Numerical Study and Analysis of Inertance-Type Pulse Tube Refrigerator

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Abstract:-This research article illustrates a numerical study of single stage coaxial as well as inline Inertance-Type Pulse Tube Refrigerator (ITPTR), which performance is mostly depending upon a regenerator. Regenerator is the significant component of a pulse tube refrigerator which has a great importance of producing cooling effect. In this present work a computational fluid dynamic (CFD) solution approach has been chosen for numerical purpose. The detail analysis of cool down behaviour, heat transfer at the cold end and pressure variation inside the whole system has been carried out by using the computational fluid dynamic software package FLUENT. A number of cases have been solved by changing the porosity of the regenerator from 0.5 to 0.9 and rest of parameter are remains unchanged. The operating frequency for all case is 34 Hz, pulse tube diameter 5mm and length is 125mm not changed for all cases. The result shows that porosity value of 0.6 produce a better cooling effect on the cold end of pulse tube refrigerator. The variation of pressure inside the pulse tube refrigerator during the process also analysed. To get an optimum parameter experimentally is a very tedious job for iterance tube pulse tube refrigerator. So the CFD approach gives a better solution which is the main purpose of the present work.

Key-Words: -Pulse tube, Refrigerator, ITPTR, Regenerator, Porosity

1. Introduction

In the history of cryogenics the invention of Pulse tube refrigerator was by Gifford and Logesworth [1] in the year of 1964S at Syracuse University. It was known as a Basic Pulse Tube Refrigerator (BPTR). Pulse tube refrigerator is used for helium liquefaction, magnetic resonance and aerospace. Pulse tube Refrigerator has the advantages of other type of cryocoolers due to its high reliability, no moving part at cold end region, low price, and low magnet interference. Mikulin [2] was the modifier who brought a modification to the BPTR by adding a small orifice valve in 1984, which is responsible for increase in efficiency. It is due to addition of small orifice causes improve in phase between velocity and temperature as a result more enthalpy flow near hot heat exchanger. Such types of Refrigerator are known as Orifice Type Pulse Tube Refrigerator (OPTR). And then after different researcher brought various new type of constructional modification to enhance the cooling capacity and cool down behaviour. Zhu [3] introduce Double inlet Orifice Pulse Tube Refrigerator (DIOPTR) in which a bypass is directly connected to reservoir inlet from the compressor outlet through which the surplus amount of fluid passes to compressor and decreases the regenerator losses. Minimum temperature of nearly 3.5K was achieved with Multi stage pulse tube refrigerator [4-6]. Beside from BPTR, OPTR and DIOPTR configuration discussed, variegated alternative configuration have been proposed. For a sample Zhu et al. [7] proposed an active-buffer pulse tube refrigerator in which more than one reservoir was attached. 80 K temperature was achieved with 11% of Carnot efficiency. Pan et al. [8] experimentally studied a 4 valve pulse tube refrigerator and archived a 57 K temperature at 100 W with an input power of 7 kW. The most recent type developed Pulse tube refrigerator is the Inertance-Type Pulse Tube Refrigerator (ITPTR). A long inertance tube is replaced instead of the small orifice in between the cold end and reservoir. The addition of inertance tube causes a batter phase relation between mass flow rate and pressure than orifice type. Cha et al. [9] first time gave a computational fluid dynamics approach to solve the single stage iterance pulse tube refrigerator using well CFD solution software FLUENT and studied the multidimensional flow effects. One major assumption of this CFD analysis using FLUENT package is the regenerator efficiency is 100 %. This is studied by Ashwin et al. [10] by considering a thermally non equilibrium medium of porous region inside the Regenerator. However there are so many fascinating investigations on mathematical as well as numerical analysis had been done by different researchers to study the multidimensional flow effect, nonlinear process, thermodynamic process and oscillatory flow.The main objective of this present paper is to study the pressure fluctuation as well as the cool down behavior at cold end due to varying in porosity inside the porous region. It is reported by the use of CFD simulation of the governing equation with the help of FLUENT software package.

