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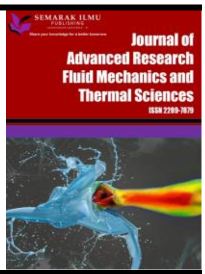

Numerical Investigation of a Heat Exchanger with Annular Fins for Cryocooler Application

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ABSTRACT

In this paper, the heat transfer characteristics of a typical hot heat exchanger used in pulse tube cryocooler have been numerically investigated. The numerical simulation is performed with the help of the commercial tool Ansys Fluent® to study the effect of different shapes and sizes of fins on the rate of heat transfer. The major parameters of the analysis include the diameter and pitch of the fins, the oscillating temperature at the inlet and outlet of the heat exchanger, and its operating frequency. A numerical investigation is conducted for the annular fin to illustrate its effect on the rate of heat transfer by calculating its Nusselt number and heat transfer coefficient. The effect of with fin and without fin on temperature and pressure variations has been studied for different angles in a cycle. The vortex development mechanisms within the gap between fins have been analyzed and compared with the vortex development mechanisms without fin. Results will be helpful for better design of heat exchangers for cryocoolers.

1. Introduction

Ever since the introduction of cryocoolers by Gifford and Longsworth [1], there has been regular attempts to improve its performance [2]. An investigation of natural convective flow in a cryogenic tank was conducted numerically using the liquid-vapor mixture model [3]. A numerical performance study of a PCM-based multitube heat exchanger integrated with two innovative fin designs is conducted in order to enhance the heat exchanger design [4]. In the past decades, the uses of cryocoolers have been increased considerably for both military and space-based applications. A typical cryocooler consists of a linear compressor, a transfer line, an aftercooler, a pulse tube, cold

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and hot heat exchangers, a phase shifting mechanism (an orifice valve/an inertance tube/a double inlet valve with a buffer in series) and a regenerator. Amongst these, the linear compressor plays the vital role for the pressure wave generation. Besides the linear compressor, heat exchangers are essential for improving the performance of the cryocooler by releasing the excess heat that arises during the compression [5]. Thus, optimization of the performance of heat exchangers concerning to their geometric parameters could enhance the performance of the cryocooler [6,7].

The resultant flow from the linear compressor can be characterized as oscillatory flow. Unlike commonly observed unidirectional flow for both fluids in most heat exchangers, there exists an oscillatory flow in one fluid that flows inside the heat exchangers used in the cryocooler. Outside the heat exchanger, steady air flow occurs [8,9]. Sometimes, water flow is also arranged to enhance the cooling rate [10-12]. Analysis of the heat transfer processes of this type of complex heat exchanger using Ansys Fluent is the target of this study. Te major methods of external cooling of those heat exchangers used in most practical applications are either by natural convection cooling or by forced convection cooling. The former is mostly applicable due to its characteristics of no moving parts, vibration free operation and nonexistence of any maintenance activity. But, as elucidated by many researchers, the rate of heat transfer by natural convection is minimal. Therefore, in order to enhance the heat transfer whilst maintaining vibrations free and maintenance free operation, the use of fins was suggested by Yüncü and Anbar [13] and Yildiz and Yüncü [14]. Furthermore, the effect of annular fins on the cylindrical surfaces was studied by Yildiz and Yüncü [14], in which they have illustrated the effect of fin spacing on the rate of heat transfer. Effect of different parameters on the rate of heat transfer processes of heat exchangers have been studied in previous references [15-19].

The typical geometrical and operating parameters of the fin like its shape, size, pitch, frequency, etc. play a vital role in determining the effective area of heat transfer. Thus, the current investigation illustrates the effect of aforementioned vital parameters on the thermo-fluidic characteristics of the heat exchanger.

2. Problem Formulation

Figure 1 shows the schematic of a single-stage pulse tube cryocooler including the location of heat exchangers. The axisymmetric computational domain of the heat exchanger is drawn in Figure 2. The computational geometry consists of a 2D axisymmetric cross-section of the heat exchanger with fins, surrounded by air. The heat exchanger gas flow path is filled with a porous media through which the pulsating fluid flows. The fins are of uniform thickness of 2 mm and a constant pitch of 2 mm. Copper is chosen as the solid material for heat exchanger canister and fin because of its higher thermal conductivity. In most practical application, copper is also used for making such heat exchanger [8-12]. The hollow heat exchanger canister is filled with copper mesh to enhance the rate of heat transfer. It is operated at 5 Hz frequency with a pressure rating of 25±5 bar.

