CFD Analysis of Stirling Cryocoolers

Thesis submitted in partial fulfilment of the requirements for the degree
of

Bachelor of Technology (B. Tech) In

Mechanical Engineering

By

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Certificate of approval

This is to certify that the project entitled, **"CFD analysis of Stirling Cryocoolers"** being submitted by ***Mr. Anurag Singh*** and ***Mr. Prashant Singh*** has been carried out under my supervision in partial fulfilment of the requirements for the Degree of **Bachelors of Technology (B. Tech)** in Mechanical Engineering at National Institute of Technology Rourkela, and this work has not been submitted elsewhere before for any other academic degree/diploma.

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ABSTRACT

The application of cryocoolers are in various fields of modern day applications for adequate refrigeration at specified temperature with low power input, long lifetime, high reliability and maintenance free operation with minimum vibration and noise, compactness and light weight. The demand of Stirling cryocoolers has increased due to the ineffectiveness of Rankin cooling systems at lower temperatures. With the rise in applications of Stirling cryocoolers, especially in the field of space and military, several simulations of Stirling cryocoolers were developed. In this project, so far a detailed analysis has been carried out on previous developed model with computed lengths of different sections of the cryocooler. The previous models have been developed by solving governing partial differential equations mathematically. The regenerator efficiency had been taken 100% in the previous analysis and with performing Schmidt analyses taking sinusoidal variations of pressure and volume, previous model geometry parameters were calculated. The CFD analysis has been done in this project taking the previous model and a UDF for the movement of the piston has been developed in C++ language. The regenerator parameters have been calculated from the results of the analysis.

Simulations are done on Stirling cryocooler with adiabatic conditions at the cold space side and later with a heat load of 0.25 W at cold end and their results were compared. An attempt has also been made in determining the optimum frequency of operation of the proposed model of Stirling cryocooler by comparing the minimum cool down temperature attained by them with different frequencies of operation.

LIST OF FIGURES:

CHAPTER 1

INTRODUCTION

Cryogenics is the branch of study that deals with physical phenomena that occurs at very low temperature range, which can be the lowest theoretically attainable temperature i.e. absolute zero, 0 K. Cryogenics is a branch of mechanical engineering that has several applications which operates in the temperature ranges below 120 K. In cryogenics the basic processes involves refrigeration, liquefaction, transport & storage of cryogenics fluid & the subsequent behaviour of the participating entities at very low temperatures.

1.1 CRYOCOOLERS – A BRIEF DESCRIPTION

Cryocooler is a cooling machine which operates in a temperature range less than 120 K with a small refrigerating capacity. According to classification given by Walker there are two types of cryocooler: recuperative type and regenerative type. The former includes the joule Thomson cryocooler and Brayton cryocooler [1]. The latter includes the Stirling type and Gifford-McMahon type cryocooler. These cryocoolers are mainly used for cooling of infrared sensors in the missile guided systems and satellite based surveillance, as well as in the cooling of superconductors and semiconductors. The cryocoolers can also be used in other applications such as cryopumps, liquefying natural gases, cooling of radiation shields, SQUID(Super Conducting Quantum Interference Device), Magnetometers, SC Magnets, Semiconductor fabrication etc.

1.2. TYPES OF CRYCOCOOLERS

Cryocoolers can be classified on the basis of operating cycle and heat exchanger.

1.2.1 On the basis of types of operating cycles

Open cycle cryocoolers:

These cryocoolers use cryogenic fluids which are in liquid state and either in subcritical or supercritical state & also solid cryogens which are stored as high pressure gas with a Joule-Thomson expansion valve.

Closed cycle cryocoolers:

Closed cycle cryocoolers provide refrigeration effect at very low temperatures and reject heat at very high temperatures. They are mechanical cryocoolers. A few examples of closed cycle cryocoolers are Stirling cryocoolers, Brayton cycle cryocoolers, closed cycle Joule-Thomson cryocoolers etc. In these there are two working fluids, one which will be working in the cycle and the other one which will be coming in direct contact of the space to be cooled [2].

1.2.2 Types of heat exchangers

1. Recuperative Cryocoolers

These are similar to direct current electrical systems because the direction of flow of the working fluid is unique. The compressor and expander have separate inlet and outlet valves to maintain the flow direction. Valves are necessary when the system has any rotary or turbine components.

In rotary motion of components there are maximum chances for back flow of the working fluid, so in order to avoid that valves are necessary [3]. Working fluid since forms the important part of the cycle, that's why the efficiency of the cryocooler depends upon it. Recuperative cryocoolers can be scaled to any size for specific output which is their main advantage. They are classified into valve less and with valves type of cryocoolers. A few examples of recuperative cryocoolers can be seen in Fig.1 below [2]. Larger volumes of working fluids can be used in these systems because of steady pressure oscillations; as a result these fluids flow anywhere except for locations where there are larger radiation heat leaks due to additional volumes at the cold end. Due to expansion and cooling of the working fluid inside the cryocooler, there can be "pipe cold" at different locations. Vibration of compressor is reduced greatly as the cold end of the cryocooler is separated from the compressor part. Electromagnetic interference is also reduced as there is a lot of distance between cold space and compressor part of cryocooler. There can be traces of oil in the working fluid which are needed to be removed & which can be done from oil removal equipment implanted at the hot end of the cryocooler. Any traces of oil in the working fluid will freeze and will clog the system.

2. Regenerative Cryocoolers

There is oscillatory motion of the working fluid inside these type of cryocoolers in which it oscillates in cycles and while moving through the regenerator part of the cryocooler, working fluid exchanges heat with the wire mesh. These cryocoolers are similar to alternative current electrical system since the working fluid oscillates in the flow channel. The regenerator has a very high heat capacity which stores the heat during one half of the cycle and then in other half it gives it back to the working fluid. These Cryocoolers cannot be scaled up to large sizes but they are very efficient because of their very low heat transfer loss. There are oscillating pressures and mass flows in the cold head and there is a phase relationship between the pressure variations and the mass flows. The oscillating pressure can be generated with a valve less compressor as shown in Fig.1 for the Stirling and pulse tube cryocoolers, or with valves that switch the cold head between a low and high pressure source, as there in a Gifford-McMahon cryocooler. Gifford-McMahon cryocooler has compressor with inlet and outlet valves which are used to generate high- and low-pressure in the system. There is oil lubrication in the cryocooler so it is required to install an oil removal equipment which will be installed at the high pressure line. The use of valves greatly reduces the efficiency of the system. There can be any type of pressure oscillation in a pulse tube cryocooler but they use generally compressors with valves. The compressors which are with valves are modified for air conditioning and they are used primarily for commercial applications. The main heat exchanger in regenerative cycles is called a regenerator. In a regenerator, incoming hot gas transfers heat to the matrix of the regenerator, where the heat is stored for one half of the cycle and then in the second half of the cycle the returning cold gas, flowing in the opposite direction through the same channel, absorbs heat from the matrix and returns the matrix to its original temperature as it was at the start of the cycle. Regenerator is a stacked mesh of wire screens which is having a high heat transfer capacity and generally it is made up of steel.

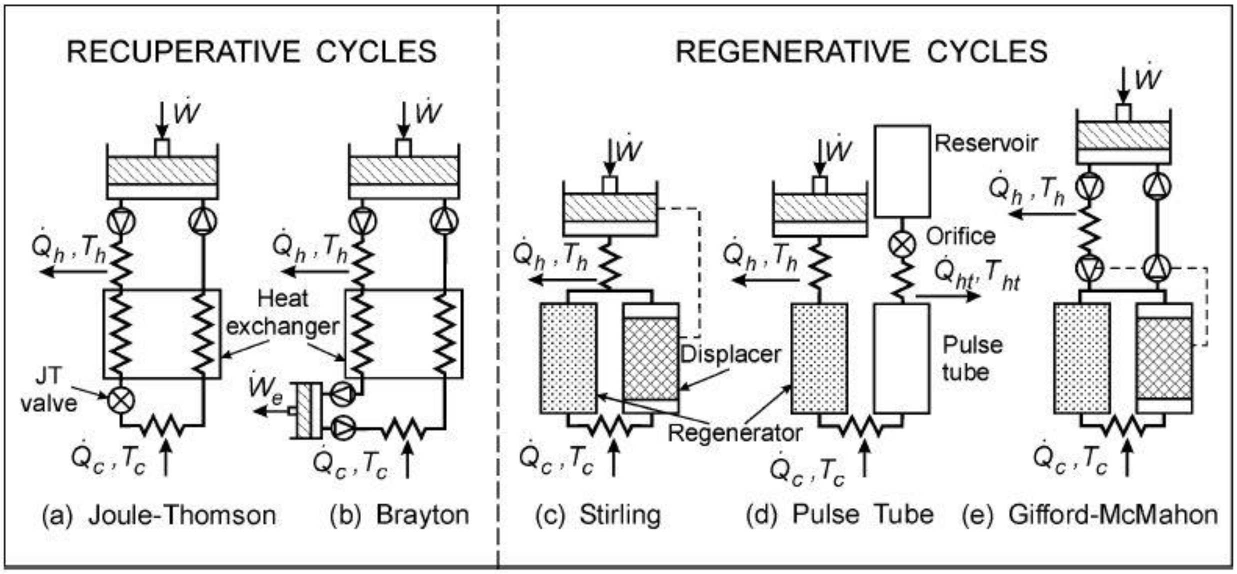


Fig 1: Different types of Recuperative and Regenerative Cryocoolers

1.2.3 Description of Different Cryocoolers

1. Joule Thomson Cryocooler

The Joule-Thomson cryocoolers works on the principle of expansion of gases from high pressure state to low pressure state at constant enthalpy. The expansion occurs with no heat input or production of work, thus, the process occurs at a constant enthalpy. The heat input occurs after the expansion and is used to warm up the cold gas or to evaporate any liquid formed in the expansion process. Enthalpy is independent of pressure at a constant temperature for an ideal gas but for real gases, enthalpy changes with pressure. Thus, cooling in a JT expansion occurs only with real gases and at temperatures below the inversion curve. At very low temperatures the cooling increases for a given pressure change and it's maximum around the critical point. Generally nitrogen and argon are used in joule Thomson cryocoolers. 20 M Pa pressure or more Than that is used on the high pressure side and which is needed to obtain a good cooling effect. Joule Thomson cryocoolers doesn't contain any moving component due to which a rapid cool down rate is obtained in it. Joule Thomson cryocoolers are used in missile guiding systems because of their rapid cool down rate. When joule Thomson cryocooler operates in an open cycle mode it lasts for few days only till the gas evaporates fully from it. Miniature finned tubing is used for the heat exchanger. Gas flows from the high pressure bottle through an explosive valve and then it gets out to the atmosphere.

When joule Thomson cryocooler works in a closed cycle mode, its efficiency is low because of the clogging by moisture at the small orifice. Joule Thomson cryocoolers are nowadays used with mixtures of different gases as its working fluid. This mixture of gases lowers the freezing point of fluid as a whole by having gases with higher boiling points.

2. Brayton Cryocoolers

Brayton cryocoolers are also referred as reverse-Brayton cycle as cooling occurs by the expansion of gas and by which it does work. According to the First Law of Thermodynamics the heat absorbed with an ideal gas in the Brayton cycle is equal to the work produced. This process is then more efficient than the JT cycle and it does not require very high pressure ratio. Brayton cryocoolers are generally used in large liquefaction plants. Turbines with approximately of 6 mm of diameter on shafts of 3 mm diameter are those systems which were reviewed by McCormick *et al.*8 for space applications of cooling of infrared sensors. Turbines which are having speeds of 2000 to 5000 rev/s are generally used in Brayton cryocooler system. Centrifugal compressors which are providing a pressure ratio of approximately 1.6 with a low side pressure of 0.1 M Pa are used with these systems. The working fluids which are used in the turbo-Brayton cryocoolers is usually neon when operating above 35 K, but helium is used for lower temperatures. Brayton cryocoolers are having very low vibration of their rotating parts in a system with turbo expanders and centrifugal compressors. Low vibration of Brayton cryocoolers make them suitable for use with sensitive telescopes in satellite applications.

