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INVESTIGATION OF THE THERMAL PERFORMANCE OF CRYOGENIC REGENERATOR AS A POROUS STRUCTURE

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ABSTRACT

An efficient cryocooler is one of the essential requirements for cooling of Infrared (IR) sensors to low temperatures in high resolution night vision systems. A regenerator is an imperative component of cryocooler which has a significant effect on cooling performance of a cryocooler. In this research work a Computational Fluid Dynamic (CFD) methodology based on thermal equilibrium modelling approach has been implemented to analyze the thermal performance of a regenerator under different design conditions. The regenerator was modeled as a porous media with time varying boundary conditions in the commercial software package FLUENT by incorporating the effects of temperature dependent physical properties of both, matrix material and working fluid. Simulations were conducted at different cyclic flow velocities and effect of these variations on temperature swing, pressure drop, in-efficiency, capacity ratio and number of heat transfer units of the regenerator have been studied for design of better regenerators. The results show that any increment in the velocity of flow, increase the temperature swing, pressure drop and in-efficiency while decrease the capacity ratio and number of heat transfer units of the regenerator with fixed geometry. In this research work, it is also concluded that the thermal performance of regenerator strongly depend upon the temperature dependent physical properties of both matrix material and working fluid.

INTRODUCTION

Cryocooler is an integral part of thermal imaging systems used in military operations, which cools down the Infrared (IR) sensors to maintain high accuracy of these sensors. Essentially, the cryocooler performance depends on the efficiency of a regenerator. A regenerator consists of an array of porous matrix material which is exposed to oscillating flow of a working fluid. Hot and cold fluids pass through the regenerator and exchange heat with matrix material periodically. An efficient regenerator for cryocooler is a challenge to design, and has been a focus of special consideration for the last decade or so due to complex flow behavior inside the porous matrix material. Recent studies on regenerator design approaches have shown Computational Fluid Dynamics (CFD) methodology to be the best amongst the available options, since it allows performance analysis under periodic flow conditions at the least cost.

Many cryogenic engineers and scientists around the globe have modelled and analyzed regenerator as a porous structure [1-3]. Suzuki and Muralidhar [4] using a local thermal non-equilibrium model investigated pulsating flow inside a porous media and indicated that at low Reynolds number, effectiveness/efficiency of regenerator increases with an increase of frequency but decreases with increase of frequency at high Reynolds number. Tao et al. [5] analyzed different regenerators for pulse tube cryocooler and concluded that cooling performance of cryocooler enhances by increasing the specific heat and density of regenerator matrix material but decreases with an increase of thermal conductivity. Conrad et al. [6] numerically modeled the regenerator as a two dimensional porous media to measure the hydrodynamic

