CFD Simulation of a Small Stirling Cryocooler Using Non-Thermal Equilibrium Model

A Thesis Submitted in Partial Fulfillment of the Requirements for the Degree of

Bachelor of Technology

In

Mechanical Engineering

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May 2013

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Certificate of approval

This is to certify that the project entitled, **"*CFD Simulation of a Small Stirling Cryocooler Using Non-Thermal Equilibrium Model*"** being submitted by ***Mr. Abhishekh Kumar Jha*** has been carried out under my supervision in partial fulfillment 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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ACKNOWLEDGEMENT

The writing of this dissertation has been one of the most academic challenges I have ever had to face. Without the support, patience, guidance of the following people this graduation would not have been completed. Here I eagerly like to owe my deepest gratitude to them.

First, I would like to show my greatest gratitude to my advisor Prof. R. K. Sahoo at National Institute of Technology Rourkela for his endless support and inputs which led to my successful graduate journey. Fruitful and rewarding discussions with him on numerous occasions have made this work possible. I would also like to thank Mr. Sachindra Rout, PhD for supporting me and exposing me to the research field of cryocoolers since the beginning of this project. With his inputs day and night, weekdays and weekends, in every occasion he has been there to share his knowledge with me and to motivate me to further excel in my research. I thank you again Mr. Sachindra Rout.

I am thankful to my friends Rudra and Sagar for their help and motivation from backstage all through the journey of my life till date. I especially would like to thank Mohana Das, who always supported me with love and warmth all through the journey of this bachelor's degree. I thank her from the bottom of my heart.

Lastly, my work towards the completion of the degree wouldn't have been possible without my unsung supporters. They are the members of my family, my greatest asset of all. Without them and their support there is no way I would be standing here today. There is no way to repay them for their love. Thank you again, Papa (Radheshyam Jha), Mommy (Anjani Jha) and brothers (Anand and Ashish). Words can never express my gratitude for my family.

CONTENTS

ABSTRACT

Stirling Cryocoolers serve in a number of advanced technological applications to cool a variety of important components, such as infrared (IR) detectors, high temperature super conductors (HTSC), and cryogenic catheters. With the advancement in applications of cryocoolers, several simulations of such cryocoolers were also developed. These types of simulations can save a lot of time and money as it provide an accurate analysis of the performance of the cryocooler before actually manufacturing it. In this study, the commercial computational fluid dynamic (CFD) package Fluent and Gambit was utilized for modeling the entire Stirling cryocooler that includes a compressor, an after cooler, a transfer tube, a regenerator that is represented as porous medium and cold and warm heat exchangers. The regenerator is the key element of Stirling cycle cryocoolers. It can be seen that enactment of the regenerator straightly affects the cryocooler performance. Therefore, any improvement on the regenerator will lead to a more efficient cryocooler. This project represents completely new type of numerical computational fluid dynamic (CFD) approach for making it more realistic to the porous media inside the regenerator of a Stirling refrigerator. The available commercial software package FLUENT which is used for solving Computational fluid dynamics (CFD) has the capability to define a porous media and to solve the governing equation for this region.

But one problem arises is that inside the porous media region the software consider the fluid medium temperature and solid matrix medium temperature remains same in any spatial location which is impractical in real case. So to avoid this impractical situation we made an attempt to make a non-thermal equilibrium medium inside the regenerator for which there is provision for directly defining it in ANSYS 14. It is a non-thermal equilibrium model therefore there are separate energy equations for each phase within the domain (N fluid phases plus one solid phase). Additionally, it does not make any particular assumption on the solid material properties. The thermal non-equilibrium condition predicts a higher cold heat exchanger temperature compared to thermal equilibrium.

LIST OF FIGURES:

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NOMENCLATURE

TE	thermally equilibrium
NTE	non- thermal equilibrium
E_f	total fluid energy
Es	total solid medium energy
ρ_f	fluid density
ρ_s	solid medium density
γ	porosity of the medium
k_f	fluid phase thermal conductivity
k_s	solid medium thermal conductivity
h_{fs}	heat transfer coefficient for the fluid / solid interface
A_{fs}	interfacial area density, i.e., the ratio of the area of the fluid / solid interface and the volume of the porous zone
T_f	temperature of the fluid
T_s	temperature of the solid medium
S_f^h	fluid enthalpy source term
S_s^h	solid enthalpy source term

CHAPTER 1

INTRODUCTION AND BACKGROUND

1.1. CRYOGENICS

Cryogenics is the science that addresses the production and effects of very low temperature close to the lowest theoretically attainable temperature (absolute zero, 0K equivalent to -273.15°C or -459.67°F). In engineering, cryogenics can be best described as an application which operates in the temperature range from absolute zero to 120K.

1.2. CRYOCOOLERS

A cryocooler is a device used to cool the environment and anything inside it to extremely cold temperatures. Typically used in scientific and engineering applications, it is designed to achieve temperatures well below those reached by standard appliances. There are numerous fields in which cryocoolers play a vital role. These include medical, automotive, and aerospace applications, use in scientific research and military operations, and more which are mentioned below. In the system of cryocooler, gas is typically circulated through a closed cycle to absorb heat from the interior of the device and transfer it to the outside environment. This gas may be hydrogen, helium, or some other gas or mixture of gases. The ability of the device to cool its interior environment depends largely on the thermodynamic properties of the gas circulating through the system.

1.3. TYPES OF CRYOCOOLERS

The classification of cycrocooler is based on the types of operating cycle and the type of heat exchanger.

1.3.1. On the basis of operating cycles

- *Open cycle cryocoolers*

These types of cryocoolers consume stored cryogens either in subcritical or supercritical liquid state. Solid cryogens are stored as high pressure gas with a Joule- Thomson expansion valve.

- *Closed cycle cryocoolers*

Closed cycle cryocoolers provide cooling at cryogenic temperature and reject heat at very high temperatures. These types of cryocoolers are also known as mechanical cryocoolers. A few examples are Stirling cryocoolers, closed cycle Joule-Thomson cryocoolers, Brayton cycle cryocoolers etc. In these types, there are two working fluids, one which will be working inside the cycle and the other one which will be coming in direct contact with the space to be cooled [1].

1.3.2. On the basis of heat exchangers used

- *Recuperative cryocoolers*

These are analogous to DC electrical systems since the direction of flow of the working fluid is in one direction only. Because of this characteristic, the compressor and expander have isolated inlet and outlet valves to maintain the flow direction. Valves are necessary in this configuration unless the system has any turbine or rotary elements. The enactment of these cryocoolers predominantly depends upon the type of working fluid or refrigerant used. The most leading advantages of these cryocoolers are that it can be scaled to any size (i.e. even upto the order of few MWs). They can be further classified into valve less and with valves cryocoolers. Also it includes the joule Thomson cryocooler and Brayton cryocooler [2].

- *Regenerative cryocoolers*

These cryocoolers are analogous to AC electrical system since the working fluid oscillates in the flow channel. As there is a single flow channel, the expander and compressor do not need any valves in this case.The regenerator, which has a very high heat capacity, stores the heat for half a cycle and then releases it back to the working fluid. These cryocoolers cannot be ascended to large sizes however, these are very efficient because of very low heat transfer loss. Liquid helium is commonly used as working fluid in these types of cryocoolers.

1.4 APPLICATIONS OF CRYOCOOLERS [3]

I. Military applications

- Infrared sensors for guiding the missile guidance and also to get the clear vision at night.
- Infrared sensors for surveillance (satellite based)
- Gamma ray sensor for observing nuclear action
- Superconducting magnets to sweep mine

II. Environmental

- Infrared sensors for getting the atmospheric readings(satellite based)
- Infrared sensors to monitor if pollution takes place
- For Cryotrapping the air samples that locate at remote locations

III. Medical applications

- Superconducting magnets for MRI systems
- SQUID magnetometers for heart and brain studies
- Blood and semen storage
- Cryosurgery

IV. Commercial applications

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

V. Transport applications

- Superconducting magnets for observing maglev trains
- Infrared sensors for getting the clear vision of aircraft at night

VI. Industrial and Commercial

- Superconductors for communication at high-speed
- Semiconductors for the computers that act at high-speed
- Cryopumps for the manufacture of semiconductor
- Low-level moisture sensors for ultrapure gases
- Infrared sensors to process the monitoring sector

1.5. STIRLING CRYOCOOLER

Stirling cryocoolers are regenerator type cryocoolers which work on the basis of Stirling cycle. An ideal Stirling cycle consists of two isothermal process and two constant volume processes seen in Fig.1. The detailed explanation of Stirling cycle has been done in the following section.

1.5.1.Stirling cycle

The key principle of a Stirling engine is that a fixed amount of a gas is sealed inside the engine. In Stirling cycle, Carnot cycle's compression and expansion isentropic processes are replaced by two constant-volume regeneration processes. During the regeneration process heat is transferred to a thermal storage device. The regenerator is assumed to be a reversible heat transfer device, that can be a wire or a ceramic mesh or any type of porous plug with a high thermal mass (mass times specific heat).

The ideal Stirling cycle is composed of four processes which are shown in Fig.1. In ideal Stirling cycle diagrams, heat exchangers, regenerators, connecting elements are anticipated to have zero volume.

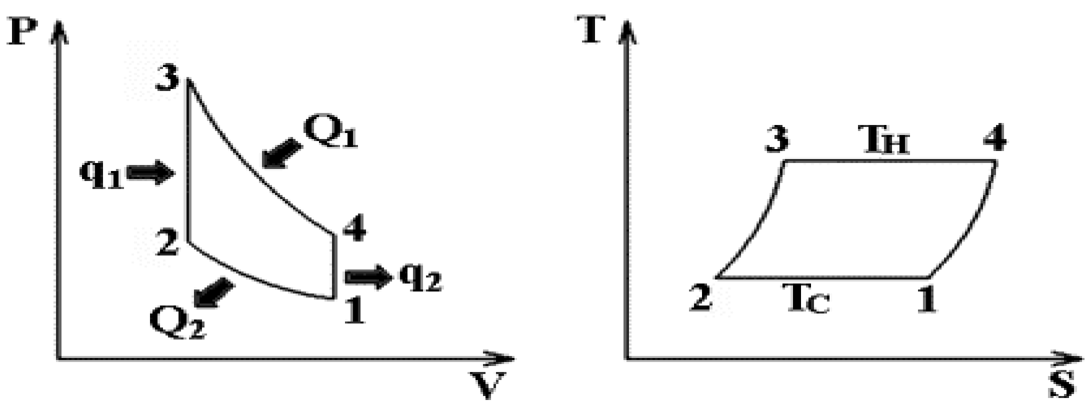

Fig.1. Thermodynamic process of an ideal Stirling Cycle

The four processes can be explained as follows:

- Isothermal Expansion:

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

- Isochoric heat Addition:

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

- Isothermal Compression:

After compressing the gas from cold end, it travels through the regenerator towards the hot end of the cryocooler. During this process it takes away the heat, stored in the regenerator.

- Isochoric heat rejection:

Having the heat from the regenerator, the gas in the hot end has now some space to be cooled.Later on the gas present in compressed state rejects heat to the ambient.

1.5.2. Design of Stirling Cryocooler

The layout of a typical Stirling cryocooler has been shown in Fig. 2. The leading components of the Stirling cryocooler are Compressor space, Piston, Aftercooler, Transfer line, Regenerator, Cold space and Hot space. Here, in this cryocooler, the piston and Regenerator are the moving components while rests are stationary. The Piston and Regenerator follow a reciprocating motion that is achieved by crank shaft mechanism coupled to a motor. They (Piston and regenerator) reciprocate with the same frequency (20Hz) of the driving motor with a phase difference of 90^0 in the motion. This phase difference in motion is themajor cause of cooling effect produced. Detailed explanation of how it produces is given below.