3. Pulse Tube Cryocoolers

In a pulse tube cryocooler there is a proper gas motion in phase with the pressure which is achieved by the use of an orifice or an inertance tube along with a reservoir volume to store the gas during a half cycle. The reservoir volume diminishes any pressure oscillations during the oscillating flow. This oscillating flow separates the cooling and heating effect. For a given frequency there is a limit on the diameter of the pulse tube in order to maintain adiabatic process. There are four steps in its cycle of operation. First the piston moves down to compress the gas in the pulse tube. Secondly this heated gas due to its higher pressure it moves into the reservoir and exchanges heat with the ambient through the heat exchanger at warm end of the pulse tube. Thirdly then piston moves up and expands the gas adiabatically in the pulse tube. Fourthly this cold, low pressure gas is pushed through the cold heat exchanger by the gas from the reservoir and this stop when the pressure in the reservoir increases to the average pressure. Regenerator cools the incoming high pressure gas before it reaches to the cold end. The gas in between the pulse tube insulates the two ends as it never leaves its position and creates a temperature gradient between the two ends. The gas in between minimizes the turbulence. Pulse tube transfers acoustic power in an oscillating gas system from one end to the other across a temperature gradient with minimum power dissipation and entropy generation. If different changes are made in geometries of pulse tube cryocooler then it increases the lower temperature limit.

1.3 APPLICATIONS OF CRYOCOOLERS

1.3.1 Military applications

- IR sensors for missile guidance and night vision
- IR sensors for surveillance(satellite based)
- Gamma ray sensor for monitoring nuclear activity
- Superconducting magnets for mine sweeping

1.3.2 Environmental

- IR sensors for atmospheric studies
- IR sensors for pollution control

1.3.3 Medical applications

- Cooling superconductors for MRI
- Cryosurgery

1.3.4 Commercial applications

- Cryopumps
- Industrial gas liquefaction
- Cooling superconductors for cellular phone base stations

1.3.5 Transport applications

- Superconducting magnets for maglev trains

1.4 STIRLING CRYOCOOLER

Stirling cryocooler is a regenerator type cryocooler. In general there are three configurations viz. alpha, beta & gamma type of Stirling cryocoolers. A Stirling cryocooler operates on Stirling cycle. The engine used helium as the working fluid. An ideal Stirling cycle consists of two isothermal processes and constant volume processes alternatively. The cycle takes place in anticlockwise direction. The isothermal processes are of contraction and expansion whereas constant volume processes are of heat addition and rejection.

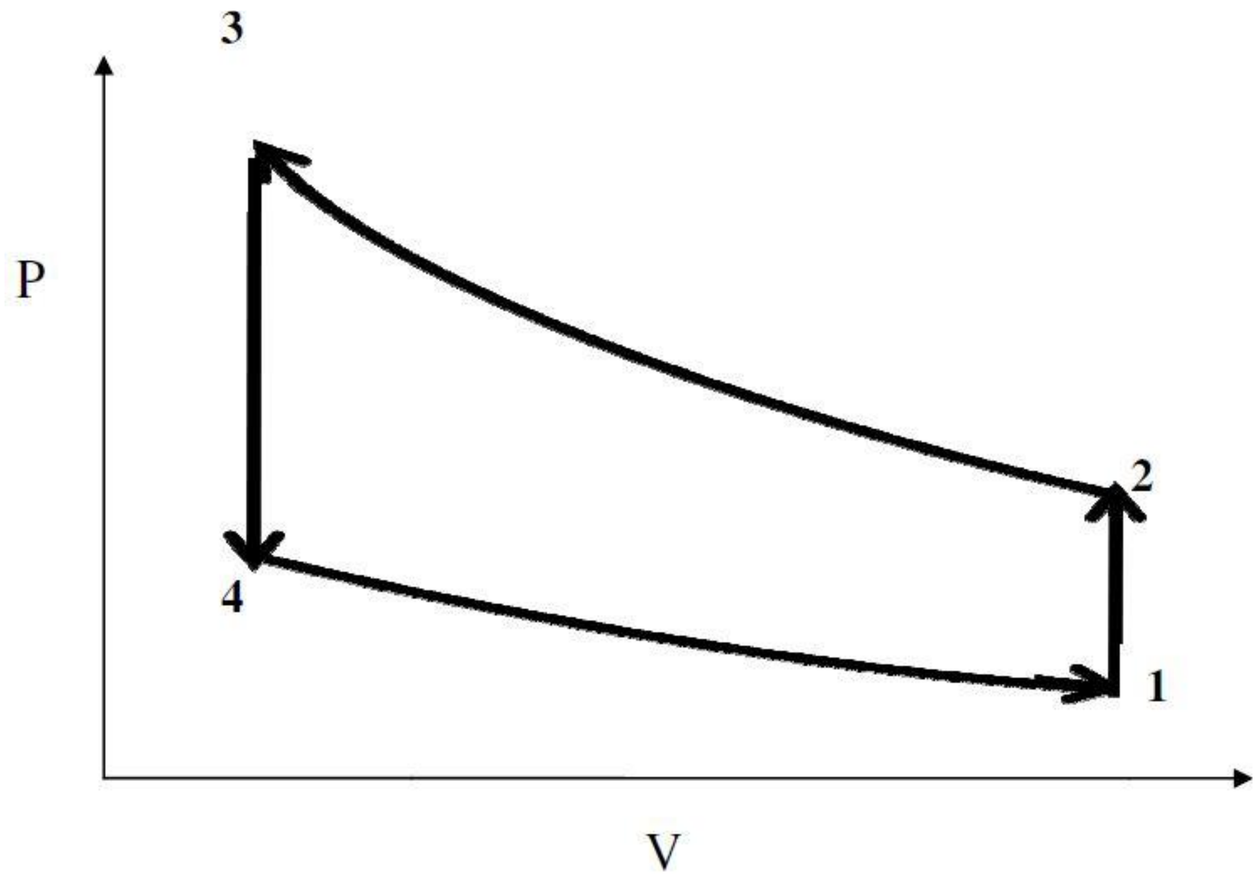


FIG 2: Thermodynamic Process of an Ideal Stirling Cycle [3]

The four processes can be explained as follows:

- Isothermal Expansion:

  The gas in the cold heat exchanger space expands as the gas from the hot heat exchanger travels through the regenerator towards the cold end at constant temperature. During this process the fluid exchanges heat with the regenerator as it gives away its heat to regenerator to store it.

- Isochoric heat Addition:

  The cold end is exposed to the space where the cooling effect is to be produced, so in this stage gas absorbs heat away from the space by keeping the volume of the cold space constant.

- Isothermal Compression:

  The gas from the cold end is now compressed and then it travels through the regenerator towards the hot end of the cryocooler, during its course it takes away heat which was stored in the regenerator.

- Isochoric heat rejection:

  The gas in the hot end is now having the heat from the regenerator and the space to be cooled. The gas now in compressed state rejects heat to the ambient.

1.4.1 Description of Stirling Cryocooler

Its main components are compressor with piston, after cooler, transfer line, hot heat exchanger, displacer (regenerator) and cold heat exchanger. Piston and Regenerator are the moving components and rest are stationary.

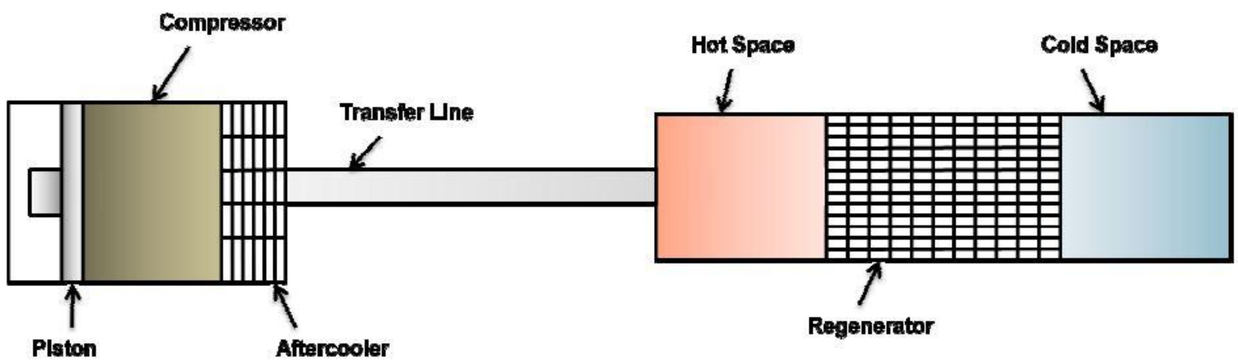


FIG 3: Schematic representation of Stirling cryocooler [3]

1.4.2 Working of Stirling Cryocooler

From above Fig.2 of Stirling cycle if the gas is at the maximum volume condition, then the gas absorbs heat from the space to be cooled while regenerator remains stationary and piston was moving towards left. In the next step the displacer moves to the right and thus forces the fluid to pass through it and while moving it absorbs heat from the regenerator at constant volume condition as the piston remains stationary. Next the piston compresses the working fluid as it moves right and this compression at the hot end is isothermal so heat is given off to the surroundings at the ambient temperature. Next the regenerator moves to left thus forcing the fluid to pass through it and during this fluid gives away heat to the regenerator while maintaining constant volume condition as piston doesn't move, so it's an isochoric heat rejection process.

Next the piston moves towards left and thus causing the fluid to expand. A pressure oscillation in the system will cause temperature to oscillate and produce no refrigeration. The motions of two parts are sinusoidal in nature and displacer motion is 90 degrees out of phase from that of piston motion. With this condition the mass flow or volume flow through the regenerator is in phase of pressure [2]. The moving piston causes both compression and expansion of the gas & net power Input is required to drive the system. The moving displacer reversibly extracts net work from the gas at the cold end and transmits it to the warm end where it contributes some to the compression work. In an ideal system, with isothermal compression and expansion and a perfect regenerator, the process is reversible [2].

Q_c represents heat which is being rejected from the compressor during isothermal compression, W_c is the work input which is equal to Q_c and this is given by [4].

$$Q_c = mRT_c ln\left(\frac{V_{max}}{V_{min}}\right) = W_c$$

Qe is the amount of heat being absorbed into the expander during isothermal expansion [4].

$$Q_e = mRT_e ln\left(\frac{V_{max}}{V_{min}}\right) = W_e$$

Then Carnot efficiency of the cryocooler is:

$$h = \frac{Q_e}{W} = \frac{Q_e}{W_c - W_e}$$

$$h = \frac{T_e}{T_c - T_e}$$

CHAPTER 2

LITERATURE REVIEW

In 1816, Dr. Robert Stirling patented a concept which states that heat can be converted to work on a hot air engine through repeated compression and expansion of the working fluids at different temperature levels. This is the base line of the research done in refrigeration till date. Stirling cryocoolers were introduced to the commercial market in 1950 for the first time as single cylinder air liquefiers and cryocoolers for IR sensors to about 80K. In the 1960s, William Beale invented free-piston Stirling engines acting as power systems and has been in continuous development process since then. The first linear free piston cryocooler was developed at Phillips laboratory Eindhoven.