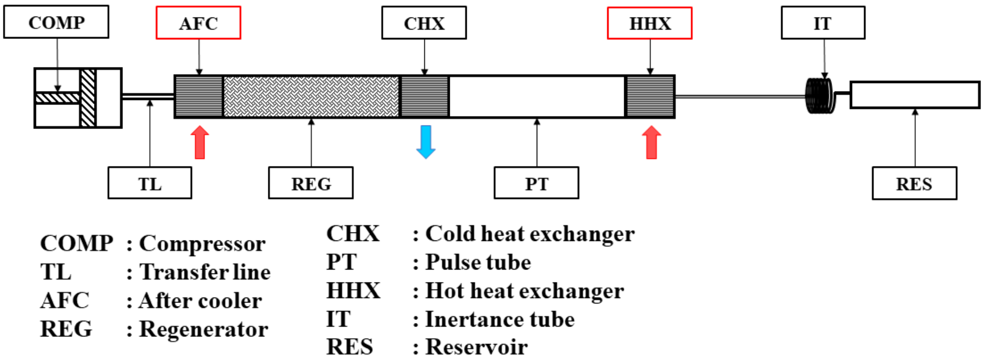


Fig. 1. Schematics of a typical single-stage inline inertance pulse tube cryocooler

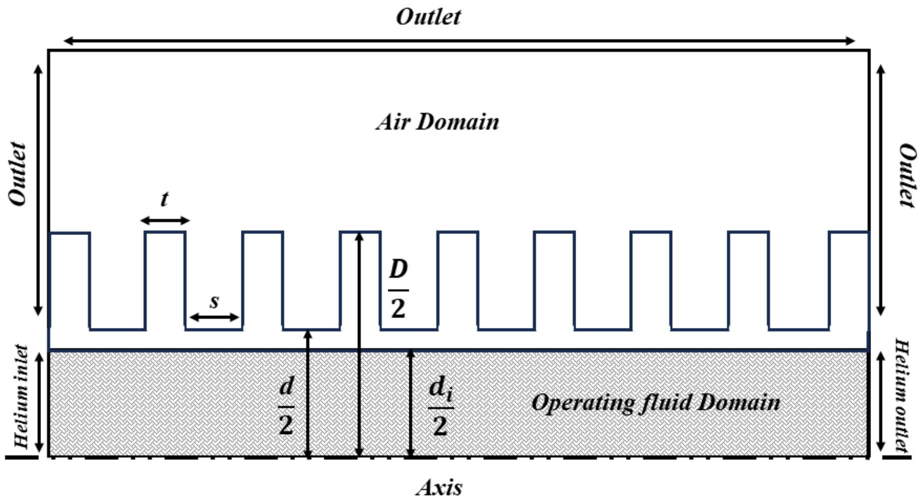


Fig. 2. Schematics of the heat exchanger computational domain (not to scale)

3. Numerical Simulation

The gas flow external to the heat exchanger canister and fins is characterized as incompressible and turbulent flow. However, the gas that flows within the heat exchanger is compressible, oscillating and laminar flow. The external flow considers the heat exchange by natural convection, whereas, internal flow considers the heat exchange by forced convection. The fundamental governing conservation equations for the fluid flow are written below. Governing conservation equations for the external fluid domain are written in Eq. (1) to Eq. (4) [20].

Continuity equation for external fluid domain:

$$\frac{\partial \rho_f}{\partial t} + \nabla.\left(\rho_f \vec{V}\right) = 0 \qquad (1)$$

Momentum equations for external fluid domain:

$$\rho_f \left[\frac{\partial \vec{V}}{\partial t} + \left(\vec{V}.\nabla\right)\vec{V}\right] = -\nabla P + \nabla.\bar{\bar{\tau}} + \rho_f \vec{f} \qquad (2)$$

The viscous stress tensor relation with the velocity gradient in the above equation is described as follows:

$$\tau_{ij} = \mu\left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i}\right) + \lambda\left(\nabla.\vec{V}\right)\delta_{ij} \tag{3}$$

Energy equation for external fluid domain:

$$\rho_f\left[\frac{\partial h}{\partial t} + \nabla.\left(h\vec{V}\right)\right] = -\frac{DP}{Dt} + \nabla.\left((k)_f \nabla T\right) + \phi \tag{4}$$

The Boussinesq approximation has been applied in the momentum equation of external fluid domain to account the changes in density and temperature, which is vital for natural convection heat transfer. Governing equations of interior fluid domain are written in following Eq. (11).

