Numerical Investigations on Flow Resistance Values for Pulse Tube Cryocoolers

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ABSTRACT

Pulse tube cryocoolers are usually modeled as one-dimensional flow fields. It has been proven that this assumption holds only for components having large L/D ratios [4]. However, during these the CFD analyses of cryocoolers, there has always been some discrepancy between the modeled and the actual operating conditions. Commonly, a pulse tube cryocooler model consists of a compressor, an aftercooler, a regenerator, a pulse tube, hot and cold end heat exchangers, a phase shift device along with a reservoir. Modelling of the regenerator, hot and cold heat exchangers using realistic input values of viscous and inertial resistances prove to be crucial towards the assumption of one-dimensional flow fields. The present work deals with the numerical investigations on the effect of these viscous and inertial resistance values for all porous media. Analyses of individual porous components, viz. regenerator along with cold end heat exchanger and aftercooler along with hot end heat exchanger are executed. The base case taken under investigations is by Cha et al. [4]. The results of resistance values for unidirectional flow and at room temperature by using Ergun's equation are validated. The case is further modified for the operating pressure suitable to employ the resistance values for oscillatory flow and at cryogenic temperatures. It is concluded that there is a substantial difference in the predictions of the cryocooler performance, mainly in terms of the no load temperature. This makes the model much more realistic.

INTRODUCTION

The pulse tube cryocoolers are highly reliable and popular especially in space applications due to their simplicity in design and absence of moving part at the cold end pulse tube. Pulse tube cryocooler works on principle of pressurization and depressurization of gas to obtain refrigeration effect at the cold end of pulse tube. Pulse tube cryocoolers are popular for the wide range of applications in the area of medicine, defense, space and industries requiring liquefaction of gases like nitrogen and oxygen. [1-3].

Many researchers have investigated the flow analysis inside the porous media like regenerator used in the cryocooler. However, very few researchers have dealt with CFD analysis of complete cryocooler that uses the appropriate resistance values in case of porous media resembling to the actual working conditions of cryocooler. It is observed that analysis is performed by oversimplified resistance values for unidirectional flow at room temperature [4-5] while practical conditions in cryocooler is oscillatory flow at cryogenic temperature.

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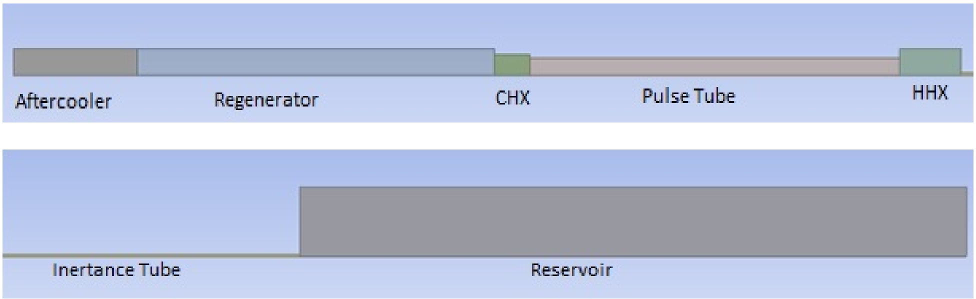

Figure 1. Model developed from Cha et al. [4]

To overcome these limitations, experimental and numerical investigations of the regenerator are performed. The viscous and inertial resistance values for oscillatory flow at room temperature for various matrix materials are determined [6-7]. Since the values obtained are at room temperature only, further investigations are done to determine these resistance values of oscillatory flow at cryogenic temperature [8]. These values are finalized by iterative adjustment to match the pressure drop from CFD modelling with that of experimentally obtained pressure drop.

Thus, it can be concluded from the literature that CFD analysis performed is not by using appropriate resistance values. Also, parametric effect of resistance values for different working conditions on cryocooler performance is not reported in depth. In view of these limitations, the objectives of present work are set towards investigations of complete cryocooler using various resistance values available in literature, and determine their effects on the performance of cryocooler.