In the 1970s, Development of linear Stirling cryocooler by Dr. G Davey, Oxford University which resulted in developments of many new components such as no contact seals, flexure springs, linear compressor etc [5]. In a typical Oxford type Stirling cryocooler the regenerator comprises of long stack of phosphorus bronze discs, the frequency of piston and displacer is usually very high in this case and is around 30 to 60 Hz. The working fluid has a very complex oscillatory motion [6]. In the 1980s, The US navy started a program, in which new designs for cryocoolers were solicited which would have a coefficient of performance within a range of 2000 to 10000 W/W [7]. In the year 1997, AFRL, Raytheon and JPL collaborated and developed a Stirling cryocooler operating in the range of 25K-120K [8]. In the same year, a light weight linear driven cryocooler was also developed for cryogenically cooled solid state Laser systems which were developed by Decade Optical Systems, Inc. (DOS Inc.) [9].

Barrett et al. has used a commercial computational fluid dynamics (CFD) software package to model the oscillating flow inside a pulse tube cryocooler. The variation of temperature and velocity vectors is represented by a 2-D axis-symmetric model along with the addition of heat flux at the cold and hot end heat exchangers [10].

Yarbrough et al. have modelled the pressure drop using CFD through wire mesh regenerator. They developed a model with the use of computational fluid dynamics for the prediction of pressure drop through the wire mesh of regenerator which eventually facilitated the development of a cryocooler system model [11].

Anjun et al. have done both the experimental and numerical study to determine the characteristics of heat transfer for cryogenic fluid helium by analyzing the physical properties which are temperature dependent in a miniature tube. They found that the heat transfer characteristics of cryogenic gas with temperature-dependent thermo physical properties (TDTP) are different from those in the ambient condition with constant thermo physical properties [12].

The earlier work done by Martini [13] was taken in practice at the University of Calgary to carry out simulations on Stirling cryocoolers using computer codes written in FORTRAN 77 to create a way to use digital simulation program for microcomputers, CRYOWEISS by Walker et al [14]. The program was used to predict the results of PPG-1 Stirling liquefier and also comparing it with the experimental results.

A CFD code, CAST (Computer Aided Simulation of Turbulent flows) was developed by Peric and Scheuerer in 1989 [15]. This computer program was used for analyzing a two dimensional unsteady state heat transfer phenomena. The code was written in FORTRAN IV. The code was widely accepted throughout the world but used mainly by the graduate students of Cleveland State University for the simulation of various components of Stirling machine.

CHAPTER 3

COMPUTATIONAL FLUID DYNAMICS

3.1 Introduction

Computational Fluid Dynamics deals with the numerical solutions of fluid flow and heat and mass transfer problems by modelling the equations for particular cases and by applying several assumptions and numerical solution techniques by which it provide us the solution. Though the solution obtained through computational modelling can't be considered as the perfect one as there are several discrepancies that might occur during simulation such as the assumptions, all the situations go in an ideal situation. The best way to study about a particular problem is by an experimental method as it gives the actual insight of what the actual situation is. No simulation gives better results than experimental methods but it still has its pros and cons. Experimentations may involve certain errors such as measurement errors which are very much significant when we deal with the nano and micro fluidics. Also cost is another most important factor that the researchers have to look after. There are some experiments which are very costly and thus they are not feasible enough to be performed in the large scale. Some use the scaled methods of experimentation but it also results in the discrepancy in the results. Another method of simulation is the numerical method. This is the most exact method to get the accurate results of literally any fluid flow problems. But the limitation of this method is that it is very much limited and can be applied only to special cases. So we see that the computational methods gives better results as compared to the other options where we have to optimize the cost and time of experimentation by compromising over the results under the safe limits of error. The most straightforward method to solve fluid flow problems is to solve the Navier Stokes equation with appropriate boundary conditions. CFD provides approximate solutions to the governing equations of fluid motion. Application of the CFD to analyze a fluid problem requires the following steps. Firstly, the mathematical equations which are the partial differential equations of second or even higher order which govern the fluid flow are modelled. These equations are then discretized using several discretization schemes based of Finite Difference Methods, Finite Volume method and Finite element analysis. The domain is divided into small cells or grids which are called meshes which can be triangular, quadrilateral or combination of both. In the final step the operating conditions and the boundary conditions are specified in order to get the solutions based on our requirements. The solution is iterative in nature and there are several controlling factors such as the least amount where we need our solutions to converge. Less the limit given, more exact will be the solution and more time it will take to solve.

The CFD simulation procedure contains three steps:

(1) Pre-processor: In this phase, geometry of the flow domain is made and the zones and boundaries are given specific types. Then the problem is setup by providing the operating conditions and boundary conditions.

(2) SOLVER: In this phase the differential equations are discretized and solved through various discretization schemes.

(3) POST PROCESSOR: The results obtained after the solution are compiled together and studied in order to arrive at a general consent.

3.2 CFD Analysis of Stirling Cryocoolers

Governing Equations:

a. Mass Conservation Equation:

$$\frac{\partial \rho}{\partial t} + \nabla.\left(\rho \vec{V}\right) = 0$$

b. Momentum Conservation Equation:

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}\left(\rho u_i u_j\right) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i}(\lambda + \mu)\nabla.\vec{V} + \frac{\partial}{\partial x_j}\left(\mu \frac{\partial u_i}{\partial x_j}\right) + \rho b_i$$

c. Energy Conservation Equation (For incompressible flow):

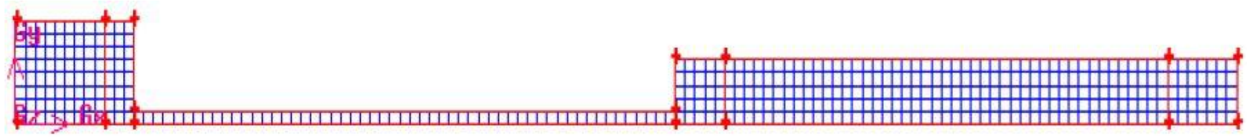
$$\rho C_p \frac{DT}{Dt} = -\left(\frac{\partial \dot{q}_x}{\partial x} + \frac{\partial \dot{q}_y}{\partial y} + \frac{\partial \dot{q}_z}{\partial z}\right) + \dot{q}_h$$

3.3 MODELING THE GEOMETRY

3.3.1 GAMBIT:

A 2 dimensional geometry of Stirling cryocooler is created in GAMBIT. The regular rectangular meshes have been taken. An axis-symmetric model has been made in order to reduce the time consumption in simulation. The main components of the design are Piston, Compressor, after cooler, transfer line, hot space, cold space, regenerator and displacer. The dimensions of various components are shown in the following table and the subsequent figure which are taken from [16].

Sl. No.	Components	Radius(in mm)	Length (in mm)
1	Compressor	6	11
2	After cooler	6	4
3	Transfer line	1	39
4	Hot Heat Exchanger	3	9
5	Regenerator	3	75
6	Cold Space	3	11
7	Piston	6	_
8	Displacer /Cold Piston	3	_



Axis-symmetric model of the Stirling cryocooler developed in GAMBIT

3.3.2 FLUENT

Fluent offers wide variety of freedom to solve any type of flow problems one could possibly come across. It provides tools and accessories to devise new technologies and also to modify or optimize the existing ones. The technology helps us to design tough simulations and products and to test them how they will be behaving in real world.

3.3.3 SETTING UP THE PROBLEM IN FLUENT:

The mesh file is incorporated in FLUENT after defining proper boundary types and zones to the geometry. The mesh is being read by the FLUENT and the grid sizing came out to be 420 cells, 996 faces, 577 nodes and 1 partition. The whole problem is then designed based on the boundary conditions and operating conditions prevailing in actual conditions for which we are designing the simulation. For this study, the above mentioned grid was scaled by a factor of 0.001 in order to convert the units to mm.

The detailed inputs given are listed as below:

Details of Solver	
Solver	Segregated
Formulation	Implicit
Space	Axis-Symmetric
Time	Unsteady
Velocity Formulation	Absolute
Unsteady formulation	1^{st} order implicit
Gradient Option	Cell Based
Porous Formulation	Physical velocity
Energy Equation	On
Details of Viscous Model	
Model	k-epsilon (2eqns)
k-epsilon model	standard
Material	Helium, air and steel
Operating Conditions	1 atm

Solution Controls	
Equations	Flow, Turbulence and Energy
Under relaxation factors	Pressure= 0.1 Density- 1 Body Forces- 1 Momentum- 0.6
Pressure velocity coupling	PISO
Skewness correction	1
Discretization scheme	
Pressure	PRESTO!
Density	First order upwind
Momentum	First Order Upwind
Turbulent Kinetic Energy	Second Order upwind
Turbulent dissipation rate	Second Order upwind
Energy	Second Order upwind
Solution Initialization	
Computation from	All zones
Gauge pressure	20 atm
Axial Velocity	0 m/s
Temperature	300 K
Turbulent kinetic energy	1
Turbulent dissipation rate	1

Pressure limits	5 atm – 35 atm
Residual Monitors	Convergence criterion
Continuity	1e-06
x-velocity	1e-06
y-velocity	1e-06
energy	1e-06
k	1e-06
epsilon	1e-06

Boundary conditions:

After cooler and hot space were chosen to have isothermal walls while rest of the walls were chosen to be adiabatic with either zero heat flux or a certain heat flux depending upon the case to be simulated. After cooler and regenerator are considered to be porous zones. The detailed boundary conditions are mentioned below.

• Piston: The piston was chosen to be an adiabatic wall and its material was steel.

• Compressor: The walls were chosen to be made of steel and were adiabatic. The compressor space was filled with Helium.

• After cooler: The walls were chosen to be isothermal, maintaining a constant temperature of 300K. The interior of the after cooler was chosen to be a porous zone. Details of the porous medium are given below which are taken from [17].

Porosity: 0.7

Viscosity: X: 9.433e+09 1/m^2; Y: 9.433e+09 1/m^2

Inertial *resistance*: X: 76090 1/m; Y: 76090 1/m

The material of the porous zone and the walls were chosen to be steel and the void spaces were filled with Helium.

• Transfer line: The walls of the transfer line are adiabatic and are made of steel. Its interior is filled with Helium.

• Hot space: The walls of the hot space are isothermal maintaining a constant temperature of 300K and is made of steel. The hot space is filled with helium.

Regenerator: The regenerator is chosen to be a porous medium with the following details [17]

Porosity: 0.7

Viscosity: X: 9.433e+09 1/m^2; Y: 9.433e+09 1/m^2

Inertial resistance: X: 76090 1/m; Y: 76090 1/m

The material of the porous zone and the walls were chosen to be steel and the void spaces were filled with Helium.

• Cold space: The space is filled with Helium and its walls are adiabatic in nature. The material of the cold space walls is steel.

• Displacer: The displacer is made of steel and is adiabatic in nature.

Description of work:

Two user defined functions for the motion of compressor and cold wall have been compiled for the sinusoidal motion between them with a phase angle of 100 degrees. There are three governing equations for this problem- Flow, Turbulence and Energy. The under relaxation factors are taken in order to get faster simulations. The solution is initialized from all zones with initial gauge pressure of 20 atm. We have been monitoring the temperature of cold wall and the temperature of cold fluid. The size of the time step was taken as 0.0007.

Initially the solutions were converging at around 100 iterations and 1 second in real time took around 2 days in virtual simulation. So keeping the time factor in mind the simulation was run for 8 days at 90 iterations for a real time run for 5 seconds and the solutions till this point converged completely. After the temperature of the cold fluid dropped to 200 K we have reduced the maximum iterations per time step to 50 and then 10 for faster simulations.

