developed a time dependent (transient) one-dimensional (1D) model for numerical simulation of DIPTR and verified the simulation result with the experimental result. Also they studied the thermal nonlinear characteristics for DIPTR. Koettig et al. [68] have demonstrated an experimental result on the direction and quantity of heat flow within a PTR of co-axial configuration. They concluded that a well adopted geometrical configuration between the regenerator and pulse tube is very much essential for getting better cooling effect and refrigeration capacity.

Banjare et al. [69] performed the 2D numerical simulation of a G-M type DIPTR (double inlet pulse tube refrigerator). They used CFD package FLUENT software. They simulated the 2D DIPTR model with various thermal boundary conditions and a sinusoidal pressure user defined function (UDF) was applied at one end of the DIPTR. They demonstrated the cooling down behavior inside the system, phase relationship between mass flow rate and pressure at pulse tube region and also the temperature profile along the wall of the cryocooler and compared the simulation data with the experimental data.

Banjare et al. [70] have performed a numerical simulation on a 3D GM type DIPTR operating under various thermal boundary conditions. They have used commercial CFD package FLUENT 6.1 software for modeling and simulation. They have applied a sinusoidal pressure UDF at one end and demonstrated the cool down behavior of the system, relationship between pressure wave and mass flow rate at the cold end of DIPTR and temperature distribution along the wall of cryocooler and validated the data with experimental results.

CHAPTER III

MODELING OF DOUBLE INLET PULSE TUBE REFRIGERATOR (DIPTR)

3.1. Introduction

DIPTR is mainly working as a Gifford-McMohan (G-M) type pulse tube refrigerator, whose working principle is different from Stirling type pulse tube refrigerator, like OPTR, by means of pressure distribution. The pressure distribution system in G-M type pulse tube refrigerator limits the working frequency (generally rotary valves are used for pressure distribution) and thus G-M type pulse tube refrigerator generally works under low frequency (1 Hz to 5 Hz). But, the oscillating amplitude is large which yields lower temperature as compared to Stirling type pulse tube refrigerators.

Here, details of geometry and boundary conditions for the CFD simulation of DIPTR by using FLUENT software are discussed. A two dimensional model of DIPTR and also meshing is created by using ANSYS software. The oscillating motion to the piston is given with writing a pressure UDF using C' programming language. Here, four UDFs of different wave forms are written. The wave forms are sinusoidal, rectangular, triangular and trapezoidal. Simulation is done by applying those four UDFs at inlet wall and then the results are compared to see which UDF yields lower temperature.

3.2. Geometry of DIPTR

The geometric design and nodalization of various components of DIPTR is done by using ANSYS Design Modeler (DM). Both quadrilateral and triangular meshing is done. The main components of DIPTR includes a compressor, a transfer line, after cooler, a regenerator, CHX, pulse tube, HHX, Double inlet valve (DI valve) and a reservoir. Piston is the only moving part in the model of DIPTR and rests are stationary.

Initially the total geometry is done as a single unit and then splitting is done to create different zones. The function of creating different zones is that, it enables to set various boundary conditions for various zones or for various components of DIPTR as required. After creating faces meshing is done using ANSYS Assembly Meshing. Mesh generation is very much important as the partial differential equations generate an infinite dimensional problem. The different options for generating mesh are triangular, quadrilateral, tri-pave etc. After mesh generation, the model is exported to fluent and then simulation is carried out.

Components	Length (mm)	Diameter (mm)
Transfer line	115	5.3
After cooler	20	22
Regenerator	210	20
Cold end heat exchanger	20	15
Pulse tube	250	15
Hot end heat exchanger	20	22
Reservoir	300	100

Table 3.1. Dimensions of various components of DIPTR.

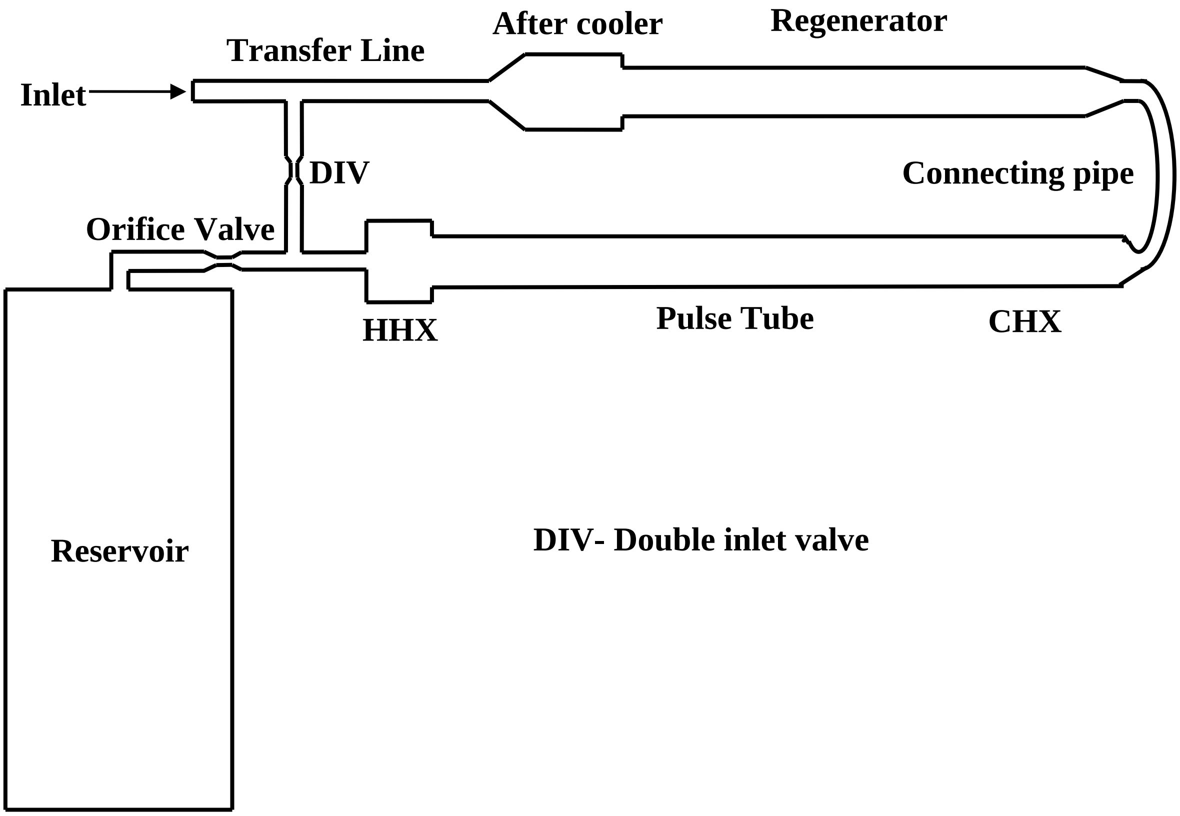


Fig. 3.1. Physical model of G-M type DIPTR.

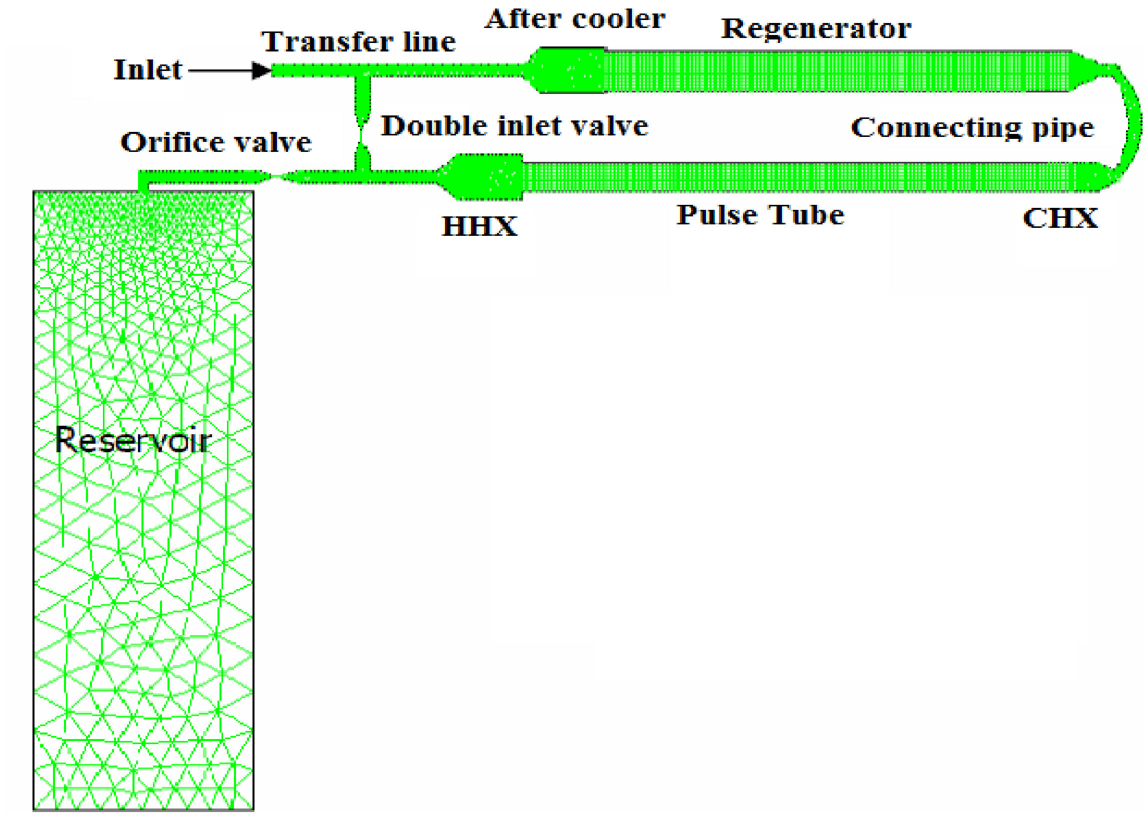


Fig. 3.2. Two-Dimensional geometry of DIPTR with Meshing.