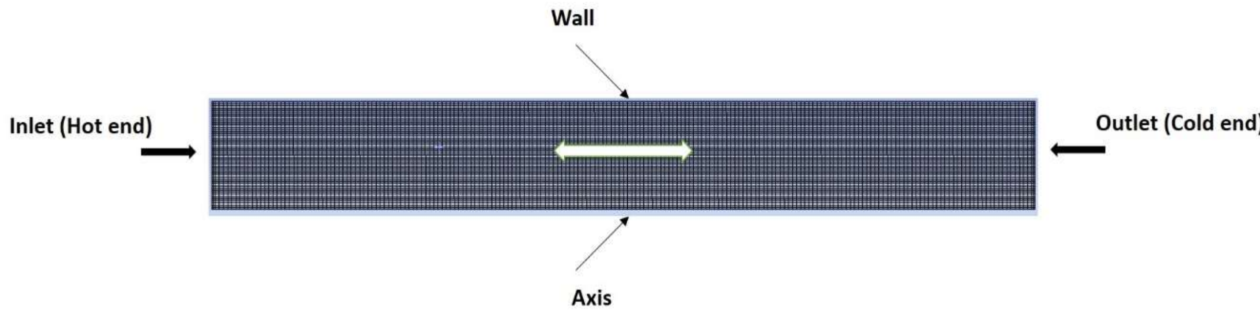

Figure 1. Regenerator mesh geometry

parameters like viscous and inertial resistance for different mesh geometries in axial and radial directions for both steady and oscillatory flow. They obtained the Darcy's permeability and Forchheimer's inertial coefficients experimentally and compared it with CFD simulation results and concluded that the Darcy's permeability and Forchheimer's inertial coefficients were usually unaffected to pressure and frequency. Landrum et al. [7] experimentally investigated the effects of average fluid pressure on hydrodynamic parameters across different regenerator fillers under steady state conditions. In their study, hydrodynamic parameters of the porous media were iteratively adjusted in FLUENT simulation to match with experimental results. Axial pressure drop at different supply pressures was also measured. A direct correlation between axial pressure drops and mass flow rates was reported. Nair and Krishnakumar [8] numerically investigated heat transfer in a wire meshes regenerator by using 2-D local thermal equilibrium porous media model. Pathak et al. [10] performed experimental and CFD based study for porous model under steady and oscillatory flow conditions to determine the Darcy's permeability and Forchheimer's coefficients. These coefficients represent the viscous losses and inertial losses in the flow respectively. The work discussed above was directed towards understanding of the hydrodynamic properties of the porous media used in regenerators. Very little attention was given to the heat transfer issues which define the performance of a regenerator. Recently, analysis of the regenerator performance for heat transfer has also attracted the interest of researchers. Costa et al. [12] used the thermal non-equilibrium porous media model to analyze the performance of a Stirling engine regenerator, and concluded that the adopted model can be used with a high confidence level for design of Stirling engine regenerators.

It is clear from the above that CFD methodology is the most popular technique towards analysis of regenerators. However, few studies have been directed towards heat transfer aspects of regenerators. The studies that have been conducted have assumed the physical properties to remain constant over the working cycle of a regenerator. However, properties like viscosity, density, specific heat and thermal conductivity of both the matrix material and working fluid vary with a change in temperature. Therefore, there is a need to incorporate variable properties in the simulations to determine the effect/ significance of these changing properties on the performance of a regenerator.

The work presented in this paper has used a CFD assisted approach to analyze the performance of a regenerator for fixed geometry, operating and hydrodynamics parameters for different mesh porosity and mass flow rates of the working fluid. Variable properties for both the matrix material and working fluid were incorporated in the working model. Details of the numerical methodology is discussed next.

MATERIAL AND METHOD

The regenerator was modeled as a porous media using the commercially available software ANSYS Fluent, which solved the volume averaged equations of mass, momentum and energy detailed below.

Conservation of mass

$$\frac{\partial}{\partial t}(\varepsilon\rho) + \nabla.(\varepsilon\rho\vec{v}) = 0 \tag{1}$$

Momentum equation

Porous media in Fluent is modelled by merging Darcy’s and Forchheimer’s term in the force expression of the momentum equation.

$$F = -\left(\frac{\mu}{\alpha}\vec{v} + \frac{1}{2}\ C\,\rho|\vec{v}|\vec{v}\right) \tag{2}$$

The first term on the right hand side of the expression is the Darcy’s term, which represents the pressure drop while the second expression models Forchheimer’sterm. The inertial resistance factor C and the permeability α for porous media required for solving this equation are provided by the user depending on the mesh geometry. Values of viscous and inertial resistance are 2.35 x 1010 1/m2 and 47000 1/m respectively for 325 SS mesh size considered in the present work [7].The volume average momentum equation is:

$$\frac{\partial}{\partial t}(\varepsilon\rho\vec{v}) + \nabla.(\varepsilon\rho\vec{v}\vec{v}) = -\varepsilon\nabla p + \nabla.(\varepsilon\tau) + -\left(\frac{\mu}{\alpha}\vec{v} + \frac{1}{2}\ C\,\rho|\vec{v}|\vec{v}\right) \tag{3}$$

Conservation of Energy

The porous media volume averaged energy equation accounts for the fluid and porous medium interactions. In this research work local thermal equilibrium has been assumed between the porous media and the working Fluid [13]. FLUENT process a single energy equation by considering local thermal equilibrium between solid and fluid. This assumption become true for the conditions in which the variation in temperature between the two phases is not large and no internal heat generated source available. This means that the fluid and matrix were assumed to be at the same temperature throughout the regenerator. Thus a single energy equation was solved by FLUENT.

$$\frac{\partial}{\partial t}(\varepsilon\rho_f\ E_f + (1-\varepsilon)\rho_s\ E_s) + \nabla.\left(\vec{v}(\rho_f\ E_f + p)\right) = \nabla.[K\ \nabla T + (\tau\,.\vec{v})] \quad (4)$$

where $K = \varepsilon\, K_f + (1-\varepsilon)\, K_s$

A 2-D axi-symmetric geometry of the regenerator was meshed in ICEMCFD (Figure 1). The simulations were conducted for an incompressible viscous working fluid using pressure based transient solver with absolute velocity formulations and considering laminar flow. Helium was used as the working fluid while porous matrix material was taken to be stainless steel.

