NUMERICAL ANALYSIS OF DIAPHRAGM TYPE PULSE TUBE REFRIGERATOR

Thesis Submitted to

National Institute of Technology, Rourkela

for the award of the degree

of

Master of Technology

In Mechanical Engineering with Specialization

in "Thermal Engineering"

by

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JUNE 2013

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CERTIFICATE

This is to certify that the thesis entitled, ***"Numerical Analysis of Diaphragm Type Pulse Tube Refrigerator"***, submitted by **Sisir Kumar Dalai** (211ME3187) to National Institute of Technology, Rourkela, is a record of bonafide project work carried out by him under my supervision and guidance. I considered it worthy of consideration for the award of the degree of Master of Technology in Mechanical Engineering with specialization in **"Thermal Engineering"** of the institute.

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DECLARATION

I certify that

a. The work contained in this report is original and has been done by me under the guidance of my supervisor.
b. The work has not been submitted to any other Institute for any degree or diploma.
c. I have followed the guide lines provided by the Institute in prepairing the report.
d. I have conformed to the norms and guide lines given in the Ethical code of conduct of the Institute.
e. When ever I have used materials (data,theoretical analysis,figures and text) from other sources, I have given due credit to them by citing in the text of the report and giving their details in the references. Further I have taken permission from the copyright owners of the sources, whenever necessary.

Sisir Kumar Dalai

ACKNOWLEDGEMENTS

During the one-year project work, I received a great deal of selfless help from a lot f people to whom I am thankful.

First of all, I would like to thank my advisor, **Prof.(Dr.) Ranjit Kumar Sahoo**, for his directions,guidance and support. His enthusiasm and inspiration lightened the road for completing this work.

I would also like to thank, **Prof.(Dr.) K. P. Maity (HOD)** , **Sachindra Kumar Rout** (Phd research scholar) and all the faculty members for all the suggestions and guidance.

I also express my sincere gratitude to my family members for their continuous encouragement,support and understanding, without which this project work would not have been possible.

Finally, I bow down to the Almighty who has made everything possible.

(Sisir Kumar Dalai)

CONTENTS

ABSTRACT

Pulse tube cryocooler (PTC) is a small-scale refrigerator capable of reaching cryogenic temperature below 123K(-150^{0}c). With intense research, it leads to advantages of no moving components at low temperature as well as its simplicity, low cost, low vibration at cold head and long life time. With its rapid development in the performance of the PTC, has been especially adopted in operations like aerospace, infrared sensors, communication, night vision equipment, cryo-pumping, SQUID and superconductivity. In this, the present study is classified into two groups which are discussed below.

In the first part of the study, illustrates a numerical study of single stage coaxial as well as inline Inertance-Type Pulse Tube Refrigerator (ITPTR), whose performance depends on a regenerator. The regenerator is the significant component of a pulse tube refrigerator which has a great importance of producing a cooling effect. In this present work a computational fluid dynamic (CFD) solution approach has been chosen for numerical purpose. The detail analysis of cool down behavior, heat transfer at the cold end and pressure variation in the whole system has been carried out by using the commercially available computational fluid dynamic software package FLUENT. A number of cases have been solved by changing the porosity of the regenerator from 0.5 to 0.9 and the rest of parameters remains unchanged. The operating frequency in all cases is 34 Hz, pulse tube diameter 5mm and length is 125mm. The result shows that a porosity value of 0.6 produces a better cooling effect on the cold end of the pulse tube refrigerator. The variation of pressure inside the pulse tube refrigerator during the process also analyzed.

In the second part of study, a new modeling approach for the performance of a pulse tube refrigerator is proposed. A single stage Stirling type diaphragm type pulse tube refrigerator is considered for CFD simulation using Fluent software. The two-dimensional geometry is taken for simulation. In this simulation the physical dimensions of all the components are kept constant and pressure input is generated from the different UDF (User Defined Function) for compressor and diaphragm. Generally, DC gas flow in pulse tube refrigerator created a crucial problem at the time of application, which will affect the cooling down performance because DC gas flow carries heat away from the HHX and deposits at the cold end reading an additional heat loss and unwanted thermal load to CHX , which degrades the cooling performance and doctorates the refrigeration stability of pulse tube cryocooler. For obtaining better temperature stability at CHX end, suppress the DC flow by eliminating the closed loop by using a diaphragm in between HHX and orifice valve. This simulation starts with an assumed uniform system temperature, and continued until steady periodic conditions are achieved by varying the pressure amplitude and getting the better stabilization temperature. The result shows that amplitude of 0.0045 produces a better cooling effect on the cold end of the pulse tube refrigerator.

Key words: ITPTR,HHX,CHX,Pulse Tube,OPTR,DPTR,DC flow

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NOMENCLATURE

Symbols

x_0	piston displacement amplitude (m)
C_p	specific gas constant, (J/kg-K)
C	inertial resistance (m^{-1})
E	total energy (JKg^{-1})
h	enthalpy (J/kg)
j	superficial velocity (m/s)
k	thermal conductivity (W/m-K)
p	pressure (N/m^2)
T	temperature (K)
t	time (s)
v	velocity (m/s)
V	volume (m^3)
S_x,S_y	momentum source terms (Nm^{-3})

Greek Symbols

ω	angular frequency (rad/s)
ψ	permeability tensors (m^2)
μ	dynamic viscosity(kg/m s)
τ	stress tensors (N/m^2)
ξ	porosity
ρ	density (kg/m^3)
ν	kinematic viscosity
τ	stress tensors (N/m^2)

Subscripts

S	solid
f	fluid
z	frequency
y	radial coordinate
x	axial coordinate

Abbreviations

PTR	Pulse tube refrigerator
BPTR	Basic pulse tube refrigerator
CHX	Cold end heat exchanger
BC	Boundary condition
DC	Direct flow
DIV	Double inlet valve
DIPTR	Double inlet pulse tube refrigerator
GM	Gifford Mc-Mahon
HHX	Hot end heat exchanger
ITPTR	Inertance tube pulse tube Refrigerator
OPTR	Orifice pulse tube refrigerator
SQUID	Supper conducting quantum interference device
TAPTR	Thermo-acoustic pulse tube refrigerator
DPTR	Diaphragm type PTR

CHAPTER 1
INTRODUCTION

1.1 General:

Cryogenics comes from the Greek word “kryos”, which means very cold or freezing and “genes” means to produce. Cryogenics is the science and technology associated with the phenomena that occur at very low temperature, close to the lowest theoretically attainable temperature. The pulse tube cooler is a very small and attractive device which can reach the temperature ranges between absolute zero to 123K (-150^{0}C) for cryogenic cooling. Now a day, cryocoolers are used in great need due to the absence of moving parts in the cold part, with respect to reliability, light weight, lifetime, low vibration and cost.

1.2 Applications of cryocoolers:

Applications of cryocoolers are:

- Industrial gas liquefaction
- Liquefaction of oxygen for hospital and home use
- Cryogenic catheters and cryosurgery
- Superconducting power applications (motors, transformers etc.)
- Infrared sensors for night-security and rescue
- Storage of biological cells and specimens
- Superconducting magnets in maglev trains
- Cooling superconducting magnets for MRI
- SQUID magnetometers for heart and brain studies
- Cryo vacuum pumps

1.3 Classification of Cryocoolers:

According to the classification by Walker (1983) there are two types of cryocoolers:

a. Recuperative type
b. Regenerative type.

Recuperative type

In this type of system the flow occurs in only one direction. So to control the flow direction the compressor and expander must have inlet and outlet valves, unless rotary or turbine compressor or expanders are used. These heat exchangers has two or more separate flow channels. The performance of the system is dependent on the properties of the working fluids used. The maximum loss of energy in this system occurs at compressor end. The main advantage of recuperative cryocoolers, however, is that they can be scaled to any size. Examples are J-T, Brayton, and Claude cryocooler as in fig.1.

Regenerative type

In this type of system flow occurs periodically as compared to recuperative cryocoolers. It resembles to AC flow in electrical system where pressure is analogous to voltage and mass flow to current. In regenerative cryocoolers during half of the cycle the regenerator receives heat from the fluid making fluid cold and in next half it returns to its previous temperature by releasing heat to the fluid. The cycle is repeated and at equilibrium regenerator is at room temperature while the other end is at the cold regenerators with the use of stacked fine-mesh screen or packed spheres. Examples of regenerative cryocoolers are Pulse Tube Refrigerator, Stirling Refrigerator and Gifford-McMahon Refrigerator as shown in Fig.1.

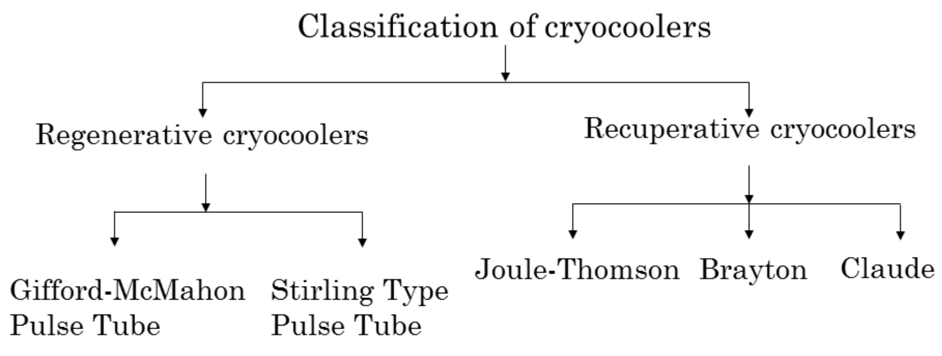


Fig.1 Classification of cryocoolers

There are many ways of classifying pulse tube refrigerator which are as follows as shown in figure.

- Based on nature of pressure wave generator:

(i) Stirling type PTR (valve less)

(ii)Gifford McMahon (GM) type PTR (with valve)

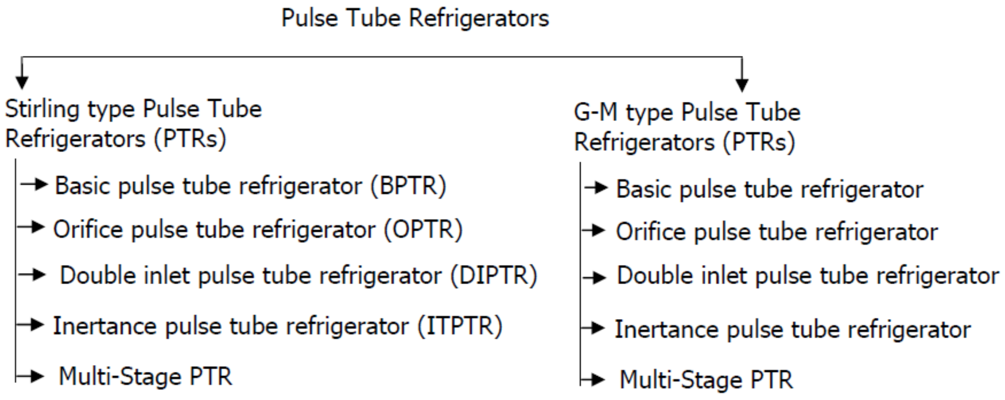


Fig.2 Types of PTR

- On the way of development:

(i) Basic Pulse Tube Refrigerator (BPTR)

(ii) Orifice pulse tube refrigerator (OPTR)

(iii) Double inlet pulse tube refrigerator (DIPTR)

(iv) Multiple inlet pulse tube refrigerator

(v) Inertance tube pulse tube refrigerator (ITPTR)

(vi) Single stage pulse tube refrigerator

(vii) Multi stage pulse tube refrigerator

(Viii) Thermoaccoustic pulse tube refrigerator (TAPTR)

- According to geometry or shape:

(i) In-line type pulse tube refrigerator

(ii) U type pulse tube refrigerator

(iii) Coaxial type pulse tube refrigerator

1.4 Principle of pulse tube refrigerator:

As discussed above, the pulse tube refrigerators (PTR) are capable of cooling to temperature below 123K.Unlike the ordinary refrigeration cycles which utilize the vapour compression cycle as described in classical thermodynamics; a PTR implements the theory of oscillatory compression and expansion of the gas within a closed volume to achieve desired refrigeration. Being oscillatory, a PTR is a non-steady system that requires a time dependent solution. However like many other periodic systems, PTRs attain quasi-steady periodic state (steady-periodic mode). In a periodic steady state system, property of the system at any point in a cycle will reach the same state in the next cycle and so on. A Pulse tube refrigerator is a closed system that uses an oscillating pressure (usually produced by an oscillating piston) at one end to generate an oscillating gas flow in the rest of the system as shown in below figure. The gas flow can carry heat away from a low temperature point (cold heat exchanger) to the hot end heat exchanger if the power factor for the phasor quantities is favorable. The amount of heat they can remove is limited by their size and power used to drive them.

