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PERFORMANCE ENHANCEMENT OF A MINIATURE STIRLING CRYOCOOLER WITH A MULTI MESH REGENERATOR DESIGN

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Abstract

A parametric study has been carried out using the software REGEN 3.3 to optimize the regenerator of a miniature Stirling cryocooler operating with a warm end temperature of 300 K and cold end temperature of 80 K. Regenerator designs which produce the maximum coefficient of performance (COP) of the system is considered as an optimized regenerator. The length and diameter of the regenerator were fixed from the cooler system requirements. Single mesh regenerators made of 200, 250, 300, 400 and 450 Stainless Steel wire meshes were considered and the optimum phase angle and mesh size were obtained. A maximum COP of 0.1475 was obtained for 300 mesh regenerator at 70^0 phase angle. Then multi mesh regenerators were considered with finer mesh on the cold end and coarser mesh on the hot end. The optimum size and length of each mesh in the multi mesh regenerator and the optimum phase angle were calculated. The maximum COP of 0.156 was obtained for 200-300-400 multi mesh regenerator at 70^0 phase angle. The COP and net refrigeration obtained for an optimized multi mesh regenerator was found to be significantly higher than that of a single mesh regenerator. Thus a multi mesh regenerator design with a proper combination of regenerator mesh size and length can enhance the regenerator effectiveness.

Keywords: Stirling, Cryocooler, Regenerator, COP, Wire mesh.

1. Introduction

Modern spacecrafts widely use cryocoolers for the thermal management of the sensors used in their payloads meant for space science, remote sensing or telecommunications. Similar coolers are used in imaging cameras in battle tanks

Nomenclatures

A	Cross sectional area of regenerator, m^2
$c_m(T)$	Heat capacity per unit volume of the matrix, J/(m^3.K)
D_h	Hydraulic diameter, m
d_w	Wire diameter, m
E	Total energy of the gas, J/m^3
$H(p,T,v)$	Heat transfer rate between the gas and the matrix, W/(m^2.K)
h	Thickness of tube containing regenerator matrix, m
L	Regenerator length, m
$k_g(p,T)$	Thermal conductivity of the gas, W/(m.K)
$k_m(T)$	Thermal conductivity of the matrix, W/(m.K)
$p(t)$	Pressure in the gas, Pa
P_r	Pressure ratio $\left(\frac{P_{max}}{P_{min}}\right)$
T_c	Inflow temperature of the gas at cold side, K
T_h	Inflow temperature of the gas at warm side, K
T_m	Temperature of the matrix, K
t	Time, s
x	Spatial coordinate, $0 \le x \le L$, m
v	Velocity of the gas, m/s

Greek Symbols

θ	Phase angle between flow and pressure at the cold end of the regenerator, deg.
$\rho(p,T)$	Density of the gas, kg/m^3
τ	Period of oscillation, s
ϕ	Porosity of the matrix

Abbreviations

CFD	Computational Fluid Dynamics
NIST	National Institute of Standards and Technology

and HTS devices also. Stirling cryocoolers are used for these applications to produce cooling in the range of 60-80K with cooling power varying from mW to a few Watts. High efficiency, compact size, light weight, low power consumption and high reliability are some of the design goals of these coolers. The performance of the Stirling cryocooler depends on the effectiveness of heat transfer and fluid dynamics of the regenerative heat exchanger used in the system. Therefore the regenerator is a vital component in the design of Stirling cryocooler. Fine metal wire mesh in a form of woven screen at different wire sizes, weave structure, mesh density and material are used for regenerator matrix.The hot and cold fluids pass in periodical operation through the permeable regenerator matrix. During the hot blow or hot period, the hot fluid will pass through the regenerator and heat will store in the solid material. During the cold blow or cold period the flow is reversed and the stored heat is recovered by the cold fluid and rejected to the hot space.