The temperature of cold fluid has been dropping exponentially to 55 K where it became asymptote to the time axis at around 73.5 second's real time. But the temperature of cold wall was still dropping continuously as the temperature of the fluid in contact with it is very less. The reason for such a big temperature drop can be accounted for the fact that we have operated the cryocooler at no load condition so the results obtained so far are under ideal conditions.

Now we are attempting to introduce heat flux of 250mW and see the changes in the refrigeration effect and the minimum temperature achieved by the cold fluid.

3.3.4 Motion of Piston and displacer

The motion of piston and displacer is governed by the User Defined Functions which are the codes written in language C, and are incorporated into fluent by using VISUAL BASIC 2010. The sinusoidal motion is considered and the piston and displacer are happened to maintain a phase angle of 100 degree during their motion. The proper phase angle is considered while writing the code. The codes were formed after research and some study from the FLUENT UDF Manual [18].

CHAPTER 4

RESULTS AND DISCUSSIONS

There are steady periodic simulations of CFD & results of which are presented & discussed in this section later. The simulations were done in fluent and the geometry of the Stirling cryocooler is made in GAMBIT. Different cases were taken into consideration by varying the frequencies, boundary conditions so that an optimum solution can be found. The geometry remains same during the simulations and only the existing conditions were altered. In the first simulation, adiabatic or no load case is taken at the cold end. In the subsequent simulation, we have taken isothermal condition with a heat load of 250 mW and then isothermal condition with different speeds of reciprocating piston. The simulations with load at cold end were done at different frequencies of 20 Hz, 30 Hz, 40 Hz and 50 Hz. No load condition at cold end means there is no heat flux through cold wall and there was one more case in which there was simulation with heat load of 250 mW at the cold end.

4.1 NO LOAD CASE WITH FREQUENCY OF 20 HZ

In this the simulation with Stirling cryocooler geometry was run and the frequency of reciprocating piston and displacer was kept at 20 Hz. No heat flux was applied to the cold end of the cryocooler. The residual monitor plot for above case is as shown below in the Fig.4 below.

4.1.1 COOLING BEHAVIOUR

The cooling behaviour of the cold end of the Stirling cryocooler is like a sinusoidal curve which is constantly decreasing. The temperature of the cold space wall and cold fluid decreases constantly. It looks like as it is given below in Fig.5.

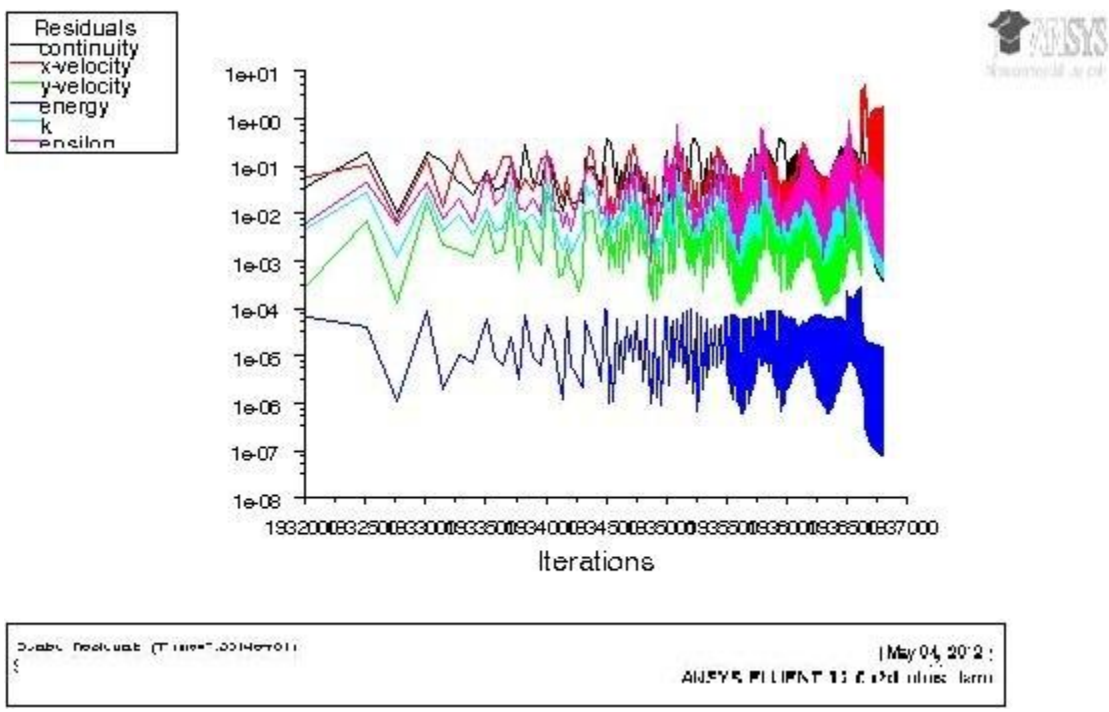


Fig 4: Residual Monitor Plot for simulation with no load condition with 20hz frequency

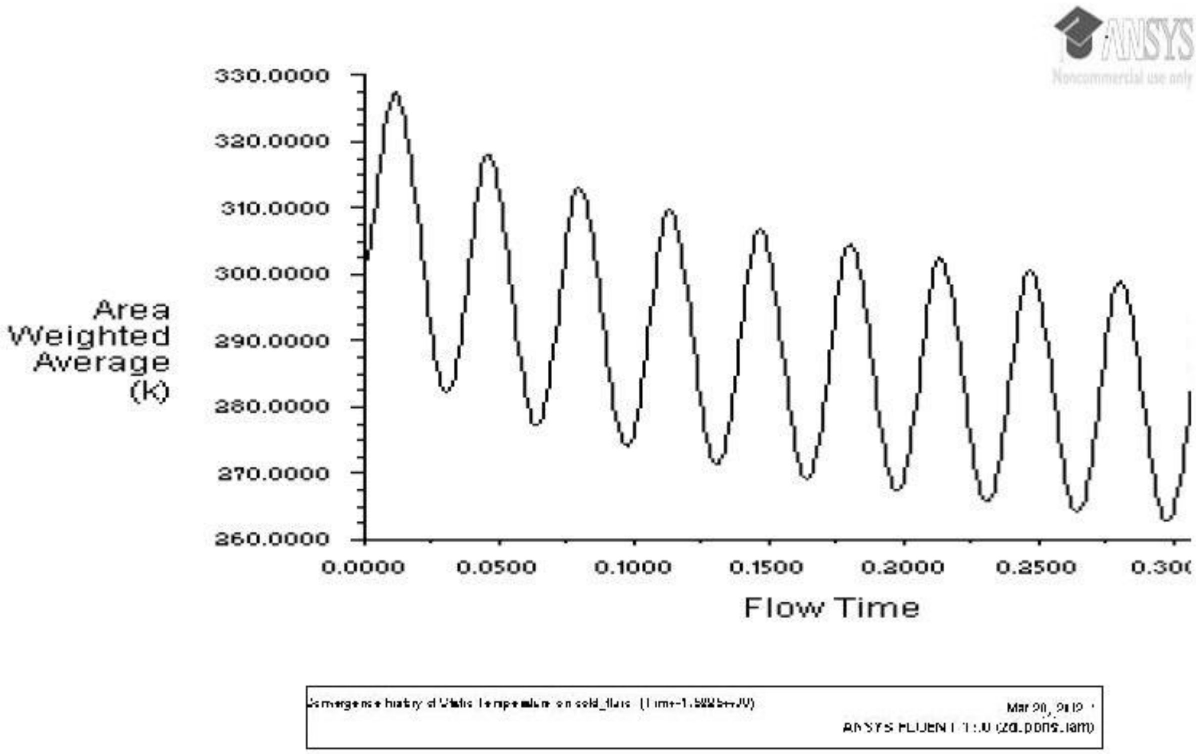
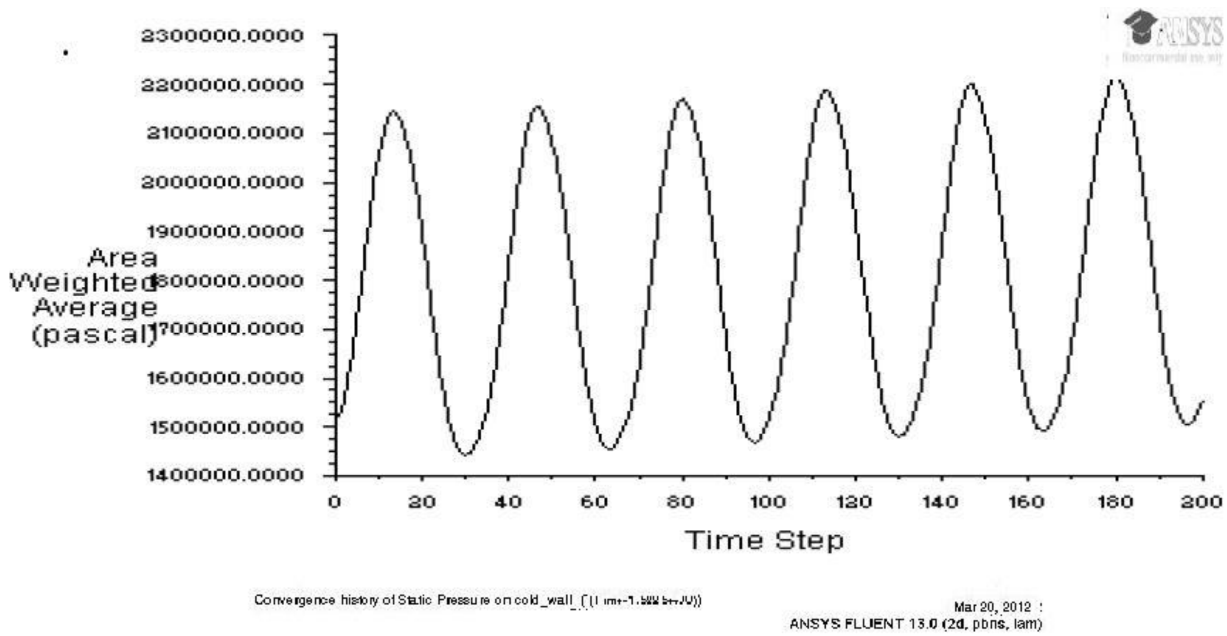


Fig 5: Sinusoidal variation of temperature of cold space

4.1.2 PRESSURE BEHAVIOUR

The pressure wave generated in the cold space and its sinusoidal variation is due to the reciprocating motion of the compressor piston. Its variation is shown below:



4.1.3 COOLING CURVE

The temperature of the cold space falls at a faster rate initially but at later stages it falls gradually and system reaches a steady state. The final temperature of steady state was reached in 73.5 s of simulation of cryocooler.

4.1.4 TEMPERATURE CONTOUR

The temperature gradient in the Stirling cryocooler in 73.5 seconds of simulation can be seen in Fig.6 given below along the cryocooler. The highest temperature is obtained at the compressor end and while the lowest temperature is obtained at the cold end of the cryocooler. The temperature at the left side of the regenerator is higher and it gradually decreases as we move towards cold space.