MODEL DEVELOPMENT

Cha et al. [4] have performed the CFD analysis of a complete cryocooler using commercially available ANSYS® Fluent software. In order to validate the methodology of numerical analysis, the model in [4] is reproduced and CFD analysis is executed. The modeling and meshing are done in ANSYS® Fluent itself. Quadrilateral type of mesh with 4319 number of nodes is used for meshing against the 4210 number of nodes as reported in [4]. A user defined function is used to give the sinusoidal pressure wave as an input to the regenerator. The dimensions, working conditions, and boundary conditions used for analysis are same as that of [4]. The model developed and meshing performed is as shown in Figure 1 and Figure 2. The working pressure and frequency are 34 bar and 34 Hz respectively for the initial simulations.

Porous media like regenerator and cold/hot end heat exchangers in cryocooler have vital impact on performance of cryocooler. It is mandatory to employ appropriate boundary conditions and the viscous and inertial resistance values for all these porous media. Porous media used for this analysis consist of stainless steel wire mesh of #325 size. For clear fluid region (non-porous media) mass and momentum equations are directly solved while volume averaging method is used for deriving

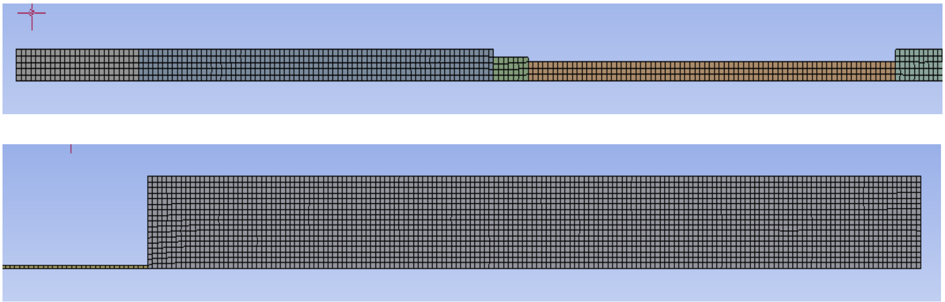

Figure 2. Meshing of model developed

these equations for a porous media. The mass and momentum equations for a porous media can be written as [9]:

$$\varepsilon \frac{\partial \rho}{\partial t} + \nabla \bullet \left(\rho \langle u \rangle \right) = 0 \quad (1)$$

$$\frac{\rho}{\varepsilon} \times \left\{ \frac{\partial \mu}{\partial t} + \frac{1}{\varepsilon} \langle u \rangle \bullet \nabla \langle u \rangle \right\} = -\nabla P + \mu \bullet \nabla^2 u - \nabla \bullet \rho \bullet \overline{u'} + S \quad (2)$$

The momentum source S is non-zero in the porous region, and is given by:

$$S = -\left(\frac{\mu}{k} \langle u \rangle + \frac{C}{2} \bullet Q \bullet \left| \langle u \rangle \right| \langle u \rangle \right) \quad (3)$$

where, u is the volume averaged velocity; C and k are the inertial resistance and permeability of the porous media, respectively.

The energy equation for the porous media can be written as

$$\left(\rho \times c_p \right)_f = \left\{ \frac{\partial T}{\partial t} + \nabla \bullet \langle T_f \rangle \times \langle u \rangle \right\} = \varepsilon \times \nabla \bullet \left(k_{eff} \times \nabla \bullet \langle T_f \rangle \right) \quad (4)$$

$$\left(\rho \times c_p \right)_f = \varepsilon \times \left(\rho \times c_p \right)_f + \left(1 - \varepsilon \right) \times \left(\rho \times c_p \right)_s \quad (5)$$

$$k_{eff} = \varepsilon \bullet k_f + \left(1 - \varepsilon \right) k_s \quad (6)$$

To analyze the effect of resistance values at different working conditions on the performance of cryocooler, various cases are made which are listed in Table 1. It is reported that the resistance values by Perrella et al. [8] are valid for the pressure range from 10 bar to 28.6 bar. To use these values, the charge pressure is reduced from 34 bar [4] to 20 bar and pressure ratio to 1.5, based on the majority of cases reported in literature. The operating frequency is kept same as that of [4] i.e. 34 Hz. All of the other boundary conditions, dimensions and working conditions are kept same for these investigations.

