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RESEARCH NOTE

CFD Simulation of a Multi-mesh Pulse Tube Regenerator

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ABSTRACT

During the last two decades, inertance tube pulse tube cryocoolers (ITPTC) applications in astronautics instruments gained momentum due to their high reliability. Moreover, significant efforts were made in order to improve ITPTCs operation. Investigations showed that most losses occur in the regenerator part. Due to complexity of physics of these losses, effects of the regenerator efficiency on the cryocooler performance were investigated in this work. Tocalculate heat transfer between solid matrix and acting fluid in the regenerator, Dual Energy Equation (DEE) model was used. Calculation of entropy flow inside the regenerator showed that almost 85% of the energy losses are due to viscous and inertial losses besides most of the energy losses occuring in its hot end. Therefore, in order to optimize the system, multi-mesh regenerator was studied. Results showed that under fixed CHX wall temperature of 150 (K), COP of the PTC with optimum multi-mesh regenerator is 1.07 times higher than the value of uniform mesh regenerator ITPTC. A precise agreement was observed between simulation results and available experimental data published in the literature.

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NOMENCLATURES

A	Heat transfer area (m^2)	$\vec{V}$	Velocity vector (m/s)
C	Inertial resistance factor	$\dot{W}_{pv}$	P-V power of compressor (W)
COP	Coefficient of performance	X_{amp}	Ampitude of piston motion (m)
D_h	Hydraulic diameter (m)	X_t	Screen transvers pitch
E	Local energy (J/kg)	x	x axis
$f_{osc,m}$	Oscillating friction factor	*2-D*	two dimensional
$\vec{g}$	Gravity acceleration (N/kg)	***Greek Symbols***	
H	Local enthalpy (J/kg)	∇	Gradient operator
h_{sf}	Convective heat transfer coefficient in Porous media (W/m^2)	ρ	Density (kg/m^3)
k	Thermal conductivity (W/m.K)	τ	Stress tensor
$\dot{m}$	Mass flow rate (kg/s)	μ	Dynamic viscosity (Pa.s)
n	Number of packed screens per length	β	Darcy permeability (m^2)
Nu	Nusselt number	ν	Piston oscillating frequency (Hz)
p	Static pressure (Pa)	ε	Porosity
Pr	Prandtl number	φ	Opening area ratio of screen
Re	Reynolds number	***Subscripts***	
Re_m	Oscillating flow Reynolds number	c	Cold heat exchanger
R_{osc}	Regenerator performance factor (m/s.K)	f	Fluid
$S''_{gen,sys}$	Entropy generation at regenerator (W)	s	Solid matrix
T	Temperature (K)	sf	Solid-Fluid interface
t	time (s)		

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1. INTRODUCTION

Ability to reach very low temperatures with a desired cooling power and high reliability due to omitting the moving part in cold head makes the pulse tube cryocooler (PTC) very useful so that they are widely used in many engineering applications i.e. astronautics, medical and electronic instruments. However, efficiency of the PTC is low and many efforts have been paid to improve that, since its invention by Gifford and Longsworth in 1963 [1].

There are many sources of energy loss in PTCs, i.e. viscous and inertial losses and losses due to conduction in solid matrices, etc. However, after four decades of experiences with PTC, it is known that the most of energy losses occur in regenerator part of the system [2]. For the last 10 to 15 years, the regenerator has been the main subject of researches in the cryocooler community due to their complicated flow phenomena of fluid-solid interactions.

Few theoretical and mathematical models have beendeveloped to understand the mechanism of oscillating flow and heat transfer in porous media. Siegel [3] investigated laminar forced convectional oscillating flow inside a tube with a uniform heat flux in the tube wall and showed that flow oscillation causes more axial energy transport in the tube. Khodadadi [4] studied the oscillating flow inside the porous media between two parallel plates. In that study, the inertia term in momentum equation and axial conduction in energy equation are neglected but all the other terms are counted. Results of that study showed that flow oscillation causes a phase lag between pressure and velocity which its value decreases by increasing fluid viscosity.

Another important part of losse in the regenerator is pressure drop which affects the flow mechanism and heat transfer between fluid and solid matrices [2]. To achieve an exact heat transfer coefficient in the regenerator, it is necessary to calculate the pressure drop precisely. There have been many efforts to introduce a good relation for friction coefficient in oscillating flow inside the regenerator [5- 13]. Organ's studies [14] showed that there are many differences between oscillating and steady pressure drops in it. Recently, the oscillating friction factors are used in some codes such as Sage Software [11].