3.3. Technique for Solution

Finite Volume Method (FVM) is one of the best differentiation technique used for simulation of DIPTR. This FVM technique solves the governing equations for mass flow rate of fluid, heat transfer and fluid flow problems and the results obtained satisfy the conservation equations of mass, momentum and energy. In this FVM, the computational domain is divided into control volumes or continuous cells where the variables are specified at centroid of the cell forming grid. Then, the differential form of governing equations are integrated over each and every control volume and then interpolation profiles are plotted which are assumed to describe the variation of concerned variable between the centroids of control volume. The schemes used for interpolation are upwind differencing, central differencing, quadratic upwind differencing and power law differencing schemes. The resulting equations thus obtained are called discretization or discretized equations. These discretized equations satisfy the conservation principle of different variables like mass, momentum and energy.

3.4. Governing equations for Pulse Tube Refrigerator

For present analysis, various components of pulse tube refrigerator are dual opposed piston-cylinder arrangement, a transfer line, a regenerator, CHX (cold end heat exchanger), pulse tube, HHX (hot end heat exchanger), orifice valve and a reservoir. Continuum based conservation equations are applied at the components of PTR system except heat exchangers and regenerator, as these are considered as porous zones. General governing equations are:

Conservation of mass:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho \vec{v}) = S$$

Where,

$\nabla =$ Gradient operator

$\rho =$ Density of gas

$\vec{v} =$ Velocity of gas (in vector form)

$t =$ Time

$S =$ Source term (in this analysis source term is taken as zero)

Conservation of momentum:

$$\frac{\partial}{\partial t}\left(\rho \vec{v}\right) + \nabla.(\rho \vec{v}\vec{v}) = -\nabla p + \nabla.\left(\tau\right) + \rho \vec{g} + \vec{F}_E$$

Where

$p =$ Static pressure

$\tau =$ Stress tensor

$\vec{g} =$ Acceleration due to gravity

$\vec{F}_E =$ Source term or external body forces.

Conservation of energy:

$$\frac{\partial}{\partial t}\left(\rho E\right) + \nabla.(\vec{v}(p + \rho E)) = \nabla.(k_{eff} \nabla T - \sum_j \psi_j h_j + (\tau.\vec{v})) + S_E$$

Where

$$E = h + \frac{v^2}{2} - \frac{p}{\rho}$$

$$h = -\int_{T}^{T_1} C_p dT$$

$$k_{eff} = k_t + k$$

Where

$k =$ Thermal conductivity of gas

$k_t =$ Thermal conductivity due to turbulence

$C_p =$ Specific heat of gas

$h =$ Local enthalpy

$T =$ Temperature of gas

$v =$ Local velocity

$\psi_j =$ Diffusion flux

$S_E =$ Source term (source term is taken as zero for this analysis)

Governing equations for PTR in 2-dimensional cylindrical polar co-ordinate system are given below:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r) + \frac{\partial}{\partial x}(\rho v_x) = 0$$

Where

$r =$ Radial co-ordinate

$x =$ Axial co-ordinate

$v_r =$ Radial direction of velocity

$v_x =$ Axial direction of velocity

Momentum equations:

In axial direction,

$$\frac{\partial}{\partial t}(\rho v_x) + \frac{1}{r}\frac{\partial}{\partial x}(r\rho v_x v_x) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho v_r v_x) = -\frac{\partial p}{\partial x}$$
$$+\frac{1}{r}\frac{\partial}{\partial x}\left[\mu r\left\{2\frac{\partial v_x}{\partial x} - \frac{2}{3}(\nabla.\vec{v})\right\}\right] + \frac{1}{r}\frac{\partial}{\partial r}\left[\mu r\left(\frac{\partial v_x}{\partial r} + \frac{\partial v_r}{\partial x}\right)\right]$$

In radial direction,

$$\frac{\partial}{\partial t}(\rho v_r)+\frac{1}{r}\frac{\partial}{\partial x}(\rho r v_r v_x)+\frac{1}{r}\frac{\partial}{\partial r}(\rho r v_r v_x)=-\frac{\partial p}{\partial r}+\frac{1}{r}\frac{\partial}{\partial r}\left[\mu r\left\{2\frac{\partial v_r}{\partial r}-\frac{2}{3}(\nabla.\vec{v})\right\}\right]$$
$$+\frac{1}{r}\frac{\partial}{\partial x}\left[\mu r\left(\frac{\partial v_x}{\partial r}+\frac{\partial v_r}{\partial x}\right)\right]-\frac{2\mu}{r^2}v_r+\frac{2\mu}{3r}(\nabla.\vec{v})$$

Where,

$$\nabla.\vec{v}=\frac{\partial v_r}{\partial r}+\frac{\partial v_x}{\partial x}+\frac{v_r}{r}$$

$v_r =$ Velocity in radial co-ordinate

$v_x =$ Velocity in axial co-ordinate

$\mu =$ Fluid viscosity (in *kg/m-s*)

It can be noted that, in the above momentum conservation equation, body forces, gravity forces and other external forces are neglected.

Energy Equation:

$$\frac{\partial}{\partial t}(\rho E)+\nabla.\left(\vec{v}(\rho E+p)\right)=\nabla.\left(k_{eff}\nabla T+(\tau.\vec{v})\right)$$

CHAPTER IV

SIMULATION AND RESULT DISCUSSION

4.1. Defining the Model

Before starting iteration using ANSYS FLUENT, the properties of the model like materials, fluent solver, fluid properties, thermal properties, operating conditions and also the boundary conditions have to be defined for solving the problems of fluid flow. The various conditions that are to be specified are furnished below.

- **Solver**

  Segregated solver, 2D planar space, implicit formulation and unsteady time.
- **Material**

  Working fluid is ideal gas (Helium) and the material for components of DIPTR is taken stainless steel.
- **Energy equation and Viscous model**

  Energy equation and k-epsilon set of equations are considered.
- **Operating Condition**

  The operating condition is chosen for simulation is atmospheric pressure i.e. 1.01325 bar.
- **Boundary Conditions**

  The boundary conditions for various components of DIPTR are same except the inlet condition. At inlet, for oscillation of piston four pressure UDFs are written of different wave forms. The different wave forms of pressure UDFs are sinusoidal, rectangular, triangular and trapezoidal.

Different Cases Components	Case 1	Case 2	Case 3	Case 4
Inlet	UDF (Sinusoidal Pressure wave)	UDF (Rectangular pressure wave)	UDF (Triangular pressure wave)	UDF (Trapezoidal pressure wave)
Transfer line wall	300 K	300 K	300 K	300 K
After cooler wall	293 K	293 K	293 K	293 K
Regenerator wall	Adiabatic	Adiabatic	Adiabatic	Adiabatic
Cold end heat exchanger wall	Adiabatic	Adiabatic	Adiabatic	Adiabatic
Pulse tube wall	Adiabatic	Adiabatic	Adiabatic	Adiabatic
Hot end heat exchanger wall	293 K	293 K	293 K	293 K
Orifice valve wall	300 K	300 K	300 K	300 K
Reservoir wall	300 K	300 K	300 K	300 K
Viscous Resistance (m^{-2})	4.15*10^(+9)	4.15*10^(+9)	4.15*10^(+9)	4.15*10^(+9)
Inertial resistance (m^{-1})	12140	12140	12140	12140
Initial condition	300 K	300 K	300 K	300 K
Cold end load	0 W	0 W	0 W	0 W
Cold end temperature*	134 K	128 K	111 K	128 K

Table 4.1. Boundary conditions and Initial conditions for various components of DIPTR

* Results obtained from simulation.

- **Limits**

  Pressure: 10 bar to 20 bar (Same for all four UDFs)

- **Relaxation factors**

  Momentum: 0.1, pressure: 0.1, energy: 0.1.

- **Discretization technique**

  Pressure- velocity coupling: PISO

Pressure: PRESTO
Density: First Order Upwind
Momentum: First Order Upwind
Energy: First Order Upwind

- **Convergence criteria**

Continuity: 0.001
X-velocity: 0.001
Y-velocity: 0.001
Energy: 1e-06
K: 0.001
Epsilon: 0.001

Defining material properties

Here, in this simulation, the working fluid is taken as Helium and for solid stainless steel is taken. Helium is taken as an ideal gas. The various properties specified are density, thermal conductivity, specific heat, diffusivity and viscosity.

Defining Porous Zone

The cold end heat exchanger, hot end heat exchanger, after cooler and regenerator are defined as porous zone. To define porous zone, various parameters required are viscous resistance, inertial resistance and porosity and the respective values are: 4.15e+09, 12140 and 0.6.