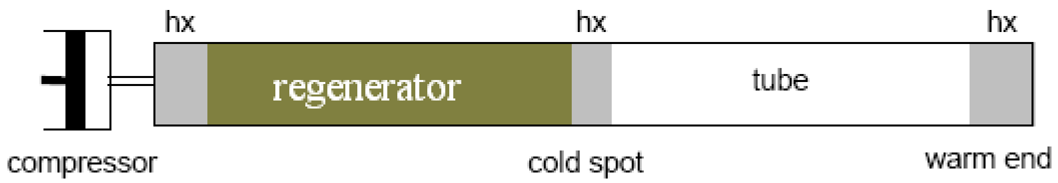


Fig.3 Basic PTR

It is necessary for the PTR for its efficiently working that:

1. The gas should reach the cold heat exchanger without carrying much heat. This is satisfied by the regenerator which traps the maximum amount of heat before fluid reaches the cold end of the system.
2. Secondly the amplitude of the fluid flow and pressure oscillations in the pulse tube should be enough to carry heat from cold point to hot point.
3. The phase relation between pressure and gas flow in pulse tube section should be such it gives maximum positive effect.

In basic pulse tube refrigerator, the flow becomes zero when the pressure reaches its peak value which causes very low cooling. And this is due to its closed end at the hot end. Hence further modification was done by incorporating orifice and a buffer or latest the narrow and long tube called inertance tube giving the maximum cooling effect which is discussed later in this thesis.

The pulse tube refrigerator is the simplest among all the cryocooler's as there are no moving parts in the cold region resulting in high reliability, long life, and low vibration at the cold tip which is an advantage over Stirling cryocooler. It does not follow Stirling cycles but as the working principle has a resemblance to Stirling cryocooler so it is also called Stirling type cryocooler. At the current stage of worldwide research, such accurate models of cryocoolers are not readily available in. Further, the complexity of the periodic flow in the PTC makes analysis difficult. Although different models are available to simulate pulse tube cryocoolers, the models have its limitations and also range of applicability. In order to accurately predict and improve the performance of the PTC system a reasonably thorough understanding of the thermo fluid-process in the system is required. One way to understand the processes is by numerically solving the continuum governing equations based on fundamental principles, without making arbitrary simplifying assumptions. The recent availability of powerful computational fluid dynamics (CFD) software [37] that is capable of rigorous modeling of transient and multidimensional flow and heat transfer process in complex geometries provides a good opportunity for analysis of PTCs. A successful demonstration of the suitability of the CFD type analysis for Stirling and Gifford-McMahon type PTCs is one of the main features of the investigation.

During the last three decades, Computational Fluid Dynamics has been an emerged as an important element in professional engineering practice, cutting across several branches of engineering disciplines. CFD is concerned with numerical solution of differential equations governing transport of mass, momentum, and energy in moving fluids. Today, CFD finds extensive usage in basic and applied research, in the design of engineering equipment, and in calculation of environmental and geophysical phenomena. For a long time, design of various engineering components like heat exchangers, furnace etc. Required generation of empirical information which was very difficult to generate. Another disadvantage of this empirical information is that, they are applying for a limited range of scales of fluid velocity temperature etc. Fortunately, solving differential equations describing mass transport, momentum and energy and then interpreting the solution to practical design was the key to the whole designing process. With the advent of computers, these differential equations were solved at a speed that increased exponentially and hence the applications of CFD became widespread.

CFD can be broadly divided in to 3 elements:

1. Preprocessor:

During preprocessing, the user

- Defines the modeling goals
- Identifies the computational domains
- Designs and creates the grid system

2.Main Solver:

The main solver sets up the equations required to compute the flow field according to the various options and criteria chosen by the user. It also meshes the points created by the preprocessor which is essential for solving fluid flow problems. The process broadly includes selecting a proper physical model, defining material properties, boundary conditions, operating conditions, setting up solver controls, limits, initializing solutions, solving equations and finally saving the output data for post process analysis.

3. Post processor:

It is the last but a very important part of CFD in which the results of the simulation are examined and conclusions are drawn. It allows user to plot various data in the form of graphs, contours, vectors etc.

Fluent 6.3.26:

Fluent is a computer program that is used for modeling heat transfer and modeling fluid flow in complex geometries. Due to mesh flexibility offered by Fluent, flow problems with unstructured meshes can be solved easily. Fluent supports triangular, quadrilateral meshes in 2D problems and tetrahedral, hexahedral, wedge shaped meshes in 3D.

Fluent has been developed in C language and hence, provides full flexibility and power offered by the language. C language provides facilities like efficient data structures, flexible solver controls and dynamic memory allocation also. Client- server architecture in Fluent allows it to run as separate simultaneous processes on client desktop workstations and computer servers. This allows efficient execution, interactive control and complete flexibility of the machine.

1.5 Objectives of Investigation:

All pulse tube refrigerator units operate as closed systems where no mass is exchanged between the cryocooler and its environment. The only moving component is the piston (or the rotary valve) which oscillates back and forth to generate periodic pressure oscillations of the working fluid. Mostly helium is chosen as the working fluid because it offers lowest critical temperature compared to other available fluids. Accurate modelling of the pulse tube cryocooler is essential to predict its performance and thereby arrive at optimum design. At the current stage of worldwide research, such accurate models are not readily available in open literature. Further, the complexity of the periodic flow in the PTC makes analysis difficult. Although different models are available to simulate pulse tube cryocoolers, the models have its limitations and also range of applicability. The recent availability of powerful computational fluid dynamics (CFD) software that is capable of rigorous modelling of transient and multidimensional flow and heat transfer process in complex geometries provides a good opportunity for analysis of PTCs.

Present study aims to focus on the following points:

- Principle of low temperature generation in inertance tube pulse tube cryocooler numerically by varying the porosity.
- Derivation of pressure drop characteristics of wire mesh matrix and other type of porous media regenerator.
- Reducing temperature instability at CHX end and controlled DC flow suppression using a diaphragm in between the HHX and orifice.
- Find the best cooling performance using a diaphragm by varying the amplitude.
- The simulation of PTC using a commercial CFD software, FLUENT to study the flow phenomena and heat transfer characteristics in the pulse tube system. Under this simulation the present models are analyzed.

(I) 2D -contour axis-symmetric problem of inertance pulse tube.

(II) 2D-contour axis-symmetric problem using a diaphragm type pulse tube.

1.6 Literature review:

A pulse tube refrigerator is a device which is used to provide cooling at lower temperature. Gifford et al. [1-2] gave the first report of PTR (pulse tube refrigerator) in the mid-1960s and noticed that the closed end of the pipe becomes very hot when there was a pressure oscillation inside, whereas the open end towards the compressor was cool. After further studies and optimization of the geometry, they were able to reach a low temperature of 124 K at one end due to pressurization and depressurization in the hollow tube and the closed end was cooled with water. Their first design was based on a hollow cylindrical tube with one end open and the other end closed. The closed end is exposed to an ambient temperature heat exchanger, while the open end represents the cold end. As a result of the oscillatory flow field caused by the piston, the open end was subjected to an oscillatory pressure from the regenerator, causing the open end to cool. This refrigerator is commonly known as Basic Pulse Tube Refrigerator (BPTR). In his second paper he reported useful refrigeration in a pulse tube operating well below the critical pressure ratio. After experimentation by researchers, it was thought that the surface heat pumping effect occurs in a pulse tube by interaction between gas displacement along the wall surface, energy change in the gas, heat transfer from the gas to the wall and heat conduction along the wall, as a result of the periodic change of pressure of the gas [3]. This effect was considered to be the major mechanism by which heat is pumped from the cold to the hot end and a refrigeration effect generated in BPT refrigerators. Later, Radebaugh et al. [4] pointed out that enthalpy flow in the pulse tube and regenerator is important. The enthalpy flow consists of work flow without entropy flow, and heat flow with the entropy flow [5]. To improve the cooling performance of pulse tube cooler (PTC) at 20–40 K, L.M. Qiu et al. [6] employed a hybrid regenerator. They have used three-layer regenerator, which consists of woven wire screen, lead sphere and Er3Ni is optimized to enhance the cooling performance and explore the lowest attainable refrigeration temperature for a single-stage PTC. After carrying out their theoretical and experimental studies, a lowest no-load refrigeration temperature of 11.1 K was obtained with an input power of 6 kW.

M. Kasuya et al.[7] shows at the optimum phase angle, gas elements near the hot end of the pulse tube move towards the cold end during compression and towards the hot end during expansion. This gas motion is just the opposite to that required for surface heat pumping. This means that surface heat pumping is not essential for pulse-tube refrigeration and found that the improvement achieved with double-inlet pulse-tube refrigerators can be explained by their capability to reach a phase angle beyond 90°. Hofmann et al. [8] calculated the optimum phase shift between the pressure and mass flow rate for PTRs operated with different kinds of phase shifters such as the orifice system, the inertance tube, and the double inlet arrangement. The basic features are discussed first for ideal systems with no losses. Shown that for the pulse tube, the phase shift can be well estimated from the ideal model, whereas there is more discrepancy for the regenerator. It is advantageous in some cases to use pressure and volume flow phasors instead of pressure and mass flow phasors for modelling such coolers.

A significant improvement was made in 1984 by Mikulin et al. [9] and Storch,P et al [10], who introduced the orifice pulse tube refrigerator (OPTR). They modified the half open tube by connecting an orifice and a reservoir to the hot end. Due to this modification the performance of the pulse tube has been improved and for the first time it became comparable to the performance of practical cryocoolers (Stirling cycle,Gifford-McMahon and Joule-Thomson cryocoolers).Ya-Ling He et al.[11] revealed that first law numerical analysis can reveal the detailed variations of the thermodynamics, fluid flow and heat transfer parameters with time. Such information is essential for the second law thermodynamics analysis. From the first law and second law analysis of the orifice type and double-inlet type PTR, they found that the cooling performance coefficient and exergy efficiency improved has been improved.G.Q. Lu et al. [12] has been carried out on dynamical pressures of the viscous compressible flow oscillating at different locations in a Gifford–McMahon (G–M) type pulse tube refrigerator operating at cycle-steady states. Results shown that the oscillating amplitude of the pressure was largest at the hot end of the regenerator while the cycle-averaged pressure was the largest in the reservoir.J. Gerster et al. [13] described generally most pulse tube refrigerators need an additional heat exchanger at the hot end of the working space in contrast to other refrigerators with regenerators. Regenerative effect of the hot end heat exchanger acts as a loss. In that contribution the so-called hot end loss is described by means of an orifice pulse tube refrigerator and is compared with that of a four valve pulse tube refrigerator. Furthermore, FVPTRs do not need time for the heat exchange process at the hot end of the pulse tube. Therefore the period t can be shorter than at OPTRs with hot heat exchangers by Thummes et al.[14].

In this study, B.J. Huang et al. [15] proposed a new design of orifice pulse-tube refrigerator (VROPT) using a variable-resistance valve to replace the conventional orifice.

They have seen that VROPT is able to achieve on-line control by regulating the duty cycle or frequency fv of the VRO. We also show that VROPT will not loss its thermal performance as compared to conventional OPT. Bender et al [16]. have performed CFD simulations for OPTR and ITPTR [17] respectively at different frequency by using dual opposed piston compressor. They have discussed that at higher frequency turbulence and recirculation of fluid is observed, which deteriorates the overall performance of the system. CFD simulation results showed that there is an optimum frequency for each PTR model at which it provides maximum refrigeration. Roach et al. [18] carried out a theoretical analysis of the behavior of a typical pulse tube regenerator. Assuming simple sinusoidal oscillations, the static and oscillatory pressures, velocities and temperatures are determined from a model that includes a compressed gas and imperfect thermal contact between the gas and the regenerator matrix. For realistic material parameters, the analysis reveals that the pressure and velocity oscillations are largely independent of the details of the thermal contact between the gas and the solid matrix. Only the temperature oscillations depend on this contact. De Waele [19] discussed the dynamic behavior of the temperature profiles in the regenerator and in the gas near the hot and cold ends of the tube. With some simplifying assumptions, the basic properties of the temperature profile in the regenerator and in the tube are understood. [20].