Continuity equation for internal fluid domain:

$$\frac{\partial}{\partial t}\left(\varepsilon\rho_f\right) + \frac{1}{r}\frac{\partial}{\partial r}\left(r\varepsilon\rho_f V_r\right) + \frac{\partial}{\partial z}\left(\varepsilon\rho_f V_z\right) = 0 \tag{5}$$

Momentum equations for internal fluid domain:

$$\frac{\partial}{\partial t}\left(\varepsilon\rho_f\vec{V}\right) + \nabla.\left(\varepsilon\rho_f\vec{V}\vec{V}\right) = -\varepsilon\nabla P + \nabla.\left(\varepsilon\bar{\bar{\tau}}\right) - \left(\mu\bar{\bar{\vartheta}}.\vec{j} + \frac{1}{2}\overline{\overline{C_2}}\rho_f\left|\vec{j}\right|\vec{j}\right) \tag{6}$$

where, inertial resistance (C_2) and viscous resistance (ϑ) are written by Eq. (7) and Eq. (8) respectively,

$$C_2 = \frac{3.5}{D_p}\frac{(1-\varepsilon)}{\varepsilon^3} \tag{7}$$

$$\vartheta = \frac{1}{\zeta} = \frac{D_h^2}{150}\frac{\varepsilon^3}{(1-\varepsilon)^2} \tag{8}$$

The terms inertial resistance and viscous resistance considers the additional pressure loss because of porous media.

Energy equation for the internal fluid domain:

$$\begin{aligned}&\frac{\partial}{\partial t}\left(\varepsilon\rho_f E_f + (1-\varepsilon)\rho_s E_s + \nabla.\left(\vec{V}\left(\rho_f E_f + P\right)\right)\right)\\ &= \nabla.\left(\left(\varepsilon(k)_f + (1-\varepsilon)(k)_s\right)\nabla \mathrm{T} + \left(\varepsilon\bar{\bar{\tau}}.\vec{V}\right)\right)\end{aligned} \tag{9}$$

According to Newton’s law for cooling, we can calculate the total convective heat transfer rate (Q) from the heat exchanger as follows,

$$Q = hA_{eff}(T_w - T_\infty) \tag{10}$$

The convective heat transfer coefficient (h) accounts total effective heat transfer surface area (A_{eff}), which includes the surface area of the heat exchanger ($A_{w/o\ fin}$) and the area of the fin (A_{fin}),

$$A_{eff} = N_{fin}A_{fin} + A_{w/o\ fin} \tag{11}$$

$$A_{w/o\,fin} = 2\pi dL \tag{12}$$

$$A_{fin} = t * (D - d) \tag{13}$$

Therefore, we can calculate the convective heat transfer coefficient from the previously calculated value of convective heat transfer.

$$h = \frac{Q}{A_{eff}(T_w - T_\infty)} \tag{14}$$

From this value of h, the surface Nusselt number over the fin surface can be calculated by using the following relation:

$$Nu = \frac{hD}{k} \tag{15}$$

Boundary conditions

For the porous region of the heat exchanger, a pressure inlet boundary condition is assigned by using the user defined functions as follows,

$$P_{in} = P_0 + P_a \sin(2\pi f t) \tag{16}$$

$$T_{in} = T_0 + T_a \sin(2\pi f t) \tag{17}$$

At the outlet of inner flow domain, pressure outlet boundary condition is applied. The conjugate heat transfer formulation has been enabled via thermal coupling for a set of two adjacent zones. The first one is between the inner fluid and walls of the heat exchanger, and the second conjugate condition is between the walls of heat exchanger and surrounding fluid flowing over it (air). The rest of the three sides of the surrounding (air) domain are set to the pressure outlet boundary condition as shown in Figure 2. The commercially available CFD software Ansys Fluent® has been utilized for solving the mass, momentum, and energy equations with the application of the aforementioned boundary conditions [20]. The analysis is conducted on a pressure-based solver with absolute velocity formulation. The SIMPLE algorithm has been employed for the pressure-velocity coupling, and a first-order implicit scheme is used for the transient formulation. The second-order upwind discretization schemes are used for the pressure, density, momentum, and energy formulations. The convergence criterion for the energy equation is 1E-6 and for other equations, its value is set to 1E-3 respectively.