To incorporate the temperature dependent properties of the matrix material (thermal conductivity and specific heat) and working fluid (viscosity, thermal conductivity and specific heat) in the simulations, National Institute of Standards and Technology (NIST) [15-16] data was used. This was then fed into Fluent using the best fit Polynomial curve. As the flow of working fluid in the regenerator is cyclic, the simulations must be conducted under varying boundary conditions. The heating (flow of fluid from hot end to cold end) and cooling operations (flow of fluid from cold end to hot end) are periodic in nature, and form one complete cycle. Both modes are operational for half the time period of the complete cycle, since the end of heating period marks the start of the cooling period and vice versa with an instantaneous switch from heating to cooling. To realize this, velocity inlet boundary condition was applied at both ends of the regenerator using the Sine function. An appropriate DEFINE_PROFILE user defined function (UDF) was selected from the FLUENT user manual [13], compiled and assigned at both ends of the regenerator. A Stirling cryocooler generally operates at a frequency of 45Hz, which was used to calculate the time period of the sine function employed. Temperature of the working fluid at inlet of the regenerator during heating period was set at 300K, while inlet temperature for the cooling cycle was kept at 80K. Second order upwind discretized governing equations were solved by employing the Semi Implicit Method for Pressure Linked Equations (SIMPLE) algorithm with PRESTO discretization scheme for pressure interpolation. Convergence criteria for the field residuals was kept at 10^{-6}. Table 1 summarizes the regenerator parameters used.

Table 1. Geometric parameters of Regenerator

Symbol	Definition	Value
Th	Hot End Temperature	300 K
Tc	Cold End Temperature	80 K
L	Length of regenerator	0.050 m
D	Diameter of regenerator	0.010 m
Ac	Cross section area of regenerator	0.000079 m2
Aff	Free flow area of regenerator	0.000055 m2
Am	Matrix material area	0.000024 m2
Ac	Convictive heat transfer area	0.134694 m2
dw	Diameter of mesh screen wire	0.000035 m
α	Porosity of matrix material	0.7
Dh	Hydraulic diameter of regenerator	0.000082 m
λ	Heating/Cooling flow period	0.011 sec
f	Operating Frequency	45 Hertz
SS	Stainless steel matrix material	
He	Helium working Fluid	

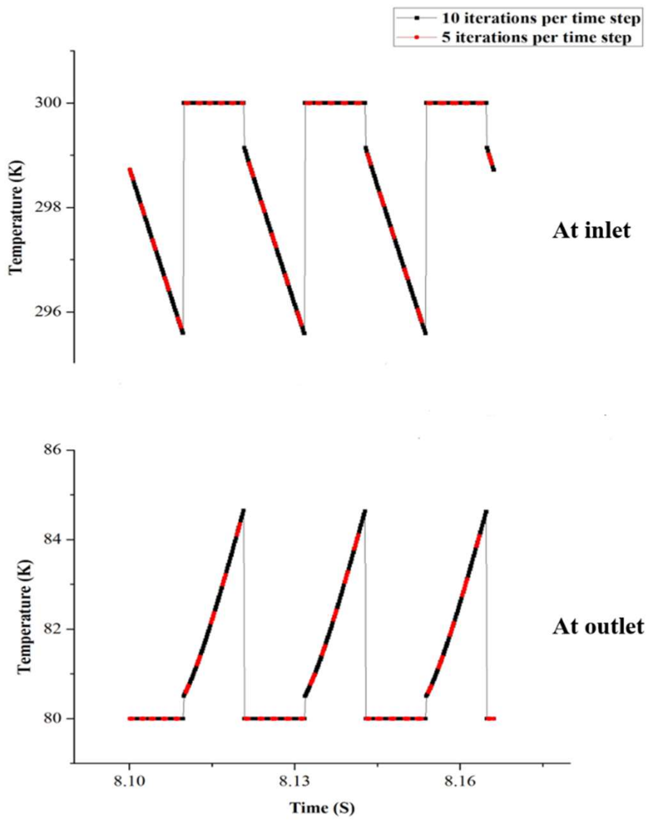

Figure 2. Temperature variation with time at inlet and out of regenerator for 5 and 10 iterations per time step

Since the simulations were conducted under varying boundary conditions, 5 inner iterations per time step were performed for all the results presented with a time step of 0.1ms. This number was chosen to be the minimum required to attain 'convergence' of the solution and was identified by comparing the simulation results for different inner iterations. A comparison of temperature profiles for simulations conducted using 5 and 10 iterations shown in Figure 2 shows that the temperature at inlet and outlet of the regenerator for both 5 and 10 iterations per time step remain same at all times. This confirms that increasing the inner iteration beyond 5 would only result in added computational overhead. Velocity of the working fluid in the regenerator was varied in the range of 20m/s to 50m/s. For fixed regenerator geometry considered in the present work, this equates to a mass flow rate variation from 0.266g/s to 0.631g/s. The results obtained from simulation are discussed next.