The regenerative heat exchanger is the major loss source in the cryocooler. It causes losses due to its limited heat transfer units, limited matrix specific heat, pressure drop across the regenerator, regenerator dead volume and axial thermal conduction. Therefore optimum design of regenerator is necessary to improve the system performance

The basic thermal and fluid relations to calculate the actual dimensions of an optimized regenerator are given by Radebaugh and Louie [1]. De Waele et al. [2] studied the non ideal gas effect of working gas on the performance of the regenerator. They have shown that the non ideal gas properties have a profound effect on the energy balance in the regenerator and the expression for the cooling power. Their study shows the dependence of the temperature profiles on the thermal properties of the working fluid. Pfotenhauer et al. [3] used the NIST software REGEN 3.2 for the optimization of a single mesh single stage regenerator operating with a warm end temperature of 300 K and a cold end temperature that varies between 60 and 80K. The regenerator was made of 400 mesh Stainless Steel and the frequency of operation varied from 30 Hz to 60 Hz. Choi et al. [4] and Nam and Jeong [5] conducted experimental investigation on screen matrix regenerator and presented new oscillatory flow models of the pressure drop in oscillating flow through regenerator under pulsating pressure. Later they revised the flow friction factor correlation to improve the oscillatory flow model of the regenerator under cryogenic temperature [6].

Several works had done on modelling of regenerator as a porous media. Cha et al. [7] conducted experimental studies to measure the pressure drop under steady - periodic (axial and lateral) flow of helium through wire mesh regenerators. Using a CFD assisted methodology, the directional permeability and Forchheimer's inertial coefficient were obtained for the tested regenerator fillers. Tao et al. [8] investigated the hydrodynamic and heat transfer performances of regenerators with meshes of different geometric and material properties. Costa et al. [9] proposed a finite volume method (FVM) based non-thermal equilibrium porous media modelling approach characterizing the fluid flow and heat transfer in a representative small detailed flow domain of the woven wire matrix to obtain the porous media coefficients. Zhao et al. [10] developed a two dimensional axis-symmetric CFD model of a miniature Stirling type pulse tube cryocooler and calculated various regenerator losses using thermal equilibrium and non thermal equilibrium models for the porous matrix

The major losses in Stirling cryocooler are due to the heat transfer and fluid flow irreversibility in the regenerator. Accurate numerical modeling of these losses requires extensive numerical simulation of coupled mass, momentum and energy conservation equations. The geometry, material selection, frequency, temperature, pressure ratio and the phase between flow and pressure will influence the performance of the regenerator. The regenerator design code REGEN 3.3 [11] developed at NIST is a powerful tool to investigate the influence of these parameters on the performance of the regenerator. ^{4}He or ^{3}He can be considered as the working fluid and the ideal gas version of either gas also can be included in the analysis. Using REGEN 3.3, we can analyze regenerators made of a variety of materials (more than 30) and flow geometries (5). Temperature dependant properties of matrix materials and user defined materials or geometry can also be included in the analysis. The boundary conditions for the older versions 3.1 or 3.2 required the mass flow to be given at both ends of the regenerator. In the new version REGEN 3.3, the mass flow and pressure at the cold end are the inputs.

In the present study, a regenerator with fixed length, diameter, pressure ratio and average pressure is considered. The length and diameter of the regenerator were fixed from the cryocooler system requirement. The effect of mesh size and phase angle between mass flow and pressure on the performance of the single mesh regenerator is studied and optimum values of these parameters are obtained. Then multi mesh regenerators having same length and diameter are considered. It is composed of different Stainless Steel wire meshes in such a way that the porous mesh is at the warm end and denser mesh at the cold end of the regenerator. The COP of cryocooler depends on the length of different meshes in the multi mesh regenerator used in the system. In this investigation, a multi mesh regenerator with 3 different types of wire meshes was optimized.

2. Method

The physical model of the regenerator used in this study is a tube filled with a porous medium. An oscillatory flow of helium passes through the void space in the porous medium or matrix. The fluid is alternatively heated and cooled as the flow direction is reversed. ^4H_e is used as the working fluid. The conservation equations for the mass, momentum and energy in a discrete form are the basis for the REGEN 3.3 model. The effect of the porous media is modelled by the addition of a friction term in the momentum equation and a heat transfer term in the energy equation. The following conservation equations are solved in the model [11].