4. Results And Discussions

The heat transfer processes for this special heat exchanger under oscillating flows involves the transfer of thermal energy between two fluids (inner fluid: helium, and outer fluid: air) and a solid surface (heat exchanger canister). Figure 3 depicts the temperature contour of the heat exchanger at four intervals (0^0, 90^0, 180^0, and 270^0) of the pulsating cycle. During the pressurization cycle, the gas is compressed within the pulse tube giving rise to an increase in temperature at its warm end. The excess heat is liberated to the ambient atmosphere at the hot heat exchanger as seen in Figure 3. Due to this, a change in air temperature is observed in between the fins of the heat exchanger. On the other hand, during the depressurization time, the temperature at the hot end of the pulse tube drops. This leads to a decrease in temperature inside the hot heat exchanger which can also be observed in Figure 3. Therefore, the temperature value of primary fluid that flows inside the heat exchanger changes in a cycle with phase angle and the heat wave propagates inside its computational domain at different magnitudes in different times.

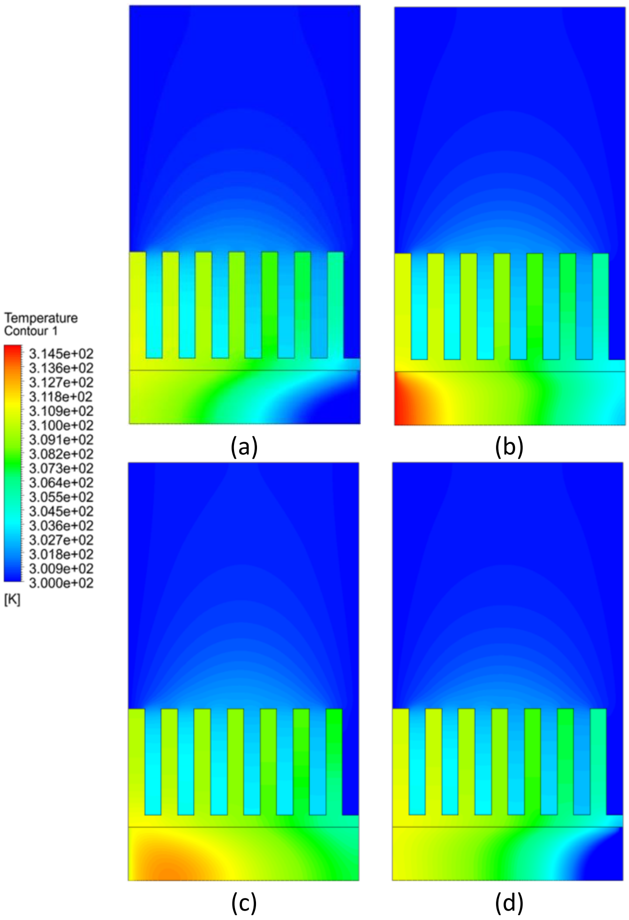


Fig. 3. Temperature distribution in the heat exchanger for different phase angles with fin

To better illustrate these heat flow phenomena, axial temperature distribution is plotted along the axial line of the heat exchanger in Figure 4 for different times. At 180°, the temperature is maximum at all points along the axial line and its value is minimum at all points along the axial line at 0° of phase angle. This rise in temperature at 180° phase is due to the propagation of heat waves generated at the warm end of the pulse tube due to gas compression in the cryocooler. Figure 5(a) to Figure 5(d) illustrate the radial temperature distribution of heat exchanger for primary fluid, solid surface, and secondary fluid domain at Fins 1, 3, 5, and 7 for 0°, 90°, 180° and 270° phase angles respectively. As Fin 1 is located closer to the warm side of the pulse tube, its temperature value is higher and the temperature value drops continuously along the axial location due to the continuous cooling from the air domain. Thus, temperature values in Fin 3, Fin 5, and Fin 7 are lower than that of Fin 1. It is pointless to say that Fin 7 is located near the exit of the heat exchanger. At 180□ angle, radial temperatures at all fins are higher than the remaining period due to the end of the pressurization phase.