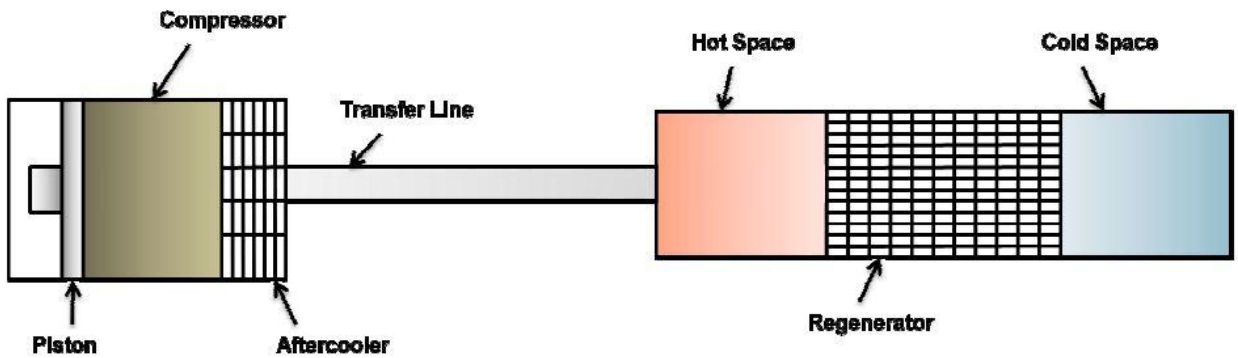

Fig.2.Schematic Representation of Stirling Cryocooler [4]

A Stirling cryocooler works on the basis of the Stirling cycle. An ideal Stirling cycle comprises of two isothermal processes and two constant volume processes as shown in Fig.1. A schematic representation of a Stirling cryocooler has been prepared in Fig.2. The stage by stage working of the Stirling cryocooler has been labeled in the Fig.3.

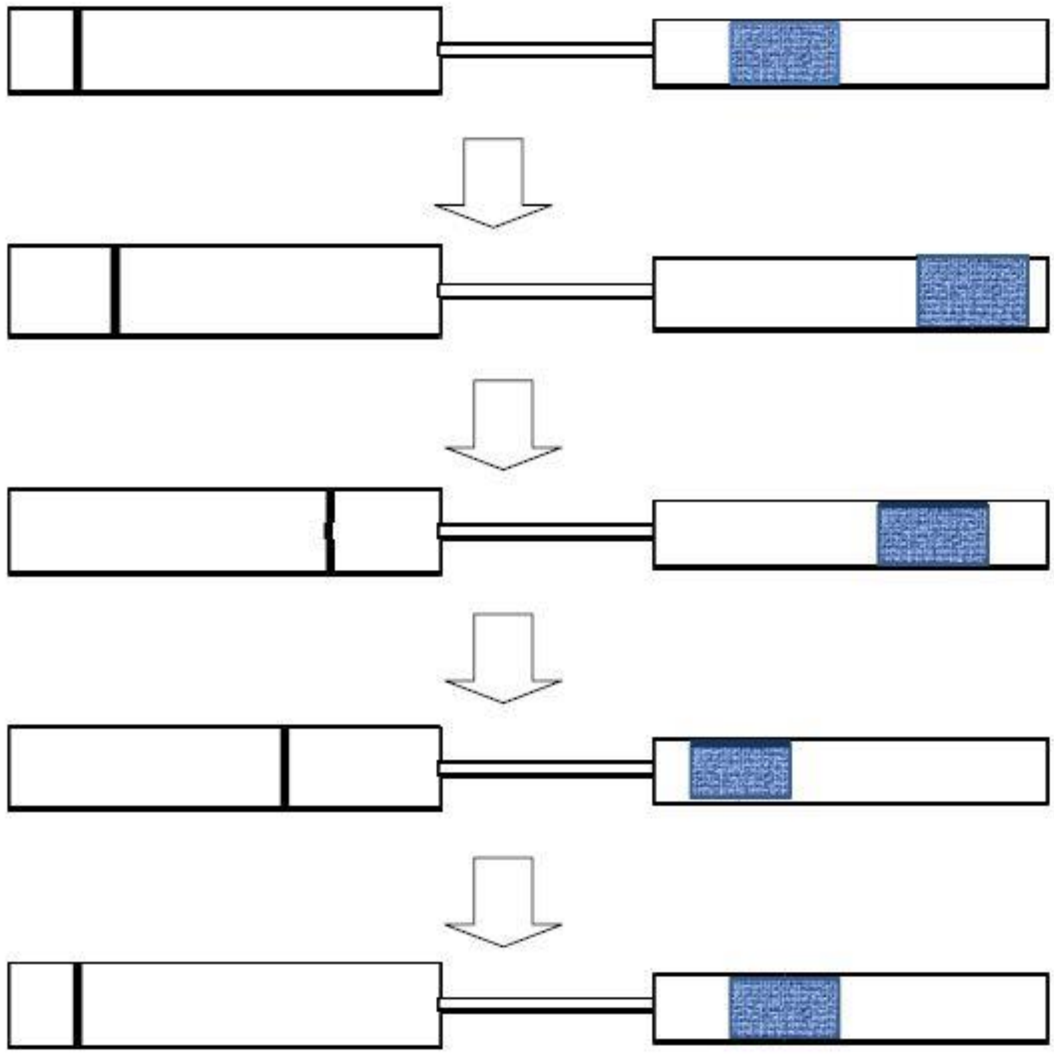

Fig.3. Step by step process of the working of a Stirling cryocooler

Firstly, let us agree that the Stirling cryocooler is in the state of maximum volume at initial time. This position denotes to state 1 in Fig.3. The working fluid remains at minimum pressure in this state and at the same time lowest temperature is accessible at the cold space. In the next stage, the regenerator moves significantly, forcing the fluid to pass through it while the piston stays almost stationary, consequently maintaining a constant volume. While the fluid passes through the regenerator,it absorbs heat from the regenerator, as the temperature of the fluid in cold space is lower than the temperature of regenerator. Subsequently as the volume remains almost constant, so that the temperature rises and henceforth its pressure also rises which takes it to state 2. This is known as "the constant volume heat addition process". Subsequently, the piston moves for compressing the working fluid to minimum volume state and the maximum pressure state where at the same time the regenerator remains almost stationary. The fluid is currently at extreme temperature and is present in the hot space, in state 3. Following, the regenerator moves towards left accordingly forcing the hot fluid to flow through it. It loses heat to the regenerator mesh as the temperature of the fluid is much higher than the regenerator temperature. Consequently, pressure also decreases at constant volume. This is signified by state 4 in the Fig.3.At this instant, the piston moves toward left causing expansion and henceforth the temperature of the working fluid reduces and comes back to the first state. This is the minimum temperature state.The expansion process originates the cooling effect. This cycle rerun itself in the Stirling cryocooler to reach a steady state and a minimum temperature state.

CHAPTER 2

LITERATURE SURVEY

2.1. EARLIER MODELS OF CRYOCOOLERS DEVELOPED

From the statements given in the previous units, cryocoolers are now being implemented in the areas of superconductor filters, infrared detectors, satellite based applications, cryopumps etc.in a large scale. In earlier 1950s for the first time, Stirling cryocoolers were introduced to the commercial market as single cylinder air liquefiers along with cryocoolers for IR sensors to about 80K. In early1960s, a free-piston Stirling engine was invented by scientist William Beale that acts as power systems in most of the cases. Since then the development of this stirling cryocoolers is increasing day by day and still it is going on in large scale. The first development of linear free-piston Stirling cryocooler was achieved at Philips laboratories, Eindhoven. Another noteworthy linear Stirling cryocooler unit was developed by Dr. G. Davey, Oxford University. In the year 1970sin Oxford University, after the invention of "Oxford-split Stirling cryocooler" it has been developing in a splendid manner which afterwards results in the progression of many new components such as no contact seals, flexure springs, linear compressor etc [5].The regenerator of Oxford type Stirling cryocooler always contains a long stack of phosphor-bronze discs. The frequency of piston and displacer used in this type of cryocooler is very high (30 – 60 Hz) and the oscillating motion of the working fluid is very complicated [6]. Currently Oxford-split Stirling cryocoolers are mostly initiated in space and flight based applications. . In 1980, the US navy started totally a new program which includes new designs for cryocoolers were solicited which would have a coefficient of performance within a range of 2000 to 10000 W/W. In addition to this, programs were received from NBS- Boulder where advancement of a nylon and fiberglass Stirling cycle cryocooler had initiated under ONR support [7].There have been rapid achievements taking place in the field of miniature free piston free displacer Stirling cryocooler over the last few decades.

In 1997, AFRL, Raytheon and JPL together developed flight based Stirling cryocoolers whose operating range was 25K – 120K. It provided refrigeration upto 3W at 60K and 1.5W at 35K [8]. Cryogenically cooled solid state Laser systems was developed by Decade Optical Systems, Inc. (DOS Inc.) in the year 1997[9]. It has been established that all mechanical cryocoolers gained a sufficient amount of cooling capacity with unprecedentedly low amount of power consumption for the cooling requirement of large telescopes and other applications [10].

2.2. LITERATURE REVIEW OF SIMULATIONS OF STIRLING CRYOCOOLERS

Over the past 60 years, we can appreciate the rapid improvement of cryocooler in the cryocooler design and functioning. Later,it became highly indispensable to develop computer based simulations for them. These simulations could offer a detailed analysis of the cryocooler's functioning and various characteristics before actually emerging it, which in turn saves a lot of time and money. With this commitment, the work for developing computer based simulations for cryocoolers originated. Analysis of the ideal Stirling cycle was done by Schmidt (first order approach). He made assumptions such as steady state procedure with ideal regenerators and no pressure drops during the isothermal processes.Later on Martini expanded the second order approach in large scale in which he further analyzed by adding realistic losses to the model developed by Schmidt. Realistic losses which he added were pressure drop loss, shuttle loss, pumping loss, static heat loss, and regenerator ineffectiveness[11]. He applied decoupled independent correction for the justification of various losses which have been comprehensively described in Martini's 1982 Stirling engine design manual [12]. During latter half of the decade, this exploration of Martini was approved for carrying out Stirling simulations at the University of Calgary. Later this was written in FORTRAN 77 language to create an easy for using digital simulation program for microcomputers, namely CRYOWEISS by Walker *et al* [11]. This program was used to predict the results of PPG-1 Stirling liquefier and to compare it with investigational outcomes. A third order split-Stirling refrigerator model was invented under an Independent Research program at Lockheed's Research and Development Division. For programming the model, Continuous System Simulation Language (CSSL) was used. The model had 90 nodes, maximum of which were found in the regenerator region [13]. This model was authorized against experimental results for the Lucas Stirling refrigerators, carried out by Yuan et al [14]. Far ahead, a dynamic model of a one-stage Oxford Stirling cryocooler was created that couldpredict the dynamic processes and performances for the given structure dimensions and operating conditions [5].

In 1989,a CFD code, CAST (Computer Aided Simulation of Turbulent flows) was progressed by Peric and Scheuerer for the prediction of two-dimensional flow and heat transfer phenomena[15]. The development process of the CAST code has been enlightened in detail by Ibrahim *et al* [16].

CHAPTER 3

MODELLING AND SOLUTION

3.1. CFD ANALYSIS OF STIRLING CRYOCOOLERS

3.1.1. Governing equations

I. Mass Conservation Equation:

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

II. Momentum Conservation Equation:

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

III. Energy Conservation Equation (For incompressible flow):

$$\rho c_p \frac{dT}{dt} = -\left(\frac{\partial \dot{q}_x}{\partial x} + \frac{\partial \dot{q}_y}{\partial y} + \frac{\partial \dot{q}_z}{\partial z}\right) + \dot{q}_h$$

3.1.2. Non-equilibrium thermal model

The hypothesis of thermal equilibrium between the solid medium and the fluid flow is not convenient for all simulations, as the presence of different geometric length scales (e.g., pore sizes) and physical properties of solid and liquid phases may result in local temperature differences between the phases. The non-equilibrium thermal model available for porous media in ANSYS FLUENT is based on a “dual cell” approach as it involves a second solid cell zone that overlaps (i.e., is spatially coincident with) the porous fluid zone; the two zones are solved simultaneously and are coupled only through heat transfer. ANSYS FLUENT can automatically create a solid zone that is a duplicate of the porous fluid zone. It should be noted that the duplicate zone is defined as a solid zone, and that the two zones have similar levels of mesh refinement.The conservation equations for energy are solved distinctly for the fluid and solid zones.The conservation equation solved for the fluid zone is

$$\frac{\partial}{\partial t}(\gamma \rho_f E_f) + \nabla.\left(\vec{V}(\rho_f E_f + p)\right) = \nabla\left(\gamma k_f \nabla T_f - \left(\sum h_i J_i\right) + (\bar{\bar{\tau}}.\vec{v})\right) + S_f^h + h_{fs} A_{fs}(T_f - T_s)$$

and the conservation equation solved for the solid zone is

$\frac{\partial}{\partial t}\big((1-\gamma)\rho_s E_s\big) = \nabla.\big((1-\gamma)k_s \nabla T_s\big) + S_s^h + h_{fs}A_{fs}\big(T_s - T_f\big)$

The fluid thermal conductivity k_f and the solid thermal conductivity k_s can be computed via user-defined functions

3.1.3. Heat transfer through the fluid and solid

The current solid heat transfer function allows a finite temperature difference between the solid and the fluid state. As it is a non-thermal equilibrium model, therefore there are discreteenergy equations for each stage within the domain(N fluid phanses plus one solid phase). Additionally, it does not make any supposition on the solid material properties.