A creative type of pulse tube cooler is made, so called as double inlet pulse tube cooler. Gedon et al. [21] performed an analysis to show that the DC gas flow in a double-inlet pulse tube cryocooler is due to the existence of a closed loop consisting of the regenerator, pulse tube and the double-inlet valve. Because positive and negative flows over each half cycle do not complete completely. That causes due to the phase between density and velocity fluctuations in the flow-resistive elements of the closed loop which tends to produce the DC pressure gradients and this produces heat lift degradation. Again Olson et al. [22] show DC gas flow is acoustic streaming which is caused by the difference of viscosity of gas parcel through the pulse tube in the boundary layer due to the boundary layer effect named Rayleigh DC gas flow. Ju et al. [23] investigated on the multi-bypass pulse tube refrigerator for improving the cooling performance by adding a bypass between regenerator and pulse tube middle. Which increases the pressure ratio and gas piston stroke in the coldest part of the pulse tube refrigerator. Thus the pulse tube refrigerator achieved a low refrigerator temperature. There used a hot wire anemometer to measure the dynamic gas velocity and temperature by a CTA (constant temperature anemometer) and found that there was a DC flow in the double-inlet pulse tube refrigerator. Their results shown that the temperature of

BPTR is 10-15Klower than DPTR. C. Wang et al. [24-25] noticed that DC flow effect leads to change the temperature profile when the flow circulating around the regenerator and pulse tube. Numerical and experimental analysis shows DC flow in a 4 K single-stage pulse tube cooler. Certain functions which cause DC flow, such as phase angle, valve timing and geometry of double-inlet valve. Numerical prospects show that negative DC flow from the hot end of the pulse tube to the inlet of regenerator has two positive effects on the cooler performance i.e. increasing the cooling capacity and decreasing the enthalpy flow at the pulse tube. DC flow must exist in most double-inlet versions. Also DC flow changes the amplitude of positive and negative fluid flow. DC gas flow, which are increasing the gross and net cooling capacity of the pulse tube and dissipating the enthalpy flow at the hot end of the pulse tube.

In their second paper, he developed a model to control the DC flow. They show DC flow rate is obtained by changing the flow coefficients for two flow directions of the double inlet valve. It shows DC flows varies from negative to positive .That means DC flow is defined as positive when gas flows from the regenerator inlet to the hot end of the pulse tube through bypass and negative when flows in the reverse direction. DC flow compensated to a value near zero in order to gain better performance. Minimum in the dynamic temperature in the 4K pulse tube decreases as negative flow increases. Luwei Yang et al. [26] analysed DC flow in a double-inlet version pulse tube refrigerator. Too high temperature or too low temperature near the pulse tube end with the double inlet opening caused by the DC flow using the second orifice. Second orifice is stable in lowering the refrigeration temperature. Without second orifice cooling down rate is faster than with second orifice. The only difference is that more cooling will be occurred with a second orifice than without second orifice. Second orifice is arranged according to the DC flow direction. Second orifice is effective to lower the temperature if there exists DC flow. Again Yang et al. [27] proposed a mechanism of double-inlet to lower the temperature of the pulse tube refrigerator by analysing the phase relation between pressure and the flow characteristics of double-inlet and multi-bypass, then another proposal is put forward to replace the low temperature multi-bypass valve, which is another double inlet valve at room temperature and a small tube prolonging to the low temperature part of pulse tube. Through experiment, this multi-double inlet version proved effective to lower the refrigeration temperature. Test acquired a lowest temperature of 77K with two double-inlet valves and 50K with one double-inlet valve.

Guoqiang Lu et al. [28] performed the flow characteristics of a slowly oscillating compressible flow through a metering valve, which is a main component in an orifice pulse tube refrigerator and a double-inlet pulse tube refrigerator. Results show that non symmetrical geometrical construction would play a role in the DC flow in a double-inlet pulse tube refrigerator. Wei Dai et al. [29] introduced an artificial DC flow from pulse tube cold end to the hot end in V-M type Cryocooler by adjusting element of one reservoir (which is actually the dead volume inside Displacer's rotor housing), two orifice valves and one check valve. Shown that without artificial DC flow cannot get not get the obvious cooling effect at the pulse tube cold end. DC flow can increase the cooling down the speed of the cold end, which facilitates more easiness to optimize operating parameters. M.Shiraishi et al.[30] investigated by using smoke-wire flow visualization techniques in double inlet pulse tube is superimposed on secondary flow with almost zero velocity due to the counterbalance between acoustic streaming and DC flow. Therefor in an inclined double inlet pulse tube the velocity of the overall flow in the core region is almost equal to that of gravity driven flow. So that the velocity is always affected by gravity-driven flow. Shaowei Zhu et al[31] studied the relation between the DC gas flow, valve operating time intervals, and flow patterns in the bypass of the GM type double inlet pulse tube refrigerator by using symmetric bypass. In the GM type double inlet pulse tube refrigerator with symmetric bypass, valve opening angle difference is an effective parameter for adjusting the DC gas flow. That effect depends on the flow character of the bypass. With the turbulence flow bypass, it is a rather strong parameter, with laminar flow bypass, it is a rather weak parameter. The statistical works of Y.L. He et al. [32] performed for the positive direction mass flow rate and negative direction mass flow rate during a period. And it is found that the net value through the orifice valve is near to zero. So, the DC flow rate occupies about 4.66% of the total mass flow rate in the DPTR.

Swift et al [33]. introduced a barrier method of a limp rubber balloon to suppress DC flow in the loop path in a thermoacoustic Stirling refrigerator. The balloon blocked streaming flow through the loop path whereas it was acoustically transparent. They demonstrated that the method was effective in suppressing of DC flow. Hu JY et.al. [34] introduced a membrane-barrier method for DC flow suppression in double-inlet pulse tube coolers. An elastic membrane is installed between the pulse tube cooler inlet and the double-inlet valve to break the closed-loop flow path of DC flow. The membrane is acoustically transparent, but would block the DC flow completely. Thus the DC flow is thoroughly suppressed and the merit of double-inlet mode is remained. With this method, a temperature reduction of tens of

Kelvin was obtained in our single-stage pulse tube cooler and the lowest temperature reached 29.8 K.

The double-inlet pulse tube refrigerator that has a diaphragm inserted in a bypass-tube, which enabled it to transmit a pressure oscillation whereas to obstruct a DC gas flow, has manufactured and tested by Shiraish et. al.[35]. The oscillating flow behavior inside of the refrigerator has been studied by using a smoke-wire flow visualization technique.

CHAPTER 2

MODELLING OF INERTANCE TYPE PULSE TUBE REFRIGERATOR

Most recently invented PTR is the inertance tube pulse tube refrigerator shown in Fig.4. In this type of PTR the orifice valve is replaced by a long inertance tube having a very small internal diameter. The implementation of this inductance generates an advantageous phase shift in pulse tube and produces an improved enthalpy flow. Components of such type of PTR are described below.

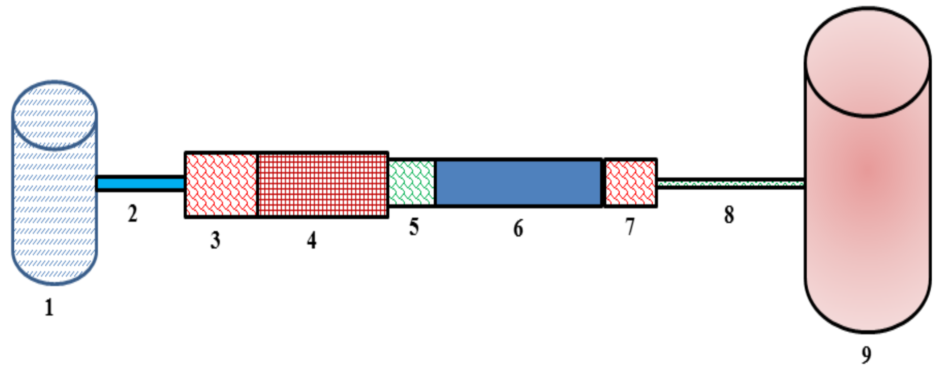


Fig. 4. Computational domain of ITPTR: 1 - compressor, 2 – transfer line, 3 - after cooler, 4 – regenerator, 5 – cold heat exchanger, 6 – pulse tube, 7 – hot heat exchanger, 8 - inertance tube, 9 - reservoir.

- **Compressor**

The main function of the compressor is to supply gas pressurization and depressurization in the closed chamber. Electrical power is applied to the compressor where this electrical work is converted into the mechanical energy associated with sinusoidal pressure waves, resulting in gas motion.

- **After cooler**

The function of the ideal after cooler is to extract all the heat that is generated in the compressor volume during the gas compression and dispose to the environment. This minimizes the warm end temperature so that the regenerator can work more efficiently and supply low.

- **Regenerator**

The heart of the pulse tube refrigerator is a regenerator. Its function is to absorb the heat from the incoming gas during the forward stroke, and deliver that heat back to the gas during the return stroke.

- **Cold heat exchanger(CHX)**

Cold heat exchanger inside the pulse tube refrigerator is the equivalent of the evaporator in the vapor compression refrigeration cycle. Inside the cold heat exchanger the refrigeration load is absorbed by the system.

- **Pulse tube**

The pulse tube is the most critical component of the whole refrigerating system. The main objective of the pulse tube is to carry the heat from the cold end to the warm end by an enthalpy flow. By imposing a correct phase difference between pressure and mass flow in the pulse tube by phase shifting mechanisms, heat load is carried away from the CHX to the HHX.

- **Hot Heat Exchanger (HHX)**

In every periodic cycle the gas rejects heat of compression at the hot end exchange. Receiving the enthalpy flow from the pulse tube, the heat load at a higher temperature is rejected to the environment.

- **Inertance Tube**

It is a long thin tube that introduces the possibility of an additional phase shift between the pressure and mass flow rate in the pulse tube section.

- **Reservoir**

The surge volume used as a closed buffer reservoir to allow for small pressure variations resulting from the oscillating mass flow.

2.1 Problem Description:

Computational fluid dynamics or CFD is the analysis of systems involving fluid flow, heat transfer and associated phenomenon such as chemical reactions by means of numerical simulations. The technique is very powerful and spans a wide range of industrial and non-industrial application areas. The availability of fast computers equipped with very large memories allow for remarkably precise numerical simulations. The main CFD tools are PHONICS, FLUENT, FLOW3D and STAR-CD etc. The fluent is one of the most highly expected CFD codes. Fluent is a state-of-the-art computer program for modeling fluid flow and heat transfer process in complex engineering problems. With the help of this one can enervate code and set boundary condition by User Defined Functions (UDF). Fluent has also a dynamic meshing function. This function allows the user to create deforming mesh volumes such that applications involving volume compression and expansion can be modeled. Thus in view of Fluent's versatility, its capability for solving the compression and expansion volume, allowing UDF boundary conditions and modeling capability for porous media, it is selected for the simulation of the Stirling type ITPTR.

The pressure waves in a Stirling-type pulse tube refrigerator is generated directly by a valve less compressor. Thus, a Stirling-type pulse tube refrigerator usually works at high frequencies (10Hz-50Hz). The dimensions of the Stirling type inertance tube pulse tube refrigerator (ITPTR) re taken from literature, except the compressor is replaced by a dual opposed piston model in the present simulation. Fig. 5 illustrates the 2-dimensional physical geometry of the ITPTR. The ITPTR is therefore modeled in a 2-dimensional axis-symmetric co-ordinate system.