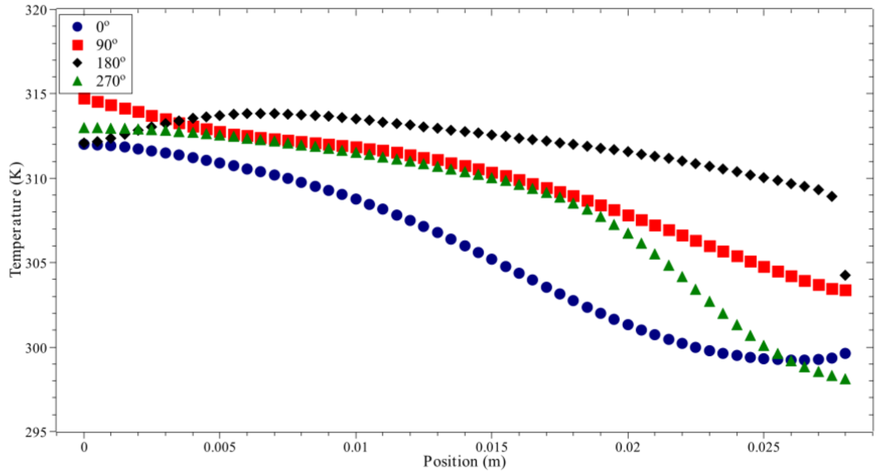
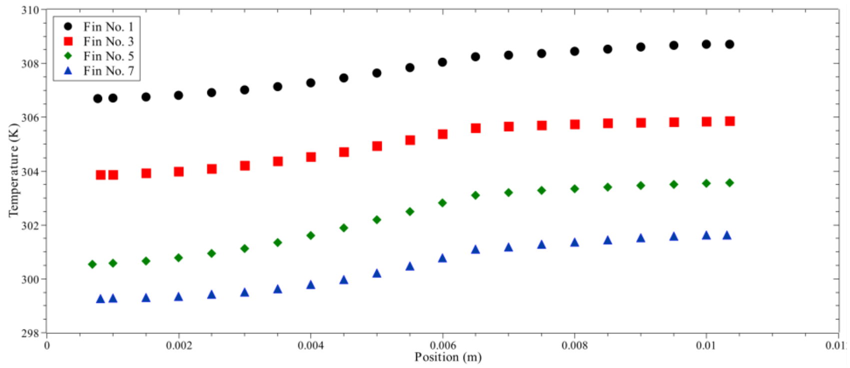
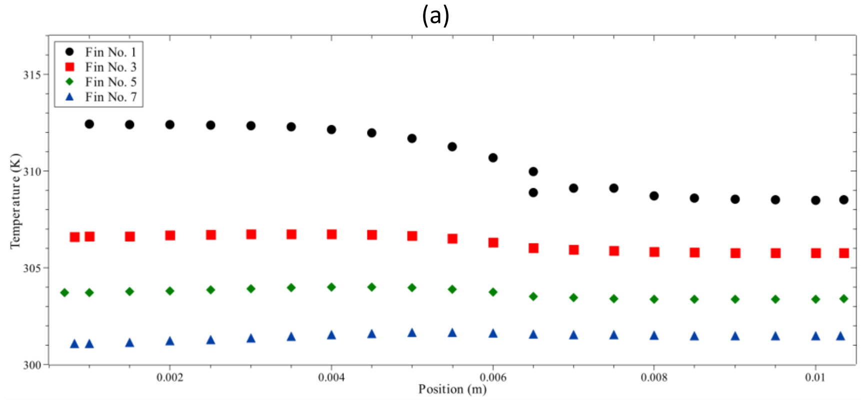


Fig. 4. Axial temperature variation of inner fluid for different phase angles



(a)