Experimental investigations are costly and time consuming. Furthermore, there are many performance details that cannot be easily measured; hence, numerical simulation approach provides comprehensive details which are very useful to understand the flow physics in the PTC; hence, some of the commercial softwares have been recently used to simulate the PTC [15]. However, few 2-D simulations have used DEE model to simulate the heat transfer between fluid and solid matrix in the regenerator [2, 16].

In this study a 2-D axisymmetric CFD model is used to simulate a single stage multi-mesh regenerator ITPTC. Fluent software is used to do the simulations. In order to model the piston motion, Fluent's dynamic meshing function is used. A User Defined Function (UDF) was developed in C programming language to simulate the piston cylinder effect. Both Local Thermal Equilibrium (LTE) and DEE models were used to simulate the heat transfer in the regenerator. Fluent uses LTE model for heat transfer between fluid and solid matrix in porous medium. Therefore, to apply DEE model to the regenerator a User Defined Source (UDS) was developed to add a source term to the energy equation of gas and to simulate the energy equation of solid matrix.

2.MATHEMATICAL MODELING, GOVERNING EQUATIONS & CFD SIMULATION CONDITIONS

According to correlations of Brereton and Mankbadi [17] which show the turbulent and laminar regimes border lines for oscillating internal flows, flow is laminar for all parts of the simulated PTC. Thus laminar flow formulation is used in the simulations. Besides, the thermal resistance of all the walls is neglected since these resistances are negligibly small except the wall thickness of the pulse tube section which is extremely important since the heat is transported by the thickness walls using the theory of the surface heat pumping [18]. Hence, the thickness of pulse tube wall is considered.

The continuum based conservation of mass, momentum and energy equations along with the equation of state of the working fluid are as follows [19]:

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

$$\frac{\partial \varepsilon\rho}{\partial t} + \nabla.(\varepsilon\rho\vec{V}\vec{V}) = -\varepsilon\nabla p + \nabla.(\varepsilon\tau) + \rho\vec{g} - (\frac{\mu}{\beta}\vec{V} + \frac{1}{2}\rho C\left|\vec{V}\right|\vec{V}) \tag{2}$$

$$\begin{aligned}&\frac{\partial}{\partial t}(\varepsilon\rho_{\,f}E_{f} + (1-\varepsilon)\rho_{s}E_{s}) + \nabla.(\vec{V}(\rho_{\,f}E_{f} + p))\\ &= \nabla.(\varepsilon k_{f}\nabla T + (\tau.\vec{V})) + S\end{aligned} \tag{3}$$

$$p = \rho RT \tag{4}$$

where, $S = \nabla.((1-\varepsilon)k_s\nabla T)$ and $S = h_{sf}A_{sf}(T_s - T_f)$ are belong to LTE and DEE models, respectively. For the non porous medium: $\varepsilon = 1, \beta = \infty$ and $C = 0$, whereas, for the porous mediums $0 \le \varepsilon \le 1$, $\beta \ne \infty$ and $C \ne 0$. The energy equation of solid matrix is as follow:

$$(1-\varepsilon)\frac{\partial}{\partial t}(\rho_{\,s}C_{s}T_{s}) = (1-\varepsilon)\nabla.(k_{s}\nabla T_{s}) + h_{sf}A_{sf}(T_{f} - T_{s}) \tag{5}$$

The UDS boundary condition in the regenerator is shown in Figure 1. Due to high gradient of temperature in the PTC the viscosity and thermal conductivity of the fluid are set as Sutherland formula and polynomial profile of temperature [20], respectively. There are some important sources of entropy generation in the regenerator i.e. interfacial convection, viscous and inertial, matrix and gas conduction etc. Entropy generation in the regenerator is calculated as follow [2]:

$$S''_{gen,sys} = \varepsilon \frac{k_f}{T_f^2}\left(\frac{\partial T_f}{\partial x}\right)^2 + (1-\varepsilon)\frac{k_s}{T_s^2}\left(\frac{\partial T_s}{\partial x}\right)^2 + \frac{a_{sf} h_{sf}}{T_s T_f}(T_s - T_f)^2 - \varepsilon \frac{V}{T_f}\frac{\partial p}{\partial x} \geq 0 \quad (6)$$