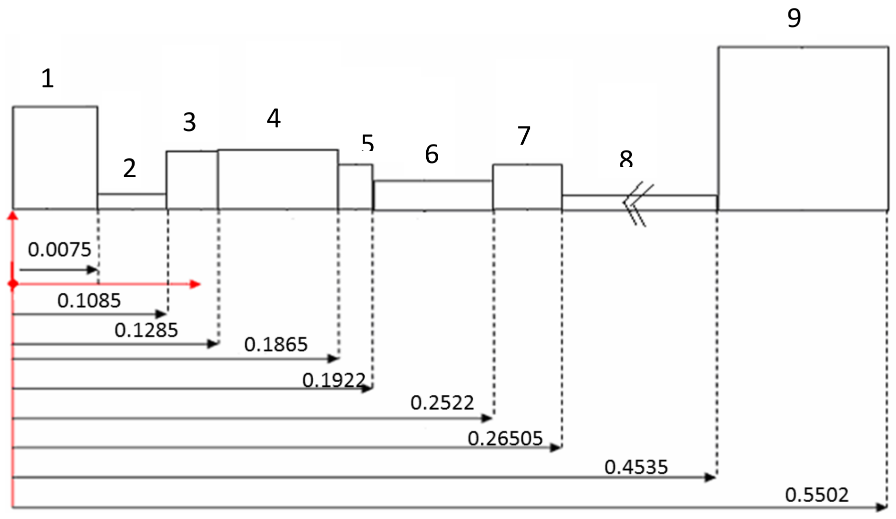
The geometric modeling and nodalization of various parts of Stirling cryocooler has been done using Gambit Design Modeler. Because of the symmetry of the model a 2D axisymmetric design was developed. Both triangular and quadrilateral meshes have been used to mesh the model. The main components of the design are Compressor, Transfer line, After-cooler, Regenerator, CHX, Pulse tube, HHX, Inertance tube, reservoir. Piston and displacer are the only moving parts of the pulse tube while the rest parts are stationary. The dimensions of the model have been illustrated in Table.1.

Table.1 Component dimensions of ITPTR

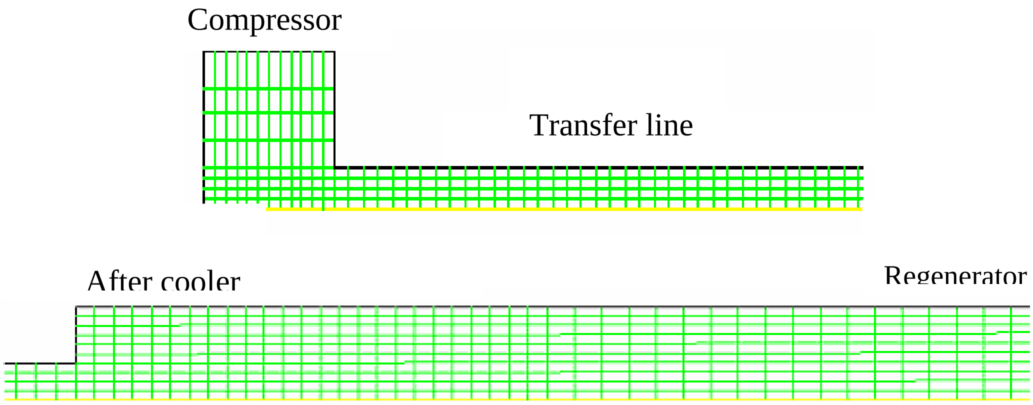
Constituent part	Diameter (m)	Length (m)
(1) Compressor	0.01908	0.0075
(2) Transfer line	0.0031	0.101
(3) After cooler	0.008	0.02
(4) Regenerator	0.008	0.058
(5) Cold heat exchanger	0.006	0.0057
(6) Pulse tube	0.005	0.125
(7) Hot heat exchanger	0.008	0.01
(8) Inertance tube	0.00085	0.684
(9) Reservoir	0.026	0.13

Initially the whole geometry is created as single components then it is splitted in different zones. The function of splitting is to divide the geometry as different components so as to enable to set different boundary conditions at different zones as needed. Once these faces/volumes are created, an actual meshing process can be initiated.

The reason for creating a mesh is that a partial differential equation generates an infinite dimensional problem and the solution must in general be sought in a finite dimensional space. There are different options for mesh generation are available in Gambit to list a few are triangular elements, hexahedral elements and quadrihedral elements. If the computational domain under consideration does not contain any complex surface the structured quad element could be used for meshing.Following figure shows the some enlarged view of the axis-symmetric geometry of the different components for the ITPTR with complete meshing. Once the model is created and exported to fluent 6.3.26, the model boundaries have to be defined.



Two-dimensional axis-symmetric geometry of ITPTR



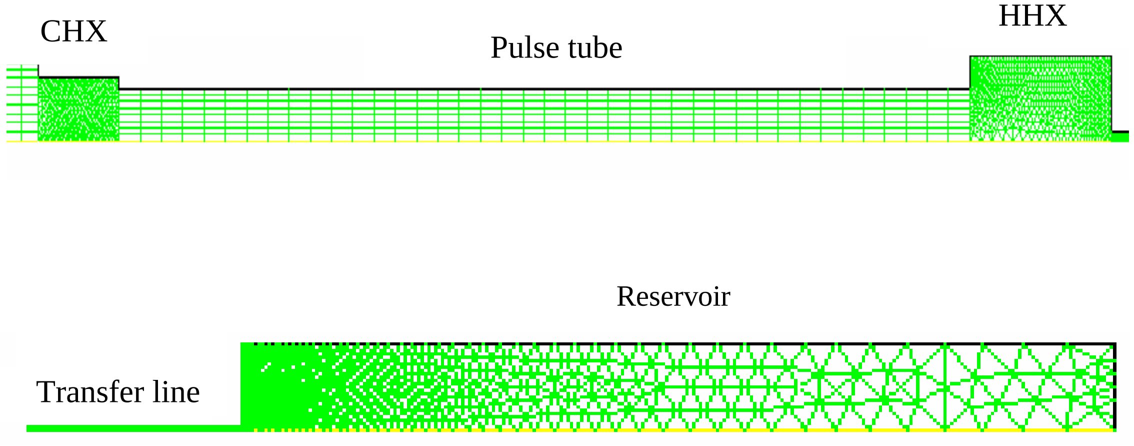


Fig. 5 2-D axis-symmetric model and meshing

2.2 Solution Technic:

The Finite Volume Method (FVM) is one of the most versatile discrimination techniques used for solving the governing equations for fluid flow and heat and mass transfer problems. The most compelling features of the FVM are that the resulting solution satisfies the conservation of quantities such as mass, momentum, energy and species. This is exactly satisfied for any control volume as well as for the whole computation domain. Even a coarse grid solution exhibits exact integral balances. Apart from this, it can be applied to any type of grids (structured or unstructured, Cartesian or body fitted), and especially complex geometries. Hence, it is the platform for most of the commercial packages like Fluent, Star-CD, and CFX etc. which are used to solve fluid flow and heat and mass transfer problems. In the finite volume method, the solution domain is subdivided into continuous cells or control volumes where the variable of interests is located at the centroid of the control volume forming a grid.

The next step is to integrate the differential form of the governing equations over each control volume. Interpolation profiles are then assumed in order to describe the variation of the concerned variables between cell centroids. There are several schemes that can be used for interpolation, e.g. central differencing, upwind differencing, power-law differencing and quadratic upwind differencing schemes. The resulting equation is called the discretized or discretization equation. In this manner the discretization equation expresses the conservation principle for the variable inside the control volume. These variables form a set of algebraic equations which are solved simultaneously using special algorithm.

2.3 Governing Equations:

The governing equations for the present 2D axisymmetric transient model are continuity equation, Momentum equation in both axial as well as in radial direction and energy equation for solid matrix inside the porous region written with no swirl assumption. Two extra source term, both axial as well as in radial direction are considered for the porous region to calculate momentum losses. For the rest of porous medium these values are assumed to be zero and solid matrix inside a porous region is considered as homogeneous.

Continuity Equation:

$$\frac{\partial}{\partial t}[\xi\rho]+\frac{1}{y}\frac{\partial}{\partial y}\left[\xi r\rho_y\right]+\frac{\partial}{\partial x}[\xi\rho_f v_x]=0 \tag{1}$$

Momentum Equation:

For axial direction:

$$\begin{aligned}&\frac{\partial}{\partial t}[\xi\rho v_x]+\frac{1}{y}\frac{\partial}{\partial x}[\xi r v_x v_x]+\frac{1}{y}\frac{\partial}{\partial y}\left[\xi y\rho_f v_x v_y\right]=-\frac{\partial\xi p}{\partial y}+\\&\frac{1}{y}\frac{\partial}{\partial y}\left\{y\mu\left(2\frac{\partial\xi v_x}{\partial x}-\frac{2}{3}\left(\vec{\nabla}\xi.\vec{v}\right)\right)\right\}+\frac{1}{y}\frac{\partial}{\partial y}\left\{y\mu\left[\frac{\partial\xi v_x}{\partial y}+\frac{\partial\xi v_y}{\partial y}\right]\right\}+S_x\end{aligned} \tag{2}$$

For radial direction:

$$\begin{aligned}&\frac{\partial}{\partial t}\left[\xi\rho_f v_y\right]+\frac{1}{y}\frac{\partial}{\partial x}\left[\xi r v_x v_y\right]+\frac{1}{y}\frac{\partial}{\partial y}\left[y\xi\rho_f v_y v_y\right]=-\frac{\partial p}{\partial y}+\\&\frac{1}{r}\frac{\partial}{\partial x}\left\{2x\mu\left(\frac{\partial v_x}{\partial x}-\frac{1}{3}\left(\vec{\nabla}.\vec{v}\right)\right)\right\}+\frac{1}{y}\frac{\partial}{\partial y}\left\{2y\mu\left[\frac{\partial v_x}{\partial y}-\frac{1}{3}\left(\vec{\nabla}.\vec{v}\right)\right]\right\}\\&+\frac{2\mu}{y}\left[\frac{\left(\vec{\nabla}.\vec{v}\right)}{3}-\frac{v_y}{y}\left(\vec{\nabla}.\vec{v}\right)\right]+S_y\end{aligned} \tag{3}$$

Where S_x and S_y are the two source term in the axial and radial direction which values is zero for nonporous zone. But for the porous zone the source term which is solved by the solver is given by the following equation.

$$S_x = -\left(\frac{\mu}{\psi} v_x + \frac{1}{2} C \rho_f \left| v \right| v_x \right) \tag{2}$$

$$S_y = -\left(\frac{\mu}{\psi} v_y + \frac{1}{2} C \rho_f \left| v \right| v_y \right) \tag{3}$$

In the above equation the first term is called Darcy term and the second term is called the Forchheimer term which are responsible for the pressure drop inside the porous zone.

Energy Equation:

$$\frac{\partial}{\partial t}\left(\xi \rho E_f + (1-\xi) \rho_s E_s \right) + \vec{\nabla}.\left(\vec{v} \left(\rho_f E_f + p \right) \right) = \vec{\nabla}.\left(k \vec{\nabla} T_f + \tau.\vec{v} \right) \tag{4}$$

Where

$$k = \xi k_f + (1-\xi) k_s \tag{5}$$

$$E_f = h - p/\rho_f + v^2/2 \tag{6}$$

2.4 Steps involved in CFD problem:

1. The physical domain is specified according to the given flow situation.

2. Various boundary conditions, fluid properties are defined and appropriate transport equations are selected.

3. Nodes are defined in the space to map it into the form of a grid and control volumes are defined around each node.

4. Differential equations are integrated over a control volume to convert them into an algebraic one. Some of the popular techniques used for discretization are finite difference method and finite volume method.

5. A numerical method is devised to solve the set of algebraic equations and a computer program is developed to implement the numerical method.

6. The solution is interpreted.

7. The results are displayed and post process analysis is done.

Defining the Model:

Before iterating, the model properties such as fluent solver, materials, thermal and fluid properties besides model operating conditions and boundary conditions have to be defined to solve the fluid flow problems. The conditions specified in this case have been mentioned below.

- **Solver**

Segregated solver, implicit formulation, axisymmetric space, unsteady time.

- **Viscous model and energy equation**

k- epsilon set of equations and energy equation were selected.

- **Material**

Helium (ideal gas) as working fluid, Steel and cooper for the frame material.

- **Operating condition**

Operating pressure was chosen to be 101325 Pa.

- **Boundary condition**

After-cooler and Hot space were chosen to have isothermal walls while the rest of the walls was chosen to be adiabatic with zero heat flux shown in table. Regenerator was chosen to be porous zones.