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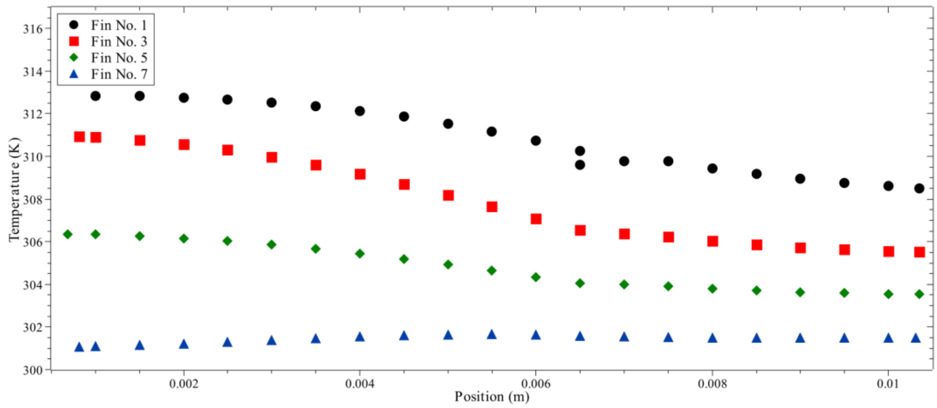
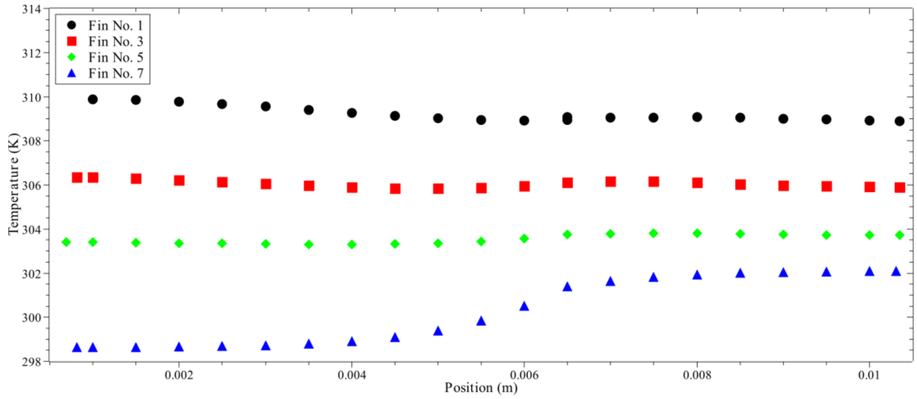


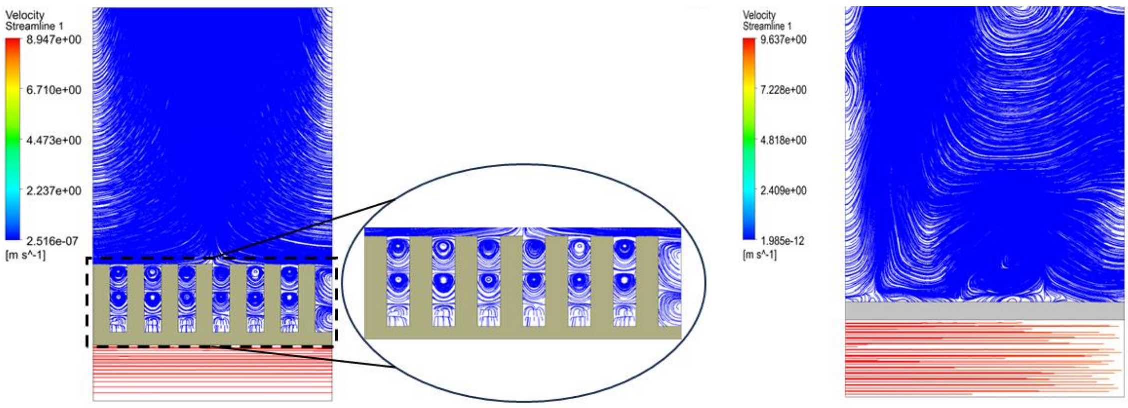
(c)



(d)

Fig. 5. Radial temperature variation at (a) 0^0 phase angle, (b) 90^0 phase angle, (c) 180^0 phase angle, (d) 270^0 phase angle

Figure 6 illustrates the velocity streamlines over the heat exchanger in which vortex formation and eddy circulations are observed to be dominant for heat transfer with annular fins. Vortices contribute to the convective heat transfer from the solid fin as the nonlinearity in air flow lifts heat from the surface resulting in better performance of the heat exchanger. The turbulence enhances heat transfer by promoting a more efficient interaction of momentum and energy between adjacent fluid layers. This leads to higher convective heat transfer coefficients as compared to laminar flow and yields better heat transfer rates



(a) (b)

Fig. 6. Streamlines in a typical heat exchanger for (a) with fin and (b) without fin

Figure 7 depicts the heat transfer coefficient of the solid fin at four different intervals of the cycle. It is a fundamental parameter used to quantify the rate of heat transfer between a solid surface and fluid as it represents the effectiveness of heat transfer across the interface between them. The heat transfer coefficient is observed to increase along the vertical length of the fin due to a larger surface area of the annular structure. Due to this, the temperature of the fluid inside the heat exchanger reduces. This results in a decrease in the heat transfer coefficient, which is observed to be the case along the horizontal axis of the fin.