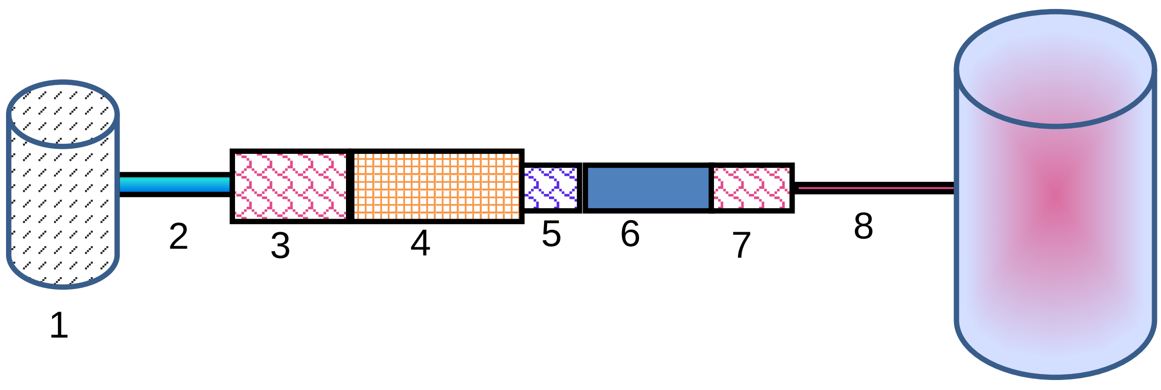

Fig.1. Schematic diagram of computational domain of ITPTR: 1 - compressor, 2 – transfer line, 3 - after cooler,4 – regenerator, 5 – cold heat exchanger, 6 – pulse tube, 7 – hot heat exchanger, 8 - inertance tube, 9 - reservoir.

Table 1. Dimensions of the ITPTR considered for the present analysis

constituent part	Diameter (m)	Length (m)	Boundary condition
(1) Compressor	0.01908	0.0075	Adiabatic
(2) Transfer line	0.0031	0.101	Adiabatic
(3) After cooler	0.008	0.02	300 K
(4) Regenerator	0.008	0.058	Adiabatic
(5) Cold heat exchanger	0.006	0.0057	Adiabatic
(6) Pulse tube	0.005	0.125	Adiabatic
(7) Hot heat exchanger	0.008	0.01	300 K
(8) Inertance tube	0.00085	0.684	Adiabatic
(9) Reservoir	0.026	0.13	Adiabatic

2. Description of Problem Geometry

Fig. 1.shows the line diagram of Inertance-Type Pulse Tube Refrigerator (ITPTR) model along with its individual associated components. The whole system consists of a compressor which produces the fluctuating pressure for the system and it is connected to a heat exchanger through a transfer line known as after cooler for extracting the heat to

surrounding generated by the compressor. The heart of the whole system is a regenerator, whose effectiveness is responsible for the cooling capacity of the pulse tube refrigerator. The cold heat exchanger is the cold region where the cooling load is supplied to the system. The heat is extracted from the system with the hot heat exchanger. In between hot heat exchanger and cold heat exchanger there is present a pulse tube which is responsible for producing the phase difference between mass flow rate and fluctuating pressure. The reservoirs store the working fluid and acting as the buffer. It is separated from cold heat exchanger either by an inertance tube for ITPTR or by an orifice in OPTR.

3. Details of Modeling

All The details modeling and meshing of the linear coaxial 2D axi-symmetric inertance pulse tube refrigerator model is done using the modeling software GAMBIT. The experimental nonlinear model is turned into equivalent present linier model. The dimensions of the whole system are shown in the Table 1. The numbers of grid points are varied from 3643 t0 5217 for the purpose of a grid independency test .To operate the compressor piston head a suitable DEFINE_CG_MOTION user define function (UDF) chosen from FLUNT user manual [13]. This UDF is written in C language, compiled and attached to the head of the piston. The two head piston oscillate about its center with the relation $a = a_{0*}\sin_*(\omega t)$. Where ‘a’is the piston displacement and $a_0 = 0.0045$m is the amplitude with a time increment of 0.0007s is assumed and the piston head velocity is related with the correlation $v = a_{0*}\omega_*\cos_*(\omega t)$. Where ‘v’ is the piston head velocity. For regenerator porous media region the parameters are taken from [10], inertial resistance 76090 m^{-1} and permeability 1.06×10^{-10} m^2. Steel is chosen as the matrix material for regenerator. The working fluid is chosen is helium and the various property (thermal conductivity, viscosity, specific heat) of the fluid are taken as temperature dependent from NIST data base. No wall thickness consider as the assumption for present analysis.