Table.2 Boundary and initial conditions for ITPTR

Study case	Boundary condition
Compressor wall	Adiabatic wall
Transfer line wall	Adiabatic wall
After cooler wall	293K
Regenerator wall	Adiabatic wall
Cold end wall	Adiabatic wall
Pulse tube wall	Adiabatic wall
Hot end wall	293 K
Inertance tube wall	Adiabatic wall
Surge volume wall	Adiabatic wall
Viscous resistance (m^{-2})	9.44e+9
Inertial resistance (m^{-1})	76090
Initial condition	300 K
Cold end load	0 W

- **Limits**

  Pressure: 20 atm to 40 atm.

- **Under relaxation factors**

  Pressure: 0.2; Momentum: 0.4;   Energy: 0.8

- **Discretization**

  Pressure-velocity coupling: PISO

  Pressure: PRESTO

  Density, momentum, energy were chosen to be First Order Upwind.

- The convergence criterion for continuity, x-velocity, y--velocity, k was chosen to be 0.001 and energy was chosen to be 1e-06.

Defining the Material properties:

This section of the input contains the option for the materials chosen as the working fluid. In this case, the working fluid is the helium and the solid is copper and steel. Properties that can be specified in this section are density, specific heat, thermal conductivity, viscosity and diffusivity.

Defining the Operating conditions:

The operating condition panel includes gravity and pressure. In horizontal axis-symmetric problem, the effect of gravity is not of much importance and hence it is neglected. The operating pressure is set to its default value of 1atm in this section to get the solution in terms of absolute value.

Defining the Porous zone:

The regenerator, after cooler, cold end heat exchanger and hot end exchanger of a pulse tube refrigerator is to be modelled using porous-media methods. A porous zone is modelled as a special type of fluid zone. To indicate that the fluid zone is a porous region, porous zone option in the fluid panel is enabled. The panel expands to show the porous media inputs. The user inputs for the porous media model are:

• Define the porous zone.
• Identify the fluid material flowing through the porous media.
• Select the solid material contained in the porous media.
• Specify the porosity of the porous media.
• Set the viscous resistance coefficients and inertial resistance coefficients, and define the direction vectors for which they apply.

2.5 Dynamic Meshing Function:

Fluent has a dynamic meshing function. The dynamic mesh model in Fluent can be used to model flows where the shape of the domain is changing with time due to the motion of the domain boundaries like in reciprocating compressor when the piston moves the domain of fluid will change with time in case of compression and expansion of fluid as shown in fig.6. This type of model could be handled in fluent by using the dynamic meshing function. The motion can be a prescribed motion or a un-prescribed motion where the subsequent motion is determined based on the solution at the current time. The update of the volume mesh is handled automatically by fluent at each time step based on the new positions of the boundaries. To use the dynamic mesh model, it is needed to provide a starting volume mesh and the description of the motion of any moving zones in the model. Fluent allows describing the motion using either boundary profiles or user-defined functions (UDFs).

Thus in view of Fluent's versatility, its capability for solving the compression and expansion of volumes, the compressor has been modelled using dynamic meshing in Stirling type pulse tube refrigerators. In Fluent different method are available for mesh update like smoothing, layering and remeshing for dynamic meshing. The compressor is modelled as a solid wall (piston) that sinusoidal oscillates in and out along a fixed stroke length. The piston and cylinder walls are nominally specified as adiabatic boundaries. Work input at the piston in the cylinder of a compressor provides the oscillating pressure that drives the cycle. In order to model the piston and cylinder, the fluent dynamic meshing function must be used. A user defined function (UDF) is developed in C programming language to simulate the piston cylinder effect. The compressor used in this simulation is a reciprocating dual opposed-piston design.

The mesh motion preview gives an information regarding motion of the mesh in either direction from its initial position with respect to time. This shows the compression and expansion process of the compressor. From initial condition mesh moves in an upward direction reaches TDC and then returns down till BDC. First set of Figure.6 (a) shows the axisymmetric geometry and the second set (b) shows the mirror view of the axis-symmetry geometry. If there is not proper matching between grid size spacing and time increment the mesh won't move. In this case fluent will show an error message for negative volume. So before starting the simulation it is necessary to check the motion of the mesh for the selected grid size. If it shows error then the grid size for compressor or time step size is changed and the process is repeated.

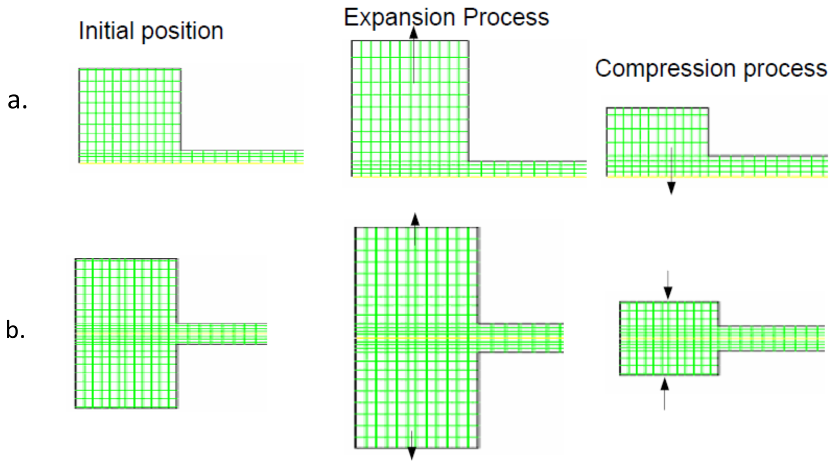


Fig. 6. Compression and Expansion of the piston

CHAPTER 3

MODELLING OF ORIFICE TYPE PULSE TUBE REFRIGERATOR

The schematic configuration of an OPTR can be viewed as a modification of the BPTR. This modification is made by including an orifice valve and a surge volume at the warm end of the BPTR, as depicted in Fig.7.Additional components create an advantage of in-phase relationship between the mass flow and the pressure within the pulse tube to enhance the heat transport mechanism. The proper gas motion in phase with the pressure is achieved by the use of an orifice and a reservoir volume to store the gas during a half cycle. The reservoir volume is large enough that negligible pressure oscillation occurs in it during the oscillating flow. The oscillating flow through the orifice separates the heating and cooling effects just as the displacer does for the Stirling and Gifford-McMahon refrigerators. The orifice pulse tube refrigerator (OPTR) operates ideally with adiabatic compression and expansion in the pulse tube. The four steps in the cycle are as follows. (1) The piston moves down to compress the gas(helium) in the pulse tube. (2) Because this heated, compressed gas is at a higher pressure than the average in the reservoir, it flows through the orifice into the reservoir and exchanges heat with the ambient through the heat exchanger at the warm end of the pulse tube. The flow stops when the pressure in the pulse tube is reduced to the average pressure. (3) The piston moves up and expands the gas adiabatically in the pulse tube. (4) This cold, low-pressure gas in the pulse tube is forced toward the cold end of the gas flow from the reservoir into the pulse tube through the orifice. As the cold gas flows through the heat exchanger at the cold end of the pulse tube it picks up heat from the object being cooled. The flow stops when the pressure in the pulse tube increases to the average pressure. The cycle then repeats. The function of the regenerator is the same as in the Stirling and Gifford-McMahon refrigerators in that it precools the incoming high pressure gas before it reaches the cold end.

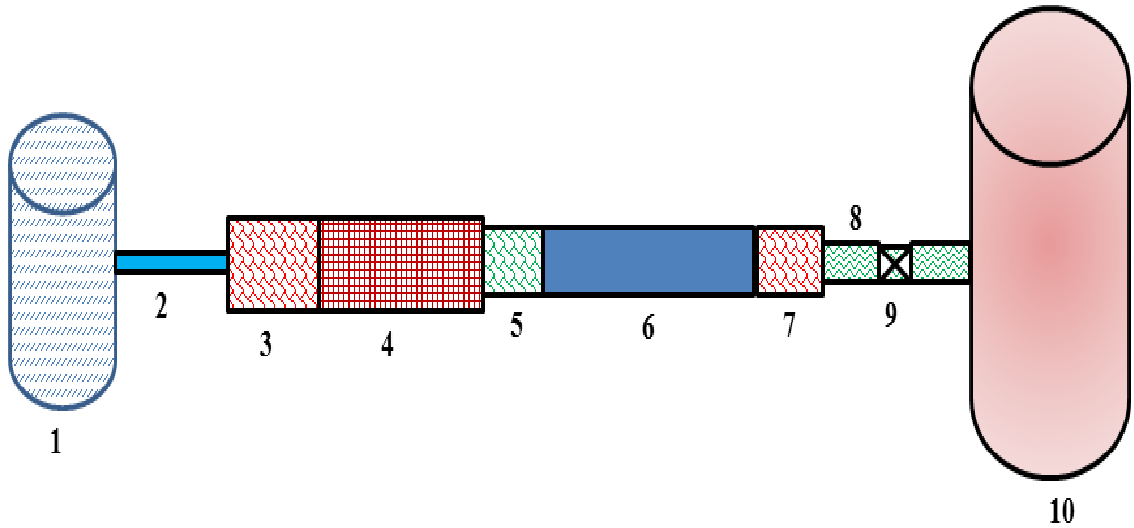


Fig. 7. Schematic diagram of computational domain of OPTR: 1 - compressor, 2- transfer tube, 3 - after cooler, 4- regenerator, 5- cold heat exchanger, 6- pulse tube, 7- hot heat exchanger,8-transfer line,9-orifice, 10 - reservoir.

A CFD study of OPTR has been reported in this chapter. The detail dimensions [36] and bcs are described in table.3 and table.4.The porous media regions are considered to be thermally non-equilibrium by using UDFs in which two separate energy equations are solved for solid matrix and fluid inside the porous region. The model is validated with experimental work by Antao et al.[36].It was found that a minimum temperature of 129K achieved when a average pressure of 18 bar, which is well match with that of experimental results by Antano et al.[36].The modeling and results are shown below.

The two dimensional view of the OPTR system is shown in Fig.7. The system consists of a compressor (dual opposed piston model), a transfer line, an after cooler, a regenerator, cold end heat exchanger, pulse tube, hot end heat exchanger, connecting tube, an orifice valve and a reservoir. The OPTR is therefore modelled in a 2-dimensional axis-symmetric co-ordinate system similar to ITPTR which is created in Gambit software. The face mode option of gambit is used to create the geometry.

Table.3 Component dimensions of OPTR

Constituent part	Radius(m)	Length (m)
(1) Compressor	0.00954	0.0075
(2)Transfer Tube	0.0085	0.04
(3) After cooler	0.0085	0.03
(4)Regenerator	0.0085	0.06
(5) Cold Heat Exchanger	0.0047	0.05
(6)Pulse Tube	0.0047	0.23
(7) Hot Heat-Exchanger	0.0047	0.03
(8)Transfer Line	0.00193	1.50
(9) Orifice	0.000255	-
(10)Reservoir	0.026	0.149

Table.4 Boundary and initial conditions for OPTR

Study case	Boundary condition
Compressor wall	Adiabatic wall
Transfer tube wall	Adiabatic wall
After cooler wall	293K
Regenerator wall	Adiabatic wall
Cold end wall	Adiabatic wall
Pulse tube wall	Adiabatic wall
Hot end wall	293 K
Transfer line,Orifice wall	Adiabatic wall
Resroir wall	Adiabatic wall
Viscous resistance(m^{-2})	9.44e+9
Inertial resistance(m^{-1})	76090
Initial condition	300 K
Cold end load	0 W

Orifice vs Double Inlet PTR:

From the above analysis and literature it is concluded that lower temperature is not achieved by using an orifice in between the HHX and the buffer. So a modification was done in [23] by adding a bypass from HHX to the regenerator inlet called double inlet PTR. Flowing of gas in that valve called as DC flow. By using double-inlet valve room-temperature gas enters the pulse tube directly without passing through the regenerator. The main contribution of the double-inlet pulse tube is to adjust the phase shift between the pressure wave and the mass flow at the hot head of the pulse tube refrigerator, and to increase their amplitudes. There are two types of DC flow such as

(i) Negative DC flow

(ii) Positive DC flow

Negative DC flow occurs when mass flow is from the regenerator inlet to the hot end of pulse tube refrigerator and treated a +ve mass flow because the pressure at the inlet of the regenerator is more than the pressure at hot end of pulse tube. Positive DC flow occurs when mass flow is from the hot end of pulse tube to regenerator inlet and treated a -the mass flow because the pressure at the hot end of pulse tube is more than the pressure at the regenerator inlet. Generally –ve DC flow increases the cooling performance of the refrigerator.