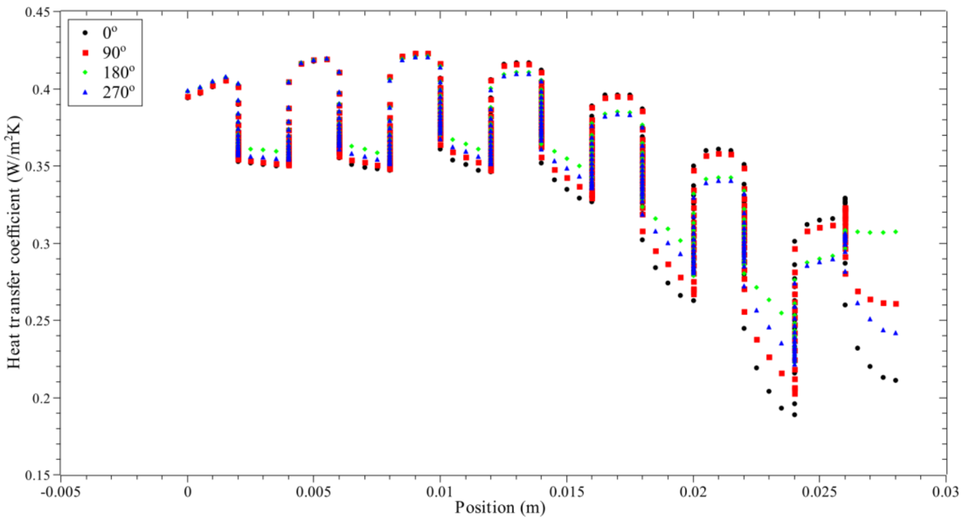


Fig. 7. Heat transfer coefficient at four different phase angles of heat exchanger for short fin

Figure 8 represents the Nusselt number along the surface of the heat exchanger. It is an important dimensionless parameter in heat transfer that relates the convective heat transfer to the conductive heat transfer rate and is used to characterize the heat transfer performance of a heat exchanger. The Nusselt number of the heat exchanger ranges from 17 to 8 which infer a substantial convective heat transfer. A similar trend in the heat transfer coefficient is observed along the horizontal axis in which the Nusselt number value reduces as the fluid inside the heat exchanger cools down. Figure 9 shows the pressure distribution at the inlet and outlet of the heat exchanger and the mass flow rate. A pressure drop is noticed in the figure, which is primarily due to the porous nature of the heat exchanger that offers additional resistance to the fluid flow.

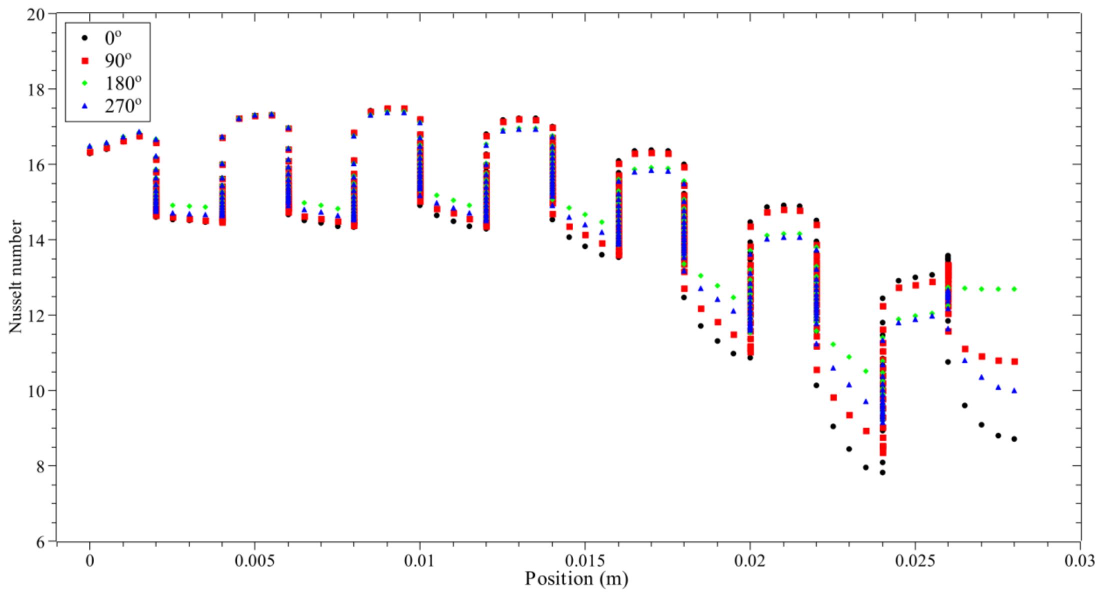


Fig. 8. Nusselt number at four different phase angles of heat exchanger for short fin