During all the simulations, helium gas is considered as an ideal gas. It is necessary to consider the variation of properties of helium and stainless steel with respect to the temperature, especially for cryogenic range and higher operating pressures. Hence, using the data from NIST WebBook [9], a higher order polynomial is fit for dynamic viscosity and thermal conductivity of helium and stainless steel.

Table 1. Various cases for Cha et al. [4] model

	Description of case	Viscous resistance (m^{-2})	Inertial resistance (m^{-1})
Case 1	Base case [4]	9.433×10^{9}	76090
Case 2	All components use resistance values of oscillatory flow at room temperature [6]	1.556×10^{10}	67000
Case 3	Only CHX and regenerator use resistance values of oscillatory flow at cryogenic temperature [8]	1.68×10^{10}	29700
Case 4	All components use resistance values of Oscillatory flow at cryogenic temperature [8]	1.68×10^{10}	29700

The polynomial is valid for the temperature range of 20-350 K. These expressions obtained for helium gas are as follows [10],

Dynamic Viscosity,

$$\mu = \left(1.195\times10^{-17}\right)T^5 - \left(1.133\times10^{-14}\right)T^4 + \left(4.1544\times10^{-12}\right)T^3 - \left(7.606\times10^{-10}\right)T^2 + \left(1.243\times10^{-7}\right)T + 1.991\times10^{-6} \quad (7)$$

Thermal Conductivity,

$$k = -\left(1.051\times10^{-12}\right)T^4 + \left(1.376\times10^{-9}\right)T^3 - \left(8.43\times10^{-7}\right)T^2 \times 0.0006\,T + 0.0235 \quad (8)$$

Using the same analogy, the expressions for stainless steel are as follows

Specific Heat,

$$C_p = \left(3\times10^{-5}\right)T^3 - 0.0218\times T^2 + 0.06089\,T - 146.96 \quad (9)$$

Thermal Conductivity,

$$k = \left(6\times10^{-7}\right)T^3 - 0.0004\times T^2 + 0.1158\,T + 1.057 \quad (10)$$

Standard k- Ɛ turbulence model with standard wall temperature is used. Aftercooler, cold and hot end heat exchangers and regenerator are modeled as porous media. Second order solution methods are adopted to have better accuracy. Local thermal equilibrium is considered between the mesh material and working fluid. The wall thickness for all components of cryocooler is neglected [11].

RESULTS AND DISCUSSIONS

Initially, validation is successfully executed to confirm the methodology. Figure 3 shows comparative cooldown curves. It can be seen that the steady state cycle average temperature obtained by simulation at cold end of pulse tube is 87.1 K against the 87 K as reported in [4]. Figure 4 shows the axial temperature distribution by simulation after normalization of the length of cryocooler between aftercooler and hot end heat exchanger, i.e. length of inertance tube and reservoir is not considered. It confirms that results obtained from present analysis are nicely in agreement with actual results reported in [4]. It is further observed that minimum temperature is achieved at the location of slightly more than half of the normalized length, which is as per the requirement of optimum performance of entire cryocooler unit.

Figure 5 shows the temperature contour plots for the cryocooler. It can be confirmed that the lowest temperature is achieved at the cold end of pulse tube with constant isotherms obtained axially, as required theoretically.

Further, investigations are done for different cases as listed in Table 1 to examine the effect of change in viscous and inertial resistance values on the performance of cryocooler for different working conditions. Initially, model using same resistance values from Cha et al. [4] is simulated for