RESULTS AND DISCUSSION

Before discussing the results, the nomenclature of terms used in this section are defined for clarity. The heat transfer performance of the regenerator is most appropriately expressed by efficiency of regenerator given by [3]

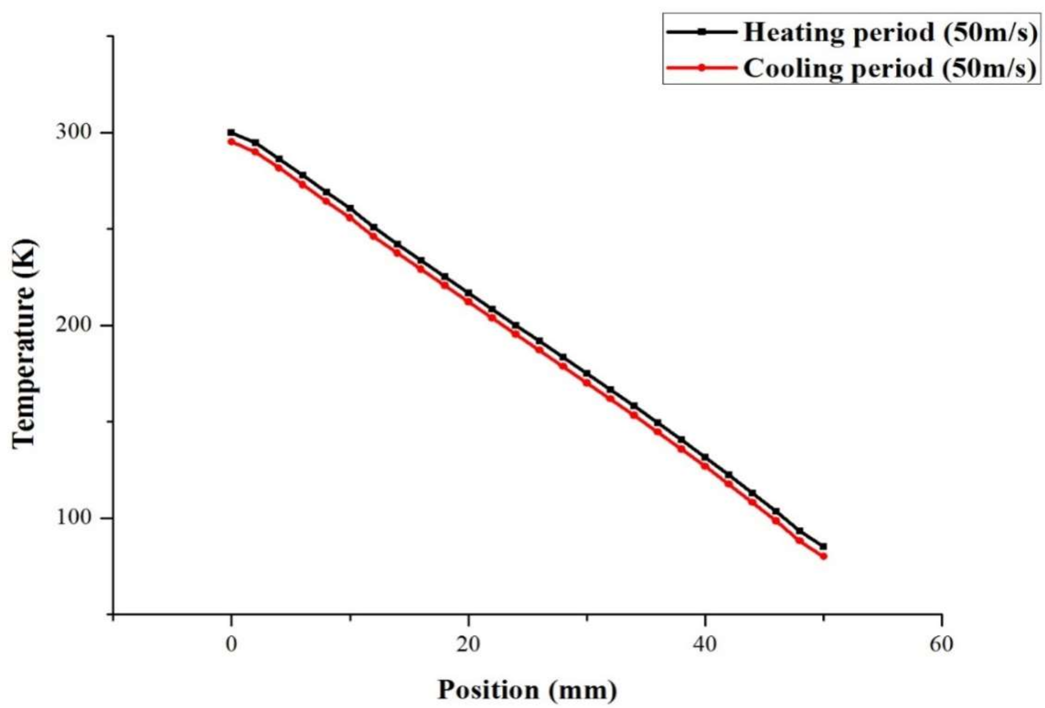

Figure 3. Temperature distribution along regenerator for heating and cooling period

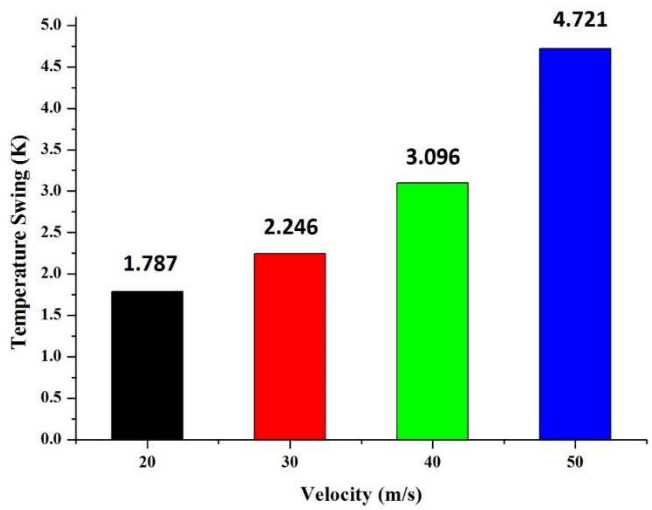

Figure 4. Temperature swing variation with velocity

$$\epsilon = \frac{(T_{in} - T_{out})_h}{(T_{h,in} - T_{c,in})} = \frac{(T_{in} - T_{out})_c}{(T_{h,in} - T_{c,in})} \tag{5}$$

However, in the literature of cryogenics, the regenerator performance is more often described in terms of inefficiency, and the same convention will be used here.