4.2. Simulation for Different Boundary Conditions

Case-1. Pressure UDF (Sinusoidal wave form)

Pressure UDF having sinusoidal wave form is as below:

```
#include "udf.h"
DEFINE_PROFILE (unsteady_pressure, thread, position)
{
Face_t f;
real t = CURRENT_TIME;
begin_f_loop (f, thread)
{
F_PROFILE (f, thread, position) = 101325.0*(10.0 + 5.0 * sin (12.56 * t));
}
end_f_loop (f, thread)
}
```

This UDF is compiled in the FLUENT software and applied as boundary condition at inlet and then simulation is carried out.

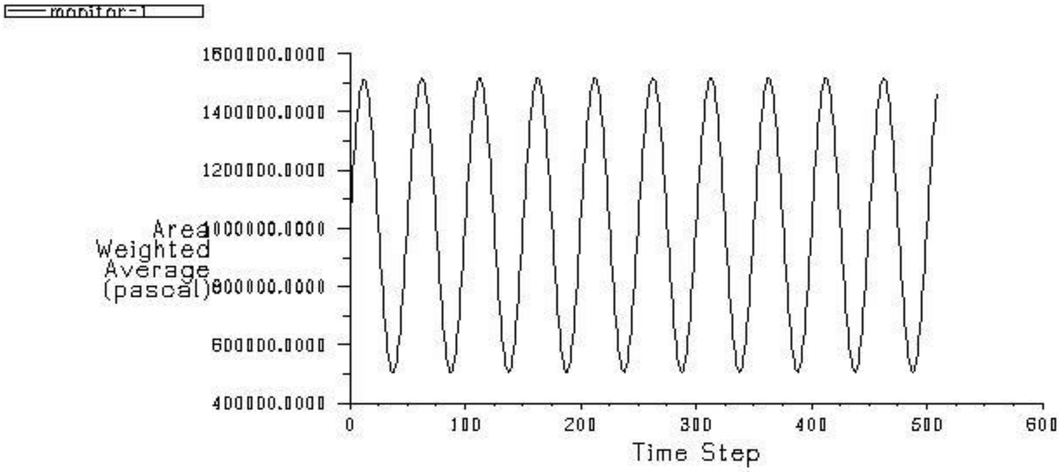


Fig 4.1. Pressure wave generated at inlet of DIPTR for Case - 1.

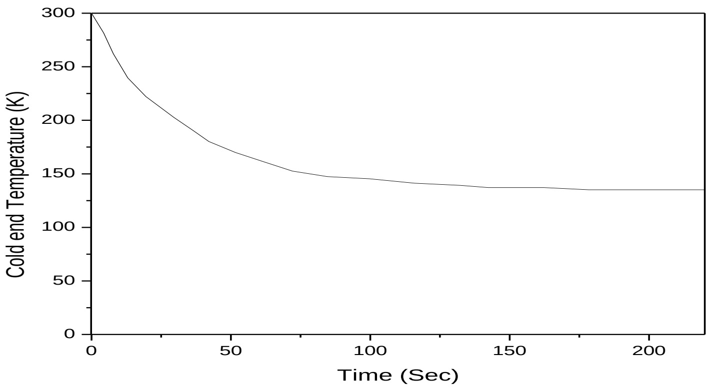


Fig 4.2. Temperature decrement behavior of DIPTR for Case - 1.

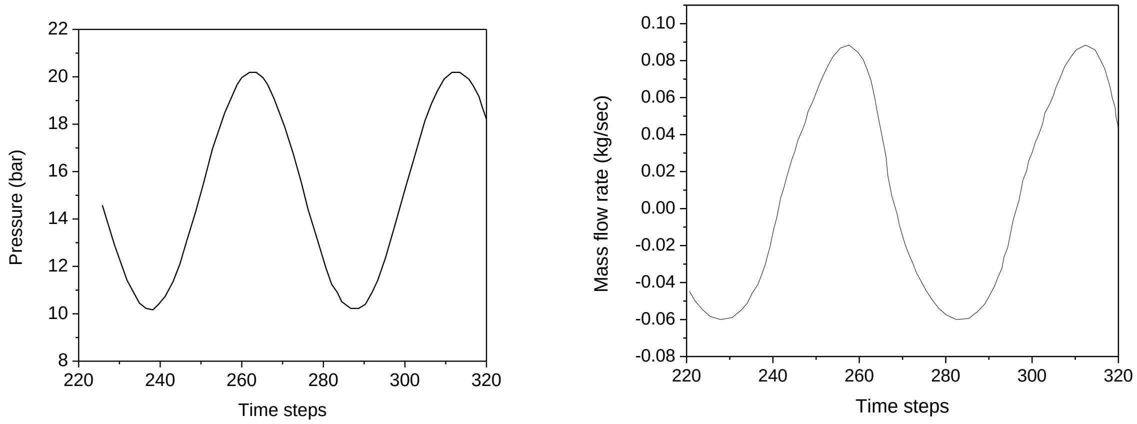


Fig 4.3. Phase relationship between Mass flow rate and Pressure at inlet of DIPTR.

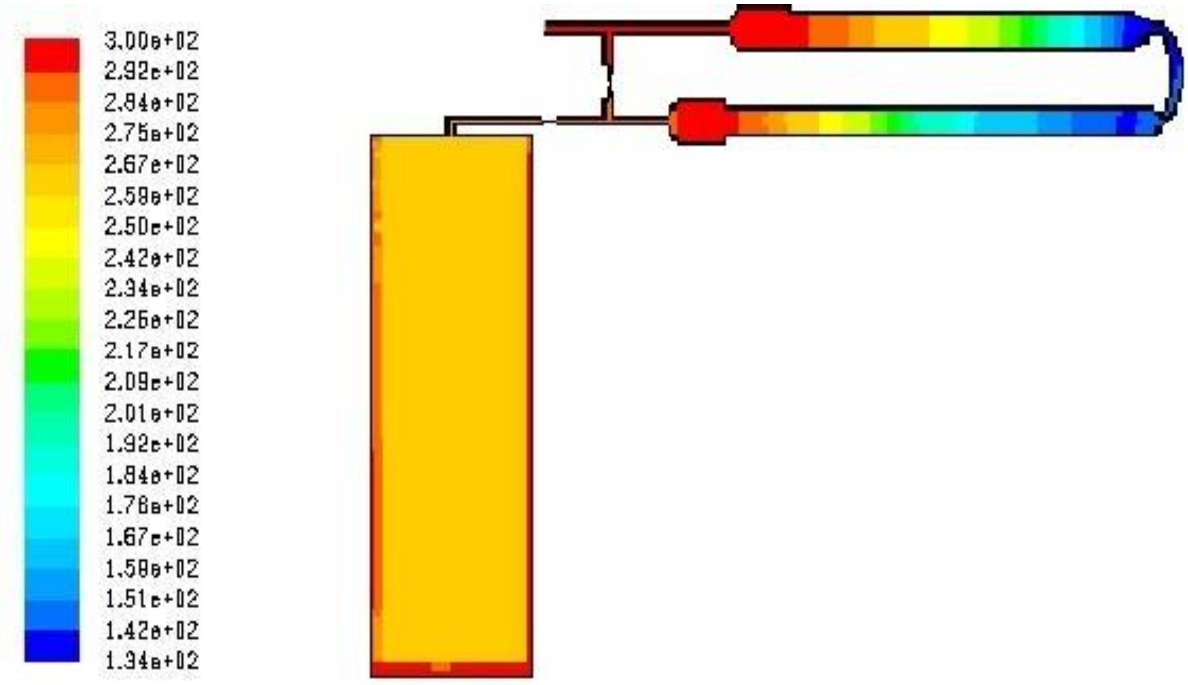


Fig 4.4. Temperature contour of DIPTR for Case-1.