Nusselt number of the oscillating flow in the regenerator is calculated as follow [21]:

For this study 51852 cells were used for meshing. Hybrid mesh (Structured and Unstructured) was selected for girding the domain. The number of cells which is located near walls in oscillating flows is more than one direction flows due to larger velocity gradient near walls. Therefore, the grid network in this study is finer.

$$Nu = \frac{k_f h_{sf}}{D_h} = (1 + 0.99(\mathrm{Re}\,\mathrm{Pr})^{0.66})\varepsilon^{1.79} \quad (7)$$

Residual amounts in convergence criteria section isset to $1\times10^{-6}$for continuity, momentum, energy and UDS equations in this CFD simulation.Discretization method for pressure and density (scalar variables) and momentum, energy and UDS (conservation equations) is second-order upwind in this study.

3. PHYSICAL MODEL, RESULTS & DISCUSSIONS

To show the simulated results based on multi-mesh regenerator, geometry and operating parameters have been selected from Harvey experimental model [2]. Schematic of simulated PTC is shown in Figure 2. Table 1 shows the characteristics of the system is studying in the present work. Piston head displacement and velocity are set as follows, respectively:

$$X = X_{amp}\sin(2\pi\nu t) \quad (8)$$

$$\frac{dX}{dt} = 2\pi\nu X_{amp}\cos(2\pi\nu t) \quad (9)$$

3. 1. Lowest Attainable Temperature & Efficiency of the PTC & Mesh Independency In order to validate the accuracy of the simulations first the PTC is simulated under adiabatic and some other fixed temperature condition of CHX wall with both LTE and DEE models and the results are compared with available numerical and experimental data as shown in Table 2. As shown in Table 2, results of simulations are in very good agreement with experimental data.

Table 3 illustrates the simulation results for different mesh type. With using fine mesh type (71037 cells),the CHX temperature just has a 0.4% reduction; so, for preventing huge calculation costs, the medium mesh type (51852 cells) was chosen for this study.

3. 2. Regenerator Characteristics In order to investigate the regenerator behavior, results of DEE model under the adiabatic CHX wall condition are considered. Figure 3 shows the pressure variations during a cycle at inlet and outlet of the regenerator. As shown in Figure 3, amplitude of the pressure decreases along the regenerator from inlet to outlet with a little phase shift.

Figure 4 shows variations of fluid and solid matrix temperature during a cycle at X/L=0.19 of the regenerator. Specific heat capacity of solid matrix is higher than fluid heat capacity.Therefore, as shown in Figure 4, amplitude of fluid temperature variations is higher than that corresponds to the solid matrix.

Besides, there are some deviations in the fluid temperature from the sinusoidal form. This deviation is due to flow returning inside the regenerator. When flow returns inside the regenerator the velocity of the fluid decreases and, consequently, convective heat transfer between fluid and solid matrix decreases.

Figure 5 shows velocity variations during a cycle for different sections of the regenerator. As shown in Figure 5, the amplitude of velocity variations in the inlet section of the regenerator is more than that of theoutlet section. The decrease in velocity amplitude is due to density increase in the outlet section of the regenerator.