4. Mathematical Modeling

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

Continuity Equation

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

Momentum Equation

For axial direction

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

For radial direction

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

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

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

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

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

Energy Equation:

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

where

$$k = \xi k_f + (1-\xi)k_s$$
$$E_f = h - p/\rho_f + v^2/2$$

5. Initial and Boundary Conditions

The governing equations for 2D model as described above are solved by Fluent. The suitable boundary condition for the present case is shown in Table 1. The mean charge pressure is 30bar at which the total system operate. The sinusoidal pressure variation inside the whole system is due to the compressor head motion which is operated with a UDF as stated above. For an ideal condition of operation, simulation started with 300K operating temperature. The working fluid is helium, assumed for simulation with temperature dependent viscosity. The wall temperature of both the cold and hot heat exchanger is maintained at 300 K while the rest component walls are operated with adiabatic. For better solution convergence 40 inner steps with the time step for iteration of 0.0004s is needed. It will take a more than 15days to reach a periodic state with the CPU configuration of 8GB of RAM and 3.1 GHz processor.

6. Numerical solution procedure

The most important factor in simulation that suitable numerical scheme. Axisymmetric, unsteady, cell based second order implicit time; physical velocity with segregated solver is taken for analysis. PISO algorithm with a PRESTO (Pressure Staggered Option) scheme for the pressure velocity coupling is used for the pressure correction equation. The PRESTO scheme uses the discrete continuity balance for a "staggered" control volume about the face to compute the "staggered" (i.e., face) pressure. This procedure is similar in spirit to the staggered-grid schemes used with structured meshes. Note that for triangular, tetrahedral, hybrid, and polyhedral meshes, comparable accuracy is obtained using a similar algorithm. The PRESTO scheme for pressure interpolation is available for all meshes in Fluent. PISO (Pressure Implicit with Splitting of Operators) algorithm is used for better convergence. For unsteady problems it is an efficient method to solve the Navier-Stokes equations.Suitable Under relaxation factors for momentum, pressure and for energy had been used for the better convergence. Quad lateral as well as triangular cells were used for the computational domain. For all equation Convergence of the discretized equations are said to have been achieved when the whole field residual was kept at 10^{-6}.

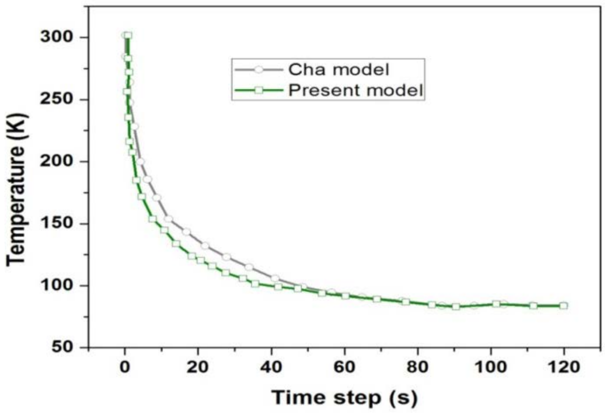

Fig. 2. Validation plot between present model and Cha et al. [9] model

7. Model validation

A model validation test is carried out against preceding published cryogenic journal by Cha et al. [9] model to figure out the accuracy of the model and solution method. It is noticeable from the Fig. 2 that the steady state temperature is reached after 120s for both the case after which there no change in temperature with respect to time. The Cha et al. model reporting a temperature of 87 K using 4200 number of cell where in the present case it is found a 86 K using 3900 cell.

8. Result and Discussion

8.1 Cool down behavior

The computational simulation werecontinuous till a steady periodic saturated state is achieved. In this condition the Facate Average (FA) temperature of the cold end achieves a steady state. After this steady state there no change in cold end temperature. In the present model it reach a temperature of 46 K with the supplied dimension, which is shown in Fig 3.

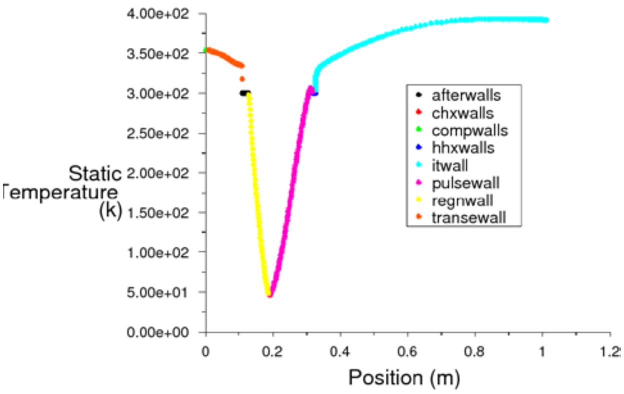

Fig.3. Axial temperature plot varying from compressor to Inertance tube after steady state temperature of 46K at cold heat exchanger