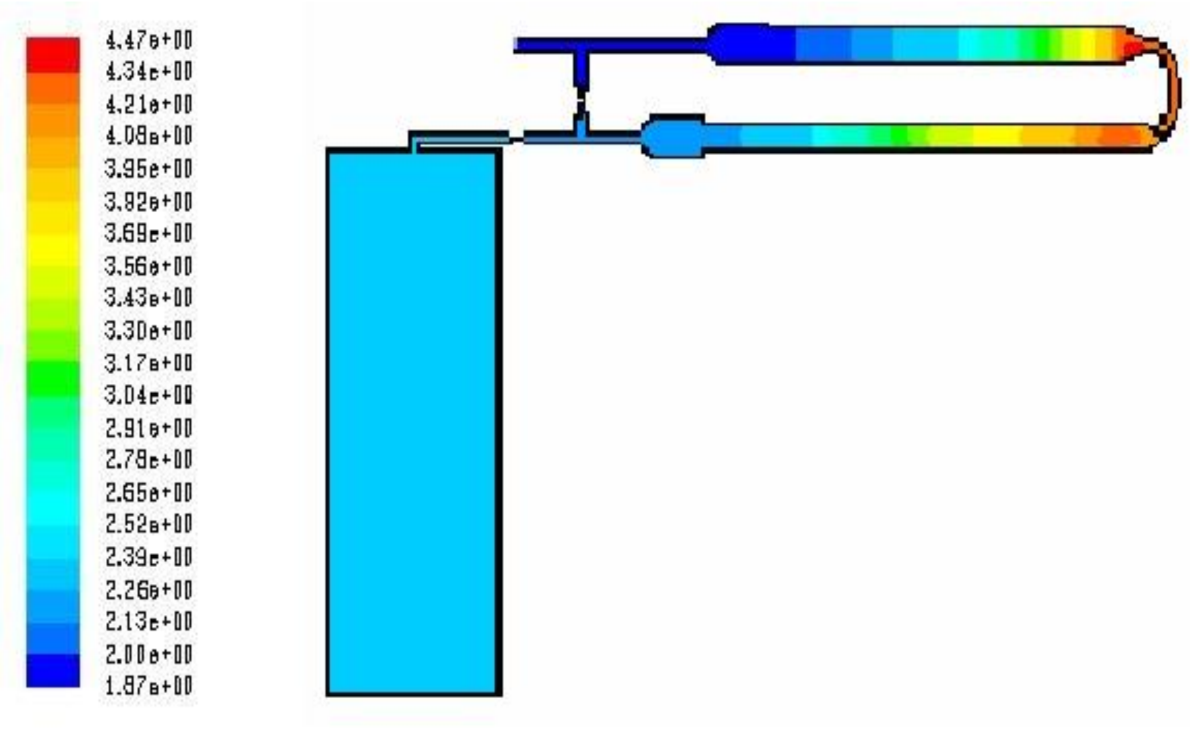


Fig 4.5. Density contour of DIPTR for Case-1.

Case-2. Pressure UDF (Rectangular wave form)

```
#include “udf.h”
DEFINE_PROFILE (unsteady_pressure, thread, position)
{
face_t f;
real t = CURRENT_TIME;
int i;
real p = 0;
begin_f_loop (f, thread)
{
for (i = 0; i <100; i++)
{
if (t < (p + 0.0025))
F_PROFILE (f, thread, position) = 20;
else if (t = = (p + 0.0025))
F_PROFILE (f, thread, position) = 15;
else if (t < (p + 0.005))
F_PROFILE (f, thread, position) = 10;
else if (t = = (p+ 0.005))
F_PROFILE (f, thread, position) = 15;
else
p = p + 0.005;
}
}
end_f_loop (f, thread)
}
```

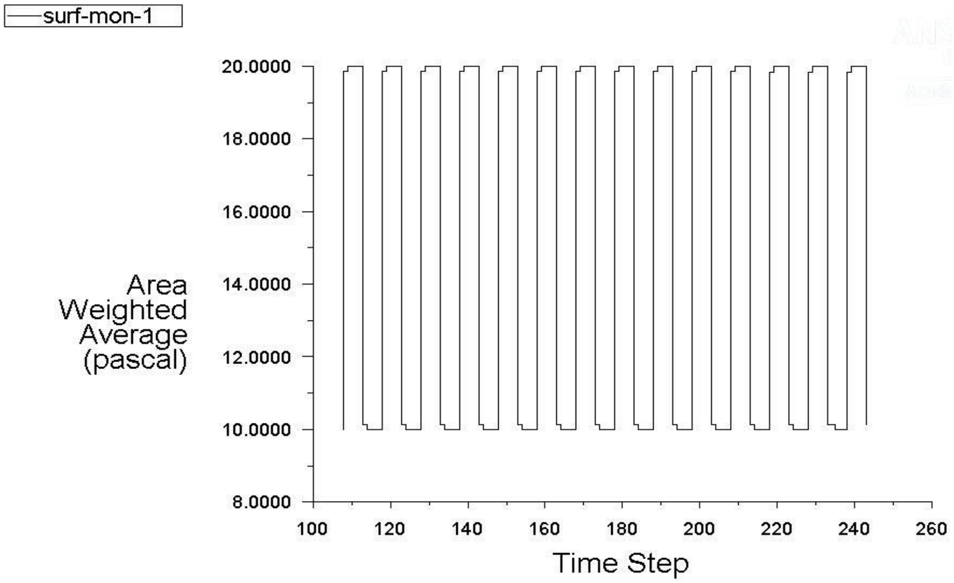


Fig. 4.6. Pressure wave generated at inlet of DIPTR for Case – 2.

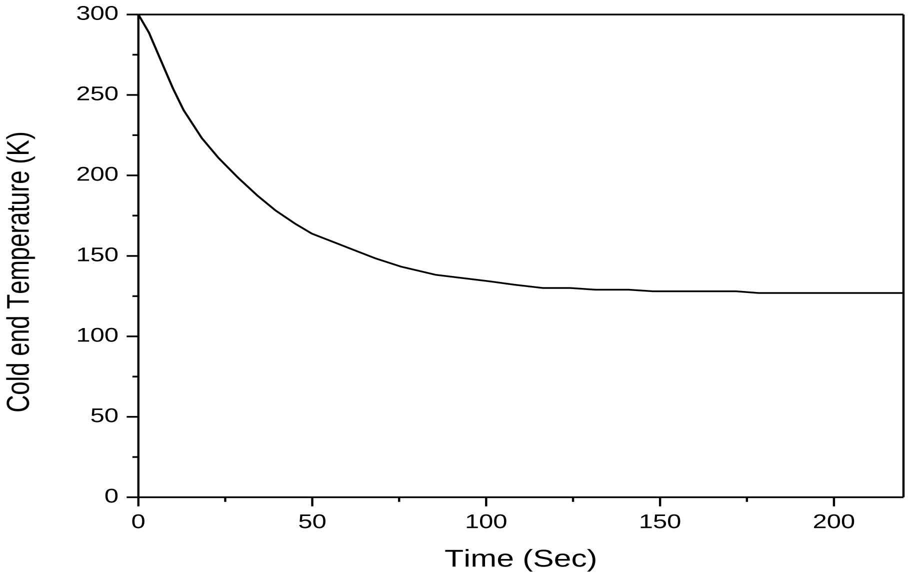


Fig 4.7. Temperature decrement behavior of DIPTR for Case - 2.

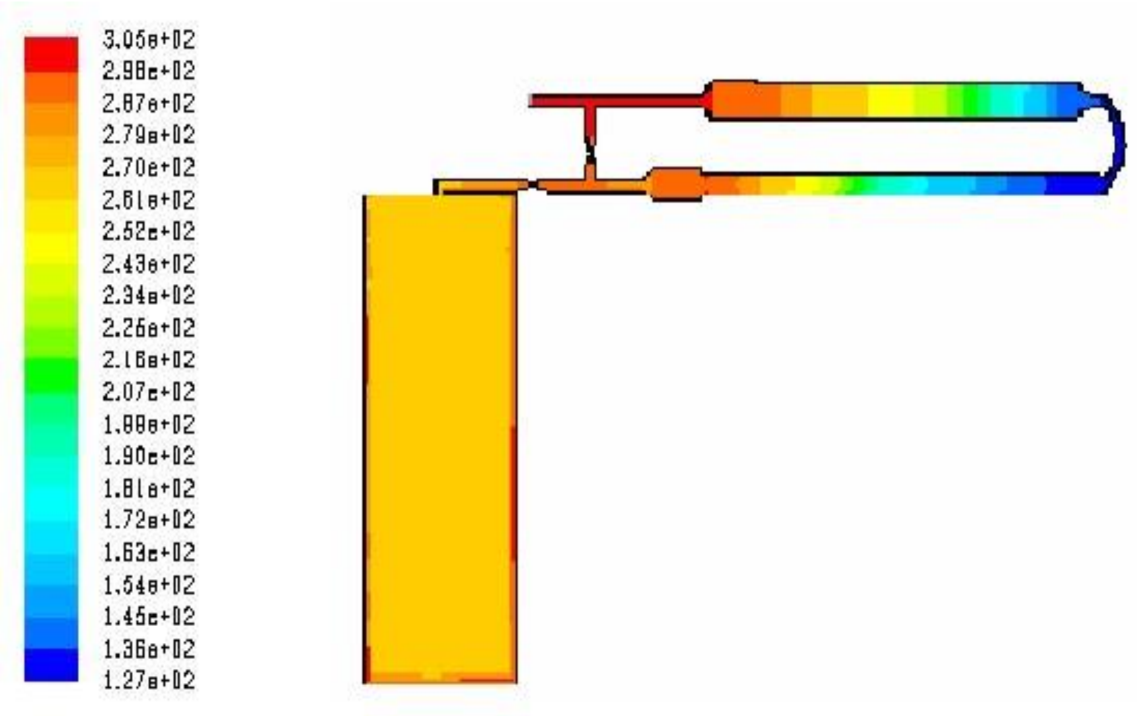


Fig. 4.8. Temperature contour of DIPTR for Case-2.