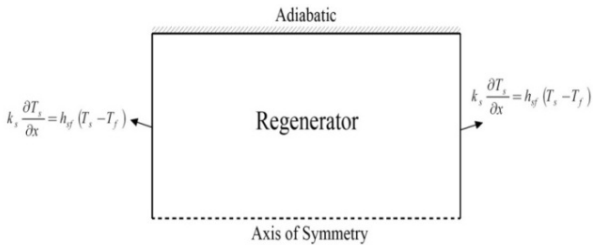


Figure 1. Boundary conditions of UDS in the regenerator

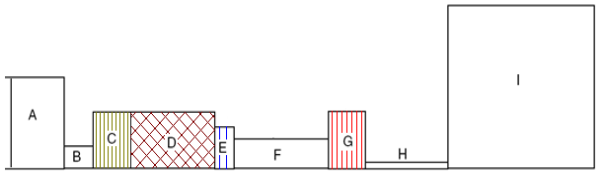


Figure 2. Schematic of the ITPTC

TABLE 1. Characteristics of the ITPTC, according to Figure 2

Component	Radius × length (mm^2)	Porous: mesh No. / ε / material	Wall thermal condition
Compressor (A)	9.54 × 7.5	-	adiabatic
Transfer Line (B)	1.55 × 101	-	adiabatic
After Cooler (C)	4 × 20	80 micron / 0.69 / copper	T=300K
Regenerator (D)	4 × 58	325 / 0.69 / steel	adiabatic
Cold Heat Exchanger (E)	3 × 5.7	80 micron / 0.69 / copper	adiabatic or fixed temperature
Pulse Tube (F)	2.5 × 60	-	adiabatic
Hot Heat Exchanger (G)	4 × 10	80 micron / 0.69 / copper	T=300K
Inertance Tube (H)	0.43 × 684	-	adiabatic
Reservoir (I)	13 × 130	-	adiabatic

TABLE 2. Simulation results

Compon.	DEE Model (present study)			LTE Model (present study)			LTE Model (Ref. 15)			Exp. (Ref. 2)
T_c (K)	88*	120	150	86.5*	100	150	87*	100	150	87*
$\dot{W}_{pv}(W)$	62.6	59.8	56.8	57.2	52.8	51.8	59.4	62.6	63.2	55
$\dot{Q}_{comp.}(W)$	-58	-55.6	-52.4	-57.4	-52.7	-49.3	-61	-63	-64	-
$\dot{Q}_{refrig.}(W)$	0	6	9.4	0	2.6	11	0	1.1	6.4	0
$\dot{Q}_{rej.}(W)$	-7.3	-9.1	-12.2	-2.5	-6.4	-10.19	-1.2	-2.8	-7.7	-
COP		0.1	0.16	-	0.05	0.21	-	0.02	0.1	-

* These values are calculated under adiabatic CHX condition.

TABLE 3.Characteristics of the ITPTC, according to Figure 2

Mesh Type	Number of Cells	CHX Temperature (K)
Coarse Mesh	16911	104
Coarse Mesh	37037	91
Medium Mesh	51852	86
Fine Mesh	71037	85.63

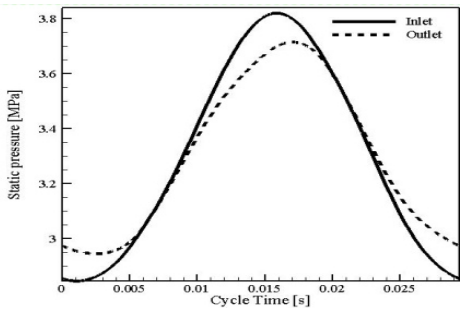


Figure 3. Pressure profile at regenerator inlet and outlet during a cycle under adiabatic CHX wall

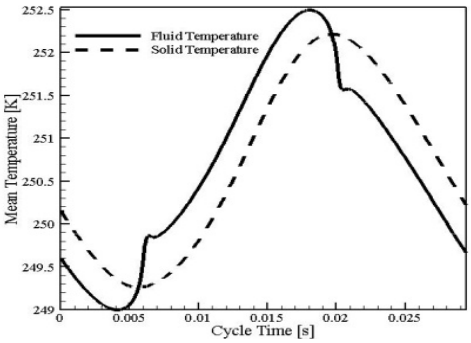


Figure 4. Fluid and solid matrix temperature profiles at X/L=0.19 of the regenerator during a cycle

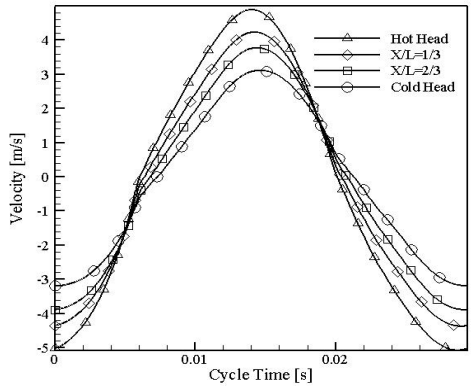


Figure 5. Velocity profiles at various sections of the regenerator

If Yuan et al. [22] correlations are used for friction factor and Reynolds number of the oscillating flow (Re_m), for the regenerator with uniform mesh#325 under frequency of 34 Hz, the friction factor reads from results of simulations as follow:

$$f_{osc,m} = \frac{102.9}{Re_m} - 0.5439 \quad \triangleleft\triangleright \quad 20 < Re_m \leq 150 \tag{10}$$

Figures 6- A, B, C and D show cyclic average of Reynolds number, Nusselt number, interfacial heat transfer coefficient and friction factor vs. Re_m along the regenerator, respectively.