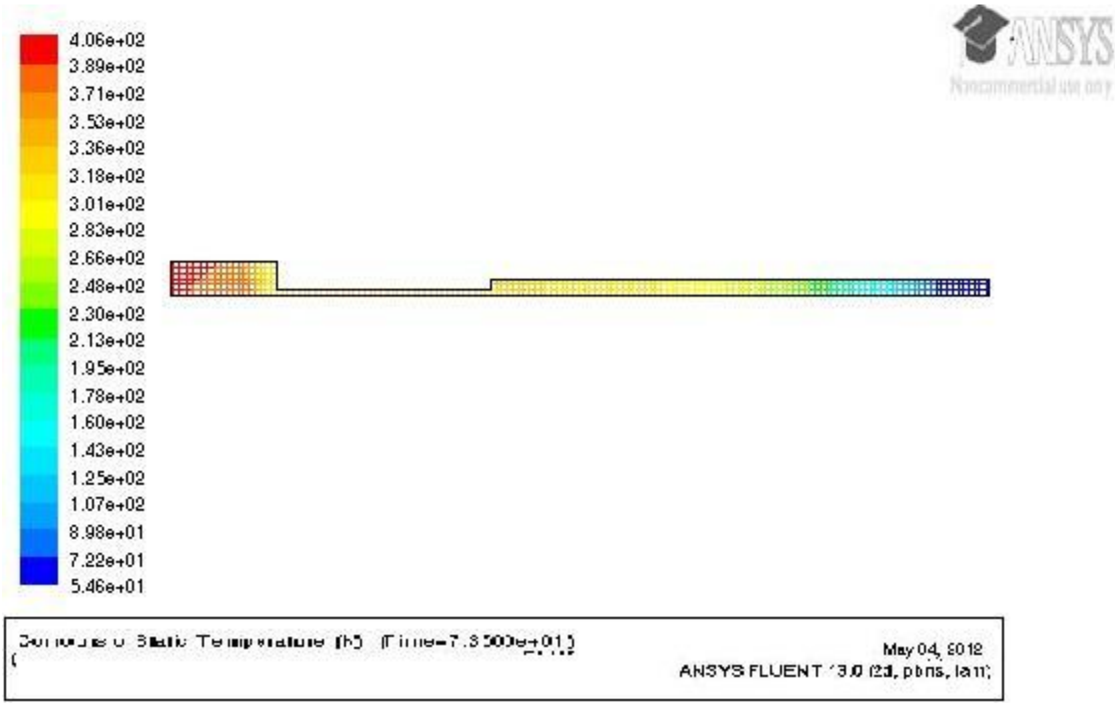


Fig 6: Temperature contour of Stirling cryocooler with no load condition at 20 Hz

4.1.5 TEMPERATURE PLOT

Following figure (Fig.7 below) shows the temperature profile along the Stirling cryocooler length. Compressor position shows the highest temperature whereas the cold space position shows the lowest temperature.

Pressure Plot – The plot of Fig.8 shows the pressure gradient along the whole length of the Stirling cryocooler and it was observed that the pressure decreases along the length of the cryocooler.

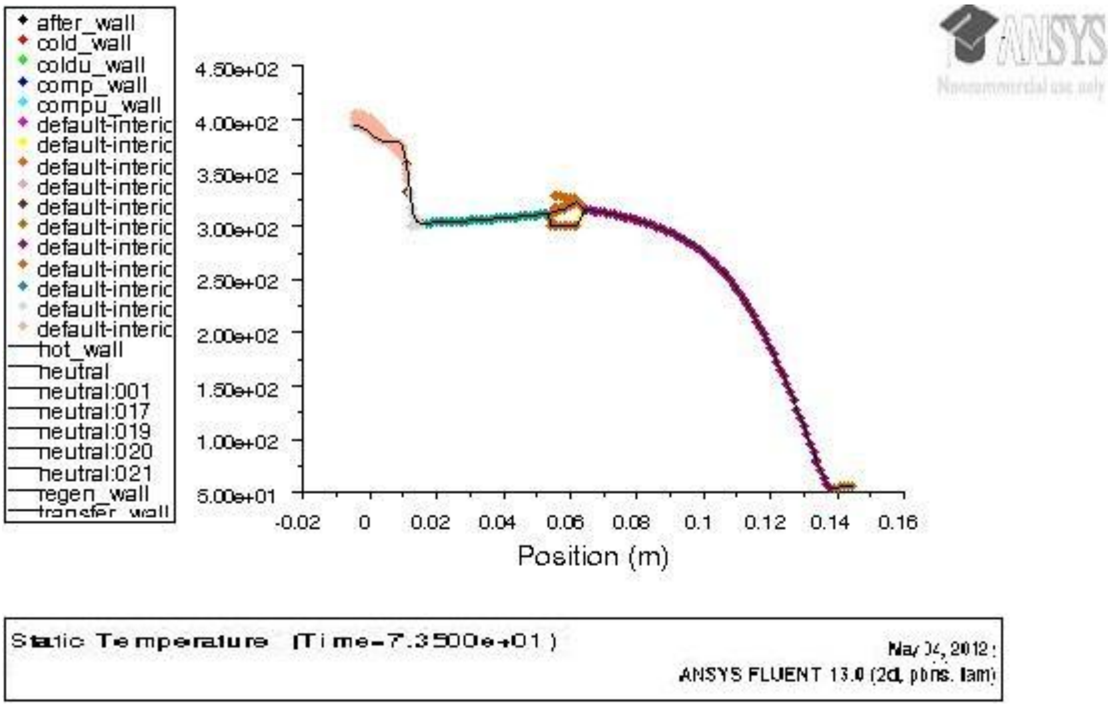


Fig 7: Temperature plot for Stirling cryocooler at no load with 20 Hz

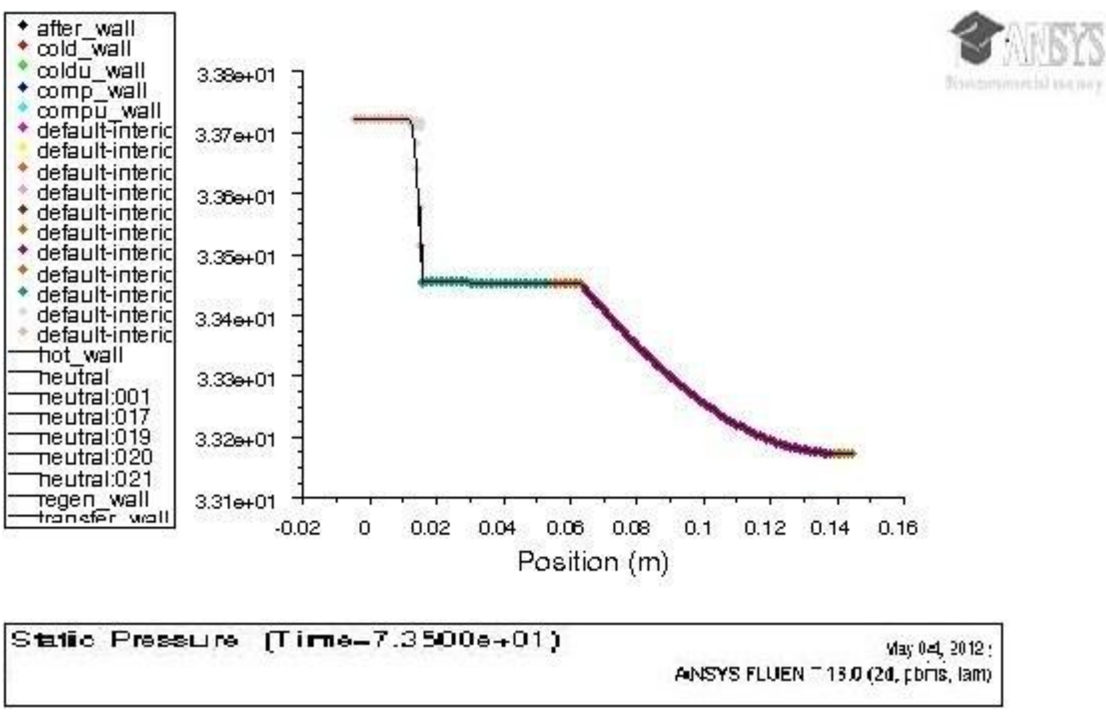


Fig 8: Pressure plot for Stirling cryocooler at no load with 20 Hz

4.1.6 PRESSURE CONTOUR

The Fig.9 below shows the variation of the pressure along the length of the Stirling cryocooler and as it was observed the pressure decreases.

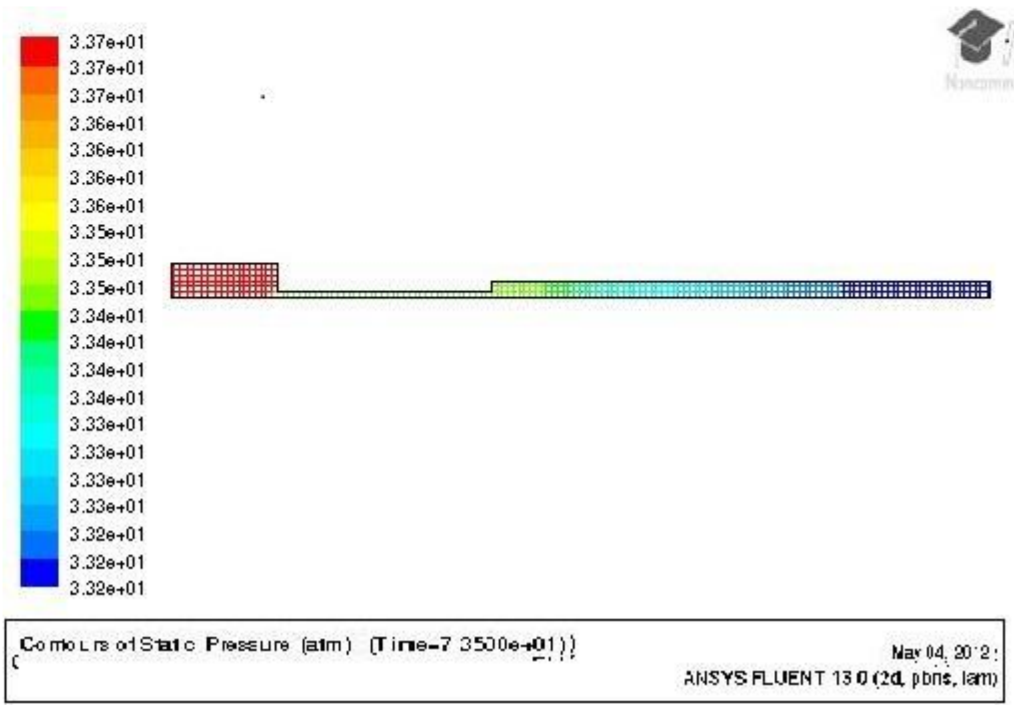


Fig 9: Pressure contour of Stirling cryocooler at no load at 20 Hz

4.1.7 VELOCITY VECTOR

The plot of Fig.10 below shows how the gas or fluid flow takes place inside the Stirling cryocooler. The arrow marks represent the velocity of the fluid particles and they are plotted in terms of velocity magnitude represented by their colours. Due to the resistance offered to the flow, the velocity of the fluid through the regenerator is very less.