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho v)}{\partial x}=0 \tag{1}$$

$$\frac{\partial \rho v}{\partial t}+\frac{\partial(\rho v^2+p)}{\partial x}-f(\rho,T,v)=0 \tag{2}$$

$$\frac{\partial \emptyset AE}{\partial t}+\frac{\partial(\emptyset A(E+p)v)}{\partial x}-\frac{\partial(\emptyset A k_g \partial T/\partial x)}{\partial x}-\emptyset A q(p,T,T_m,v)=0 \tag{3}$$

$$\frac{\partial D}{\partial t}+\emptyset A q(p,T,T_m,v)-\frac{(1-\emptyset)A k_m \partial T_m/\partial x}{\partial x}=0 \tag{4}$$

where the matrix thermal content term *D(x, T)* is defined as,

$$D(x,T)=\int_{T_{min}}^{T}(1-\emptyset)A c_m(x,T)dT \tag{5}$$

The heat transfer term $q\ (p, T, T_m, v)$ is defined as,

$$q(p,T,T_m,v)=4H(p,T,v)(T_m-T)/D_h \tag{6}$$

Sinusoidal mass flow is assumed at each end of the regenerator. Frequency, pressure ratio and the phase angle between pressure and mass flow at the cold end are given as the input parameters. The temperatures at cold and hot end of the regenerator were also given as inputs.

The analysis started with the study of the effect of phase angle and mesh size on the performance of single mesh regenerator. Then the performance of the regenerator with combination of three meshes with different mesh sizes was studied. #200, #250, #300, #400 and #450 Stainless Steel wire meshes were considered for single mesh regenerator and their combinations are used for multimesh regenerator. The regenerator is operating with a warm end temperature of 300 K and cold end temperature of 80 K. The frequency of operation is 50 Hz. The pressure ratio P_r and the average pressure P_a are 1.8 and 16 bar respectively. The working gas and the matrix material are modelled with temperature dependant thermo physical properties. The geometric properties of the wire mesh are summarized in Table 1. The hydraulic diameter of the wire mesh screen can be determined by the Eq. (7),

$$D_h = d_w \frac{\phi}{1-\phi} \tag{7}$$

Table 1. Geometric properties of the regenerator.

Type of wire screen	Wire diameter d_w [μm]	Porosity, ϕ
#200	58	.667
#250	41	.666
#300	30	.689
#400	25	.701
#450	25	.651

The output of REGEN 3.3 includes the gross and net cooling power, thermal losses, associated pV work at the cold and warm ends and many other parameters. A regenerator design which minimizes the required work supplied at the warm end to achieve a desired cooling power at the cold end is considered as an optimized regenerator. The optimization can be achieved by maximizing the co-efficient of performance (COP). The co-efficient of performance is calculated by Eq. (8),

COP=NTCADJ/PVWK0T (8)

where *NTCADJ* is the net cooling power and *PVWK0T* is the pV work at the warm end of the regenerator. The net cooling power *NTCADJ* is calculated beginning from the pV work term at the cold end of the regenerator *(PVWK1).*

This is computed from the integral,

$$\mathrm{P}VWK1 = \int_{t-\tau}^{t} \frac{\phi A v(L,t) p(t) \rho(L,t)}{\rho_c \tau} dt \tag{9}$$

where *v(L,t)* and *ρ(L,t)* are the instantaneous velocity and density of the gas at the cold end of the regenerator. *p(t)* is the instantaneous dynamic pressure and ρ_c is the gas density at the cold end temperature T_c.

The net cooling power *(NTCADJ)* is obtained from Eq. (10),

NTCADJ=GRCADJ-RGLOSS-HTFLUX-TUBECD (10)

The gross refrigeration adjusted for losses in the expansion space *(GRCADJ)* is given by Eq. (11),

*GRCADJ=(PVWK1-PRLOSS)*COOLING_MULT* (11)

PRLOSS is the correction term used to estimate the effect of pressurization on the enthalpy flux of gas and COOLING_MULT is the term used to estimate a reduced refrigerator power produced through a non-isothermal expansion.