For the fluid phases:

$$\frac{\partial \gamma \rho h}{\partial t} + \nabla.(\rho K.UH) - \nabla.(\Gamma_e K.\nabla \mathrm{H}) = \gamma S_k^h + Q_{fs}$$

For the solid phases:

$$\frac{\partial \gamma_s \rho_s C_s T_s}{\partial t} + \nabla.(\rho K_s.U_s C_s T_s) - \nabla.(\lambda K_s.\nabla T_s) = \gamma_s S_s^T + Q_{sf}$$

Where γ_s= 1-γ, and the interfacial heat transfer between the fluid and the solid, Q_{fs}, is determined using an overall heat transfer coefficient model using:

$$Q_{fs} = -Q_{sf} = h\, A_{fs}(T_s - T_f)$$

h, is the overall heat transfer coefficient between the fluid and the solid.

A_{fs} is the interfacial area density between the fluids and the solid. For the flowing of multistage purpose, this perception can be split into a fluid-independent interfacial area density and a contact area fraction between the fluid and the solid:

$$A_{fs} = A_\alpha\left(\frac{A}{V}\right)$$

Where, $A_\alpha$represents the contact area fraction of fluid $\alpha$with the solid.For single-phase flows, A_α =1.

3.2. MODELING THE GEOMETRY

3.2.1. Gambit

A 2 dimensional geometry of Stirling cryocooler was created in GAMBIT. The regular rectangular meshes have been taken into consideration.An axis-symmetric model has been prepared in order to reduce the time consumption happens in simulation. The core components of the design comprises of Compressor, Piston, after cooler, transfer line, cold space, hot space displacer and regenerator.The dimensions of various components are shown in the following table and the subsequent figure has been shown in Fig.4 [2].

Table.1. Dimensions of the various components of the proposed model of stirling cryocooler

Sl. No.	Components	Radius (in mm)	Length (in mm)
1	Compressor	8	9
2	Aftercooler	8	3
3	Transfer line	1	55
4	Hot space	5	5
5	Regenerator	5	45
6	Cold space	5	7
7	Piston	8	-
8	Displacer / Cold piston	5	-

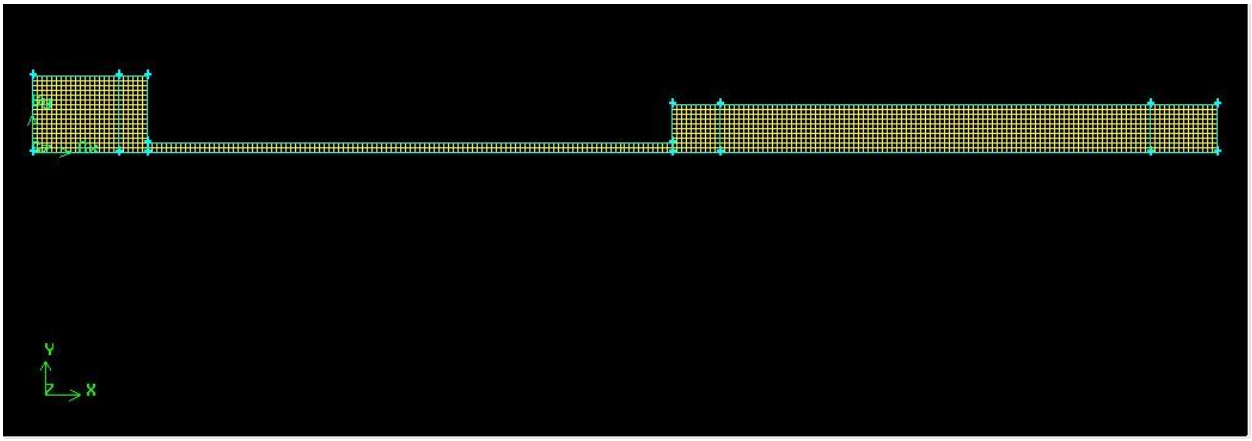

Fig.4. Axis-symmetric model of the Stirling cryocooler developed in GAMBIT

3.2.2. 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 formulate new technologies and also to modify or optimize the prevailing ones. This equipment helps us to design tough simulations and products and also add to test them how they will be behaving in real world.

3.3. SETTING UP THE PROBLEM IN FLUENT

The mesh file is incorporated in FLUENT (ANSYS14) 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 2916 cells, 6230 faces, 3316 nodes along with a partition. The entire problem that is designed at that time,depends upon 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.

Table.2. The detail of the operating and initial conditions

Detail of solver	
Solver	Segregated
Formulation	Implicit
Space	Axis-Symmetric
Time	Unsteady

Velocity Formulation	Absolute
Unsteady formulation	1st 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

3.3.1. 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. Regenerator is assumed to be porous zone. The complete boundary conditions are stated below.

• Piston:

The piston was considered to be an adiabatic wall made of steel material.

• 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/m2; Y: 9.433e+09 1/m2

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

The material of the walls was 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 taken as a porous medium with the following details [17].

Porosity: 0.7

Viscosity: X: 9.433e+09 1/m2; Y: 9.433e+09 1/m2

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. The cold space walls are made of steel.

• Displacer:

The displacer is made of steel and is adiabatic in nature.

3.4. DESCRIPTION OF THE 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 90 degrees. Three governing equations are taken in consideration for this problem- Turbulence,flow and Energy.The underneath relaxation factors are taken in order to get more rapid simulations. The solution is initialized from all zones with preliminary 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.

In the present work, three different cases were simulated,

I. Thermal equilibrium model with no load condition
II. Thermal equilibrium model with 0.5W load condition
III. Non- thermal equilibrium model with no load condition

All three models were simulated at the same frequency of reciprocating piston and displacer i.e. at 20Hz and various type of essential plots and contours were acquired which are discussed in the following section.

3.4.1. 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 taken into consideration, the piston and displacer are ensued to maintain a phase angle of 90 degree during their motion. The proper phase angle is taken into mind while writing the code. The codes were designed after some research work 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 was made in GAMBIT. Different cases were taken into consideration by varying the boundary conditions so that an optimum solution can be found. The geometry remained same during the simulations and only the existing conditions were altered. In the first case,the temperature of cold fluid of TE model at no load condition was found to be 77K where it became asymptote to the time axis at around 341 second's real time.No load condition at cold end means there is no heat flux through cold wall. 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. But, after introducing heat flux of 0.5W/m^2 to the cold wall of TE model, the cold wall temperature was found to be 85K. Again, in third case, the minimum temperature that we encountered at the cold wall for NTE model at no load condition was about 115K only. For all three cases, the frequency of reciprocating piston and displacer was kept at 20Hz.

4.1. NO LOAD CASE OF THERMALLY EQUILIBRIUM MODEL

Geometry was run in simulation with the Stirling cryocooler, the frequency of reciprocating piston and displacer was kept at 20Hz. No heat flux was applied to the cold end of the cryocooler. The residual monitor plot, cooling behavior, temperature contour and plots, and the velocity vector for above case are shown below.

4.1.1. Residual monitor plot and cooling behaviour

The cooling behaviour of the cold end and residual plot is shown in Fig.5 and Fig.6 respectively. The cooling behavior 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.

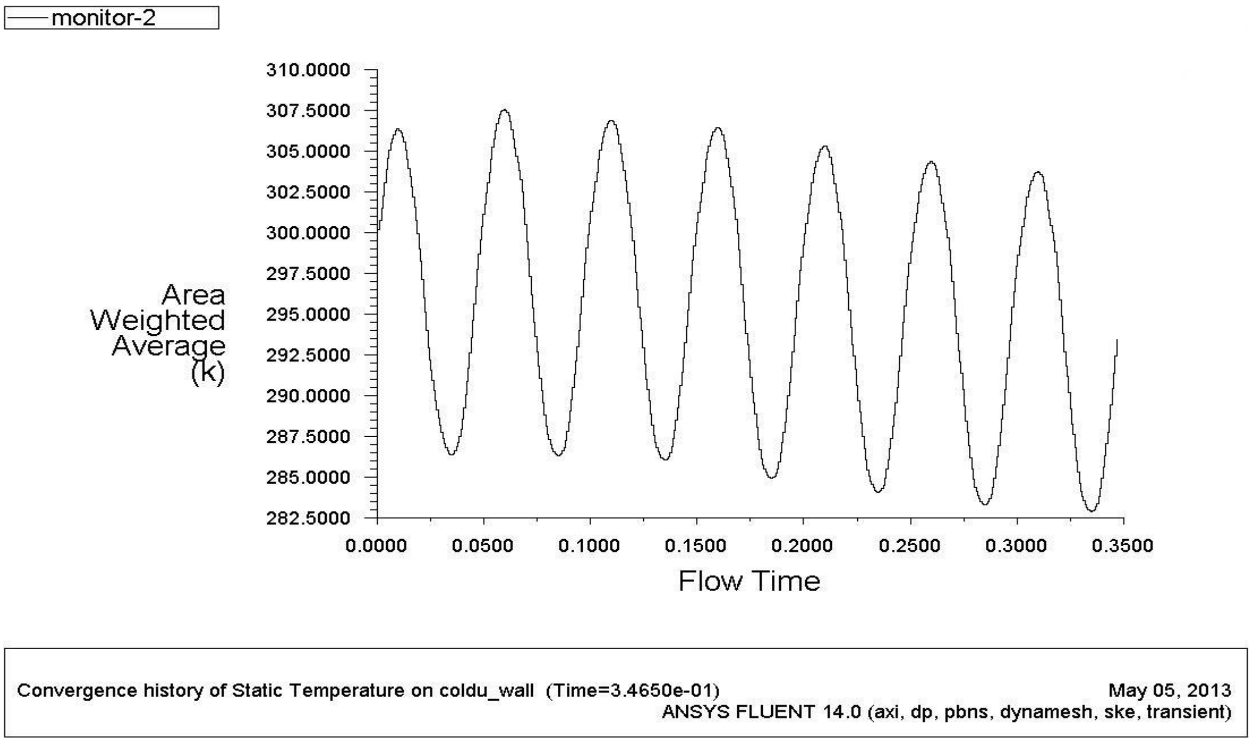

Fig.5.Sinusoidal variation of temperature of cold space of TE model at no load condition