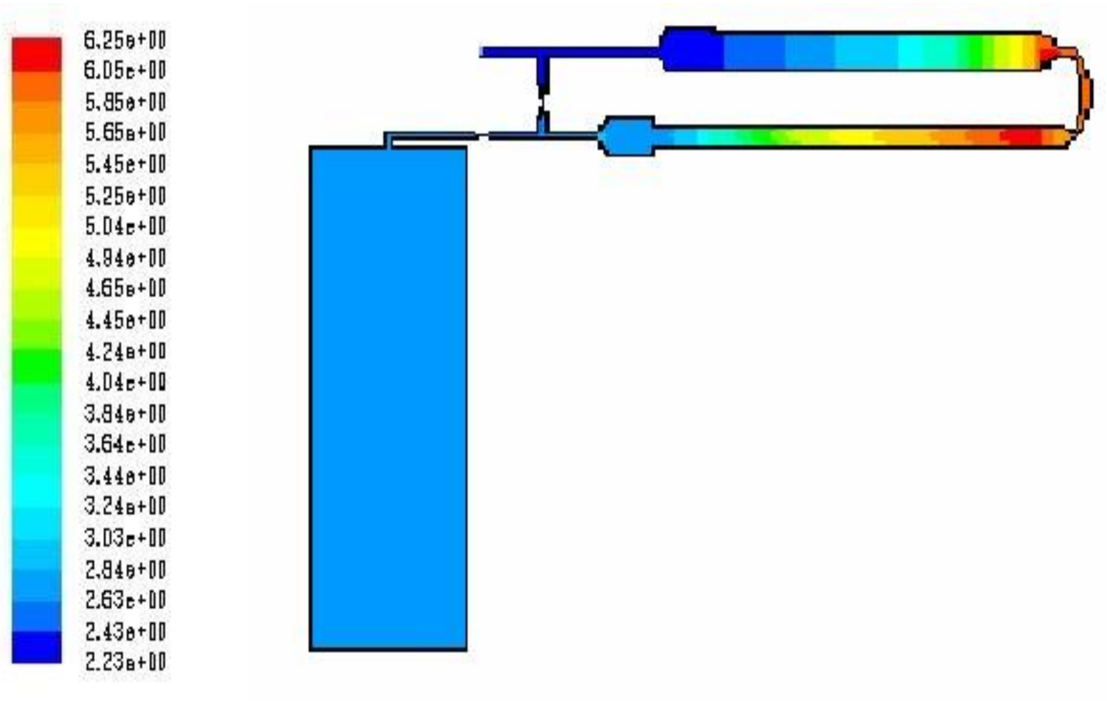


Fig 4.9. Density contour of DIPTR for Case-2.

Case-3. Pressure UDF (Triangular wave form)

```
#include “udf.h”
DEFINE_PROFILE (unsteady_pressure, thread, position)
{
face_t f;
real t = CURRENT_TIME;
int i;
real p = 0;
begin_f_loop (f, thread)
{
for (i = 0; i <100; i++)
{
if (t< (p + 0.00125))
F_PROFILE (f, thread, position) = 15 + 4000* (t-p);
else if (t < (p + 0.00375))
F_PROFILE (f, thread, position) = 20 – 4000 * (t - 0.00375 - p);
else if (t < (p + 0.005))
F_PROFILE (f, thread, position) = 10 + 4000 * (t – 0.00375 - p);
else
p = p + 0.005;
}
}
end_f_loop (f, thread)
}
```

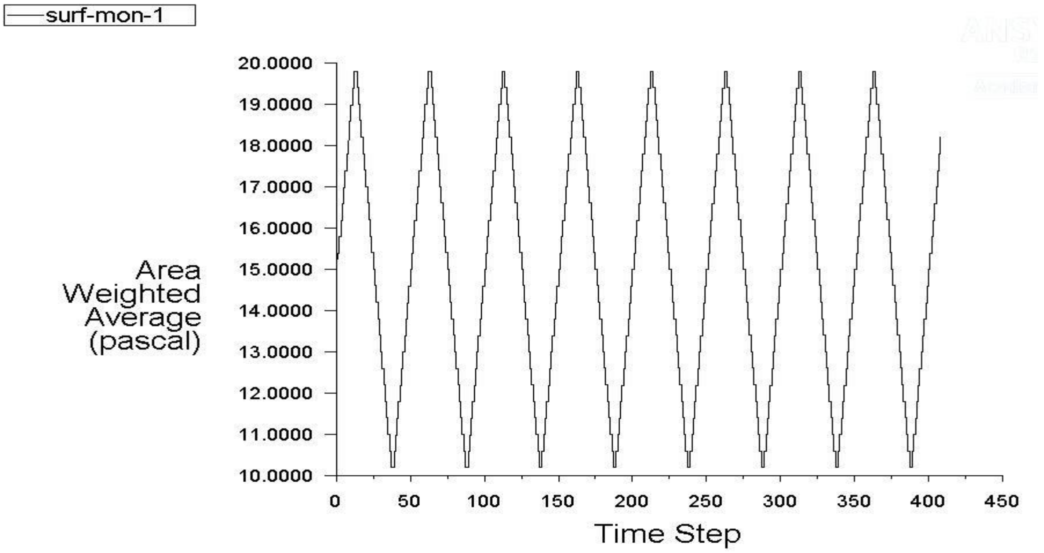


Fig 4.10. Pressure wave generated at inlet of DIPTR for Case - 3.

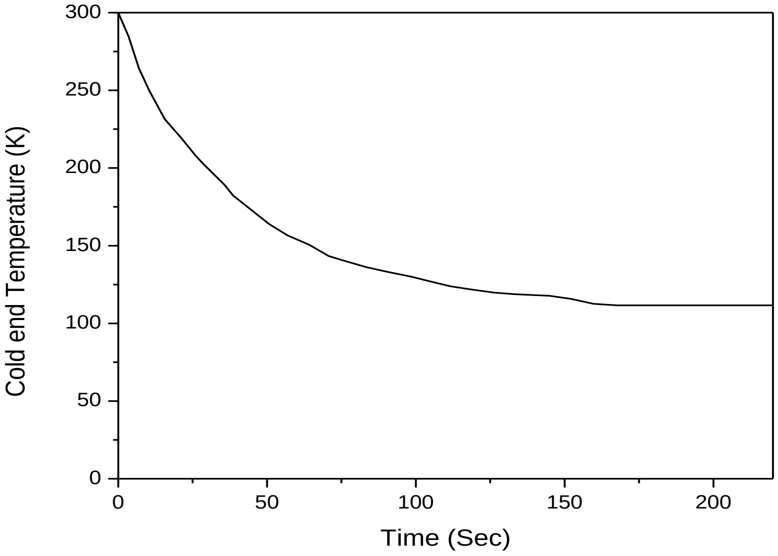


Fig 4.11. Temperature decrement behavior of DIPTR for Case – 3.

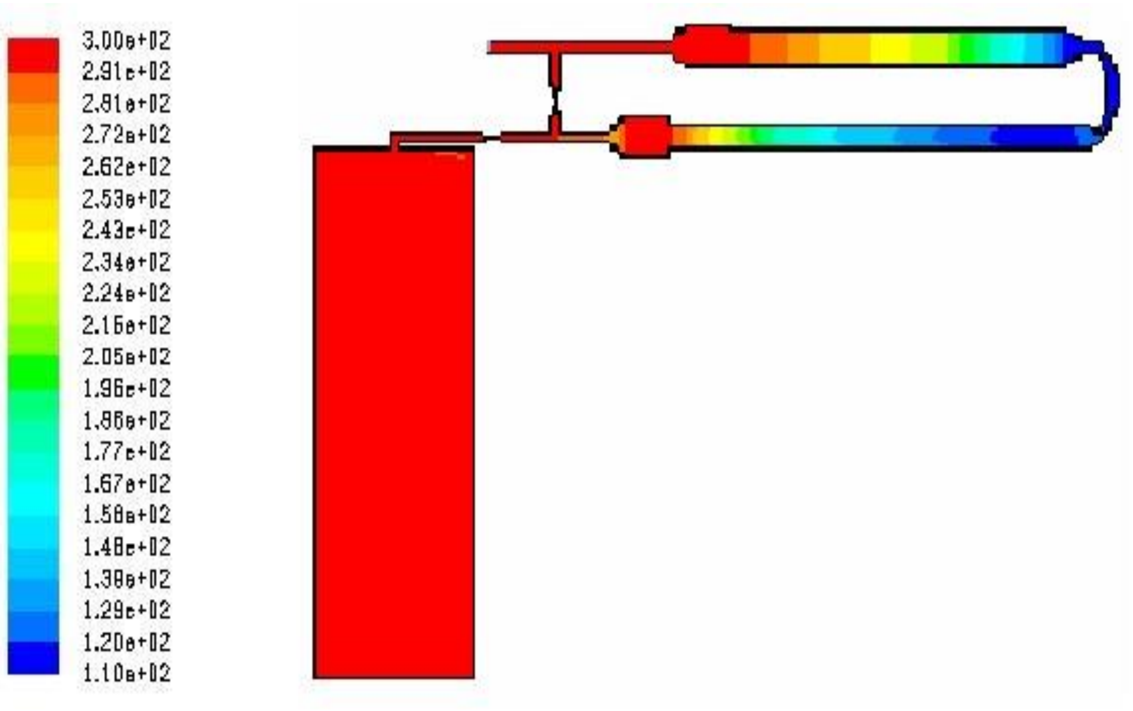


Fig 4.12. Temperature contour of DIPTR for Case - 3.