Inefficiency = 1 − efficiency

Another non dimensional parameter of importance is the Number of Heat transfer units (NTU), which expresses the non-dimensional size of the regenerator given by [3].

$$NTU = \frac{A_m h}{2(\dot{m}c_p)_f} \tag{6}$$

Matrix capacity ratio (CR) measures the thermal capacity of the matrix material relative to the minimum heat capacity of working fluid. CR is the ratio of thermal capacity of the matrix to the minimum flow stream thermal capacity [3].

$$CR = \frac{C_m}{C_{min}} = \frac{(Mc_p)_m}{\lambda(\dot{m}c_p)_{min}} = \frac{(Mc_p)_m}{\lambda(\dot{m}c_p)_f} \tag{7}$$

Figure 3 shows the temperature profile along the regenerator length for heating and cooling period at a fluid velocity of 50m/s. Recall that the simulations were set such that the hot end temperature was fixed during the heating period while the cold end temperature was fixed for the cooling period. It is evident from the Figure that during heating period, the temperature of fluid does not reach the desired temperature at cold end (fixed for the cooling period). Similarly, during reverse flow, the temperature of fluid does not reach the fixed temperature at hot end during cooling period. Actually, there is a difference in temperature during the heating and cooling period at all axial locations. This difference in temperature is referred to as the temperature swing of matrix material, and is undesirable.