By using this type PTR a lower temperature is achieved than that of OPTR. But using bypass some demerits come out which will affect the refrigeration system. Temperature instability is the biggest problem in this type of PTR due the DC flow. To avoid such type of demerits developed a new type of PTR is called diaphragm PTR.

CHAPTER 4

MODELLING OF DIAPHRAGM TYPE PTR

The geometry of diaphragm type pulse tube refrigerator is completely different from others. A diaphragm is mounted in between HHX and the orifice valve. The diaphragm looks like a cylinder piston arrangement. The system consists of a compressor (dual opposed piston model), a transfer line, an after cooler, a regenerator, cold end heat exchanger, pulse tube, hot end heat exchanger, a diaphragm, connecting tube, a needle valve and a reservoir. Figure.8 illustrates the two-dimensional physical geometry of the diaphragm PTR. Infact all the components are aligned in series and form an axis-symmetric system. The potential asymmetry caused by gravity is thus neglected. This gravity term however will be important if the order of magnitude of the acceleration becomes comparable to the other terms (such as temporal acceleration, convective acceleration etc.) in the momentum equation. The figure shows the two-dimensional geometry of the DPTR which is created in Gambit software. The face mode option of gambit is used to create the geometry.

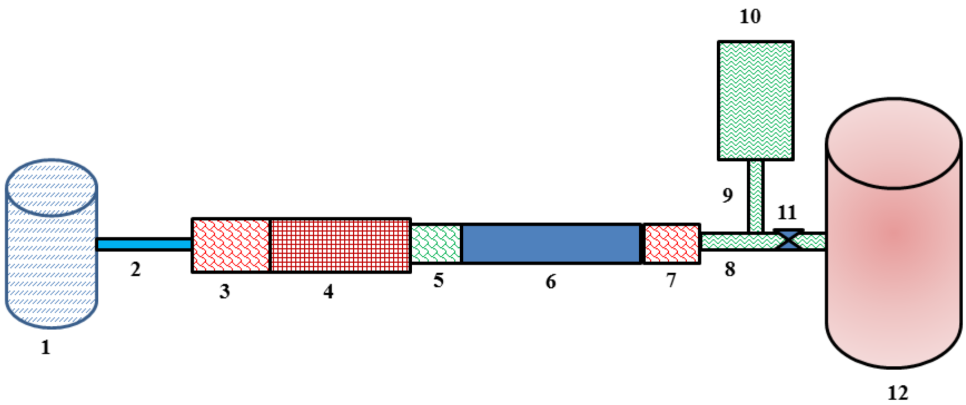


Fig. 8. Schematic diagram of computational domain of DPTR: 1 - compressor, 2- transfer tube, 3 - after cooler, 4- regenerator, 5- cold heat exchanger, 6- pulse tube, 7- hot heat exchanger,8-transfer line1, 9 – transfer line2,10- Diaphragm,11- Orifice,12-Reservoir

Axi-symmetric, two-dimensional flow was assumed everywhere in view of the axi-symmetric inline version and boundary conditions. The geometry parameters of DPTR and details are shown in Table 5.

Table.5 Component dimensions of DPTR

Constituent part	Diameter (m)	Length (m)
(1) Compressor	0.01908	0.0075
(2) Transfer line	0.0031	0.101
(3) After cooler	0.008	0.02
(4) Regenerator	0.008	0.058
(5) CHX	0.006	0.0057
(6) Pulse tube	0.005	0.06
(7) HHX	0.008	0.01
(8) Transfer line1	0.0017	0.012
(9) Transfer line2	-	0.00085
(10)Diaphragm	-	0.0034
(11) Orifice	0.000255	-
(12) Reservoir	0.026	0.13

The boundary conditions and initial of all the components are also shown in Table 6. The value of porosity, viscous and inertial resistance coefficients of the regenerator and other porous media according to literature.

Table.6 Boundary and initial conditions for DPTR

Study case	Boundary condition
Compressor wall	Adiabatic wall
Transfer line wall	Adiabatic wall
After cooler wall	293K
Regenerator wall	Adiabatic wall
Cold end wall	Adiabatic wall
Pulse tube wall	Adiabatic wall
HHX wall	293 K
Transfer line1	Adiabatic wall
Transfer line2	Adiabatic wall
Diaphragm,Orifice wall	Adiabatic wall
Rsevoir wall	Adiabatic wall
Viscous resistance (m^{-2})	9.44e+9
Inertial resistance (m^{-1})	76090
Initial condition	300 K
Cold end load	0 W

CHAPTER 5

RESULTS AND DISCUSSION

5.1 Inertance Pulse Tube Refrigerator:

A model validation test is carried out against preceding published cryogenic journal by Cha et al. [9] model to figure out the accuracy of the model and solution method. It is noticeable from the Fig.9 that the steady state temperature is reached after 120s for both the case after which there no change in temperature with respect to time. The Cha et al. the model is reporting a temperature of 87 K using 4200 numbers of cells were in the present case it is found at 86 K using 3900 cells.

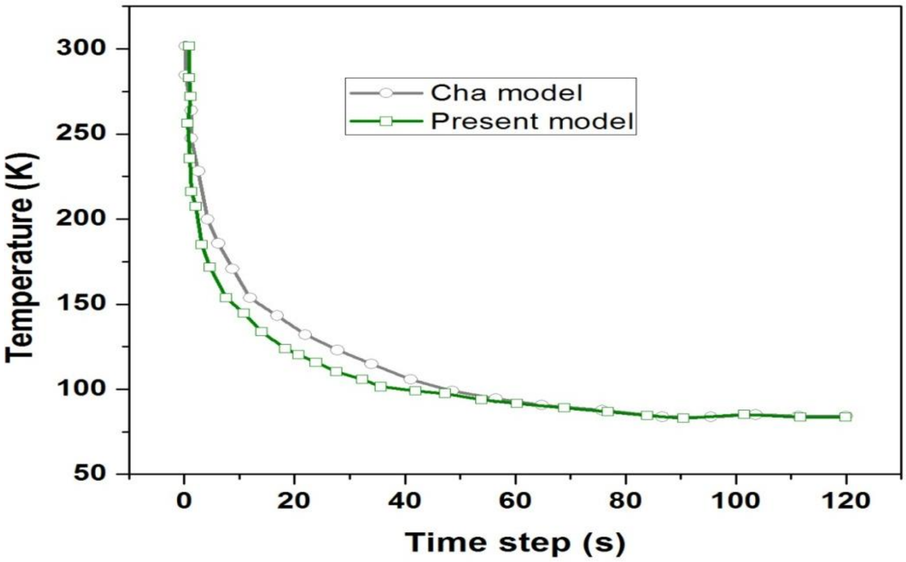


Fig. 9. Validation plot between present model and Cha et al. [11] model

The computational simulation were contineous till a steady periodic saturated state is achived. In this condition the Facate Average (FA) temperature of the cold end achieves a steady state. After this steady state there no change in cold end temperature. In the present model it reaches a temperature of 46 K with the supplied dimension, which is shown in Fig 10. The whole system wall temperature is plotted here. The compressor temperature is above the operating temperature due to compression and expansion of fluid inside it. This extra temperature generated inside the compressor is extracted to surrounding through the aftercooler. Inside the regenerator due to the presence of the extra source term causes momentum loss. In source term the presence of two terms such as Darcy and Forchheimer terms in which pressure drop is directly proportional to velocity inside the porous zone. The temperature drop contour inside the regenerator is shown in the Fig 11. It is observed from the contours that the temperature at cold end is 46.5 K.

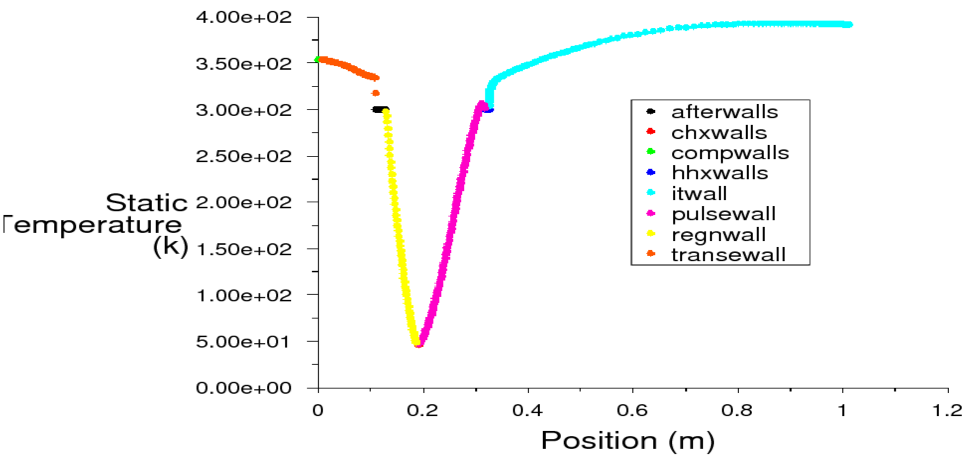


Fig.10. Axial temperature plot varying from compressor to Inertance tube after steady state temperature of 46K at cold heat exchanger

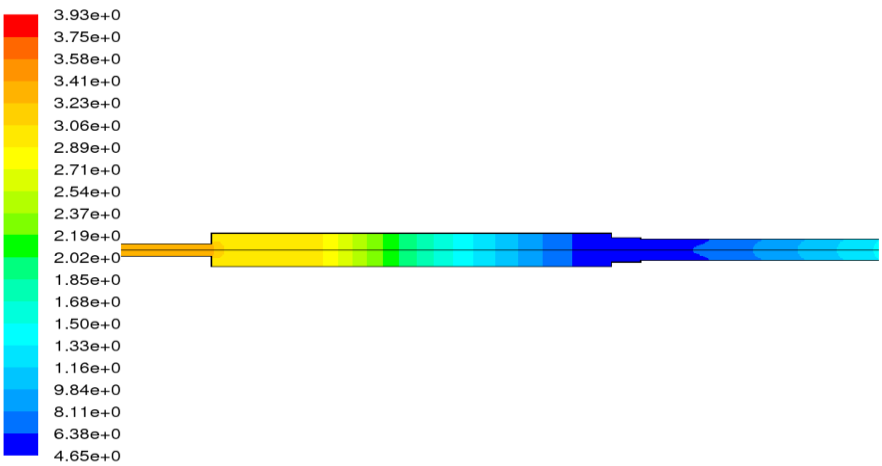


Fig. 11. Temperature Contours showing the temperature variation inside pulse tube.

The Fig [12-16] shows the pressure variation during the simulation inside the whole system. The Area Weighted Average pressure variation inside the cold heat exchanger is shown in the Fig 12. which is sinusoidal in nature. The pressure variation inside the cold heat exchanger from 35.5 bar upto 28.5 bar which can be clear from the figure. Fig 13. Shows the axial pressure variation in the total system from compressor to reservoir when the piston is presented near to the BDC of the cylinder. In that position the compressor pressure is 36 bar and the pulse tube pressure is 35 bar, where reservoir pressure is 32 bar. Fig14. Shows the axial pressure inside the total system from compressor to reservoir when the piston is presently at middle position during expansion. In that position the compressor pressure is 26.5 bar and the pulse tube pressure is 31.5 bar, where reservoir pressure is 32.5 bar. When the piston reaches its top position during expansion process, the system axial pressure variation is shown in Fig 15. It shows that the maximum pressure is inside the reservoir and minimum pressure in the compressor and pulse tube pressure is 28.5 bar. During the compression process from TDC to BDC a pressure variation plot is shown in Fig 16. From the pressure fluctuation analysis during one complete cycle it can be concluded that when there is the maximum pressure at one end at the same time the other end of the system has the minimum pressure. This was the main purpose of the study that to analyze the pressure variation through the whole system during a complete cycle.