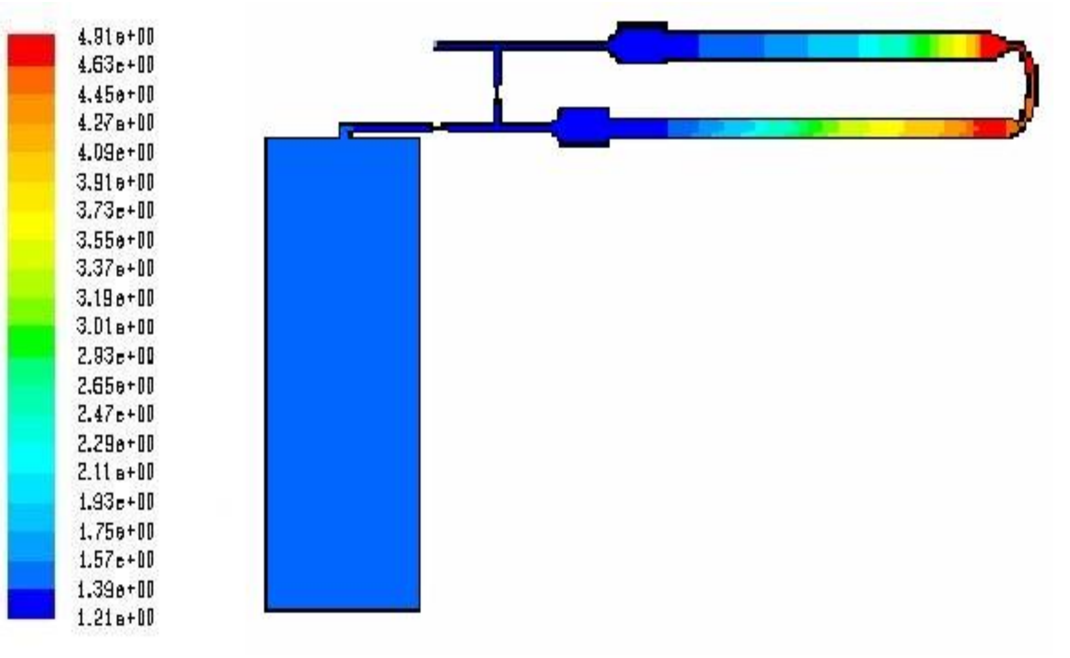


Fig 4.13. Density contour of DIPTR for Case -3.

Case-4. Pressure UDF (Trapezoidal wave form)

```
#include “udf.h”
DEFINE_PROFILE (unsteady_pressure, thread, position)
{
face_t f;
real t = CURRENT_TIME;
int i;
real p = 0;
begin_f_loop (f, thread)
{
for (i = 0; i <100; i++)
{
if (t < (p + 0.001))
F_PROFILE (f, thread, position) = 15 + 5000 * (t - p);
else if (t < (p + 0.0015))
F_PROFILE (f, thread, position) = 20;
else if (t < (p + 0.0035))
F_PROFILE (f, thread, position) = 20 – 5000 * (t – 0.0015 - p);
else if (t < (p + 0.004))
F_PROFILE (f, thread, position) = 10;
else if (t < (p + 0.005))
F_PROFILE (f, thread, position) = 10 + 5000 * (t – 0.004 - p);
else
p = p + 0.005;
}
}
end_f_loop (f, thread)
}
```

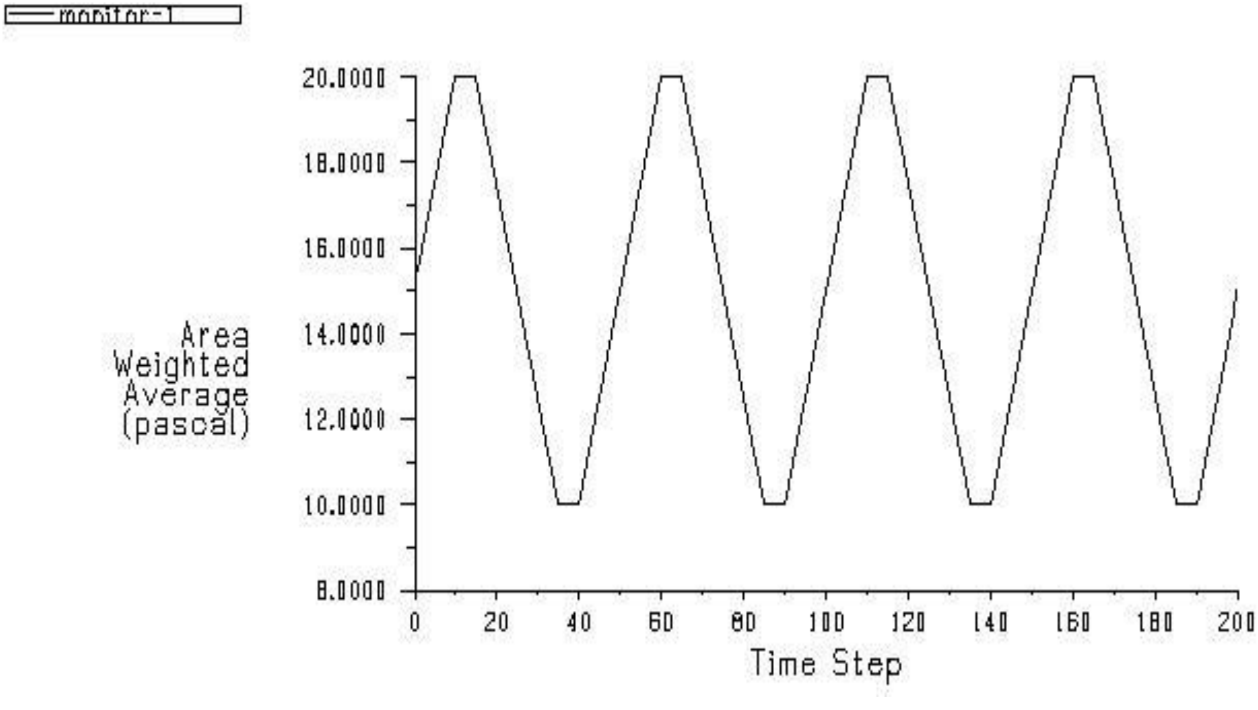


Fig 4.14. Pressure wave generated at inlet of DIPTR for Case - 4.

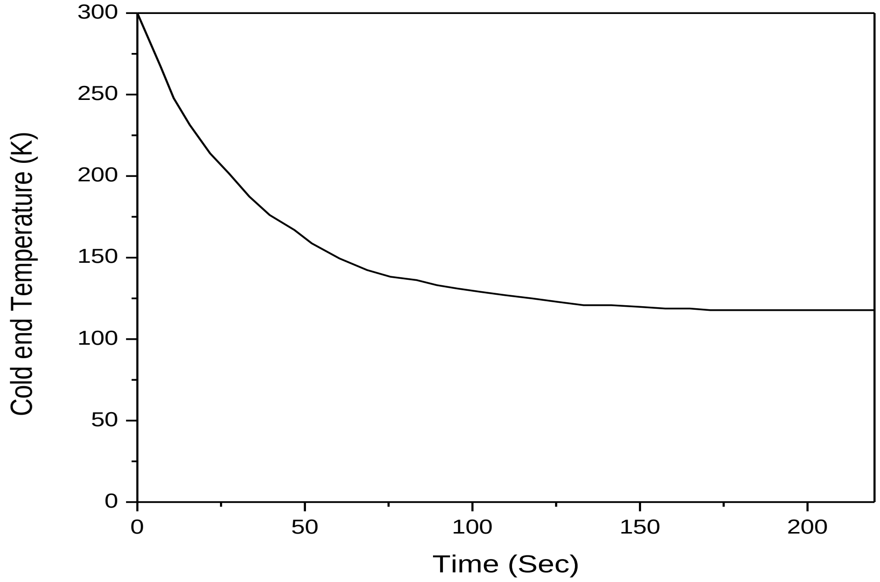


Fig 4.15. Temperature variation in CHX of DIPTR for Case - 4.

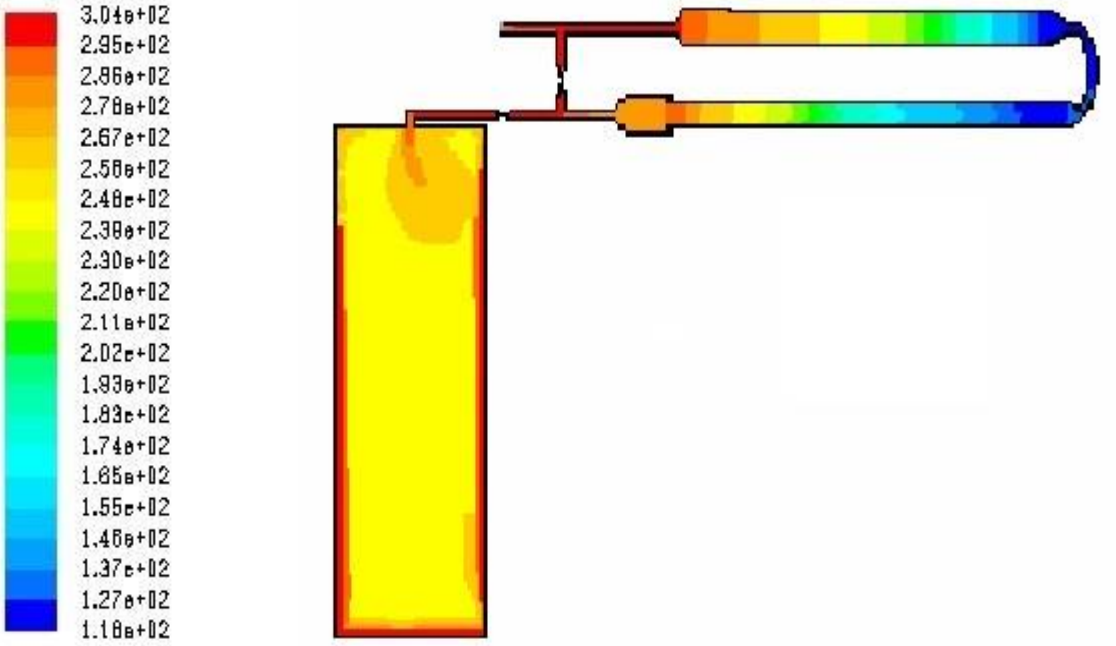


Fig 4.16. Temperature contour of DIPTR for Case - 4.