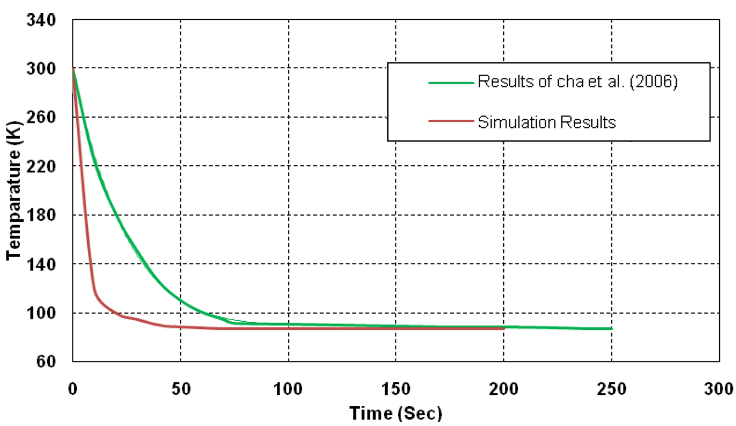

Figure 3. Comparative cooldown curve

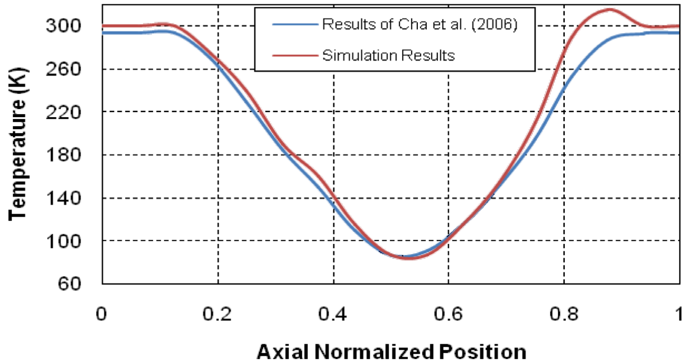

Figure 4. Axial temperature distribution along the length of PTC

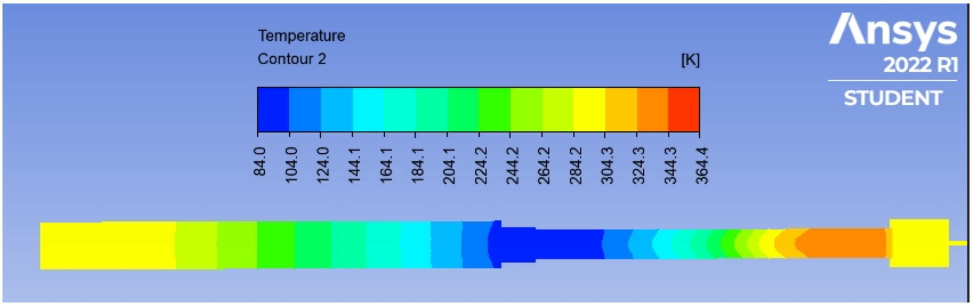

Figure 5. Temperature contour plot

the reduced pressure of 20 bar so that comparison can be made between all cases. The steady state cycle average temperature of cold end of pulse tube for all cases is listed in Table 2 given below. It is clear that the steady state cycle average temperature at cold end of pulse tube decreases when resistance values are changed from unidirectional to the oscillatory flow. Similar trend for cycle average temperature is observed when resistance values used are changed from room temperature to cryogenic temperature.

CONCLUSIONS

The major objective of this work is to investigate the effect of flow resistance values on the performance of cryocooler for different working conditions. Accordingly, a CFD model is created using ANSYS® and simulation results are successfully validated with literature to confirm the methodology for further analysis. Further, number of cases are created using appropriate resistance values for different working conditions in such a way that flow type changes from unidirectional to oscillatory and temperature from room to cryogenic range in stepwise manner. This helps to analyze the effect of each parameter separately on the performance of cryocooler. The resistance values reported in literature are valid within specific range of pressure. Accordingly, the charge pressure is adjusted to replicate the cases. The case using the resistance value for oscillatory flow

Table 2. Results for cases of Cha et al. [4] model

	Steady state cycle average temperature at cold end of pulse tube (K)
Case 1 (Base case)	111.92
Case 2	110.55
Case 3	108.19
Case 4	105.24

at cryogenic temperature yields lowest temperature among all the cases, as expected theoretically. The ideal case to resemble the actual working conditions for any pulse tube cryocooler would have after cooler and hot end heat exchanger at room temperature, while regenerator and cold end heat exchanger at cryogenic temperature range. Hence, the flow resistance values associated with such conditions would result much realistic predictions. Such case is investigated and results in the cold end temperature slightly higher than the earlier case. However, being more realistic, this result is more appreciated. Hence, it can be affirmed that the corresponding flow resistance values are to be employed for any computational simulation to predict pragmatic performance of the entire pulse tube cryocooler unit.

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