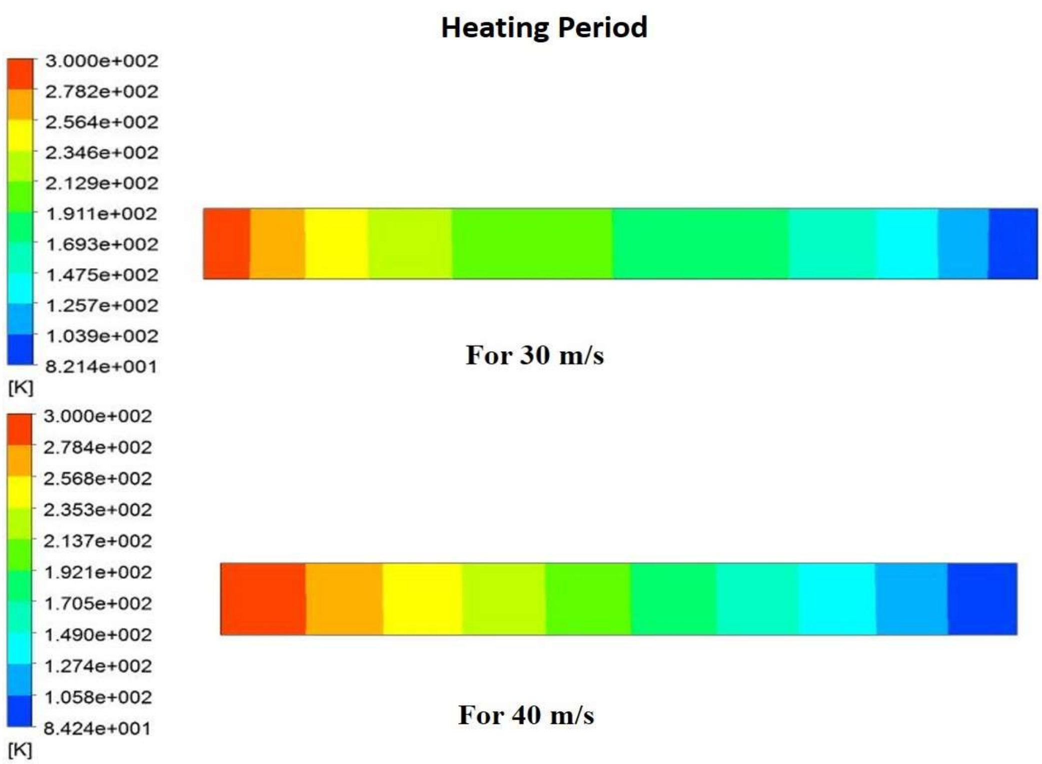

Figure 5. Temperature contours for heating period

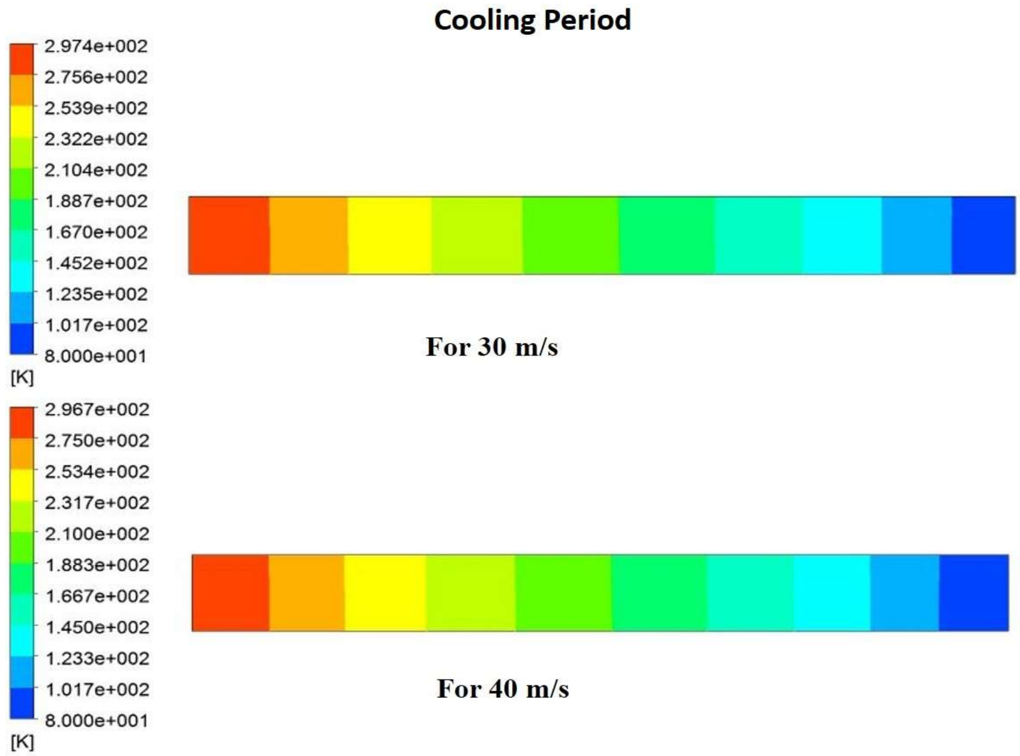

Figure 6. Temperature contours for cooling period

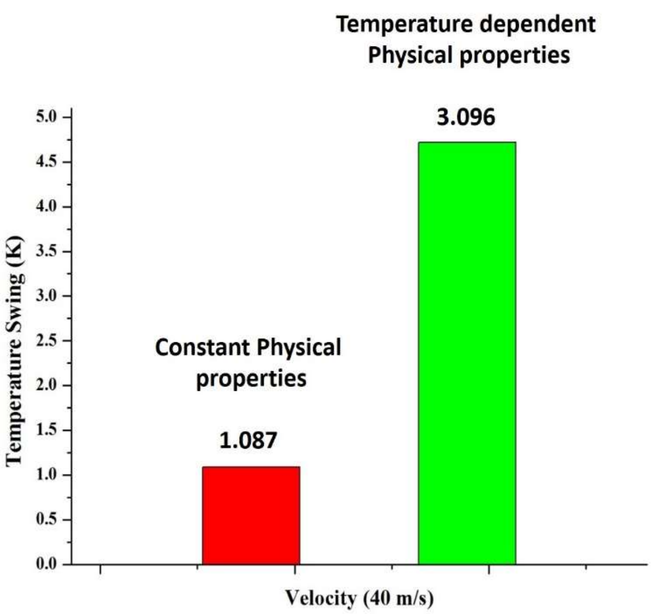

Figure 7. Comparison of Temperature swing

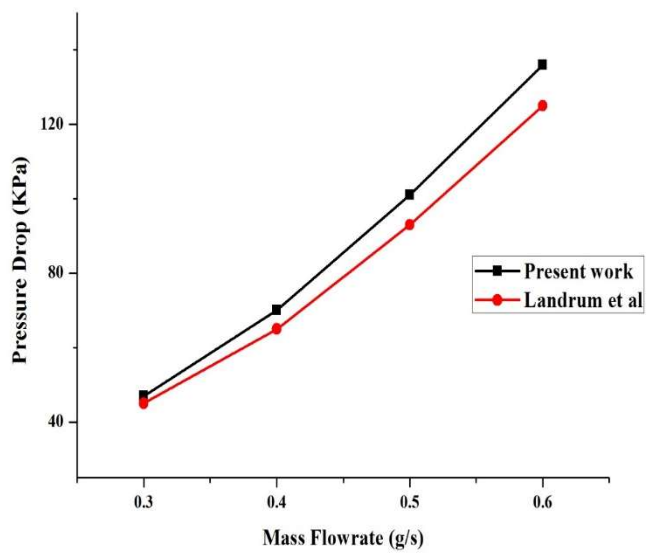

Figure 8. Pressure drop variation with mass flow rate

Table 2. Computed flow parameters from the simulation results

Parameters	Values				
	Temperature dependent properties				Constant properties
Velocity (m/s)	20	30	40	50	40
Mass Flowrate (g/s)	0.27	0.38	0.51	0.64	0.51
Reynold's number	138.9	202.81	273.95	352.34	273.95
CR	312.5	217.04	162.67	131.76	-
NTU	320	229.1	205.64	194.21	-
Pressure drop (KPa)	55.4	88.44	144.7	154.9	-
Temperature at inlet (K)	298.6	297.4	296.7	295.5	298.9
Temperature change in cooling period (K)	218.6	217.4	216.7	215.5	218.9
Temperature at outlet (K)	81.3	82.1	84.2	85.5	80.8
Temperature change in heating period (K)	217.8	217.9	215.8	214.5	219.2
In-efficiency (%) cooling period	0.636	1.182	1.500	2.045	0.50
In-efficiency (%) heating period	0.591	0.955	1.909	2.500	0.36
Difference in In-efficiency	0.045	0.227	-0.409	-0.455	0.14
Average In-efficiency (%)	0.614	1.068	1.705	2.273	0.43

The effect of fluid velocity on the temperature swing is shown in Figure 4, which compares the temperature swing at mid-length of the regenerator. The temperature swing increases from 1.787 K to 4.721 K when the velocity is increased from 20 m/s to 50 m/s, which shows that the matrix material was not able to respond to higher mass flow rates as efficiently at lower CRs.

A further elaboration of this effect has been presented in Figures 5 and 6 which compare the temperature variations along the regenerator at velocities of 30m/s and 40m/s during heating and cooling periods. A closer look at the temperature contours of heating and cooling period for the same velocity will show that the temperature change of the working fluid is different during heating and cooling periods. For instance, at 30 m/s, the fluid undergoes a temperature change of 217.9K (from 300K to 82.1K) while during the cooling period, a change in temperature of 217.4K was observed (from 80K to 297.4K. this shows that the heating cycle was more efficient than the cooling cycle. However, at 40m/s the heating cycle was less efficient than the cooling cycle since the temperature changes were noted to be 215.8K and 216.7K in heating and cooling period respectively. The cooling cycle continued to be more efficient than the heating cycle at higher velocities but the average inefficiency of regenerator is still increasing with higher velocities. A summary of the temperatures noted at all velocities tested is presented in Table 2.