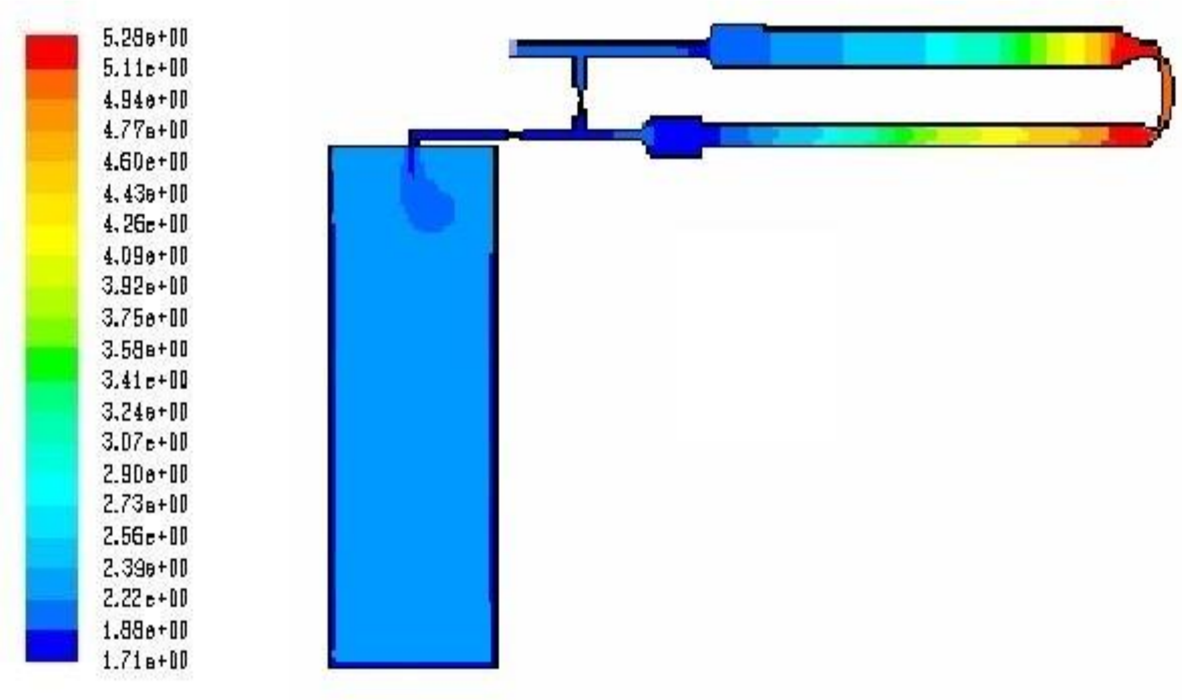


Fig 4.17. Density contour of DIPTR for Case – 4.

CHAPTER V

CONCLUSION AND FUTURE WORK

5.1. Conclusion

Simulation is carried out for DIPTR using the ANSYS FLUENT 15.0 software. All the four cases having different pressure UDFs have been considered and the cold end temperature is determined for all the four cases. From simulation result, temperature obtained at cold end of DIPTR are 134 K, 128 K, 111 K and 118 K for Sinusoidal, Rectangular, Triangular and Trapezoidal wave forms respectively.

From the above simulation result, it can be concluded that the temperature obtained using triangular pressure UDF i.e. 111 K is lower than that of the other cases. So triangular wave form of pressure is much better than all other wave forms.

5.2. Future Work

In this work, no load (0 W) condition is specified for cold end heat exchanger (CHX) of DIPTR in case of all the four pressure UDFs. Simulation can be done by specifying other boundary conditions like giving some load at cold end of DIPTR to know which UDF is more preferable.

REFERENCES

1. Matsubara, Y. "Future trend of pulse tube cryocooler research." *Proceedings of the twentieth international cryogenic engineering conference, Beijing, China*. 2004.
2. Radebaugh, Ray. "Development of the pulse tube refrigerator as an efficient and reliable cryocooler." *Proc. institute of refrigeration, London* (2000).
3. Gifford, William E., and R. C. Longsworth. "Pulse-tube refrigeration." *Journal of Manufacturing Science and Engineering* 86.3 (1964): 264-268.
4. Gifford, W. Ev, and R. C. Longsworth. "Surface heat pumping." *Advances in cryogenic engineering*. Springer US, 1966. 171-179.
5. Gifford, W.E. and Longsworth, R.C. Pulse tube refrigeration progress, Advances in cryogenic engineering 3B (1964), pp.69-79.
6. Gifford, W. E., and G. H. Kyanka. "Reversible pulse tube refrigeration."*Advances in cryogenic engineering*. Springer US, 1967. 619-630.
7. De Boer, P. C. T. "Thermodynamic analysis of the basic pulse-tube refrigerator." *Cryogenics* 34.9 (1994): 699-711.
8. De Boer, P. C. T. "Analysis of basic pulse-tube refrigerator with regenerator."*Cryogenics* 35.9 (1995): 547-553.
9. Jeong, Eun Soo. "Secondary flow in basic pulse tube refrigerators." *Cryogenics*36.5 (1996): 317-323.
10. Longsworth, R. C. "An experimental investigation of pulse tube refrigeration heat pumping rates." *Advances in Cryogenic Engineering*. Springer US, 1967. 608-618.
11. Flake, Barrett, and Arsalan Razani. "Modeling pulse tube cryocoolers with CFD." *Advances in cryogenic engineering: Transactions of the Cryogenic Engineering Conference-CEC*. Vol. 710. No. 1. AIP Publishing, 2004.
12. Jiao, Anjun, Sangkwon Jeong, and H. B. Ma. "Heat transfer characteristics of cryogenic helium gas through a miniature tube with a large temperature difference." *Cryogenics* 44.12 (2004): 859-866.
13. Ashwin, T. R., G. S. V. L. Narasimham, and Subhash Jacob. "CFD analysis of high frequency miniature pulse tube refrigerators for space applications with thermal non-equilibrium model." *Applied Thermal Engineering* 30.2 (2010): 152-166.

14. Wu, P. Y., and S. W. Zhu. "Mechanism and numerical analysis of orifice pulse tube refrigerator with a valveless compressor." *Cryogenics and Refrigeration—Proceedings of International Conference, International Academic Publishers*. 1989.
15. Richardson, R. N. "Valved pulse tube refrigerator development." *Cryogenics*29.8 (1989): 850-853.
16. Lee, J. M., and H. R. Dill. "The influence of gas velocity on surface heat pumping for the orifice pulse tube refrigerator." *Advances in Cryogenic Engineering. Vol. 35B-Proceedings of the 1989 Cryogenic Engineering Conference*. Vol. 35. 1990.
17. Kasuya, M., J. Yuyama, Q. Geng, and E. Goto. "Optimum phase angle between pressure and gas displacement oscillations in a pulse-tube refrigerator." *Cryogenics* 32, no. 3 (1992): 303-308.
18. Wang, Chao, Peiyi Wu, and Zhongqi Chen. "Numerical modelling of an orifice pulse tube refrigerator." *Cryogenics* 32.9 (1992): 785-790.
19. Kittel, P. "Ideal orifice pulse tube refrigerator performance." *Cryogenics* 32.9 (1992): 843-844.
20. Storch, Peter J., and Ray Radebaugh. "Development and experimental test of an analytical model of the orifice pulse tube refrigerator." *Advances in cryogenic engineering*. Springer US, 1988. 851-859.
21. Zhu, S. W., and Z. Q. Chen. "Isothermal model of pulse tube refrigerator."*Cryogenics* 34.7 (1994): 591-595.
22. Rawlins, W., R. Radebaugh, P. E. Bradley, and K. D. Timmerhaus. "Energy flows in an orifice pulse tube refrigerator." In *Advances in cryogenic engineering*, pp. 1449-1456. Springer US, 1994.
23. Kittel, P., A. Kashani, J. M. Lee, and P. R. Roach. "General pulse tube theory."*Cryogenics* 36, no. 10 (1996): 849-857.
24. Huang, B. J., and M. D. Chuang. "System design of orifice pulse-tube refrigerator using linear flow network analysis." *Cryogenics* 36.11 (1996): 889-902.
25. De Waele, A. T. A. M., Peter Paul Steijaert, and J. J. Koning. "Thermodynamical aspects of pulse tubes II." *Cryogenics* 38.3 (1998): 329-335.
26. Gerster, J., M. Thürk, L. Reißig, and P. Seidel. "Hot end loss at pulse tube refrigerators." *Cryogenics* 38, no. 6 (1998): 679-682.