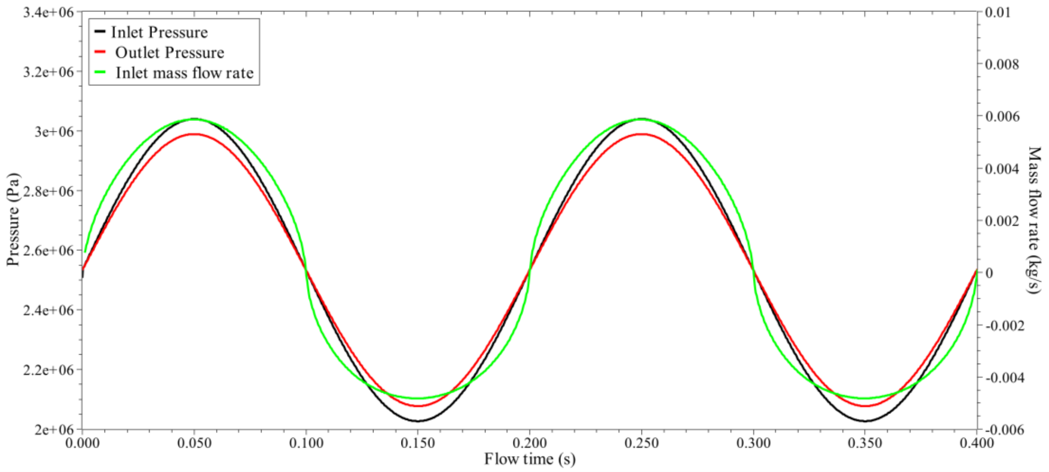


Fig. 9. Pressure and flow rate variation in the heat exchanger

Due to the lower value of porosity inside the heat exchanger, minute quantity of mass storage happens within its dead space. Thus, the magnitude of the mass flow rate at the inlet and exit of the heat exchanger remains very small and flow rate at the inlet of heat exchanger.

5. Conclusion

In this paper, numerical analysis is conducted to investigate the heat transfer processes of a typical heat exchanger used in a cryocooler. The numerical simulation is conducted by using the commercial code Ansys Fluent. The influence of various geometrical and operating parameters on the heat transfer rates has been analyzed. It is noticed that the temperature values within the heat exchanger are different durations in one cycle because of gas pressurization and depressurization. During the pressurization cycle, the gas compression inside the pulse tube leads to an increase in temperature at its warm end, causing excess heat to be liberated to the ambient atmosphere through the hot heat exchanger. This result in a change in air temperature situated between the fins of the heat exchanger. Conversely, during the depressurization cycle, the temperature at the hot end of the pulse tube decreases, causing a corresponding decrease in temperature inside the hot heat exchanger. The temperature distribution within the heat exchanger exhibits cyclic variation with a phase angle, with the maximum temperature occurring at 180 degrees. This temperature rises at 180 degrees is attributed to the propagation of the heat wave generated at the warm end of the pulse tube throughout the heat exchanger. The radial temperature distribution along the fins of the heat exchanger shows that temperature values are higher in fins closer to the warm side of the pulse tube in the cryocooler and gradually decrease along the axial location due to cooling from the air domain. Furthermore, the presence of vortex formations and eddy circulations observed in the velocity streamlines indicates the dominance of turbulent flow, enhancing convective heat transfer from the solid fins and contributing to better heat exchanger performance.

These findings highlight the significance of heat transfer phenomena in heat exchangers used in cryocoolers under oscillating flows. The results obtained in the investigation provide valuable insight into temperature variations, convective heat transfer mechanisms, and the role of vortex formations in improving heat transfer rates. The outcomes will be helpful in the design and optimization of heat exchangers for applications involving oscillating flows, leading to enhanced heat transfer performance and improved overall system efficiency.

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