The whole system wall temperature is ploted here. The compressor temperature is above the operating temperature due to compression and expainsion of fluid inside it. This extra tepmerature generated inside the compressor is extracted to sorroundig through the aftercooler. Inside the regenerator due to presence of extra source term causes momentum loss. In source term the presence of two term such as Darcy andForchheimer terms in which pressure drop is directly popertional to velocity inside the porous zone. The temperature drop contour inside the regenaror is shown in the Fig 4. It is observed from the contures that the temperature at cold end is 46.5 K.

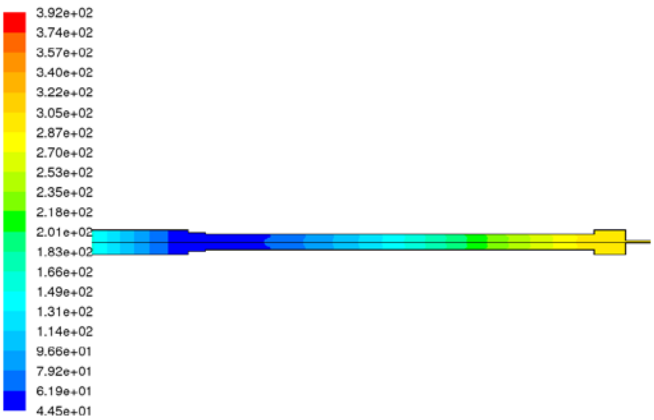

Fig.4. Temperature Contours showing the temperature variation inside pulse tube.

8.2 Varying pressure during the simulation

The Fig [5-9] shows the pressure variation during the simulation inside the whole system. The Area Weighted Average pressure variation inside the cold heat exchanger is shown in the Fig 5. which is sinusoidal in nature. The pressure variation inside the cold heat exchanger from 35.5 bar upto 28.5 bar which can be clear from the figure. Fig 6.shows the axial pressure variation inside the total system from compressor to reservoir when the piston is presence near to the BDC of cylinder. At that position the compressor pressure is 36 bar and the pulse tube pressure is 35 bar, where reservoir pressure is 32 bar. Fig7. shows the axial pressure inside the total system from compressor to reservoir when the piston is presence at middle position during expansion. At that position the compressor pressure is 26.5 bar and the pulse tube pressure is 31.5 bar, where reservoir pressure is 32.5 bar. When piston reaches to its top position during expansion process, the system axial pressure variation is shown in fig 8. It shows that the maximum pressure is inside the reservoir and minimum pressure inside the compressor and pulse tube pressure is 28.5 bar. During compression process from TDC to BDC a pressure variation plot is shown in Fig 9. From the pressure fluctuation analysis during one complete cycle it can be concluded that when there is maximum pressure at one end at the same time the other end of system has the minimum pressure. This was the main purpose of the study that to analyze the pressure variation through the whole system during a complete cycle.

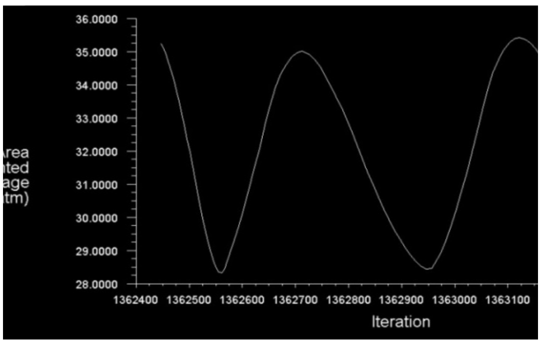

Fig. 5. Area Weighted Average Pressure inside cold heat exchanger vs. iteration curves.