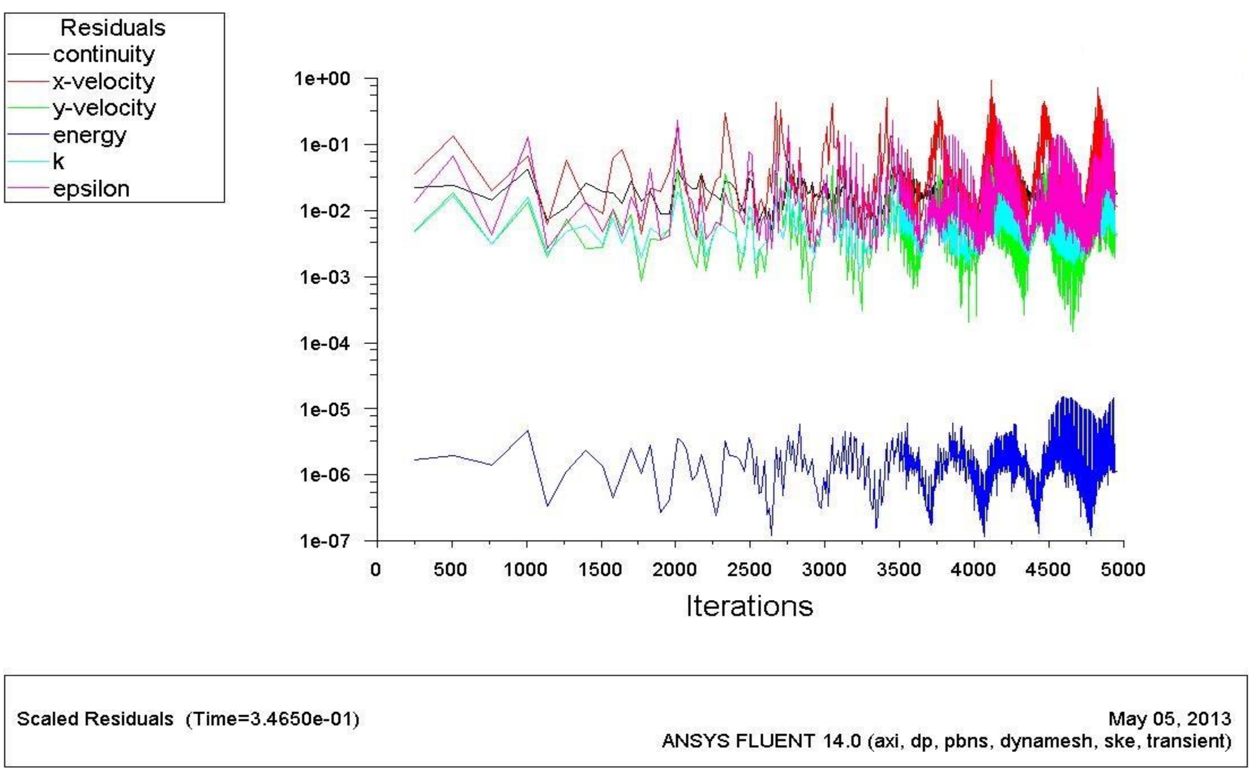

Fig.6. Residual Monitor Plot of TE model at no load condition

4.1.2. Temperature contour

The temperature contour of the Stirling cryocooler after 341.5 seconds of simulation can be seen in Fig.7. The uppermost temperature is achieved at the compressor end and while the lowest temperature is obtained at the cold end of the cryocooler. Thetemperature at theleft side ofthe regenerator is higher and it gradually decreases as we move towards cold space.

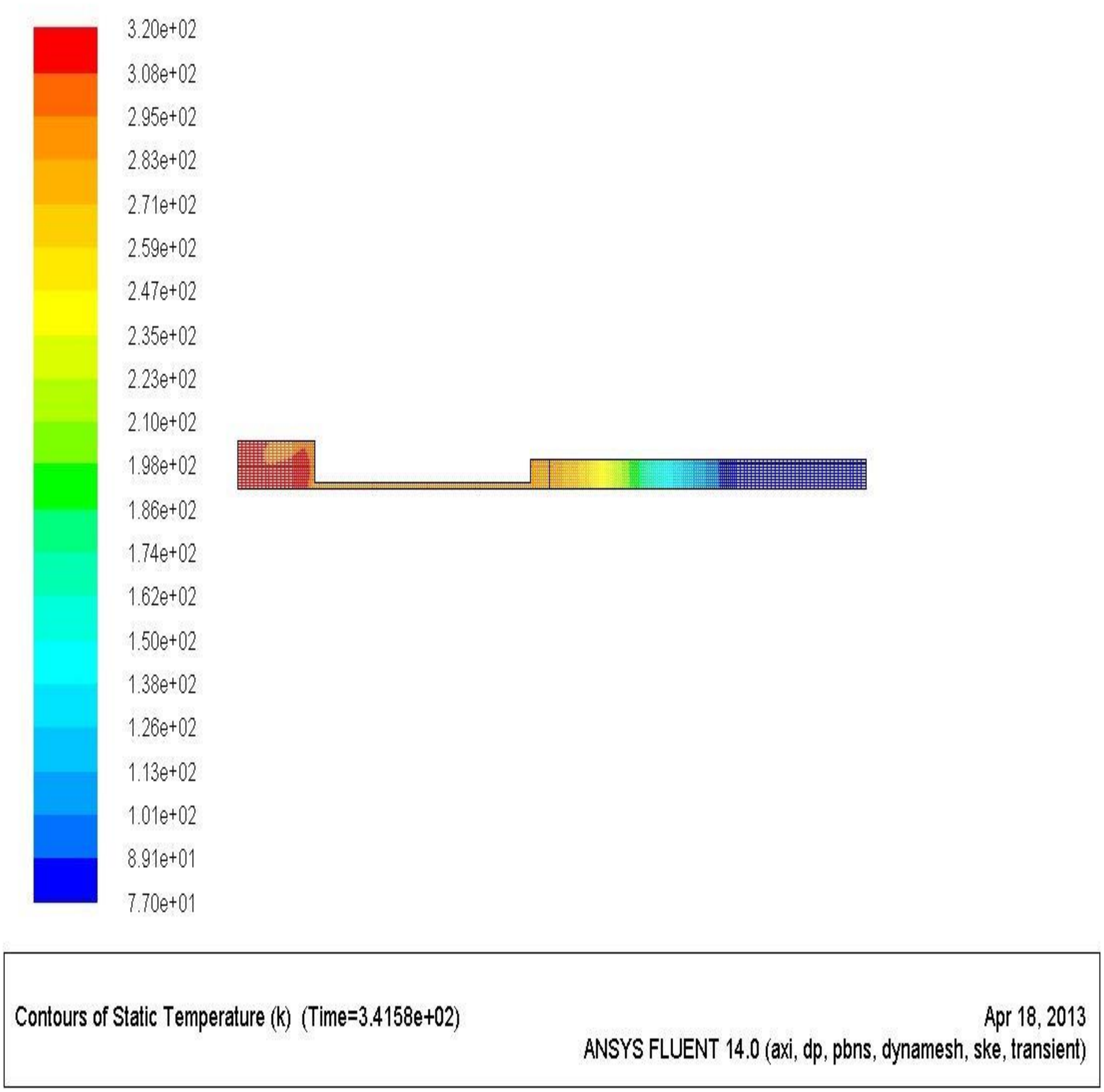

Fig.7. Temperature contour of TE model at no load condition

4.1.3. Temperature plot

Fig.8 shows the temperature profile along the Stirling cryocooler length. Compressor position shows the uppermost temperature whereas the cold space position shows the lowermost temperature.

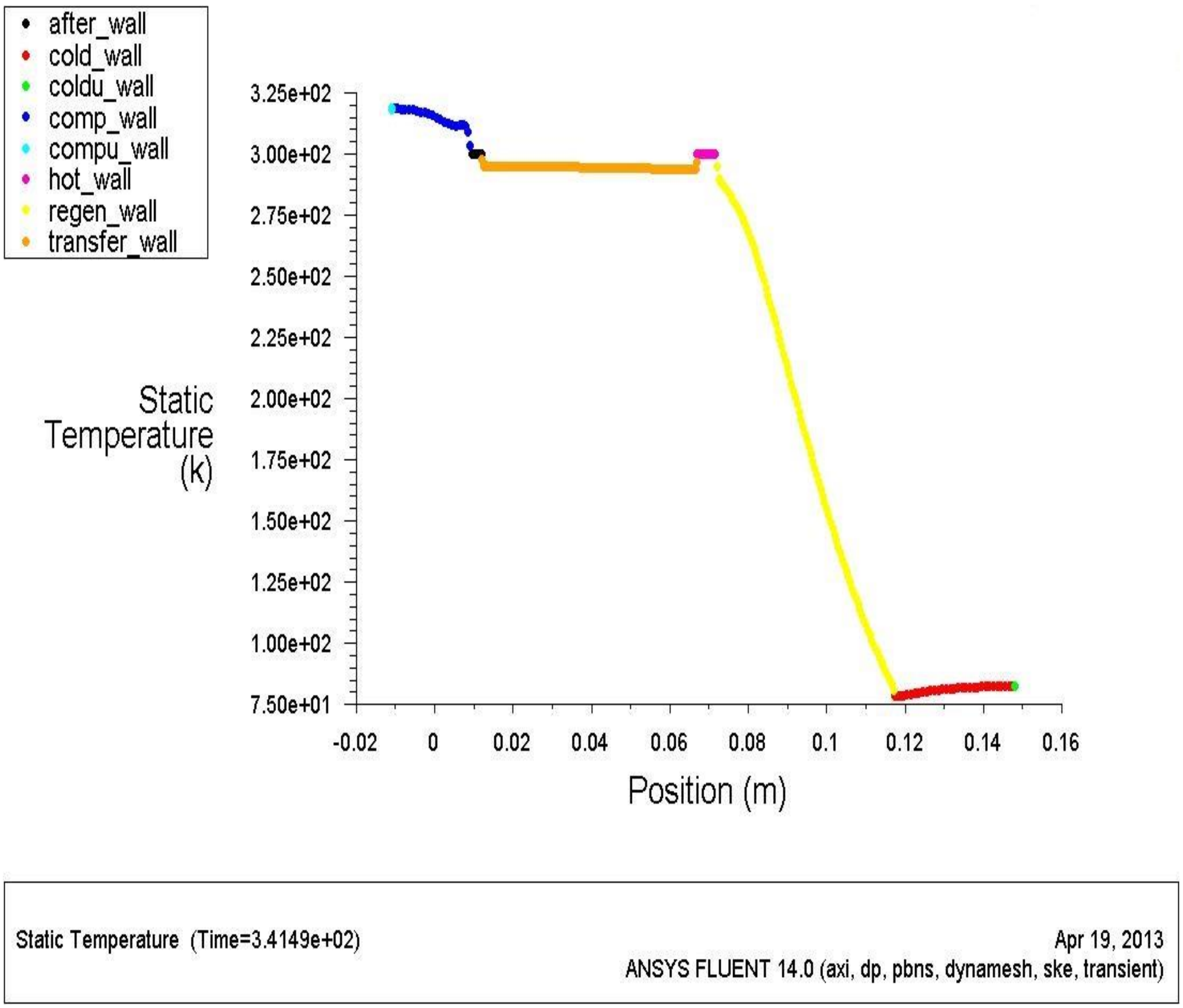

Fig.8. Temperature plot of TE model at no load

4.1.4. Velocity vector

Fig.9 below shows how the 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.