As shown in Figures 6- A to C, the values of Reynolds and Nusselt numbers increase along the regenerator from the hot towards the cold end; whereas, the convective heat transfer coefficient decreases along

the same line. The decrease of the density of the fluid and the increase of the viscosity in the cold end are the main reasons that triggerthe increase of the Re and Nu. However, a decrease of convective heat transfer coefficient in the cold end is due to a very sharp decrease of helium thermal conductivity.

Figure 7 shows cyclic average of different sources of entropy generation along the regenerator. As shown in Figure 7, the most part of entropy generation (almost 85% of total) belongs to viscous and inertial effect [23, 24]. Furthermore, the entropy generation in the hot end of the regenerator is more than the values corresponding to the cold end. Accordingly, by decreasing the viscous and inertial entropy generation in the hot end of the regenerator, its operation and efficiency improves. This is the main goal of using multi-mesh porous medium in the regenerator parts which is investigated in coming section.

3. 3. Multi-mesh Regenerator As before mentioned most of the regenerator losses occur in its hot end [25]. The current investigation is prompted to investigate the effects of multi-mesh regenerator on the decrease of these losses. To reach this goal, the value of the last term of Equation(6) (viscous and inertial losses) must be decreased in the hot end of the regenerator. According to the parameters of this term, effect of variations is investigated here. For a perfect stacking of ε square-mesh screens in which the weaving causes no inclination of the wires and screen layers are not separated, ε and φ can be calculated as follow [26]:

$$\varepsilon = \frac{4X_t - \pi}{4X_t} \tag{11}$$

$$\varphi = \frac{(1 - X_t)^2}{X_t^2} = (1 + \frac{4}{\pi}(\varepsilon - 1))^2 \tag{12}$$

For stationary density and mass flow rate at each section of the regenerator one can write:

$$\dot{m} = \rho V \varphi A \rightarrow \varphi V = \frac{\dot{m}}{\rho A} = cte \tag{13}$$

Instituting Equation (12) into the Equation (13) yields:

$$V = \frac{\dot{m}}{\rho A}(1 + \frac{4}{\pi}(\varepsilon - 1))^{-2} \tag{14}$$

$$\frac{\partial V}{\partial \varepsilon} = \frac{\dot{m}}{\rho A}(\frac{-8}{\pi}(1 + \frac{4}{\pi}(\varepsilon - 1)^{-3}) \tag{15}$$

$$\varepsilon = 0.6 \rightarrow \frac{\delta V}{\delta \varepsilon} = -21.55 \frac{\dot{m}}{\rho A} \tag{16}$$

Therefore, increasing of ε causes an intense decrease in *V*, causing *εV* to decrease too. Moreover, the term *∂p/∂x* is always negative, and its variations versus *ε* can be evaluated as follow:

$$|\frac{\delta p}{\delta x}| = \frac{\mu}{\beta}|V| + \frac{1}{2}\rho C |V.V| \tag{17}$$

$$\beta = \frac{2l}{33.6n}\frac{\varphi}{\varepsilon} = \frac{2l}{33.6n}\frac{(1 - \frac{4}{\pi}(1 - \varepsilon))^2}{\varepsilon} \tag{18}$$

$$C = 0.377n(\frac{\varepsilon}{\varphi})^2 = 0.377n(\frac{(1 - \frac{4}{\pi}(1 - \varepsilon)^2}{\varepsilon})^2 \tag{19}$$

According to the Equations (17) to (19), if the hot end porosity is properly increased so that Darcy Permeability (β) is increased and Inertial Resistance Factor (C) is decreased, the amount of viscous and inertial losses will be decreased. Two different modes of multi mesh regenerators whose characteristics are listed in Tables 4 and 5 have been considered in the present work.