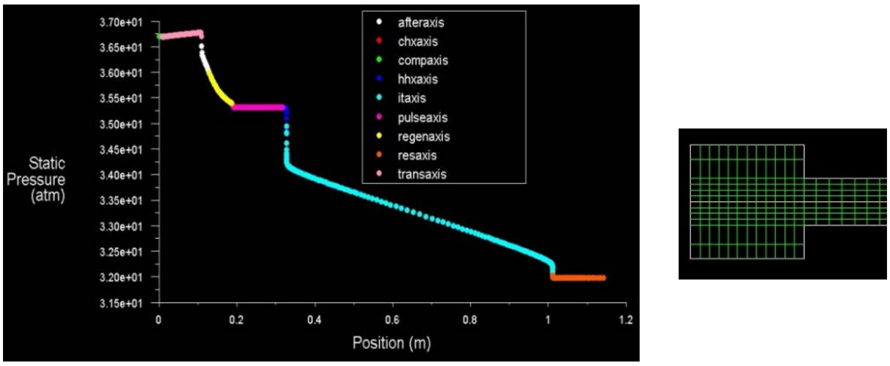

Fig.6. Axial Pressure inside the ITPTR, when piston is near the BDC

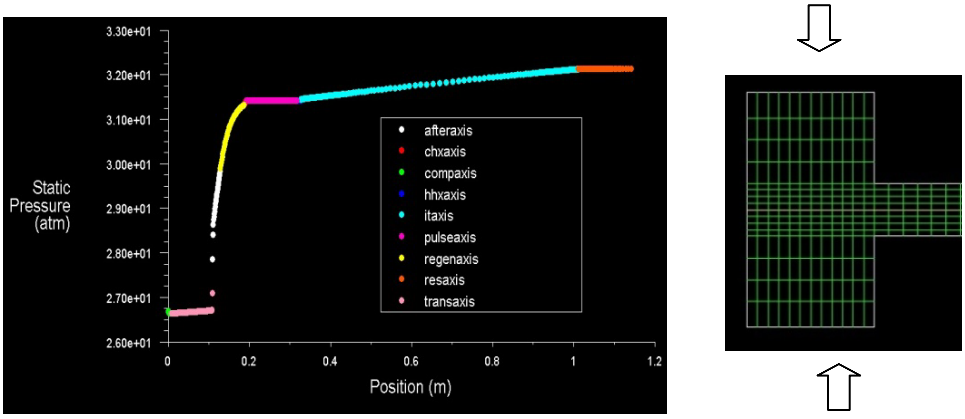

Fig.7. Axial Pressure inside the ITPTR, when piston is in middle position during compression

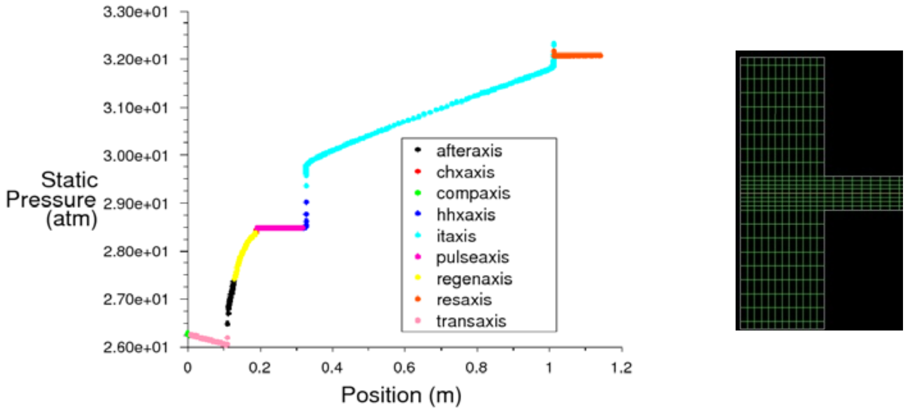

Fig.8. Axial Pressure inside the ITPTR, when piston is near the TDC

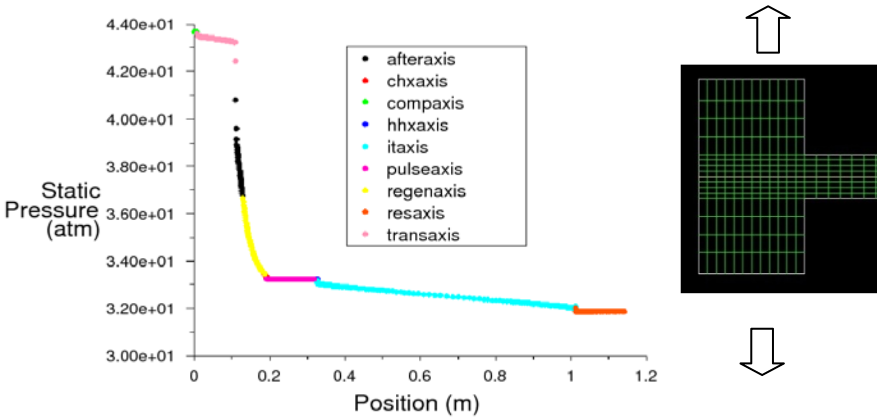

Fig.9. Axial Pressure inside the ITPTR, when piston is in middle position during expansion

8.3 Varying Porosity during the simulation

The most important part of the pulse tube refrigerator is a regenerator. The pulse tube refrigerator performance is depending upon the effectiveness of regenerator. In the present work a number of case studies have been carried out to investigate for which value of porosity it will give a better cooling temperature at the cold end. From the result of simulation it was found that at the porosity value of 0.6 the optimum result found relative to other value which is shown in the Fig 10.