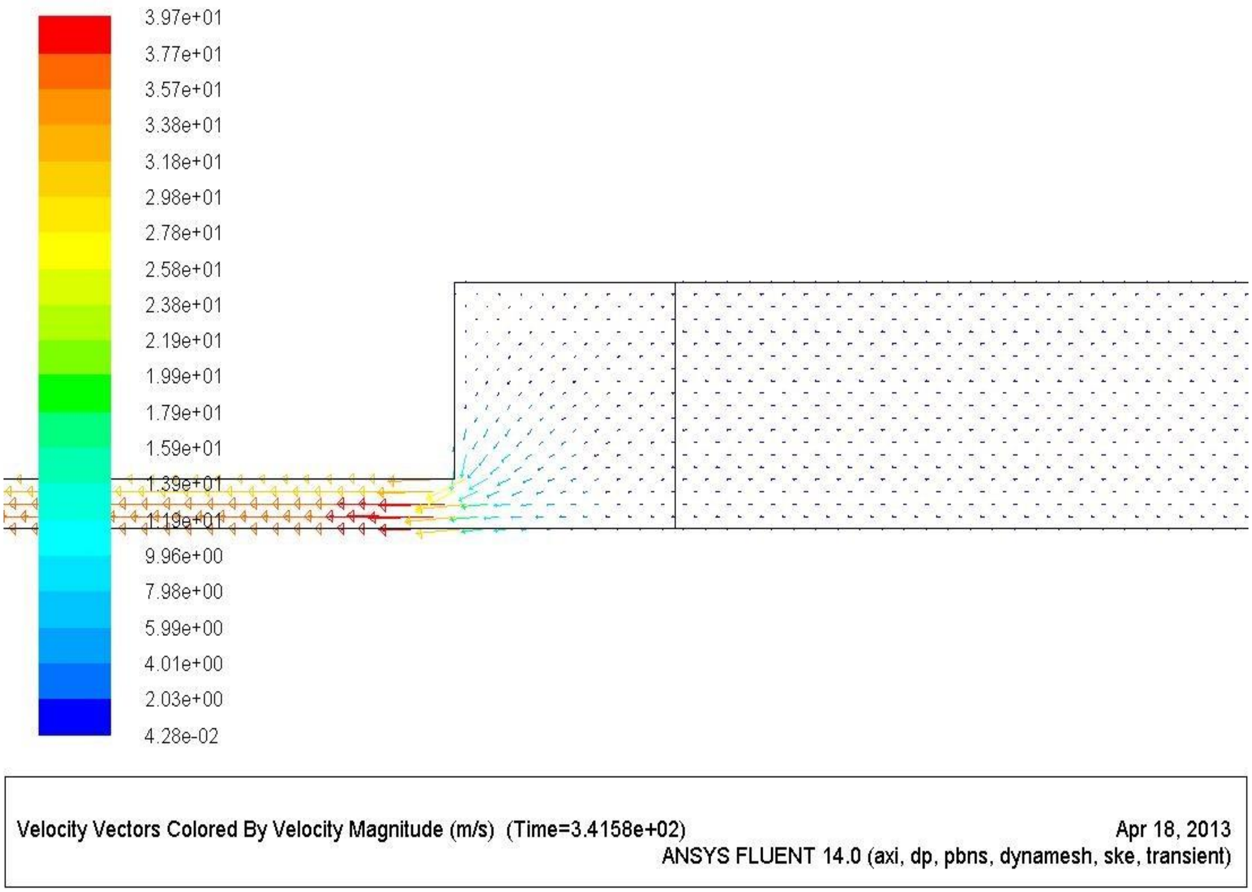

Fig.9. Velocity vector profile of transfer line and regenerator of TE model at no load condition

4.2 THERMALLY EQUILIBRIUM MODEL WITH THE LOAD OF 0.5 W

In this case the heat flux of 0.5 W/m^2 was introduced on the cold wall of TE model and the simulation was carried out. The minimum temperature we obtained was about 85K which was greater than that of no load condition. Also, the frequency of the reciprocating piston and displacer was kept at 20Hz.

4.2.1. Residual plot and cooling behaviour

Fig.10. shows the residual plot of each parameter i.e. how they are varying with time and the errors are lying within the range defined before the simulation.

Fig.11. shows how the temperature varies with time and as the simulation goes on, the temperature of the cold end decreases further.

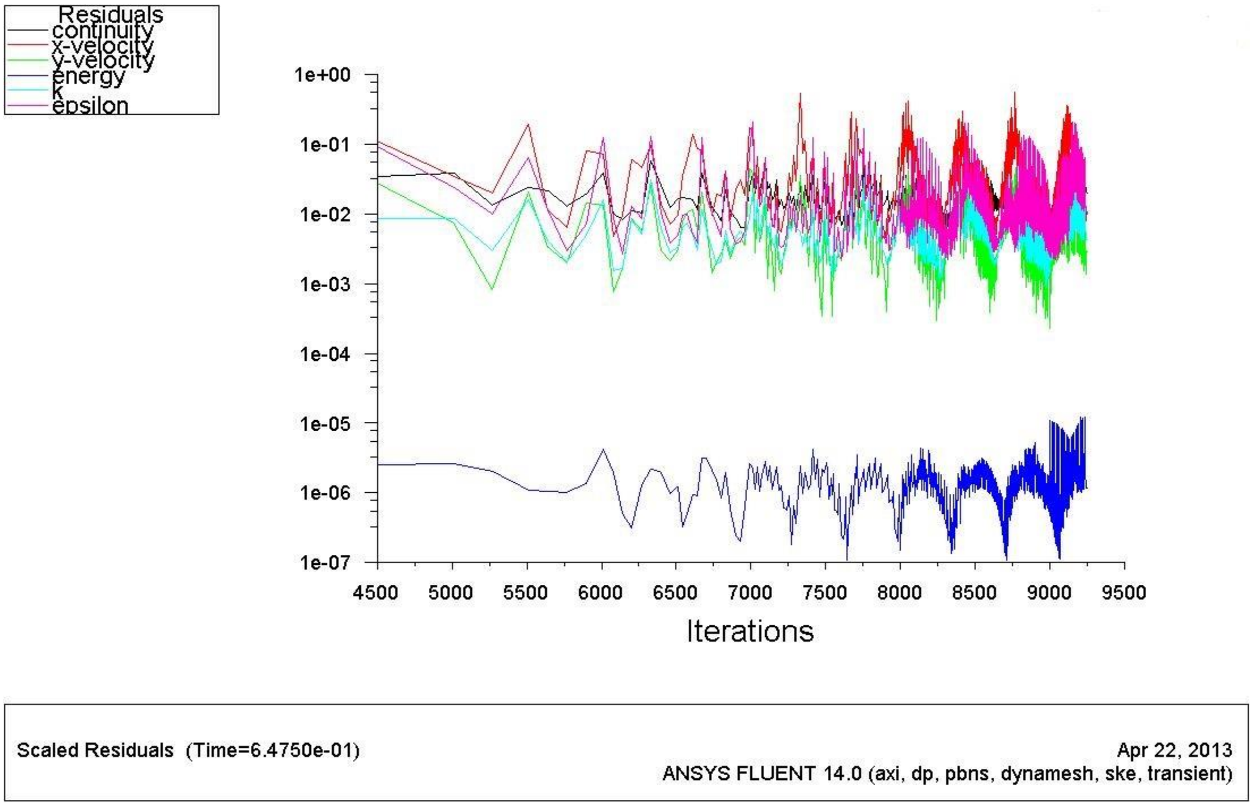

Fig.10.Residual plot of TE model at 0.5W load

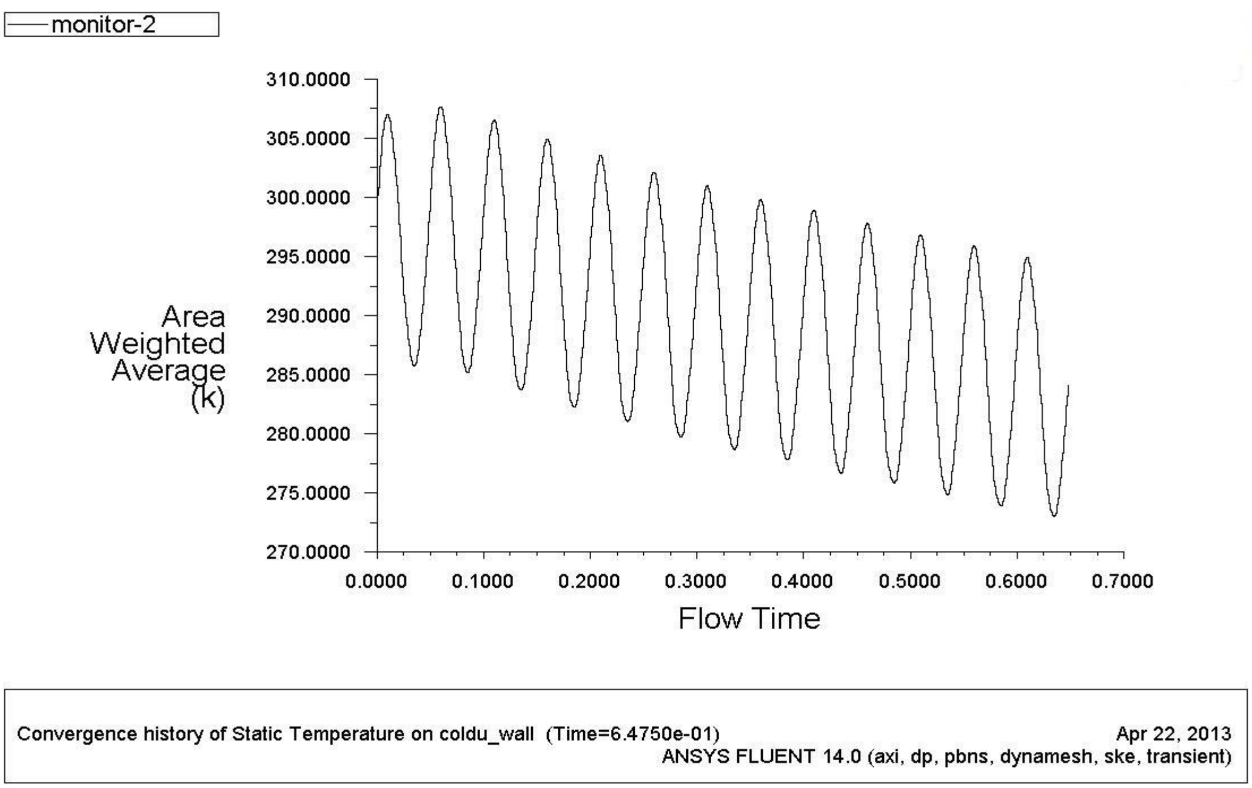

Fig.11. Sinusoidal temperature variation of TE model at 0.5W load

4.2.2. Temperature contour

The temperature contour shows different values of temperature in different region of the cryocooler. The uppermost and lowermost values of temperature are shown by red and blue colours correspondingly and in between different colours are used to represent different values of temperature. This is shown in Fig.12 below.

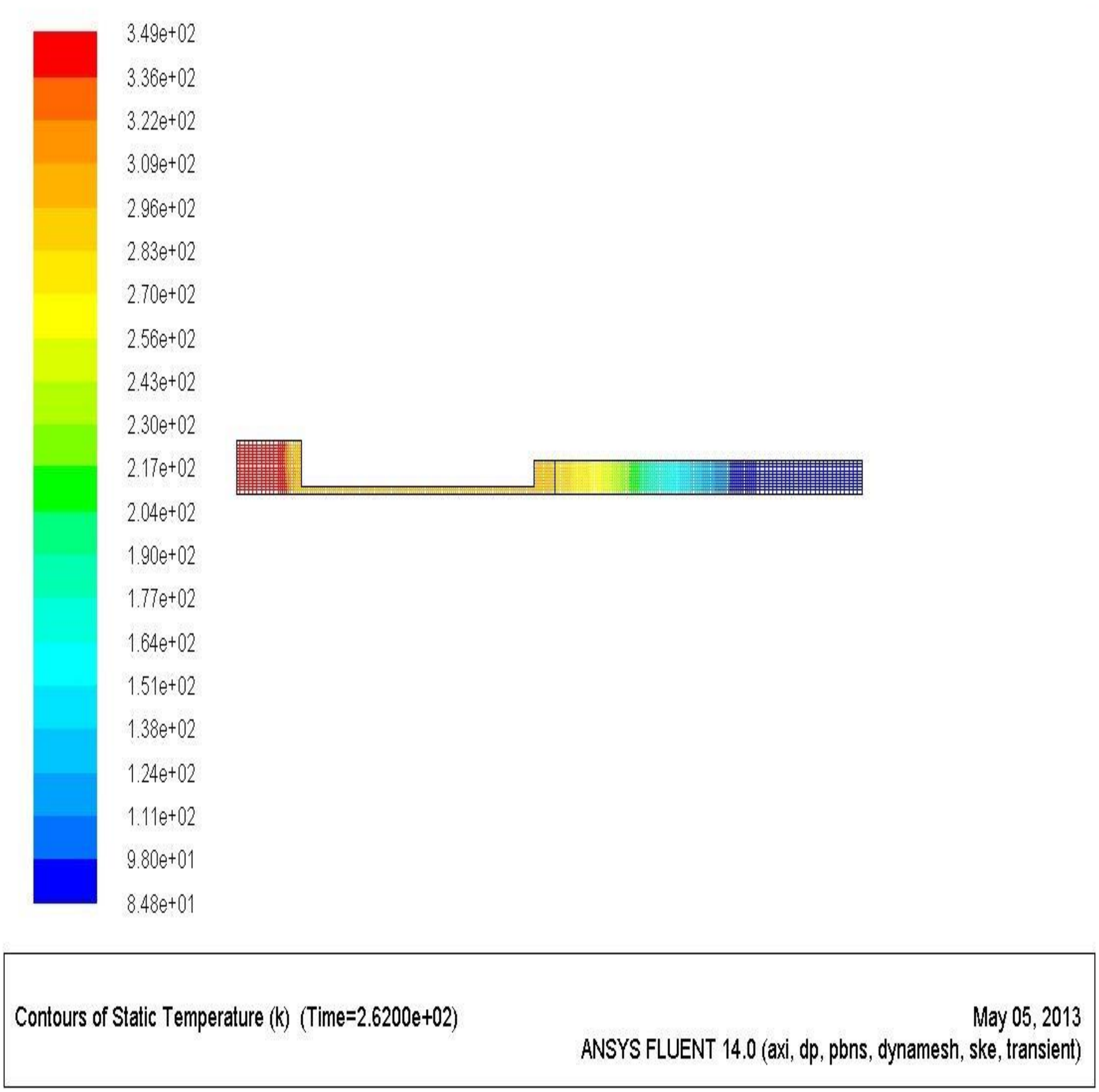

Fig.12.Temperature contour of TE model at 0.5W load

4.2.3. Temperature plot

Fig.13illustrates the temperature curves, showing the variation of temperature along the axial length of the Stirling cryocooler.