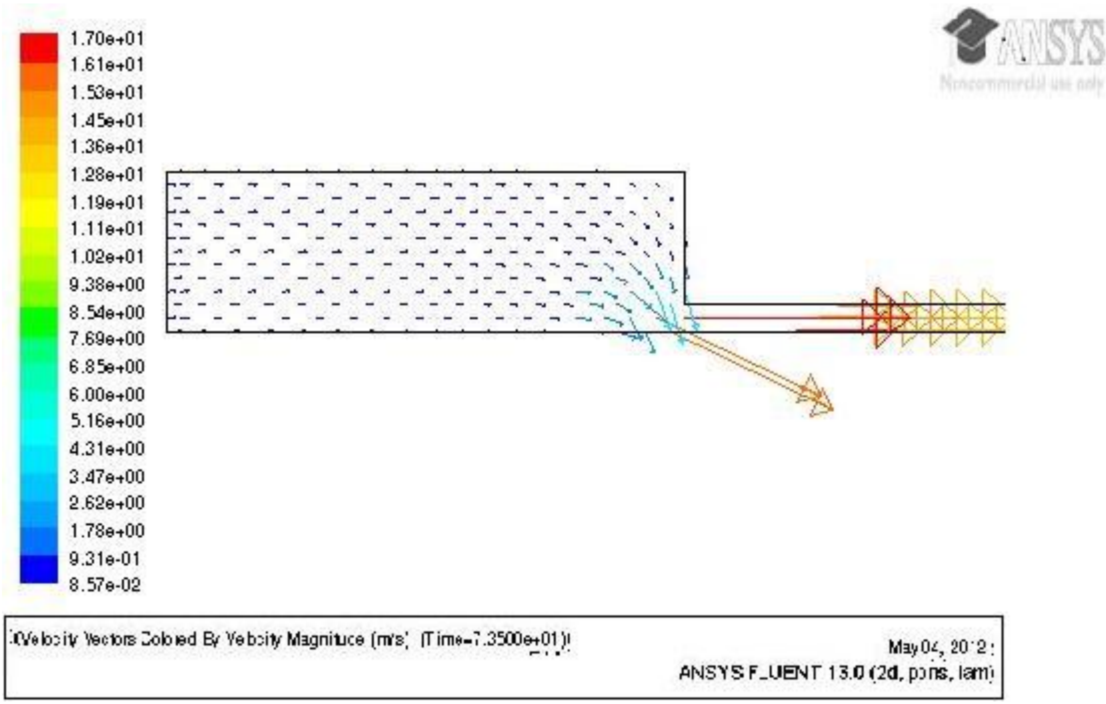


Fig 10: velocity vector profile of Stirling cryocooler at no load at 20 Hz

4.2 LOAD CASE WITH 30 Hz

4.2.1 RESIDUAL PLOT

In this the residual monitors of each parameter is shown i.e. how it is varying with time and the errors are lying within the range defined before the simulation. This is shown in the Fig.11 below.

4.2.2 TEMPERATURE VARIATION CURVE

In this the temperature profile shows how the temperature varies with time and as the simulation goes on, the temperature of the cold end decreases further. This is shown in the plot below in Fig.12

4.2.3 PRESSURE AND TEMPERATURE CONTOUR

There are pressure and temperature contours which shows the pressure and temperature values in different region of the cryocooler. The high and low values of pressure and temperature are shown by red and blue colours respectively and in between different colours are used to represent different values. These are shown in Fig.13 and Fig.14 below respectively.

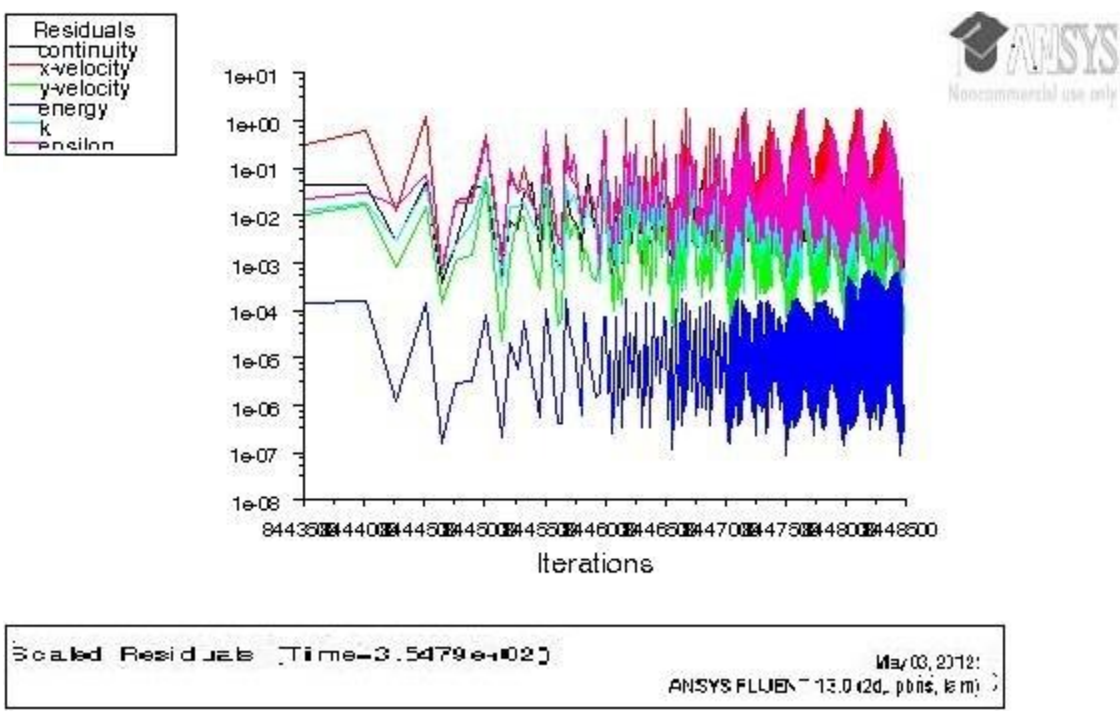


Fig 11: Residual plot for Stirling cryocooler at 0.25 W load with 30 Hz

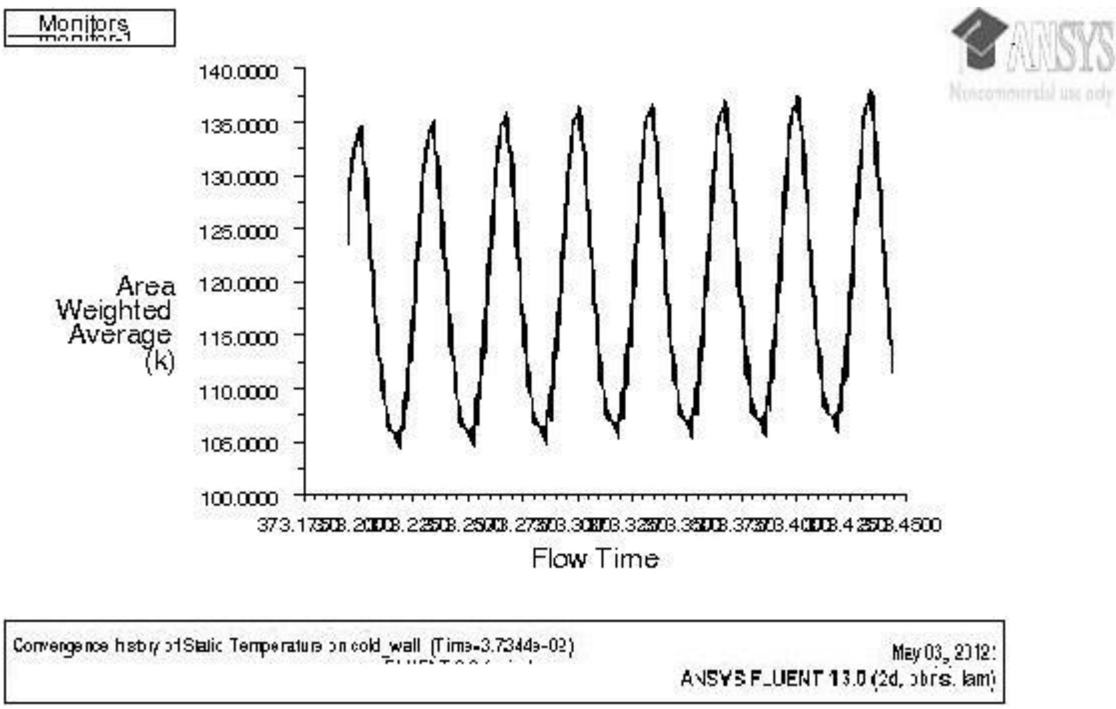


Fig 12: Temperature variation curve of Stirling cryocooler at 0.25W

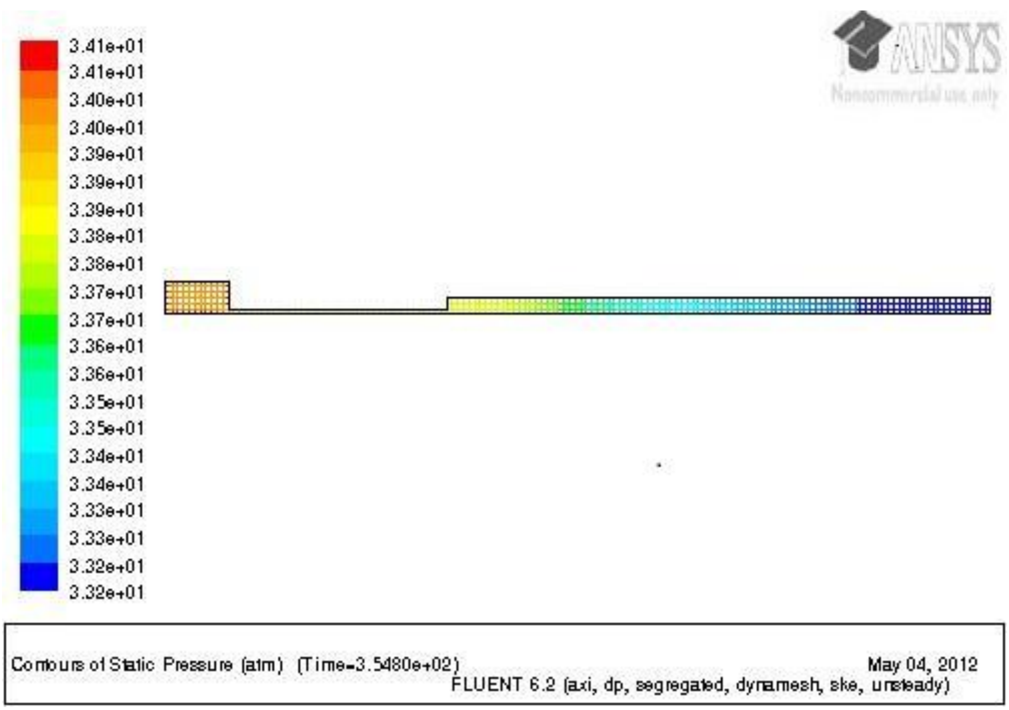


Fig 13: Pressure contour of Stirling cryocooler with 0.25 W Load

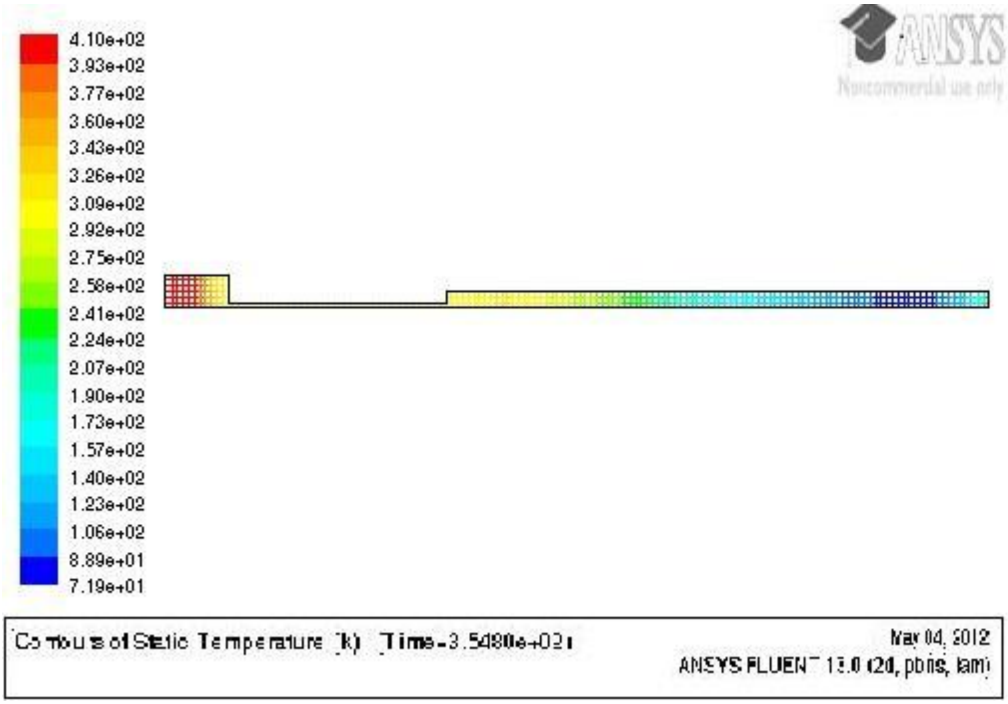


Fig 14: Temperature contour of Stirling cryocooler with 0.25 W Load

4.2.4 PRESSURE AND TEMPERATURE PLOT

These are pressure and temperature curves which are showing variation of pressure and temperature along the axial length of the Stirling cryocooler. The curves are shown in Fig.15 and Fig.16 respectively below.