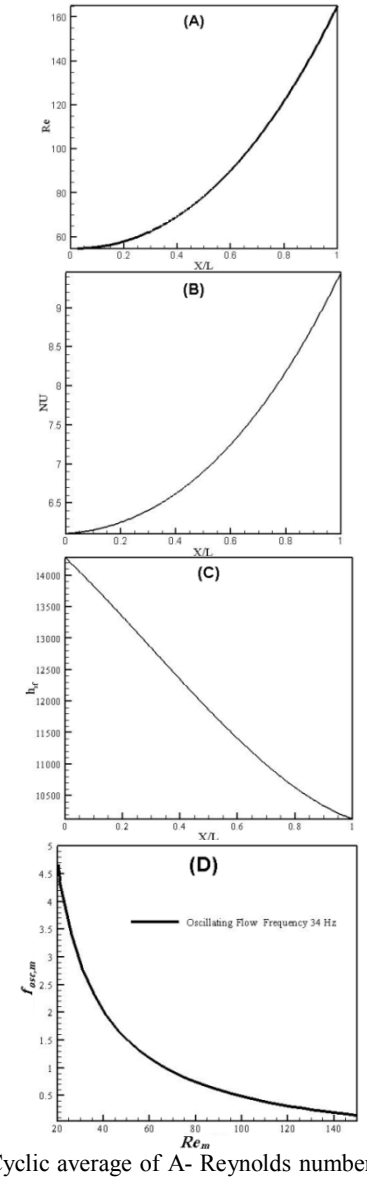


Figure 6. Cyclic average of A- Reynolds number, B- Nusselt number, C- convective heat transfer coefficient, and friction factor versus Re_m along the regenerator

As shown in Tables 4 and 5, porosity is decreased along the regenerator from the hot towards the cold end. This distribution is to balance the light area density of the hot end by higher area density in the cold end to reach an optimum operation of the regenerator [27]. Simulations were done under the fixed CHX wall temperature of 150 (K) for multi-mesh regenerators.

Figures 8- A, B and C show the pressure profile in the inlet and outlet of the regenerator during a cycle for uniform mesh and multi-mesh "A" and "B" regenerators, respectively. As shown in Figures 8- A to C the pressure drop of multi-mesh regenerator "A" and uniform mesh regenerator are semi-equal but the multi-mesh regenerator "B" has less pressure drop along the regenerator. Table 6 shows pressure drop and phase lag values for different regenerators.

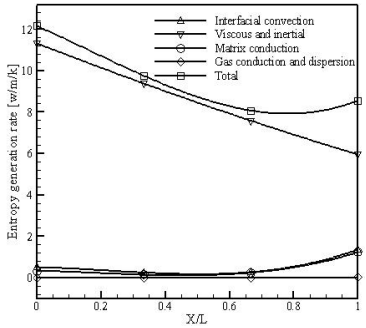
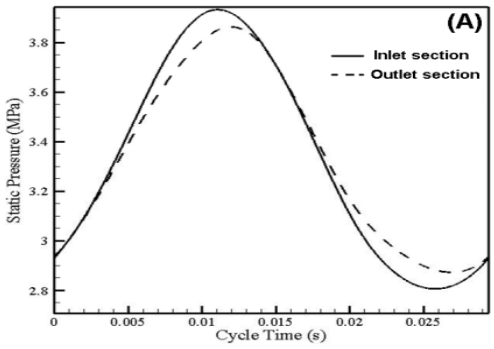
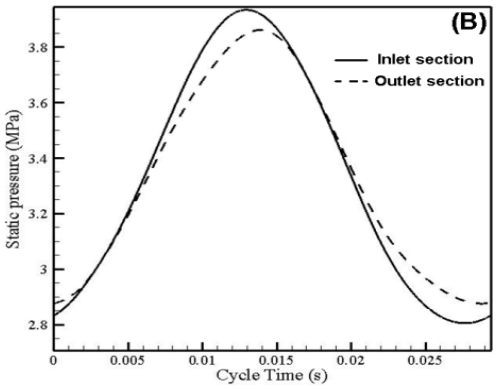


Figure 7. Cyclic average different sources of entropy generation along the regenerator





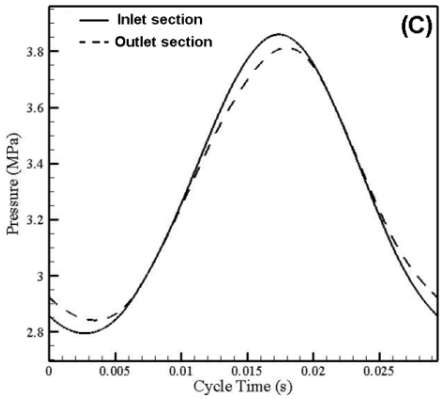


Figure 8. Pressure oscillation in the inlet and outlet sections of A- uniform mesh regenerator, B- multi-mesh "A" regenerator and C- multi-mesh "B" regenerator