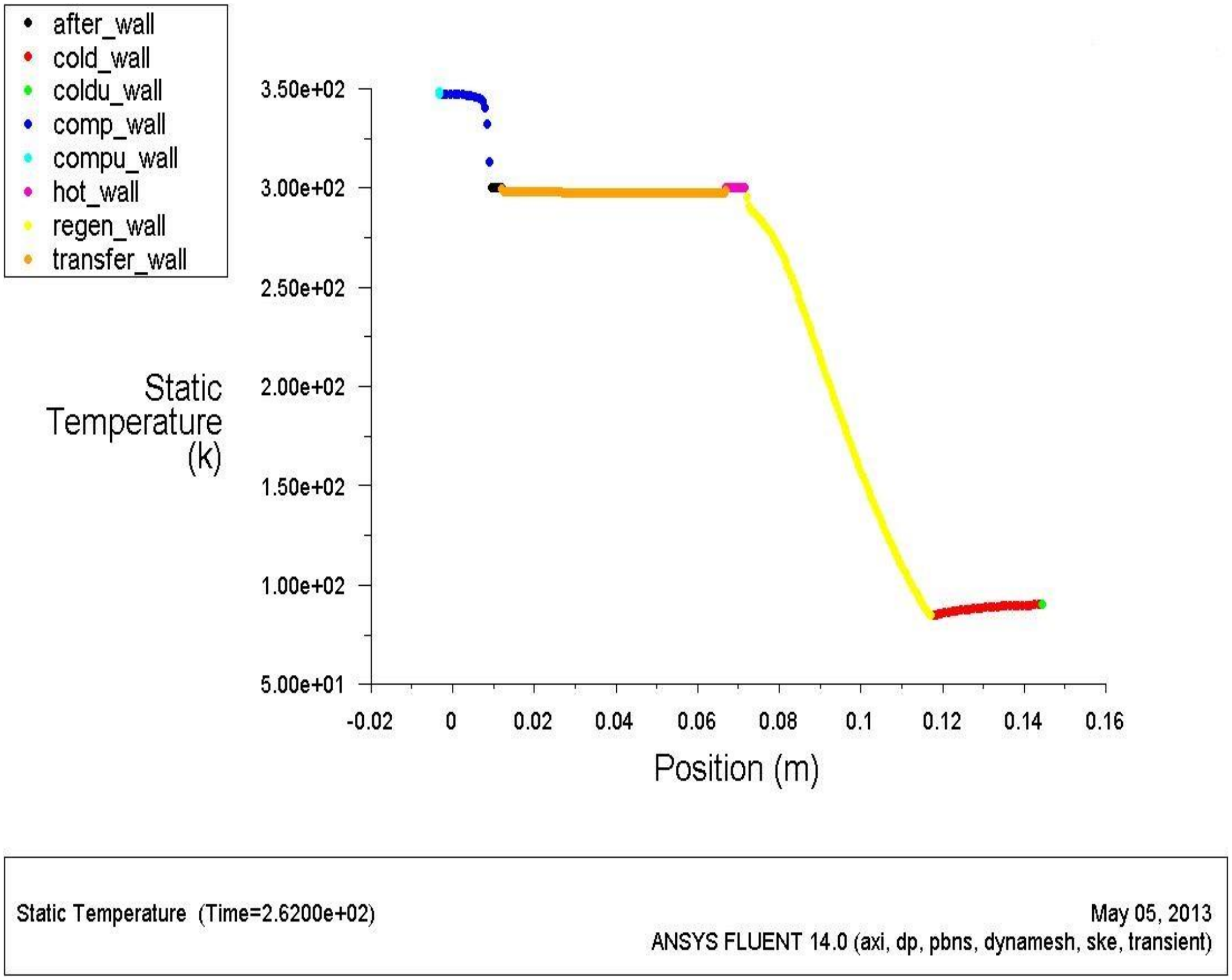

Fig.13. Temperature plot of TE model at 0.5W load

4.2.4. Velocity vector

The Fig.14 below shows how the fluid particles move inside the transfer line and regenerator of the Stirling cryocooler during simulation. The arrows here symbolize the directional vectors of the fluid particles whereas the magnitude of the velocity at each point is indicated by different colours whose demarcations are shown in the left side of the figure.

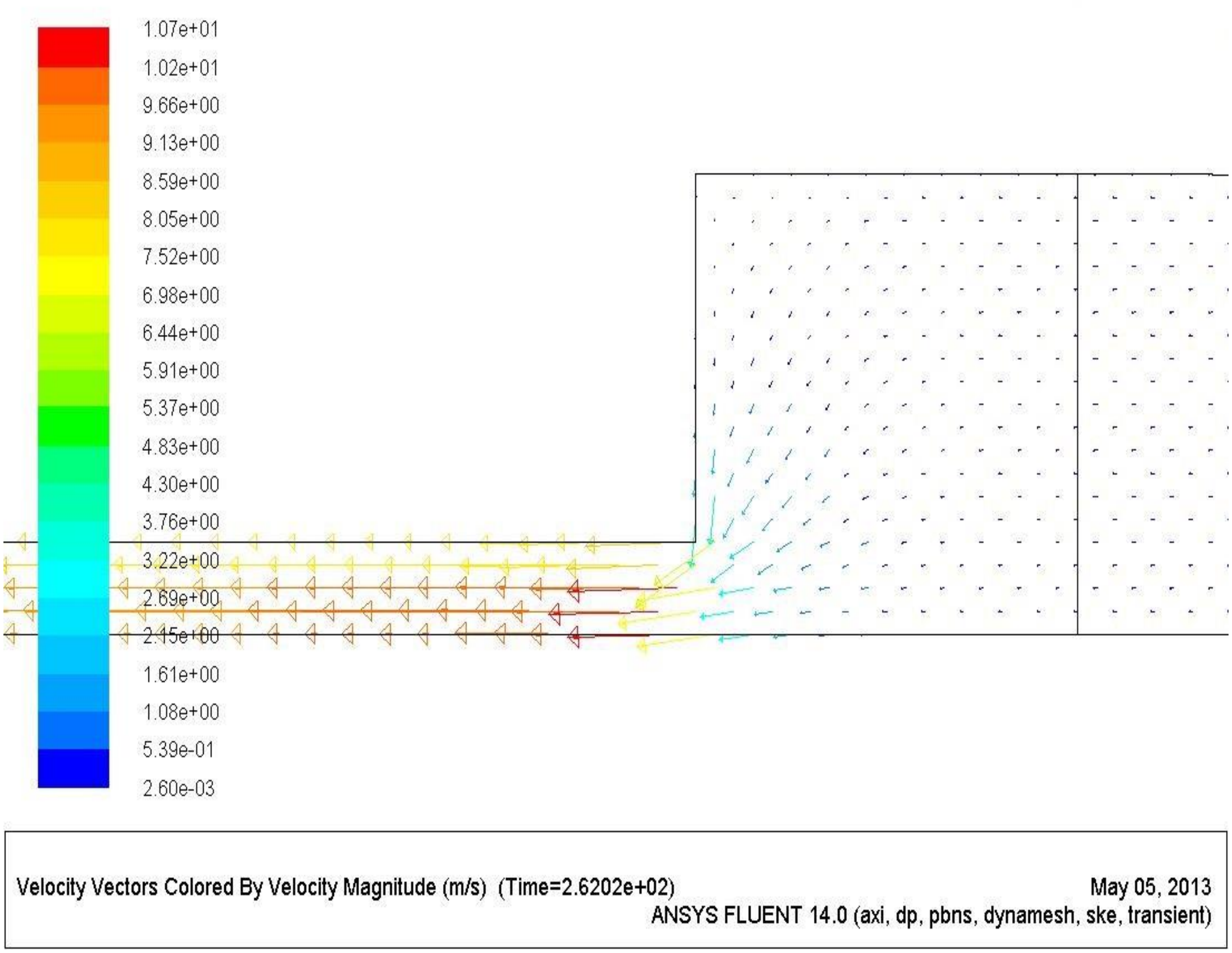

Fig.14. Velocity vector profile of transfer line and regenerator of TE model at 0.5W load condition

4.3. NON-THERMAL EQUILIBRIUM MODEL WITH NO LOAD

Again same geometry was simulated at the same 20Hz frequency by changing the thermal model to non-equilibrium thermal model which is relevantly available in ANSYS 14. No heat flux was applied to the cold end of the cryocooler. The residual monitor plot, cooling behavior, temperature contour and plots, and the velocity vector for above case are shown below.

4.3.1. Residual monitor plot and cooling behaviour

The cooling behaviour of the cold end and residual plot is shown in Fig.15 and Fig.16 respectively. The cooling behavior 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 with time.

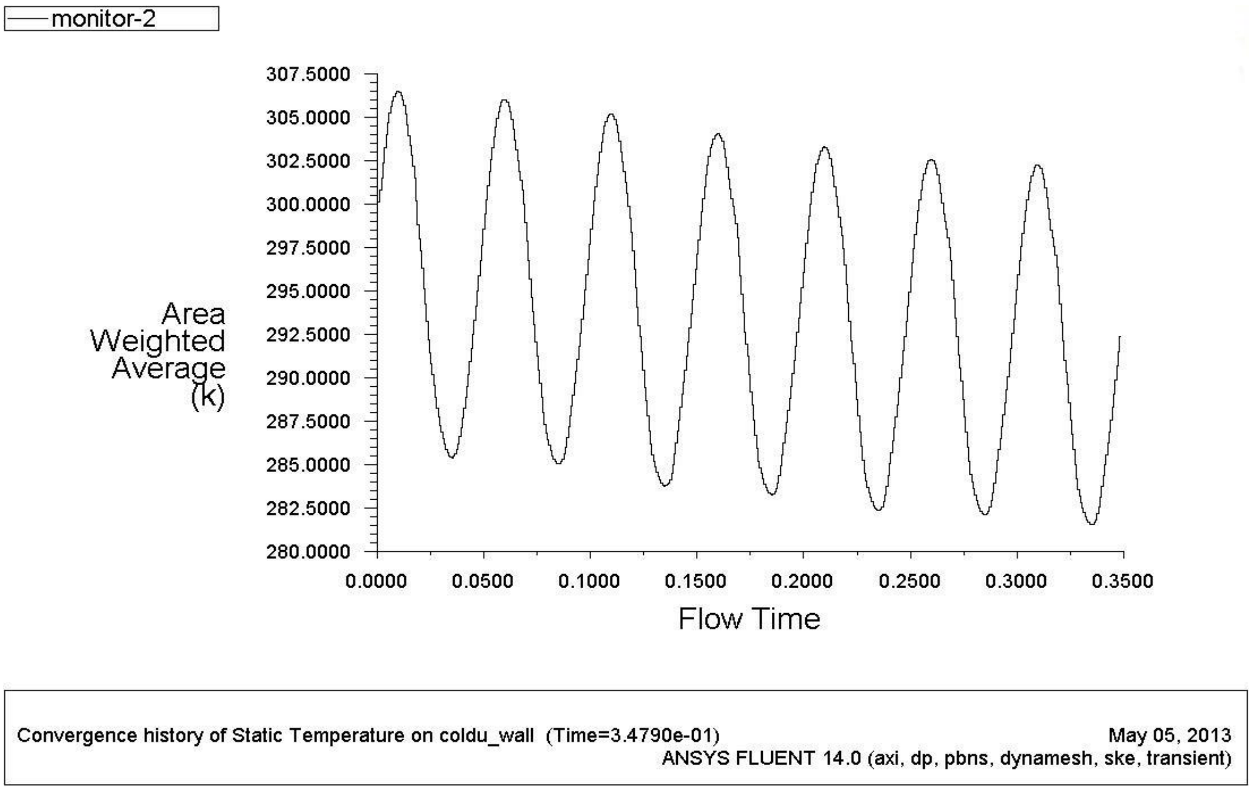

Fig.15. Sinusoidal variation of temperature of cold space of NTE model at no load condition