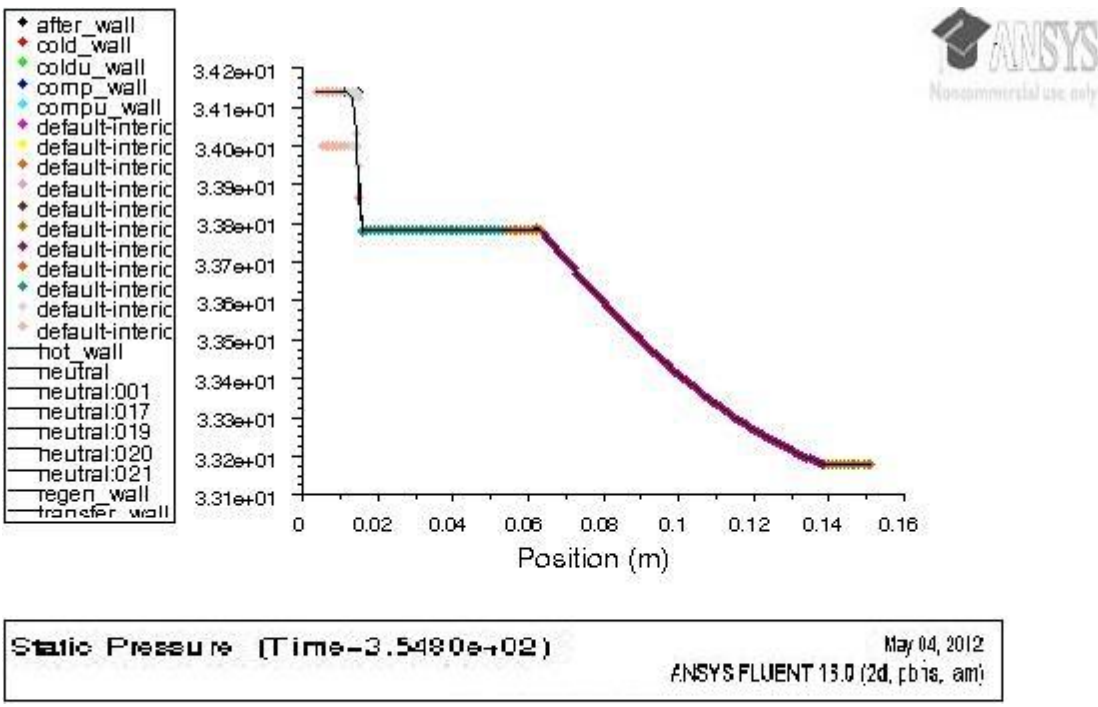


Fig 15: Pressure plot of Stirling cryocooler at 0.25 W Load

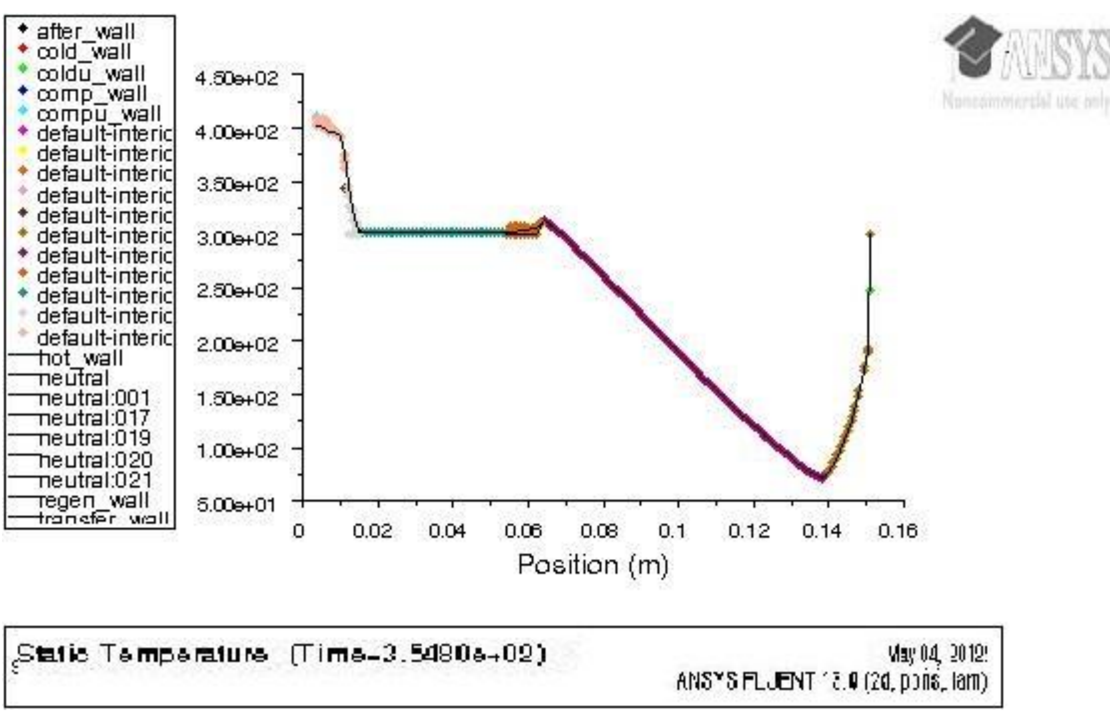


Fig 16: Temperature plot of Stirling cryocooler at 0.25 W Load

4.2.5 VELOCITY VECTOR

The Fig.17 below shows how the fluid particles move inside the Stirling cryocooler during simulation. The arrows are the directional vectors of the fluid particles and the magnitude of the velocity at each point is represented by different colours whose demarcations are shown in the left side of the figure.

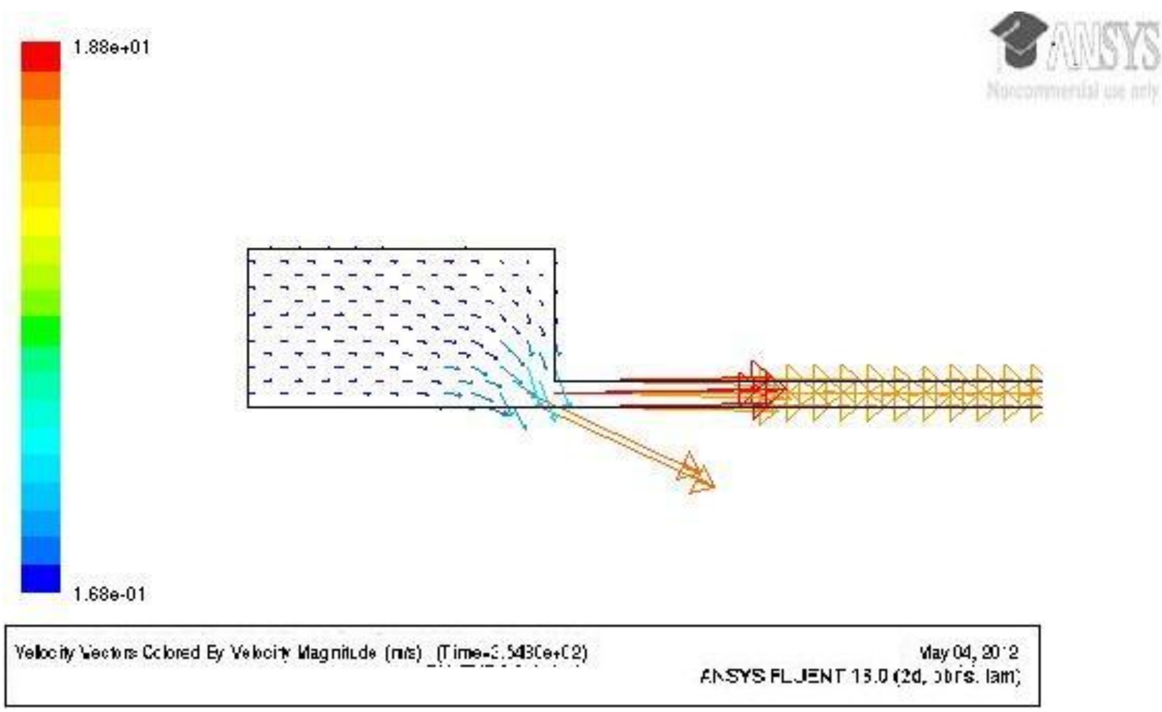


Fig 17: Velocity vector plot of Stirling cryocooler at 0.25 W Load

4.3 COMPARISON OF COOLING BEHAVIOUR IN NO LOAD CASE AND 0.25W LOAD CASE

The simulations were performed with both the cases of no load and with a load of 0.25 W at the cold space with 20 Hz frequency of working motors. In first case of no load condition at the cold space, the temperature obtained was little bit higher than that obtained in the full load condition. The minimum temperature reached in no load case was 70 k and that with load condition was 80 K. The figure Fig.18 below shows the comparison between the two cases.

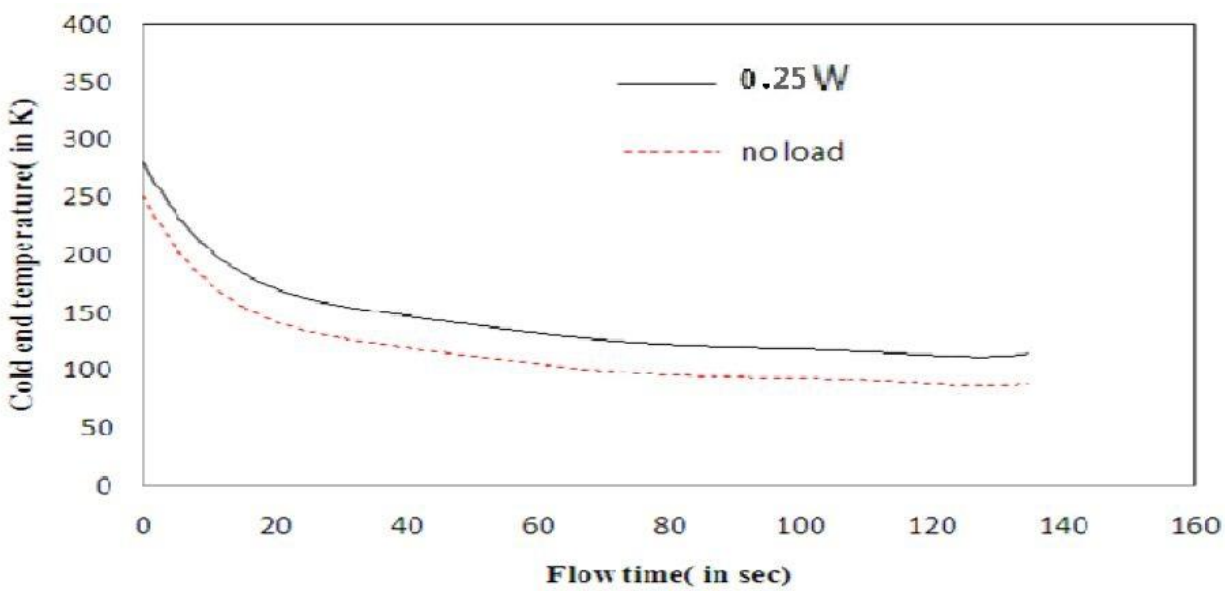


Fig 18: Cold space Temperature Vs flow time plot at no load and 0.25 W load condition

4.3.1 Optimum Frequency of Operation

The operation of Stirling cryocooler takes place at different frequencies of reciprocating motion and its cooling capacity depends upon it. The simulations were done with frequencies values of 20 Hz, 30 Hz, 40 Hz & 50 Hz & with load at the cold space. The figure Fig.19 below shows the variation of variation of cold space temperature with frequency of operation & the minimum temperature to be obtained at a frequency close to 30 Hz.