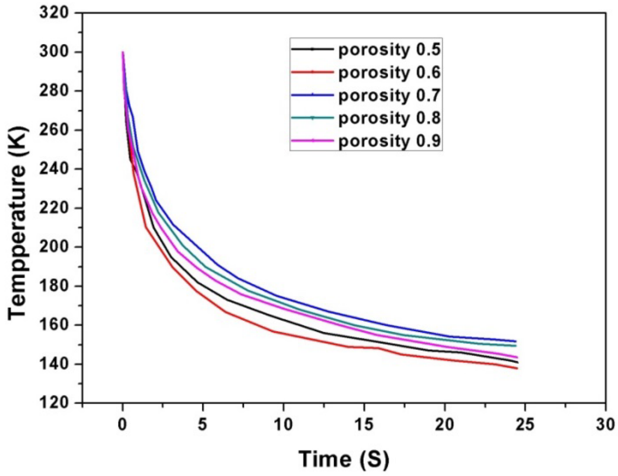

Fig.10. Cool down temperature vs. time for different porosity inside regenerator.

4 Conclusion

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

References

[1] W Gifford, and R.LongsworthPulse-tube refrigeration .Trans ASME, J EngInd (series B) 1964, 86, pp. 264–268.

[2] E. Mikulin, A. Tarasov, and M. Shrebyonock .Low-temperature expansion pulse tubes, Advances in Cryogenics Engineering 1984, 29, pp. 629-637.

[3]Z. Shaowei, W. Peiyi and C.Zhonggi, .Double inlet pulse tube refrigerator: an important improvement, Cryogenics 1990, 30, pp. 514–520.

[4]Y. Matsubara, and J.L. Gao, Multi-stage Pulse Tube Refrigerator for Temperature below 4K. Proceeding of 8th Cryocooler Conference, 1994, pp. 345.

[5]S. Zhu, Active-buffer Pulse Tube Refrigerator. Proceedings of 16th International Cryogenic Engineering Conference / International Cryogenics Materials Conference, 1996, pp. 291.

[6]C. Wang, S. Q. Wang ., J. H. Caiand Z. Yuan, Experimental study of multi-bypass pulse-tube refrigerator. Cryogenics, 1995. 35(9),pp. 555-558.

[7]S. Zhu, Y. Kakimi, and Y. Matsubara, Investigation of active-buffer pulse tube refrigerator. Cryogenics, 1997. 37(8),pp. 461-471.

[8]H. Pan, A. Hofmann, and L. Oellrich, Single-stage 4-valve and active buffer pulse tube refrigerators. Cryogenics, 2001. 41(4),pp. 281-284.

[9]JS Cha andSM Ghiaasiaan .Multi-dimensional flow effects in pulse tube refrigerators. Cryogenics 2006, 46, pp. 658-665.

[10]TR Ashwin ,GSVL Narasimham, S. Jacob.CFD analysis of high frequency miniature pulse tube refrigerators for space applications with thermal non- equilibrium model.Cryogenics 2010, 30, pp.152-166.

[11]L. Chen, Y. Zhang, E. Luo, T Li and X Wei.CFD analysis of thermodynamic cycles in a pulse tube refrigerator, Cryogenics 2010, 50, pp. 743-749.

[12]S. K. Rout, D. P. Mishra, D.N. Thatoi and AK Mishra, Numerical Analysis of Mixed Convection through an Internally Finned Tube," Advances in Mechanical Engineering. Advances in Mechanical Engineering. 2012(ID 918342), pp. 10 pages,

[13]Fluent 6.3 User's Guide, Fluent Inc.

[14]S. K. Rout, D.N. Thatoi, A. K Mishra, andD. P. Mishra, CFD Supported Performance Estimation of an Internally Finned Tube Heat Exchanger Under Mixed Convection Flow. Procedia Engineering, 2012. 38(0), pp. 585-597.