27. de Boer, P. C. T. "Optimization of the orifice pulse tube." *Cryogenics* 40.11 (2000): 701-711.
28. Roach R., A Simple Modeling Program for Orifice Pulse Tube Coolers NASA Ames Research Center (2004).
29. Mikulin, E. I., A. A. Tarasov, and M. P. Shkrebyonock. "Low-temperature expansion pulse tubes." *Advances in cryogenic engineering*. Springer US, 1984. 629-637.
30. Radebaugh, Ray, James Zimmerman, David R. Smith, and Beverly Louie. "A comparison of three types of pulse tube refrigerators: new methods for reaching 60K." In *Advances in Cryogenic Engineering*, pp. 779-789. Springer US, 1986.
31. Liang, Jingtao, Yuan Zhou, and Wenxiu Zhu. "Development of a single-stage pulse tube refrigerator capable of reaching 49 K." *Cryogenics* 30.1 (1990): 49-51.
32. Zhang, X. B., L. M. Qiu, Z. H. Gan, and Y. L. He. "CFD study of a simple orifice pulse tube cooler." *Cryogenics* 47, no. 5 (2007): 315-321.
33. Antao, Dion Savio, and Bakhtier Farouk. "Computational fluid dynamics simulations of an orifice type pulse tube refrigerator: Effects of operating frequency." *Cryogenics* 51.4 (2011): 192-201.
34. Gardner, D. L., and G. W. Swift. "Use of inertance in orifice pulse tube refrigerators." *Cryogenics* 37.2 (1997): 117-121.
35. Roach, Pat R., and Ali Kashani. "Pulse tube coolers with an inertance tube: theory, modeling, and practice." *Advances in cryogenic engineering*. Springer US, 1998. 1895-1902.
36. Zhu, Shaowei, and Yoichi Matsubara. "Numerical method of inertance tube pulse tube refrigerator." *Cryogenics* 44.9 (2004): 649-660.
37. Dai, Wei, Jianying Hu, and Ercang Luo. "Comparison of two different ways of using inertance tube in a pulse tube cooler." *Cryogenics* 46.4 (2006): 273-277.
38. Cha, J. S., S. M. Ghiaasiaan, P. V. Desai, J. P. Harvey, and C. S. Kirkconnell. "Multi-dimensional flow effects in pulse tube refrigerators." *Cryogenics* 46, no. 9 (2006): 658-665.
39. Banjare, Y. P., R. K. Sahoo, and S. K. Sarangi. "CFD Simulation of Orifice Pulse Tube Refrigerator." (2007).

40. Banjare Y.P., Sahoo R.K., Sarangi S. K.”CFD Simulation of Inertance tube Pulse Tube Refrigerator. 19th National and 8th ISHMT-ASME Heat and Mass Transfer Conference JNTU College of Engineering Hyderabad, India January 3-5, 2008. Paper No. EXM-7, PP34.
41. Rout S.K., Mukare R., Choudhury B.K., Sahoo R.K., Sarangi S.K. “ CFD simulation to optimize single stage pulse tube refrigerator temperature below 60K”. International conference on modeling, optimization and computing (ICMOC 2012). April 10-11, 2012. Procedia Engineering 38 (2012), 1524-1530.
42. Rout, S. K., A. K. Gupta, B. K. Choudhury, R. K. Sahoo, and S. K. Sarangi. "Influence of porosity on the performance of a pulse tube refrigerator: a CFD study." *Procedia Engineering* 51 (2013): 609-616.
43. Shaowei, Zhu, Wu Peiyi, and Chen Zhongqi. "Double inlet pulse tube refrigerators: an important improvement." *Cryogenics* 30.6 (1990): 514-520.
44. Ju, Y. L., C. Wang, and Y. Zhou. "Numerical simulation and experimental verification of the oscillating flow in pulse tube refrigerator." *Cryogenics* 38.2 (1998): 169-176.
45. Wang, C., P. Y. Wu, and Z. Q. Chen. "Numerical analysis of double-inlet pulse tube refrigerator." *Cryogenics* 33.5 (1993): 526-530.
46. Mirels, Harold. "Double inlet pulse tube cryocooler with stepped piston compressor." *Advances in cryogenic engineering*. Springer US, 1994. 1425-1431.
47. Zhu, Shaowei, Shin Kawano, Masahumi Nogawa, and Tatsuo Inoue. "Work loss in double-inlet pulse tube refrigerators." *Cryogenics* 38, no. 8 (1998): 803-807.
48. Kirkconnell, Carl Scott. "Numerical analysis of the mass flow and thermal behavior in high-frequency pulse tubes." (1995).
49. Zhu, Shaowei, and Zhongqi Chen. "Enthalpy flow rate of a pulse tube in pulse tube refrigerator." *Cryogenics* 38.12 (1998): 1213-1216.
50. Wang, C., G. Thummes, and C. Heiden. "Control of DC gas flow in a single-stage double-inlet pulse tube cooler." *Cryogenics* 38.8 (1998): 843-847.
51. Yuan, J., and J. M. Pfotenhauer. "Thermodynamic analysis of active valve pulse tube refrigerators." *Cryogenics* 39.4 (1999): 283-292.
52. Xu, M. Y., A. T. A. M. De Waele, and Y. L. Ju. "A pulse tube refrigerator below 2 K." *Cryogenics* 39.10 (1999): 865-869.

53. de Boer, P. C. T. "Characteristics of the double inlet pulse tube." *Cryogenics*43.7 (2003): 379-391.
54. Chokhawala, M. D., K. P. Desai, H. B. Naik, and K. G. Naryankhedkar. "Phasor analysis for double inlet pulse tube cryocooler." *Advances in cryogenic engineering* 45, no. A (2000): 159-166.
55. Banjare Y.P. , Sahoo R.K., Sarangi S.K. Sarangi "Numerical analysis of double inlet pulse tube refrigerator" National conference on RECT, GGDU Bilaspur(C.G.) Chhattisgarh, India ,2007.
56. Zhang, X. B., L. M. Qiu, Z. H. Gan, and Y. L. He. "Effects of reservoir volume on performance of pulse tube cooler." *International journal of refrigeration* 30, no. 1 (2007): 11-18.
57. Razani, A., C. Dodson, B. Flake, and T. Roberts. "The Effect of Phase-shifting Mechanisms on the Energy and Exergy Flow in Pulse Tube Refrigerators." In *Advances in cryogenic engineering: Transactions of the Cryogenic Engineering Conference-CEC*, vol. 823, no. 1, pp. 1572-1579. AIP Publishing, 2006.
58. Ju, Y. L. "Thermodynamic analysis of GM-type pulse tube coolers." *Cryogenics*41.7 (2001): 513-520.
59. Cai, J. H., Y. Zhou, J. J. Wang, and W. X. Zhu. "Experimental analysis of double-inlet principle in pulse tube refrigerators." *Cryogenics* 33, no. 5 (1993): 522-525.
60. Wang, C., P. Y. Wu, and Z. Q. Chen. "Theoretical and experimental studies of a double-inlet reversible pulse tube refrigerator." *Cryogenics* 33.6 (1993): 648-652.
61. Luwei, Yang, Zhou Yuan, and Liang Jingtao. "Research of pulse tube refrigerator with high and low temperature double-inlet." *Cryogenics* 39.5 (1999): 417-423.
62. Gan, Z. H., G. B. Chen, G. Thummes, and C. Heiden. "Experimental study on pulse tube refrigeration with helium and nitrogen mixtures." *Cryogenics* 40, no. 4 (2000): 333-339.
63. Kasthurirengan S., Jacob S., Karunanithi R., Nadig D.S. and Behera Upendra. "Indigenous Development of Rotary valve for Cryocooler Applications", Journal of Instrument Society of India,2001
64. Kasthurirengan S., Jacob S., Karunanithi R., Nadig D.S. and Behera Upendra. "A six watt single stage pulse tube refrigerator operating at 77K", Journal of Instrument Society of India,2001

65. Kasthurirengan S., Jacob S., Karunanithi R., Development and studies on convection free single stage pulse tube cooler operating at 77K, Final Technical Report, April 2000, Centre for cryogenic Technology IISC, Bangalore.
66. Roy, P. C., S. K. Sarangi, and P. K. Das. *‘Some theoretical and experimental studies on pulse tube refrigeration*. Diss. MS Thesis, Department of Mechanical Engineering IIT, Kharagpur, 2004.
67. Lu, G. Q., and P. Cheng. "Numerical and experimental study of a Gifford-McMahon-type pulse tube refrigerator." *Journal of thermophysics and heat transfer* 17.4 (2003): 457-463.
68. Koettig, T., S. Moldenhauer, M. Patze, M. Thürk, and P. Seidel. "Investigation on the internal thermal link of pulse tube refrigerators." *Cryogenics* 47, no. 3 (2007): 137-142.
69. Banjare, Y. P., R. K. Sahoo, and S. K. Sarangi. "CFD simulation of a Gifford–McMahon type pulse tube refrigerator." *International Journal of Thermal Sciences* 48.12 (2009): 2280-2287.
70. Banjare, Y. P., R. K. Sahoo, and S. K. Sarangi. "CFD simulation and experimental validation of a GM type double inlet pulse tube refrigerator."*Cryogenics* 50.4 (2010): 271-280.