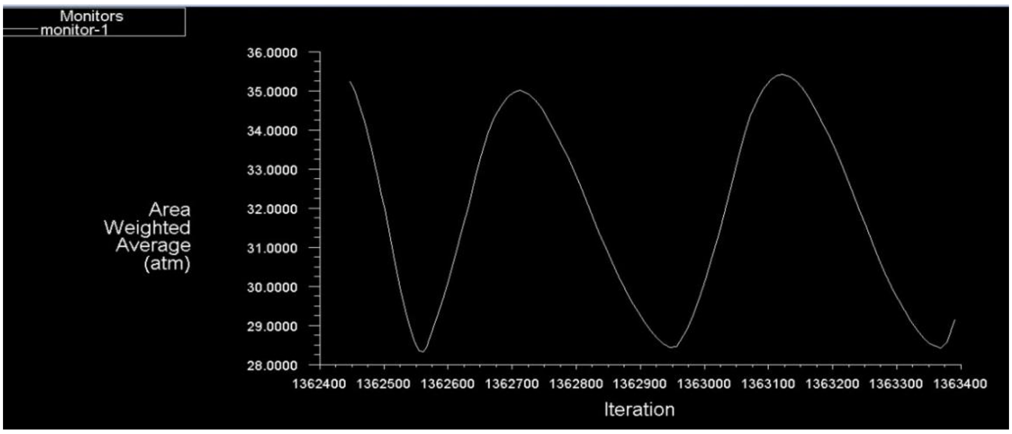


Fig. 12. Area Weighted Average Pressure inside cold heat exchanger vs. iteration curves

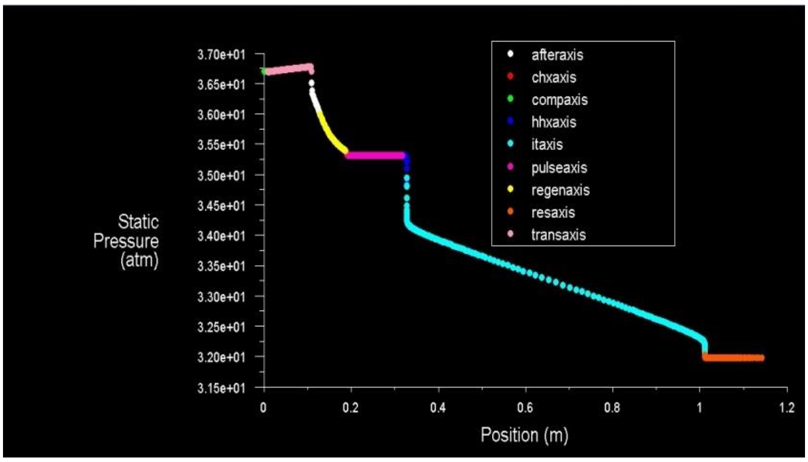


Fig.13. Axial Pressure inside the ITPTR, when piston is near the BDC

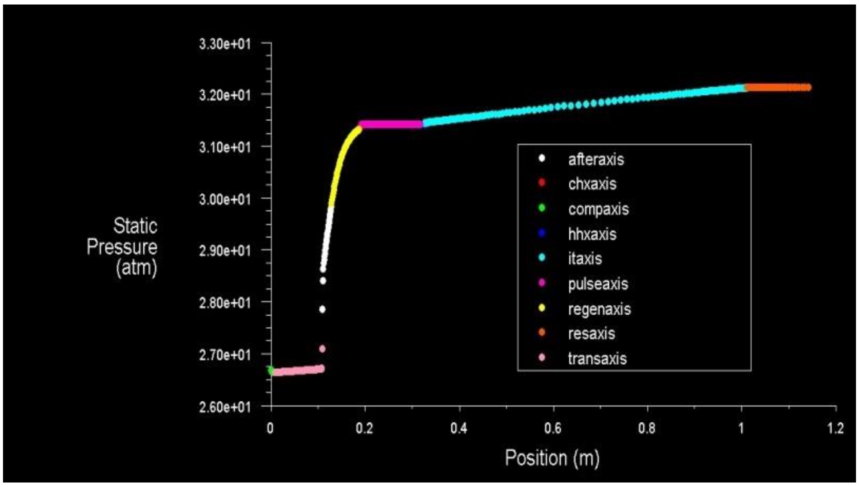


Fig.14. Axial Pressure inside the ITPTR, when the piston is in middle position during compression

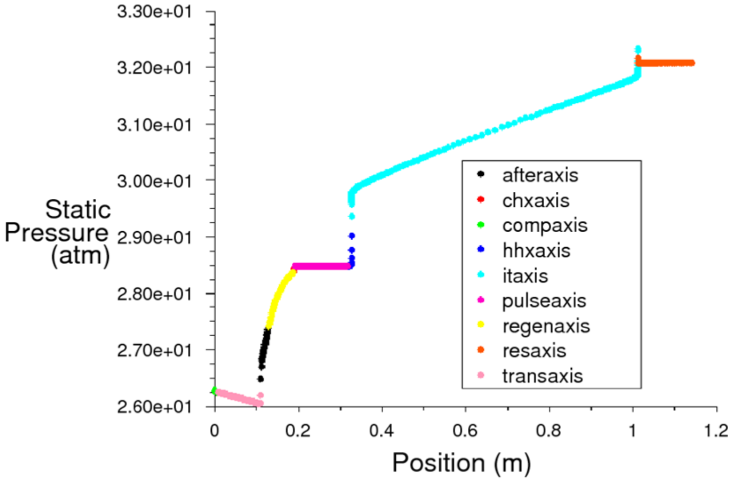


Fig.15. Axial Pressure inside the ITPTR, when piston is near the TDC

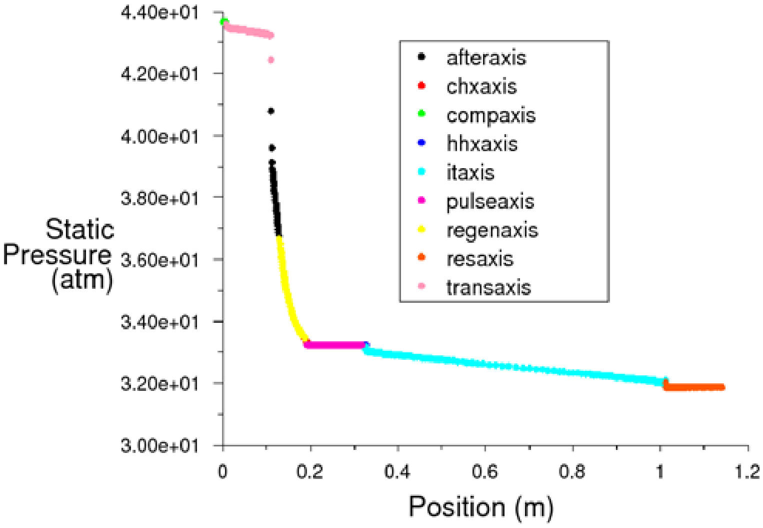


Fig.16. Axial Pressure inside the ITPTR, when the piston is in middle position during expansion

The most important part of the pulse tube refrigerator is a regenerator. The pulse tube refrigerator performance is depending upon the effectiveness of the regenerator. In the present work a number of case studies have been carried out to investigate for which value of porosity it will give a better cooling temperature at the cold end. From the result of simulation it was found that in the porosity value of 0.6 the optimum result found relative to another value which is shown in the Fig 17.

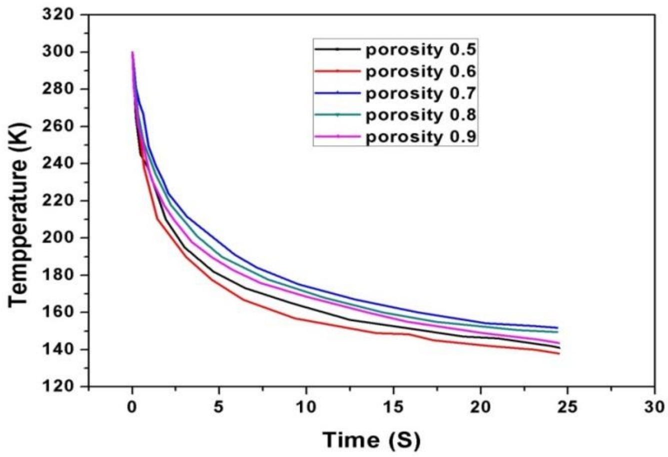


Fig. 17. Cool down temperature vs. time for different porosity inside regenerator

5.2 Orifice Type Pulse Tube Refrigerator:

Validation:

In order to determine the accuracy of the algorithm and the method of solution, a validation study is conducted against the previously published experimental work of Antao et al.[36] with thermal non-equilibrium of regenerator. Both the experiments and the computations show fairly similar trends shown in figure, i.e. lower cycle-averaged gas temperatures at higher values of mean operating pressure of 18 bar.

It is found cold end temperature reaches nearer to 129 K. Fig.19 shows the axial temperature variation from after cooler wall to hot heat exchanger wall because this is the region at which it gives the minimum temperature. The plot is taken after a steady state is achieved. So at any time the temperature remains nearly constant with small fluctuations in any position. Near the cold heat exchanger wall it is found that 129 K where after cooler wall and the hot heat exchanger wall is at 300 K.In Fig.20, it is observed from the contures that the temperature at cold end is 129 K. The gas temperature filled shown one-dimensional structure (a linear temperature variation from the cold end to worm end) with three distinct zones. That is the cold zone near the cold heat exchanger, a warm zone near the hot heat exchanger and a buffer zone in the center which is isolating the two ends.

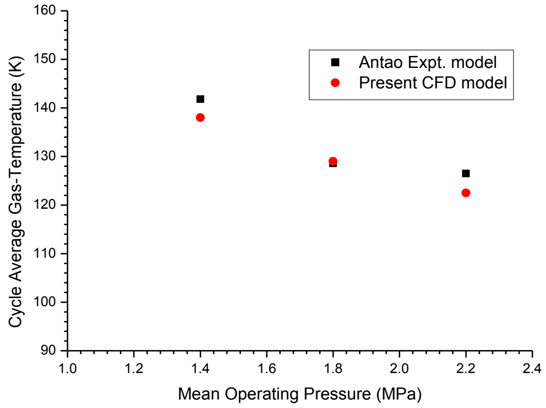


Fig.18 Comparison of near-quasi-steady cycle-averaged experimental

Computational model

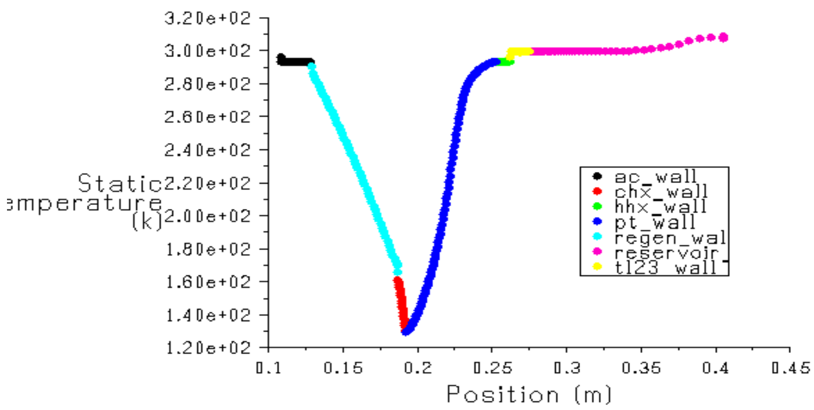


Fig.19 Axial temperature plot varying from after cooler to the reservoir wall after steady state

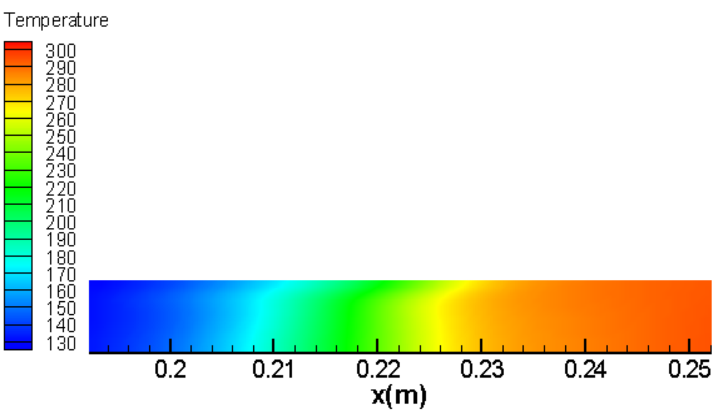


Fig. 20. Temperature Contours showing the temperature variation inside pulse tube

5.3 Diaphragm Type Pulse Tube Refrigerator:

A simulation of the Stirling cryocooler was run. The frequency of reciprocating motion of the piston was specified to be 34Hz. No heat flux was applied to the cold end of the Stirling cryocooler. The residual monitor plot for the above simulation is shown in Fig21. The residual monitor shows no signs of divergence.Fig.22 shows gas temperature decreases rapidly at the start of the iteration ; however the decreasing in temperature exponential over iteration/time. The ultimate cold end temperature of 99.5 K is obtained after steady has been reached.Fig.23 and Fig.24 shows the axial temperature variation from after cooler wall to HHX wall and temperature contour in the puse tube. Fig.25 depicts the density contours respectively under steady periodic conditions on various components. Fig. 26 shows the variation in temperature at the cold end due change of diaphragm amplitude. It shows that when the amplitude is 0.0045m, it gives better cooling performance. If we change this value then the cooling performance changes due to increase in phase angle between the pressure and mass flow rate.