The loss due to regenerator ineffectiveness is given by Eq. (12),

RGLOSS=ENTFLX-PRLOSS (12)

ENTFLX is the integral average of the enthalpy flux at the cold side of the regenerator over one cycle and *HTFLUX* is the heat flux due to thermal conduction through the cold side of the matrix. *TUBECD* is the thermal conduction through the tube containing the regenerator matrix. The material of the tube is assumed to be Stainless Steel with a steady state temperature profile and the thermal exchange between the tube and the matrix is ignored. *TUBECD* is given by the formula,

$$TUBECD = -\frac{\sqrt{4\pi A}h}{L}\int_{T_C}^{T_H}\sigma(T)dT \qquad (13)$$

Here, *σ(T)* is the thermal conductivity of Steel. The effective thermal conductivity of the matrix is smaller than the bulk thermal conductivity of the matrix material due to the reduced contact surface of the different layers of matrix material [12]. The effect of porosity on thermal conductivity can be incorporated in REGEN analysis using a thermal conduction correction factor.

From Eqs. (10), (11) and (12), the net cooling power can be calculated as,

NTCADJ=PVWK1-ENTFLX-HTFLUX-TUBECD (14)

The procedure used for the calculation of hot end pV work is similar to that used for cold end pV work. But in this case the velocity and density are evaluated at the warm end and the pressure at the hot end is obtained by adding the pressure drop to the pressure at the cold end.

For small values of pressure ratio, the pV work at the cold end, *PVWK1* can be approximated by Eq. (15),

$$W_{accoustic} = \frac{1}{2}RT_C\dot{m}_C\frac{P_d}{p_0}cos\theta \qquad (15)$$

where R is the gas constant and $\dot{m}_c$ is the cold end mass flow rate. The phase angle between the flow and the pressure affects the flow amplitude, and therefore the regenerator loss, for a given acoustic flow. The minimum loss is expected to occur for flow in phase with the pressure, but that phase can occur at only one location of the regenerator, which should be near the midpoint. For that phase to occur near the midpoint, the phase at the cold end has to be found out. In a Stirling cryocooler this phase is set by the motion of the displacer. For a fixed value of cold end pV work, the cold end mass flow rate ($\dot{m}_c$) and the phase angle at the cold end *(θ)* are calculated. To initiate the analysis, these values are used as the input parameters. The fixed values of parameters used in the study are given in Table 2.

Table 2. Fixed values of parameters.

Cold end Temperature, T_C	80 K
Hot end Temperature, T_H	300 K
Regenerator tube diameter	8.6 mm
Pressure ratio, P_r	1.8
Average pressure, P_a	16 bar
Length of regenerator, L	52 mm
Regenerator tube thickness, h	0.1 mm

The pV work required at the cold end is fixed as 5W. In the first stage of investigation, the performance of various single mesh regenerators were studied at different phase angles and the regenerator which give the maximum COP and the corresponding phase angle were obtained. In the second stage, the performance of multi mesh regenerators made of different wire mesh matrix was investigated at the above phase angle. In a multi mesh regenerator, the regenerator matrix was divided in to three sections and the regenerator was optimized to get the maximum COP by adjusting the length and matrix parameters of each sections. The orientation of a multi mesh regenerator is shown in Fig. 1.

As given in the user guide of REGEN 3.3, the accuracy of the solution and the cyclic steady state can be monitored through the ratio EHTDIF/EHTFLX. The value of *EHTDIF* is the maximum difference in the enthalpy plus heat flux integral over the length of the regenerator and *EHTFLX* is the integral average of the sum of the enthalpy flux and the heat flux at the cold side of the regenerator. If the value of *EHTDIF* is large compared to *EHTFLX* and is not reduced by running more cycles, the solution will be a poor solution. If the value is less than 10% of *EHTFLX*, the solution is considered acceptable

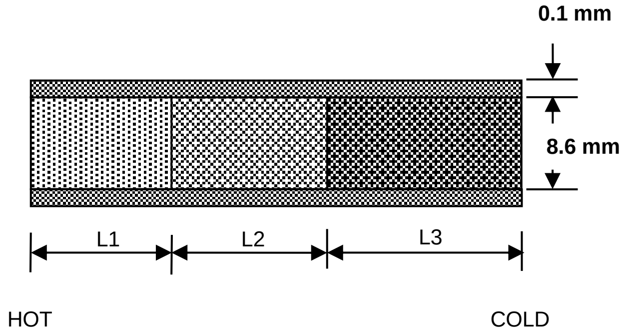

Fig. 1. Multi mesh regenerator.