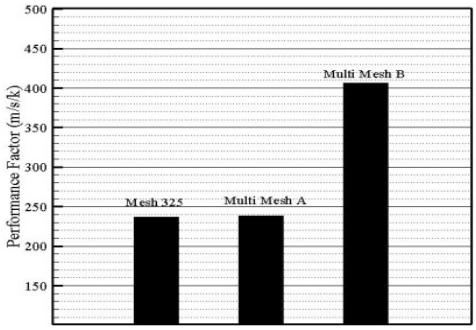


Figure 9. Performance factor of different regenerators

TABLE 4. Multi-mesh "A" properties

Properties	{hot head}0 ≤ x ≤ L/3	L/3 ≤ x < 2*L/3	2*L/3 ≤ x < L
Mesh type	SS spherical 60 micron	SS Mesh 325	SS Mesh 400
ε	0.7173	0.6961	0.692
β (m^2)	1.543×10^{-10}	1.06×10^{-10}	1.005×10^{-10}
C	68506.2	76090	103002.8
Area density (m^{-1})	28300	44444.44	48503.94
D_h (m)	152×10^{-6}	63.9×10^{-6}	57.1×10^{-6}

TABLE 5. Multi-mesh "B" properties

Properties	{hot head} 0 ≤ x ≤ L/3	L/3 ≤ x < 2*L/3	2*L/3 ≤ x < L
Mesh type	UMN	SS Mesh 325	SS Mesh 400
ε	0.9	0.6961	0.692
β (m^2)	1.86×10^{-7}	1.06×10^{-10}	1.005×10^{-10}
C	241.14	76090	103002.8
Area density (m^{-1})	493.83	44444.44	48503.94
D_h (m)	7290×10^{-6}	63.9×10^{-6}	57.1×10^{-6}

TABLE 6. Pressure drop and phase lag values along different regenerators

Multi-mesh "B"	Multi-mesh "A"	Uniform mesh	Components
$X_{\Delta p}$ (KPa)	50	76.5	80
ΔΦ	6.12 °	12.24 °	12.24 °

In order to study multi-mesh regenerator behavior, cyclic average pressure drop and fluid temperature along the multi-mesh regenerator "B" is illustrated in Figures 11 and 12, respectively. As shown in Figure11, the pressure drop is omitted in the first third span of the regenerator due to the use of media with higher porosity in this area. Besides, Figure 12 shows that there is a slight fluid temperature decrease in the hot end due to little inter-phase heat transfer area. It seems that in this area the fluid heat is lost as conduction and advection instead of storing in the solid matrix.

To investigate the operation performance of regenerators, performance factor, Rosc (ratio of inter-phase heat transfer to pressure drop), which is read from the Equation (20), is calculated.

$$R_{osc} = \frac{a_{sf} h_{sf}}{L_r} \quad (20)$$

Figures 9 show Rosc of different regenerators. According to Figures 9, theRosc value of multi-mesh regenerator "B" is almost 1.7 times higher than the value of mesh 325 and multi-mesh "A". Besides, as shown in Figures 10, the COP value of multi-mesh "B" regenerator PTC is almost 1.07 times higher than that of the others.

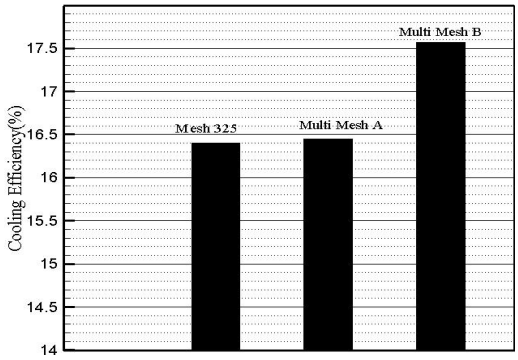


Figure 10. Different PTCs cooling efficiency

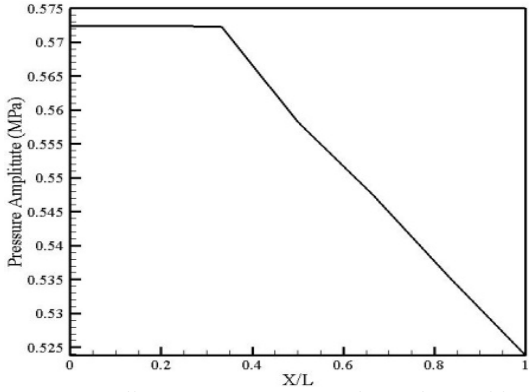


Figure 11. Cyclic average pressure along the multi-mesh regenerator "B"