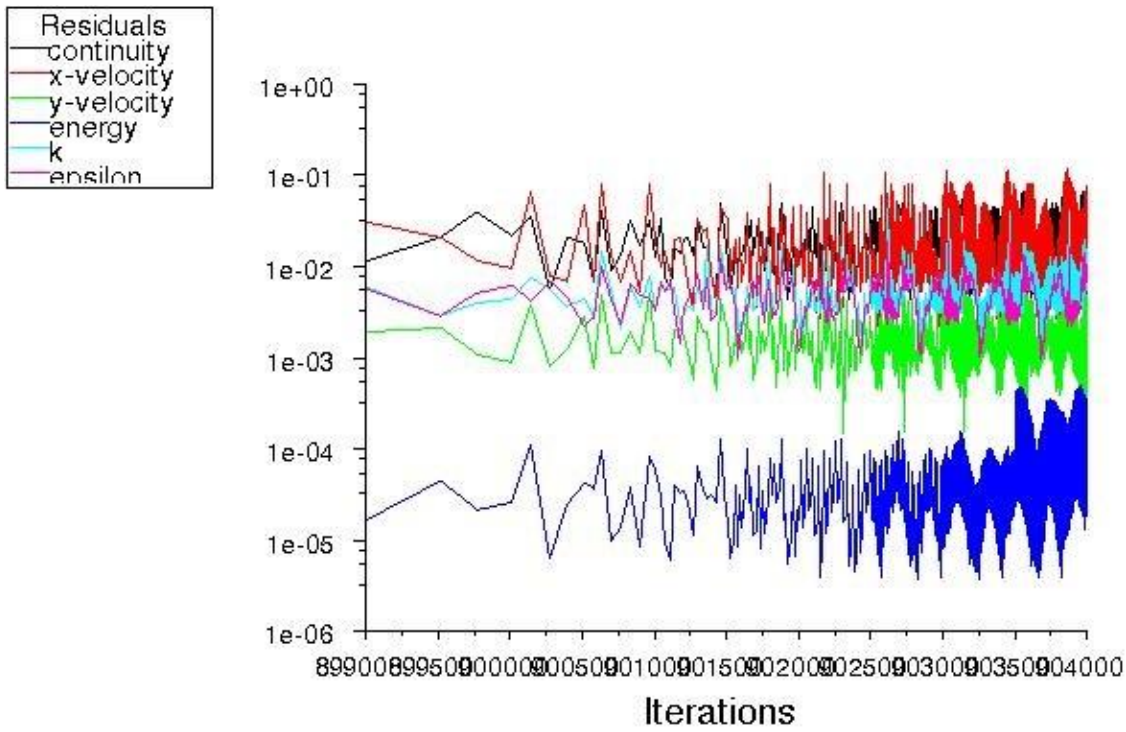


Fig. 21 Residual plot

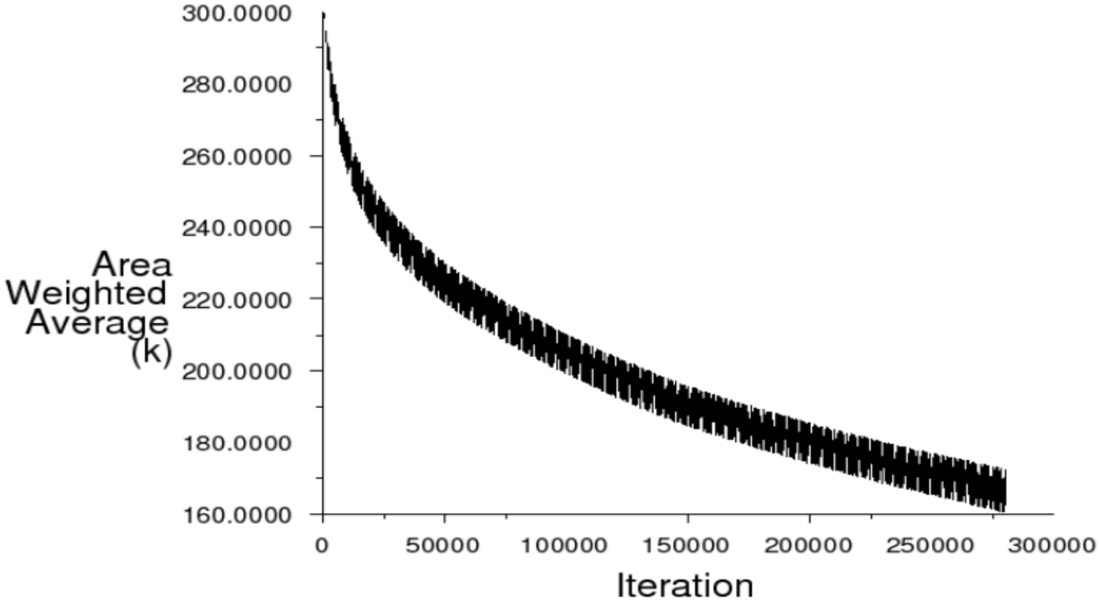


Fig.21. Cooling behaviour at the begining of simulation

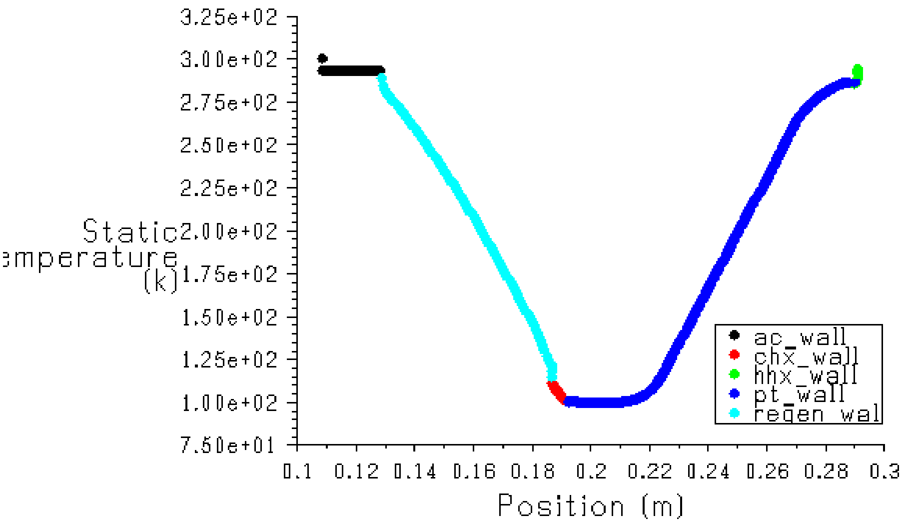


Fig.22. Axial temperature plot varying from after cooler to HHX wall after steady state

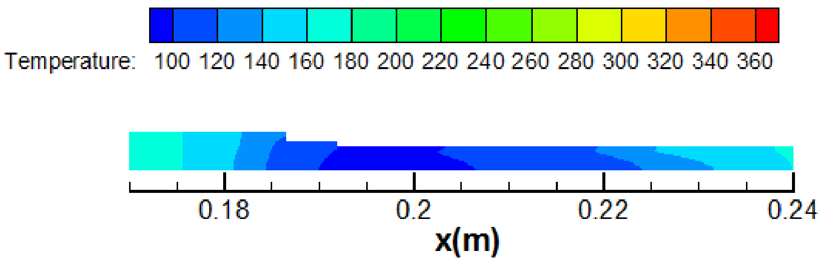


Fig.23. Temperature Contours showing the temperature at cold end of pulse tube

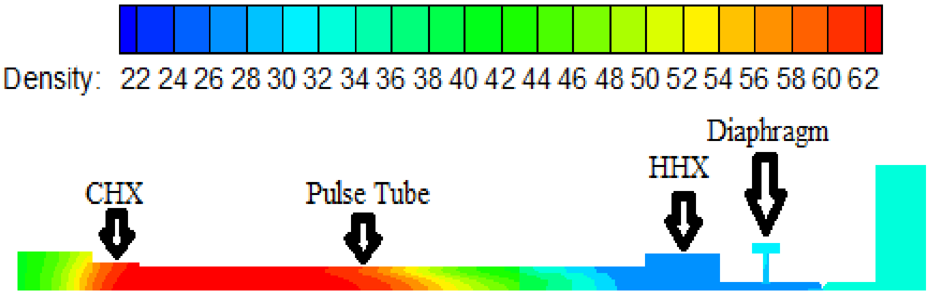


Fig.24 Density contour of DPTR

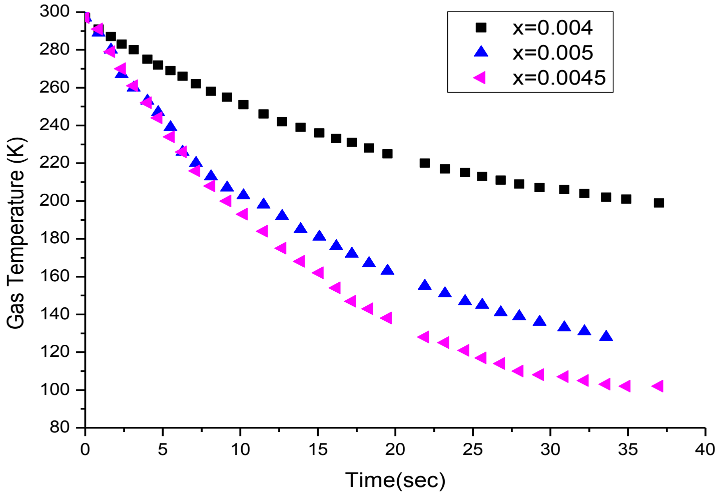


Fig.25. Effect of amplitude on Gas temperature

CHAPTER 6

CONCLUSION

A numerical approach is presented in this work to study the multidimensional flow characteristic of a pulse tube refrigerator. In the first part of the study the results obtained by numerical analysis and previously published paper work have been compared. It can be concluded that the numerical method provides reasonably accurate estimation of responses. Therefore, the method can be adopted to predict the responses before going for actual experiment. It may save time and cost of experimentation. The pressure variation inside the Inertance type pulse tube refrigerator plays a vital role which is discussed in above. The porosity value 0.6 at which it gives a better result. The proposed model can be used for selecting ideal process starts to improve the performance of PTR.

In the second part of study developed an axisymmetric time-dependent computational fluid dynamics model was used to simulate the flow and temperature fields in an OPTR at various values of mean operating pressure. A thermal non-equilibrium model was used to predict the differences in the gas and solid temperatures in the porous media zones. The computational results show trends similar to the experimental [36] performance of the OPTR where higher mean pressures result in lower temperatures.

From the result it is found that DPTR gives lower cooling temperature than OPTR and temperature stability at cold end; which is the main demerits of DIPTR and also there is no DC flow loss due to the absence of DC flow.

Recommendations:

Some possible modification for the study of DPTR recommended further related studies. The modifications are

a. It would be interest to carryout an investigation to optimize the diaphragm shape. This will helpful to give a more accurate results.
b. The above study can be done experimentally and the results of experimental study can be compared with numerical analysis to get more optimize results.

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