To understand the effect of varying physical properties on the simulation results, a comparison of the temperature swing for fixed and variable properties at a velocity of 40m/s is provided in Figure 7. It is evident that the temperature swing values change considerably when the more realistic variable properties model is used. Hence, if CFD methodology is to be used for designing of regenerators, the simulations must consider variations of properties with temperature**.**

Figure 8 shows the variation of pressure drop across the regenerator matrix. The pressure drop across the regenerator increases with increase of mass flowrate (velocity) of fluid. A comparison with Landrum et al. [6] shows that the CFD methodology adopted was able to predict the pressure drops in the porous structure with a high level of precision. As mass flowrate varied from 0.3g/s to 0.6g/s, pressure drop across the regenerator increased from 47KPa to 136 KPa.

The results suggest that for fixed regenerator geometry, increasing the velocity of the working fluid increases the inefficiency of the regenerator. This is due to a decrease in CR and NTU with increasing mass flow rates. However, Figure 9 shows that increase in mass flow rate decreases the CR more significantly compared to the NTU at higher mass flow rates. Thus the inefficiency can be reduced by adjusting the CRs if high mass flow rates are desired. This can be done by adjusting the porosity of the matrix material. Since the pressure drop increases across the regenerator with increase of velocity, decreasing the porosity will result in further pressure drops. The results of simulations for different porosities of matrix material are shown in Figure 10 where the difference between the pressures values at inlet and outlet is equivalent to the pressure drop for that regenerator. It is evident that when the porosity of the regenerator increases, the pressure drop decreases. The porosity of regenerator matrix material was changed from 0.5 to 0.9 causing a reduction in pressure drops from 130.8KPa to 93.9KPa with 40m/s flow velocity.