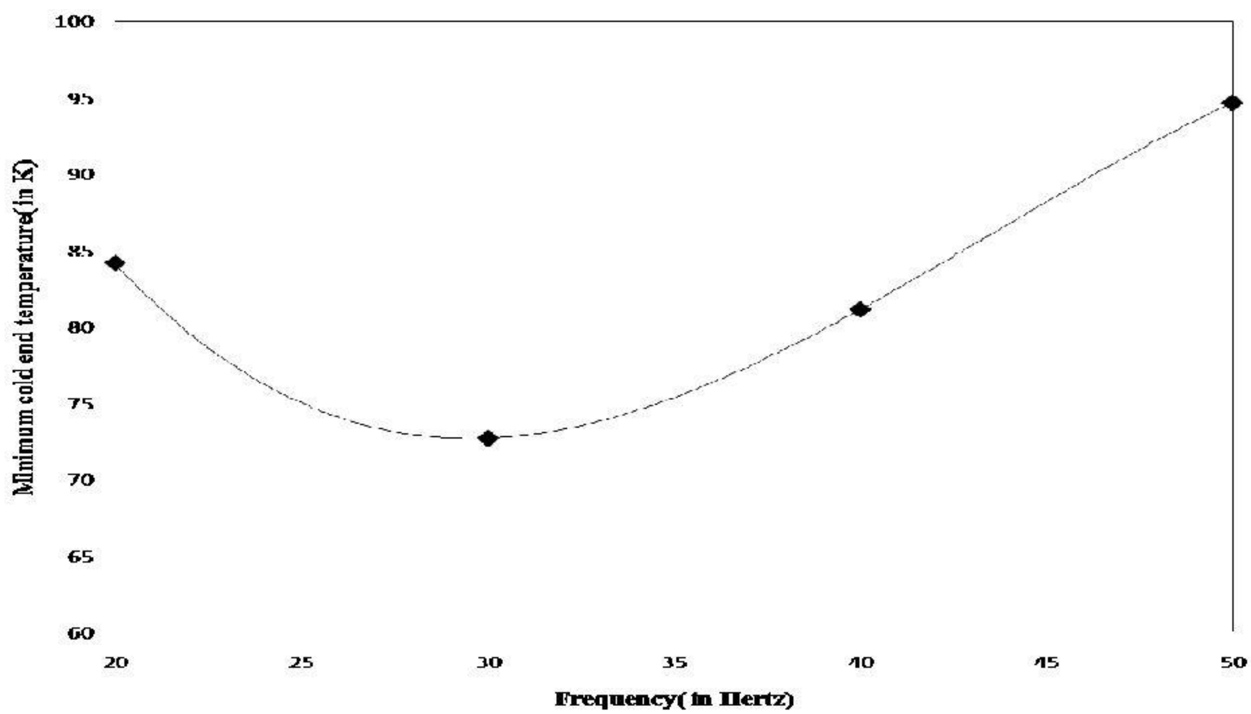


Fig 19: Variation of minimum cold end temperature attained with different frequencies of reciprocating motion

CHAPTER 5

CONCLUSIONS

CFD Simulations of the Stirling cryocooler were successfully developed. There were a lot of requirements while simulating a case of Stirling cryocooler. The motions of both piston and displacer were obtained by hooking up UDF'S to the compressor part and the displacer part. There are different cases of simulations of Stirling cryocooler. For no load case at the cold end the minimum temperature attained was around 70 k with 20 Hz of operating frequency and 80 k with 0.25 W of Load with 30 Hz of operating frequency. The comparison between the two cases shows that the minimum temperature attained is lowest in case of no load but with load condition the minimum temperature increases. Also the effect of frequency of operation on the minimum temperature was that with frequency of operation around 30 Hz, the minimum optimum temperature was obtained.

SUGGESTIONS FOR FUTURE WORK

1. There are various areas of Stirling cryocooler on which improvements can be done. The most important issue is of losses with Stirling cryocooler. There are various losses like pressure drop losses due to friction across the regenerator and then also due to filling of regenerator void volume during pressurization and depressurization of the regenerator. There is also regenerator thermal loss due to heat transfer to the surrounding by it [19]. There are also shuttle losses and acoustic losses. Shuttle loss is the heat loss due to relative motion between the displacer and it's housing which is due to the temperature difference created between them [20]. Thermal acoustic losses are due to the oscillating flow of the working fluid i.e. helium inside that gap. Due to difference in the heat capacities of the two bounding walls and due to the relative motion of the two walls, there is a heat transfer loss [21]. This loss depends upon pressure amplitude and flow velocities of the working fluid.

A proper Schmidt analysis of above mentioned losses can be done and by including their effects in the simulations, results for an optimized geometry of regenerator can be obtained, so that minimum temperature of refrigeration can be obtained while including all the constraints [22]. This can be done by making a non equilibrium model of the case. All the above simulations which are done in this report are in equilibrium model. The thermal conductivity of steel material of regenerator and other parts of the cryocooler also varies with temperature in all the zones [24], so that should also be considered while writing the UDF'S.

2. The simulations which are done in this report are made by moving the last edge of the cold end in an exact phase relationship with that of the piston. But in reality the displacer is moved to and fro inside the cylinder. Also in all the above simulations there is no gap taken between the displacer and the regenerator housing but in actual case there is a gap of very small thickness. So all the above constraints are need to be included and then further simulations can be done for obtaining the results for more practical case. The gap model and displacer motion can be modelled separately first and then after obtaining positive results, they can be brought up together to create a single model for both the cases.

3. The boundary conditions which are taken in the case where there is no load in the cold space are not totally justified. The temperature of the cold wall was taken to be constant but in the actual case the temperature will vary. Also in both the cases of load and no load at the cold space the temperature of the regenerator wall is taken constant but in actual case it will vary due to some heat loss at the regenerator wall. Also the thermal conductivity of the porous material and the cold space wall are taken to be constant but in actual case it will vary with temperature, so these can be included and then further simulations can be done.

References:

[1] Y.P. Banjare 2009, "Theoretical and Experimental Studies on Pulse Tube Refrigerator", PhD Thesis, National Institute of Technology Rourkela, India.

[2] Amrit Bikram Sahoo 2010,"CFD Simulation of a Small Stirling Cryocooler", M.Tech Thesis, National Institute of Technology Rourkela, India.

[3] Ray Radebaugh 2000, "Pulse Tube Cryocoolers for Cooling Infrared Sensors", Proceedings of SPIE, the International Society for Optical Engineering, Infrared Technology and Applications XXVI, Vol. 4130, pp. 363-379.

[4] RICOR Cryogenic and Volume Systems En Harod Ihud 18960 Israel.

[5] Park SJ, Hong YJ, Kim HB, Koh DY, Kim JH, Yu BK, Lee KB, "The effect of operating parameters in the Stirling cryocooler", Cryogenics 42 (2002) 419–425.

[6] Cun-quan Z, Yi-nong Wu, Guo-Lin J, Dong-Yu L, Lie X, " Dynamic simulation of one-stage Oxford Stirling cryocooler and comparison with experiment", Cryogenics 42 (2002) 577–585.

[7] Nisenoff M, Edelsack EA, "US Navy program in small cryocoolers".

[8] Price K, Reilly J, Abhyankar N, Tomlinson B, "Protoflight Spacecraft cryocooler performance results", Cryocoolers 11, Kluwer Academic Publishers (2002) 77-86.

[9] Price K, Reilly J, Abhyankar N, Tomlinson B, "Protoflight Spacecraft cryocooler performance results", Cryocoolers 11, Kluwer Academic Publishers (2002) 77-86.

[10] Sugita H, Sato Y, Nakagawa T, Murakami H, Kaneda H, Keigo E, Murakami M, Tsunematsu S, Hirabayashi M, SPICA Working Group, "Development of mechanical cryocoolers for the Japanese IR space telescope SPICA", Cryogenics 48 (2008) 258–266.

[11] Yarbrough A., Flake B.A , Razani A, "Computational fluid dynamics modelling of pressure drop through wire mesh" (AIP Proceeding) (Adv in Cryo Engg. 49(2004) pp1338-45.

[12] Jiao, Anjun, Jeong, Sangkwon and Ma, H. B. "Heat transfer characteristics of cryogenic helium gas through a miniature tube with a large temperature difference" , Cryogenics,44(2004), pp. 859-866.

[13] Martini, W, "Stirling Engine Design Manual" (1982) Second Editions, Martini Engineering, 2303 Harris, Washington, 99352, USA.

[14] Walker G, Weiss M, Fauvel R, Reader G, "Microcomputer simulation of Stirling cryocoolers", Cryogenics 29 846–849.

[15] Ibrahim MB, Tew Jr. RC, Zhang Z, Gedeon D, Simon TW, "CFD modeling of free piston stirling engines", NASA/TM—2001-211132 IECEC2001–CT–38.

[16] Yatin Chhabra, V. Devaraj 2011, "Mathematical and Optimization Analysis of a Miniature Stirling Cryocooler", B.Tech Thesis, National Institute of Technology Rourkela, India.

[17] T.R. Ashwin, G.S.V.L. Narasimham, Subhash Jacob, "CFD analysis of high frequency miniature pulse tube refrigerators for space applications with thermal non-equilibrium model", Applied Thermal Engineering 30 (2010) 152–166.

[18] Fluent Inc. 2006, Fluent 6.3 UDF Manual, Centerra Resource Park 10 Cavendish Court Lebanon, NH 03766.

[19] David Berchowitz, Israel Urieli 1984, Stirling Cycle Engine Analysis, Amazon.com .

[20] Ranjit Kr. Sahoo, Prof. Sunil Kr. Sarangi, Design and Analysis of Integral Stirling Cryocoolers.

[21] Vincent Kotsubo, Gregory Swift, "Thermo acoustic Analysis Of Displacer Gap Loss in a Low Temperature Stirling Cryocooler", Ball Aerospace and Technologies Boulder, CO 80027 USA.

[22] Ho-Myung Chang, Dae-Jong Park, Sangkwon Jeong 2000, Effect of gap flow on shuttle heat transfer, Cryogenics 40 159-166.

[23] S.J. Park, Y.J. Hong, H.B. Kim, D.Y. Koh, J.H. Kim, B.K. Yu, K.B. Lee 2002, The effect of operating parameters in the Stirling cryocooler, Cryogenics 42 419–425.

[24] E.D. Marquardt, J.P. Le, Ray Radebaugh, 11th International Cryocooler Conference June 20-22, 2000 Keystone, Co. Cryogenic Material Properties Database, National Institute of Standards and Technology Boulder, CO 80303.