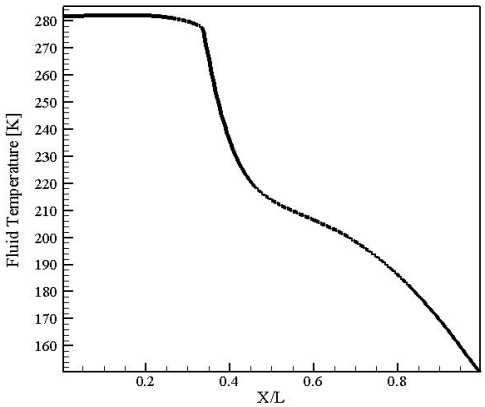


Figure 12. Fluid temperature variation along the multi-mesh regenerator "B"

4. CONCLUDING REMARKS

A single stage Stirling type ITPTC was investigated using Fluent software. In order to study the regenerator behavior and its effects on the performance of the PTC, both DEE and LTE models of heat transfer in porous medium were used. Comparison of results of DEE and

LTE models showed that DEE model is better than LTE onebecause it can show the sources of energy losses correctly.

For increasing performance of the regenerator, multi-mesh regenerator was studied. The porosity of medium in the hot end of the regenerator was increased to reduce the viscous and inertial losses in this area. On the other hand the porosity of its cold end was decreased to reach the proper heat transfer area between fluid and solid matrix. Two cases of multi-meshes named multi-mesh "A" and "B" were chosen. Results of simulations for fixed CHX wall temperature of 150 (K) show that R_{osc} of multi-mesh "B" and "A" and single mesh regenerators are 406, 239 and 238 (m/s.K), respectively. Corresponding COP of PTCs with multi-mesh "B" and "A" and uniform mesh regenerators are 0.1755, 0.164 and 0.164, respectively.

It is concluded that:

- Using DEE model of heat transfer in the regenerator part of the PTC is very useful because of its numeric power in revealing the amount of different energy loss sources.
- Proper use of multi-mesh regenerator improves R_{osc} of the regenerator, causing COP of the PTC to increase as well.

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CFD Simulation of a Multi-mesh Pulse Tube Regenerator

RESEARCH NOTE

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چکیده

طی دو دهه اخیر، سردساز لوله ضربانی لوله اینرتنسی به دلیل قابلیت اعتماد بالا کاربرد بسیار وسیعی در صنایع هوا فضا یافته است. به همین جهت تلاش‌های زیادی در جهت توسعه این سیستم صورت گرفته است. بررسی‌ها نشان داده است که بیشترین تلفات این سیستم در قسمت بازیاب که محیطی متخلخل است صورت می گیرد. به دلیل پیچیدگی فیزیک این تلفات، اثرات بازده بازیاب بر عملکرد سردساز با استفاده از روش CFD مورد بررسی قرار گرفت. جهت محاسبه انتقال حرارت جابجایی در محیط بازیاب از هر دو روش تک معادله‌ای و دو معادله‌ای بهره گرفته شد. محاسبه نرخ تولید انتروپی در محیط متخلخل بازیاب نشان داد که ۸۵٪ تلفات بازیاب ناشی از تلفات ویسکوز و اینرسیال است و همچنین مقدار این تلفات در انتهای گرم بازیاب اتفاق می‌افتد. بنابراین، به منظوربهبود عملکرد سردساز روش بازیاب با مش چندگانه مورد مطالعه قرار گرفت. نتایج حاصل از شبیه سازی‌ها تحت شرایط دمایی ۱۵۰ کلوین دیواره CHX مقدار ضریب عملکرد سردساز با بازیاب با مش چندگانه بهینه ۱/۰۷ برابر سردساز ساده است. مقایسه نتایج شبیه سازی‌ها با مقادیر آزمایشگاهی و مقادیر عددی منتشر شده در نشریات درستی نتایج شبیه سازی حاضر را تایید کرد.

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