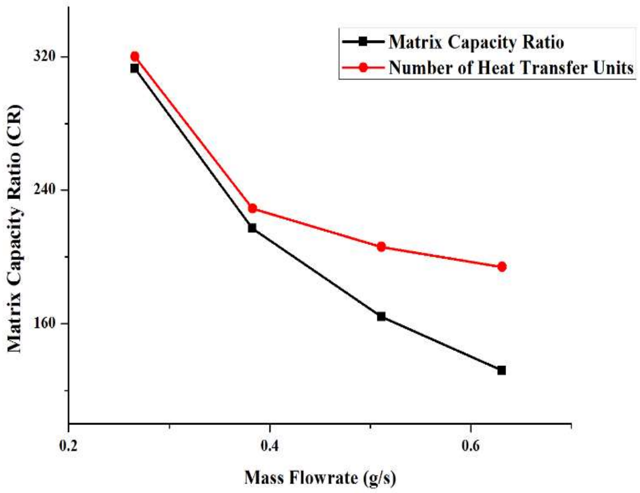

Figure 9. Variation of dimensionless number with mass flowrate

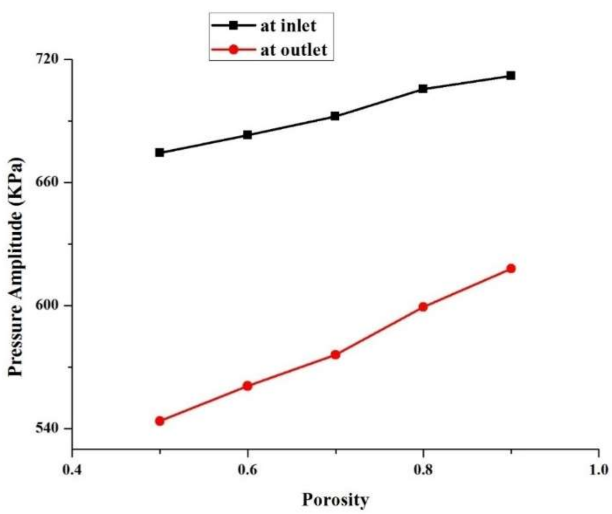

Figure 10. Pressure amplitude variation at inlet and outlet of regenerator with porosity

CONCLUSIONS

In this research study, a cryogenic regenerator as a porous structure is simulated in FLUENT under periodic flow conditions for constant and temperature dependent physical properties of matrix material and working fluid in order to analyze the thermal performance of regenerator. For the fixed regenerator geometry, the effects of velocity/mass flowrate were investigated on the performance of regenerator. The results showed that any increment in the velocity/mass flowrate of the working fluid lead to increase the temperature swing, pressure drop and in-efficiency of regenerator but decreases the CR more significantly compared to the NTU. Therefore the inefficiency of regenerator can be reduced by adjusting the CRs if high mass flow rates are preferred. When the effects of temperature dependent physical properties of both matrix material and working fluid are incorporated in the simulation, the temperature swing and inefficiency of regenerator significantly increased from the values of temperature swings and inefficiency of regenerator with constant physical properties, thus thermal performance of regenerator also strongly depend upon the temperature dependent physical properties of both matrix material and working fluid. The thermal performance of a fixed geometry regenerator can be augmented by adjusting the velocity/mass flowrate of working fluid and selection of matrix material. The computational results acquired from this investigation can be use with great confidence to design and optimize an efficient regenerator for cryocooler applications.

NOMENCLATURE

Ac	regenerator cross section area(m2)
Am	convictive heat transfer area(m2)
C	inertial resistance factor
Cm	heat capacity of matrix material (J/K)
Cmin	minimum heat capacity of working fluid (J/K)
Cpf	Specific Heat of Working fluid(J/Kg-K)
Cpm	specific heat of matrix material(J/Kg-K)
D	regenerator diameter (m)
Dh	regenerator hydraulic diameter (m)
dw	wire diameter (m)
E_f	total fluid energy (J/kg)
E_s	total solid energy (J/kg)
f	frequency (Hz)
h	heat transfer coefficient (W/m2 K)
K_f	thermal conductivity of working fluid(W/m K)
Ks	thermal conductivity of solid matrix (W/m2 K)
L	length of regenerator matrix (m)
M	mass of regenerator 9Kg)
$\dot{m}$	mass flowrate (Kg/s)
p	static pressure (Pa)
Pr	Prandtl number
Re	Reynolds number
rh	regenerator hydraulic radius (m)
T	temperaure of working fluid (K)
t	time (s)

Tin	temperure at inlet (K)
Tout	temperaure at outlet (K)
Th, in	temperaure at inlet during heating period (K)
Tc,in	temperature at outlet during cooling period (K)
V	inlet/outlet velocity magnitude (m/s)
α	permeability (m2)
μ	fluid dynamic viscosity (Pa s)
ρf	fluid density (kg/m3)
ρs	solid density (kg/m3)
ε	regenerator matrix porosity
ρ	density of working fluid (kg/m3)
∇	gradient operator
τ	stress tensor (Pa)
λ	Heating/cooling period (s)
ϵ	efficiency of regenerator

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