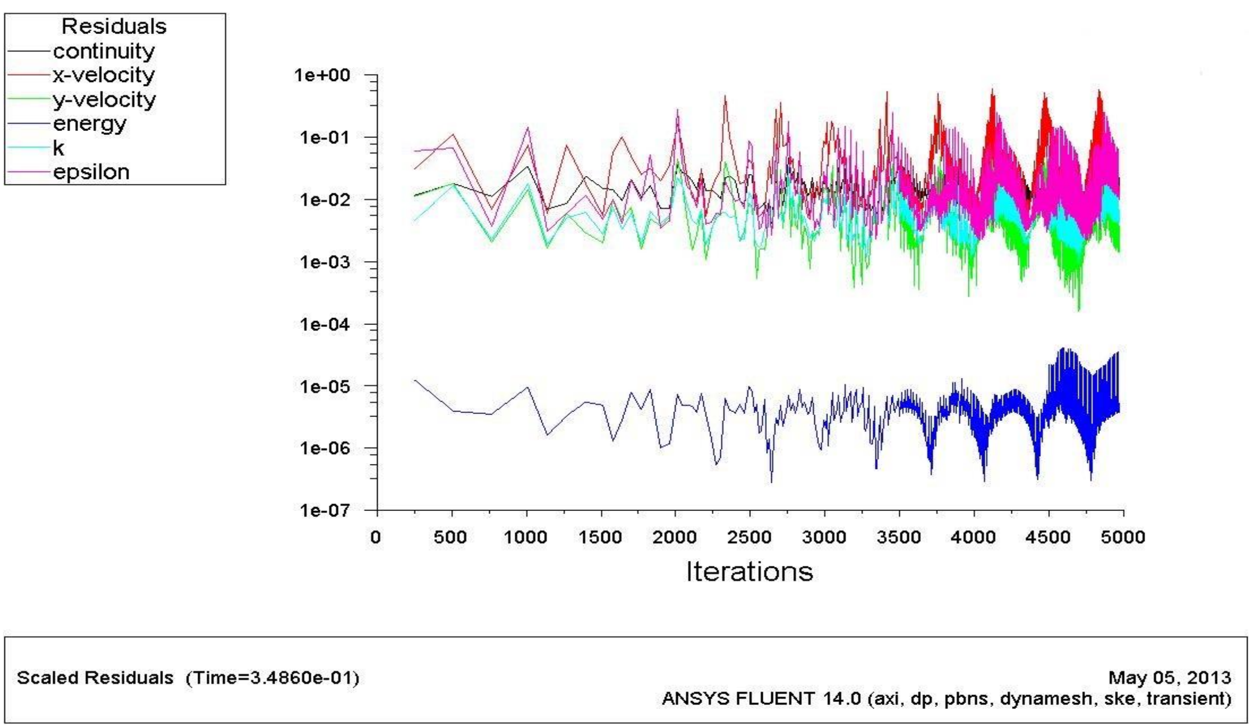

Fig.16. Residual Monitor Plot for simulation of NTE model at no load

4.3.2. Temperature contour

The temperature contour shows the temperature values at different region of the cryocooler. The higher and lower values of temperature are shown by red and blue colours respectively and in between different colours are used to represent different values. This is shown in Fig.17.

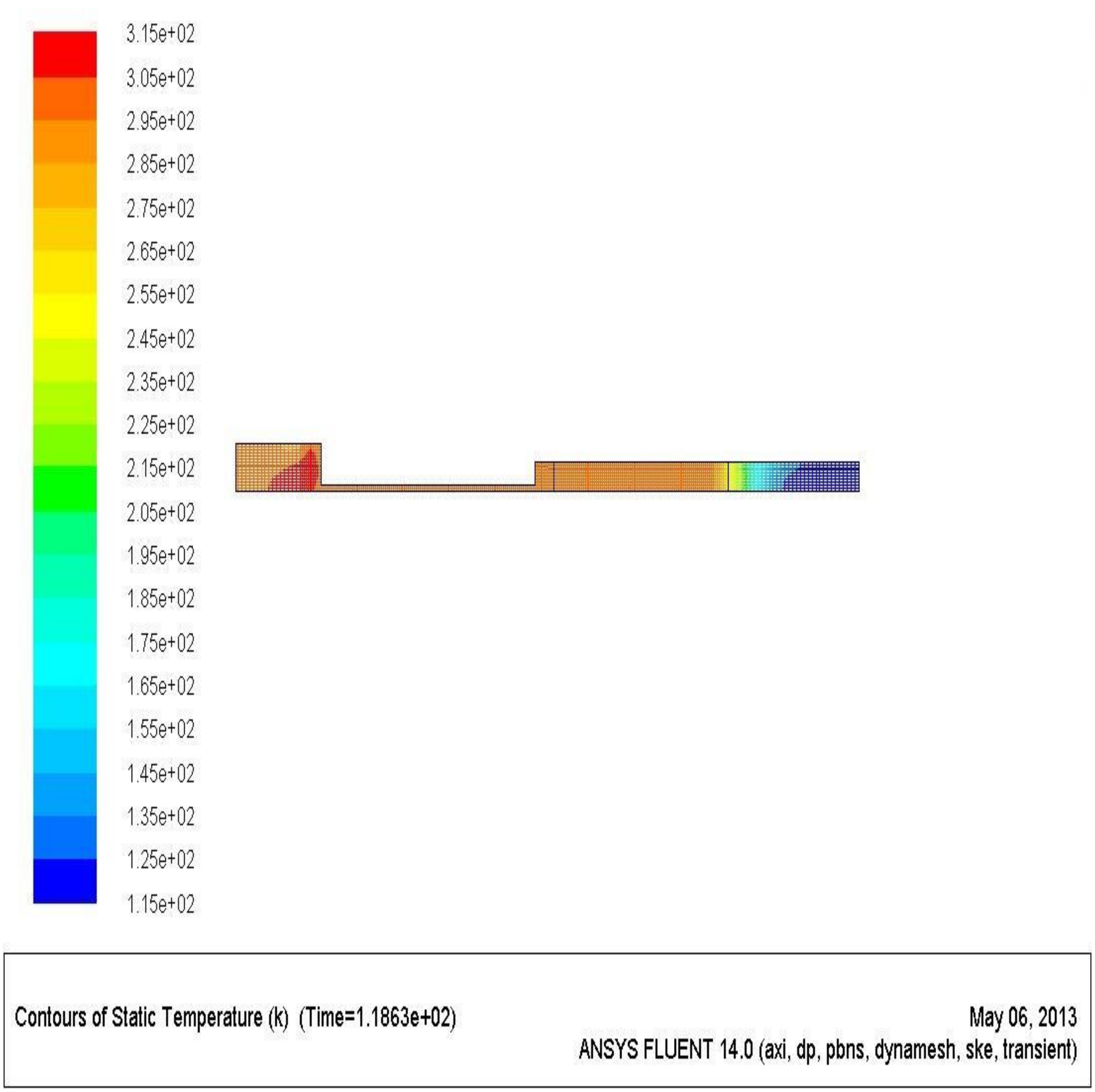

Fig.17. Temperature contourof NTE model at no load condition

4.3.3. Temperature plot

Fig.18illustrates the temperature curves, showing the variation of temperature along the axial length of the stirling cryocooler.

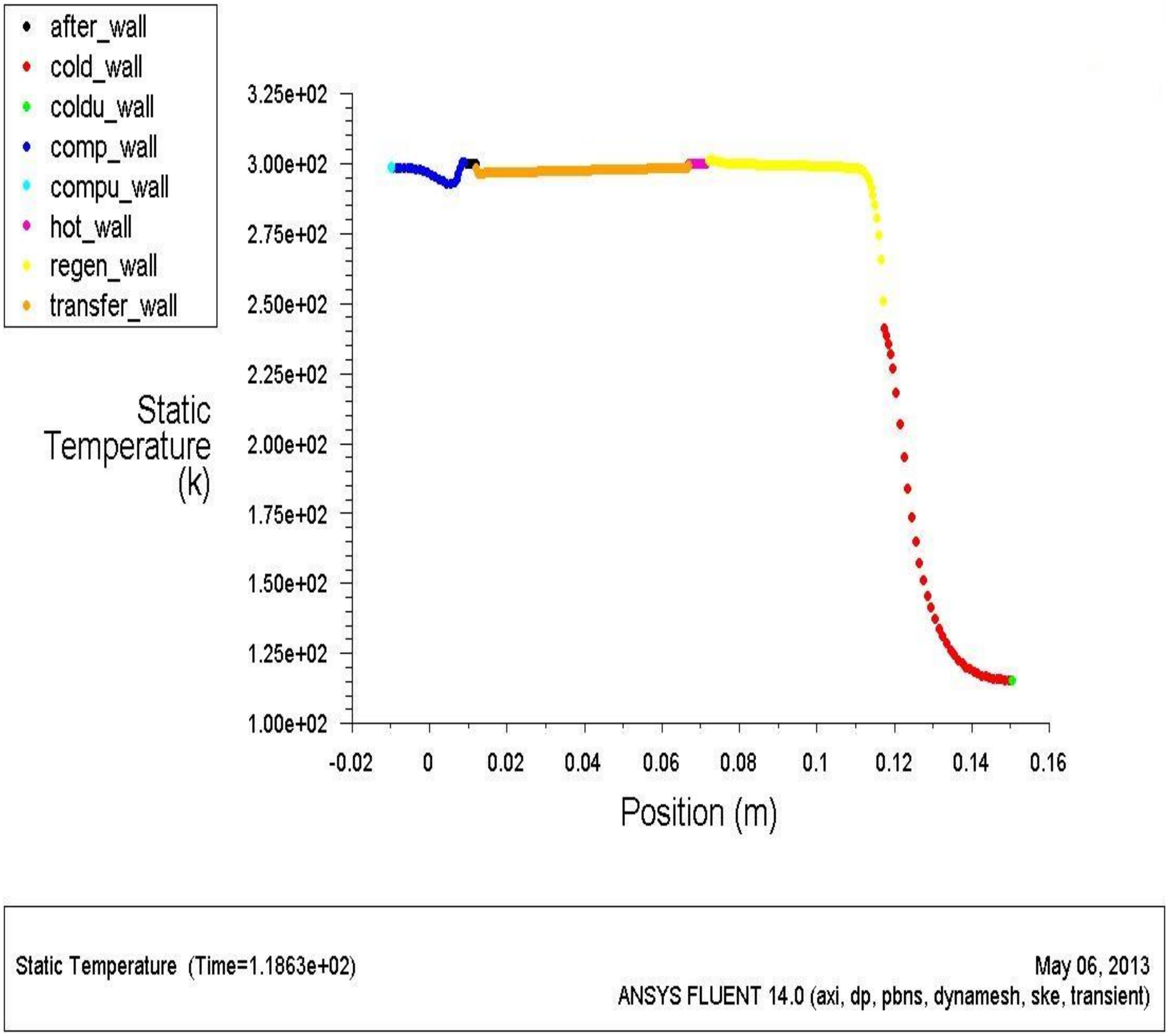

Fig.18. Temperature plot of NTE model at no load condition

4.3.4. Velocity vector

Fig.19 below shows how the fluid flow takes place inside the transfer line and regenerator of Stirling cryocooler. The arrow marks used here represent the velocity of the fluid particles and they are plotted in terms of velocity magnitude which is represented by their colours. The velocity of the fluid through the regenerator is very less due to the resistance offered to the flow.

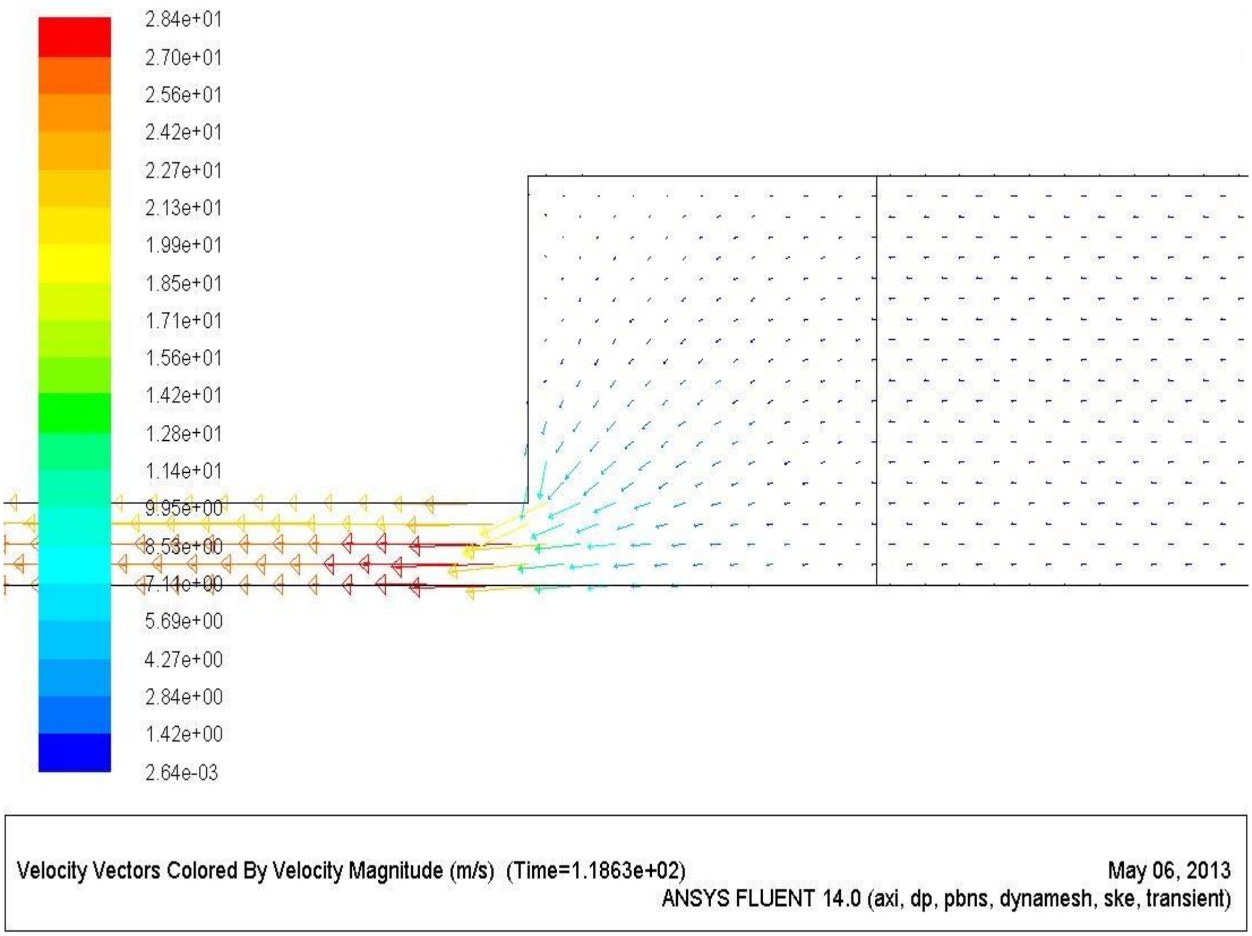

Fig.19.Velocity vector profile of transfer line and regenerator of NTE model at no load condition

4.4. COMPARISON OF TE MODEL'S NO LOAD AND LOAD CASE WITH NTE MODEL'S NO LOAD CASE

The simulations were performed in all three cases with the same 20Hz frequency of reciprocating piston and displacer. In first case of thermally equilibrium (TE) model atno load condition, the temperature obtained was lower than that obtained in the 0.5W load condition and that of non-thermal equilibrium (NTE) model at no load condition. The minimum temperature reached with TE model at no load case was 77K andthat with load condition was 85K.While for

NTE model at no load condition the minimum temperature attained was about 115K. The thermal equilibrium model exhibits rapid cooling followed by leveling off of the temperatures, while the thermally non-equilibrium model shows comparatively gradual cooling and stabilization of temperatures [17].The final temperature reached with thermal equilibrium model is lower than that of the thermally non-equilibrium model. This is due to the fact that the thermally non-equilibrium model implicitly takes into account the thermal losses in the regenerator. The thermal losses occurring in the porous zones are mainly responsible for the decrease in performance.The figure Fig.20 below shows the comparison between the three cases.

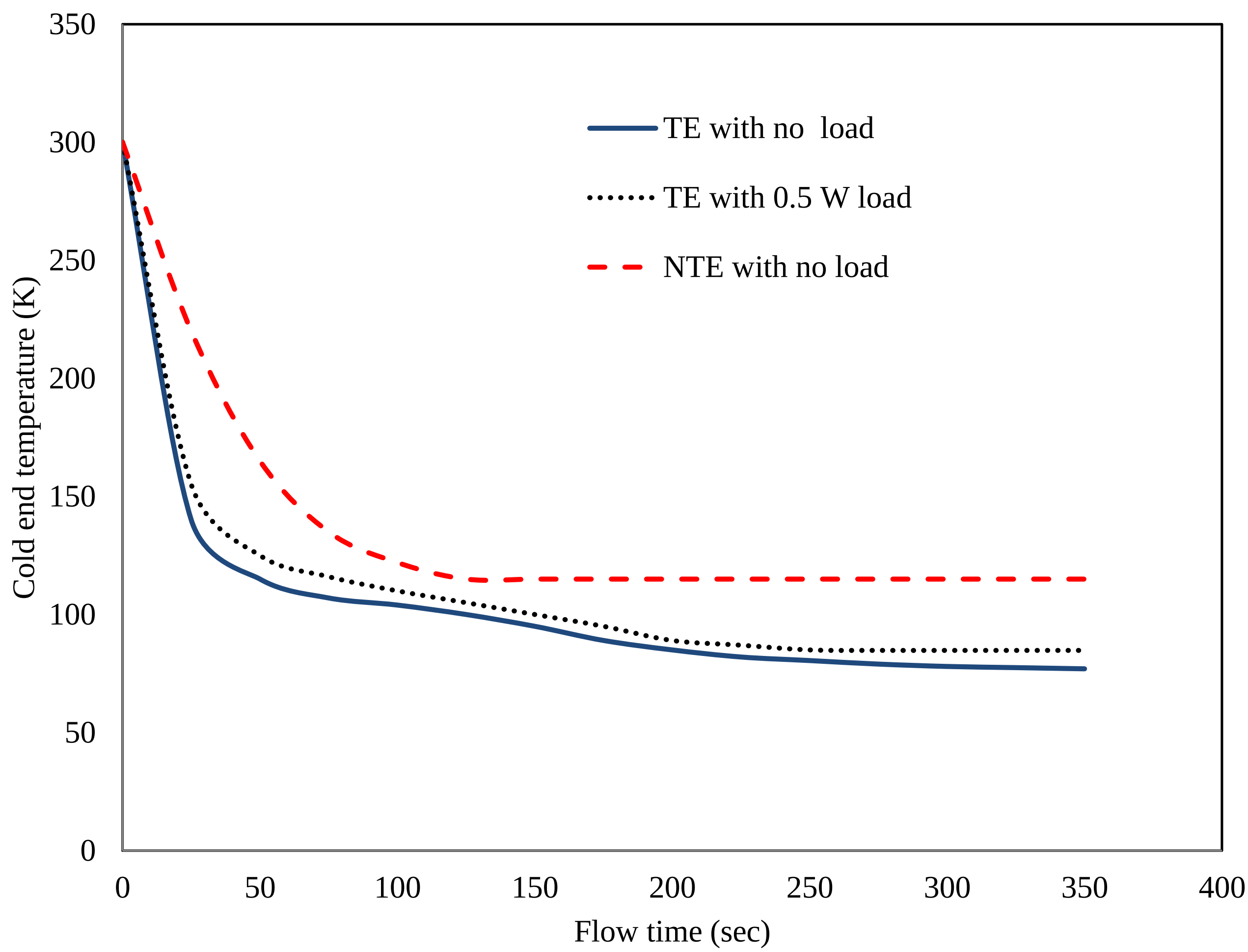

Fig.20. Cold end temperature Vs flow time plot at no load and 0.5W load condition for TE and NTE model.

CHAPTER 5

CONCLUSIONS

CFD is a powerful tool for solving a wide variety of industrial problems. Commercial general-purpose codes have the potential to solve a very broad spectrum of flow problems. CFD Simulations of the Stirling cryocooler were established properly. The motions of both piston and displacer were obtained by hooking up UDF'S to the compressor part and the displacer part at 20Hz of frequency. The results of three different cases were observed during and after the simulations of Stirling cryocooler. For TE model, the minimum temperature attained at the cold end was around 77K at no load condition with 20Hz of operating frequency and 85Kat 0.5W of load condition at the same operating frequency. Similarly, third case of NTE model at no load condition was also simulated and the minimum temperature obtained was about 115K. The comparison between the three cases shows that the minimum temperature attained is lowest in case of TE model at no load condition, and with the load condition the minimum temperature increases.

Suggestions for future work

1. Advancement of stirling cryocooler can be done in numerous fields in many ways. Firstly, the losses caused by the stirling cryocooler are now becoming the most imperative issue to consider about. Losses resembling pressure drop losses due to friction across the regenerator and also due to filling of regenerator void volume during pressurization and depressurization of the regenerator raised. Regenerator thermal loss takes place due to heat transfer to the surrounding by it [19]. Other losses we can consider are shuttle losses and acoustic losses. Shuttle loss is the heat loss occurs due to relative motion between the displacer. Its casing comes in to light due to the temperature difference created between them [20]. Thermal acoustic losses arise due to the oscillating flow of the working fluid i.e. helium inside that gap. There is a heat transfer loss [21] takes place due to the difference in the heat capacities of the two bounding walls and due to the relative motion of the two walls, this loss rest on pressure amplitude and flow velocities of the working fluid. A proper Schmidt analysis of above mentioned losses can be done thoroughly. By including their effects in the simulations, outcomes for an optimized geometry of regenerator can be achieved. As a result, minimum temperature of refrigeration can be attained while including all the constraints [22]. This can be prepared by making a non-equilibrium model of the case.

The entire simulations in this report performed in equilibrium model. The thermal conductivity of steel material of regenerator and other parts of the cryocooler also diverges with temperature in all the zones [23], so it should also be considered while writing the UDF'S.

2. The simulations performed in this report are completed by moving the last edge of the cold end in an exact phase relationship with that of the piston. But the truth is the displacer used to move to and fro inside the cylinder. Authentically a gap of very small thickness is maintainedbetween the displacer and the regenerator housing which is not considered in the above simulations performed. So all the above constraints required being included and then additional simulations can be gone through for obtaining the results of more concrete case. For obtaining this case first The gap model and displacer motion can be modeled separately 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 are considered in such case where there is no load in the cold space, and they are not totally justified. In reality the temperature will fluctuate but for performing this particular experiment the temperature of the cold wall was assumed to be constant. Similarly in both the cases of load and no load at cold space the temperature of the regenerator wall is assumed to be constant but in genuine case it will differ as some heat loss is produced at the regenerator wall. Same as the previous assumptions in this case also the thermal conductivity of the porous material and the cold space wall are taken to be constant but in real case it will diverge with temperature. These cases can be so these can be incorporated and therefore the further simulations can be performed.

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