3. Results and Discussion

The most important performance parameter of interest is the coefficient of performance (COP). Regenerators made of different wire mesh size are analyzed and the COP is plotted as a function of phase angle between the mass flow and the pressure at the cold end. From Fig. 2, it is clear that the COP of #300 is higher than the COP of all other meshes at all phase angles. The maximum COP of 0.1475 is obtained for a 300 mesh regenerator at 70^0 phase angle. Therefore a mesh size of 300 and phase angle of 70^0 can be considered as the optimum values for the single mesh regenerator.

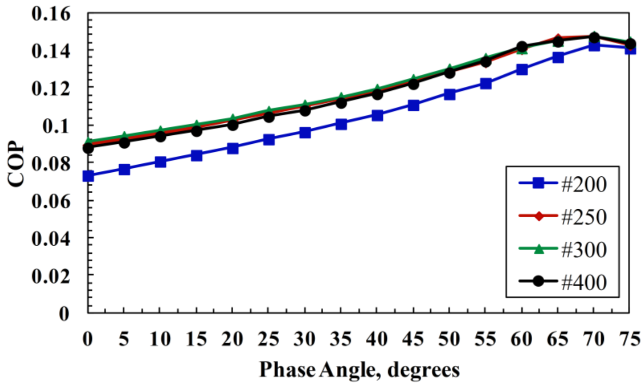

Fig. 2. COP of single mesh regenerators.

The variation of net refrigeration and hot end pV work with variation of phase angle is plotted for the #300 regenerator in Fig. 3. With an increase in the phase angle, the net cooling power increases and required hot end pV work decreases. The minimum value of hot end work and maximum value of cooling power are obtained at 70^0 phase angle.

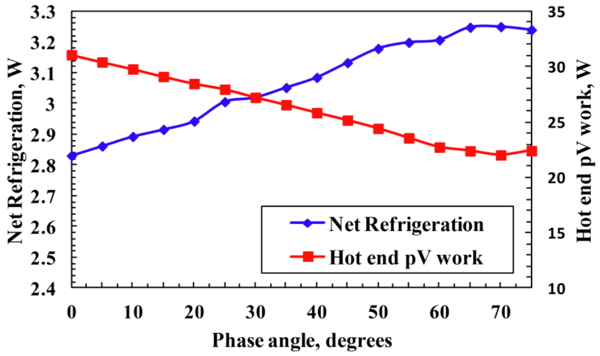

Fig. 3. Hot end pV work and net refrigeration of #300 regenerator.

Now, instead of using a single mesh for the entire length of regenerator, different combinations of 200, 250, 300, 400 and 450 meshes are considered for the regenerator. The multi mesh regenerator is constructed in such a way that finer mesh is used at the cold end and coarser mesh is used at the hot end. The COP of multi mesh regenerator was evaluated for length L3=7 mm, L3=17 mm, L3= 27 mm and L3= 37 mm. For each value of L3, different combinations of lengths L1 and L2 are considered and the COP, net cooling power and hot end acoustic power are evaluated. The COP of different combinations of multi mesh regenerators at 70^0 phase angle are plotted in Fig. 4. The maximum COP of 0.156 was obtained for #200-#300-#400 regenerator with sub lengths L1=5 mm, L2=10 mm, and L3= 37 mm.

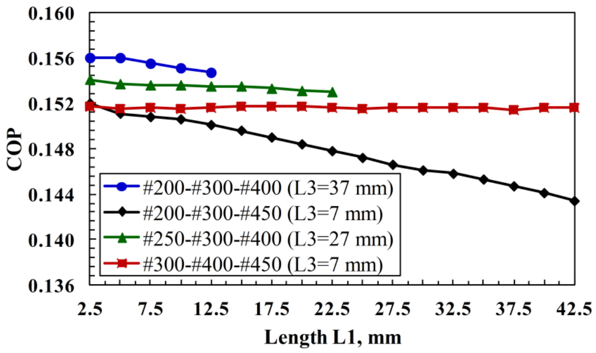

Fig. 4. COP of Multi mesh regenerator (L1+L2+L3=52 mm).

The COP of the system with #300 and #200-#300-#400 (L1=5 mm, L2=10 mm, L3=37 mm) regenerators are plotted in Fig. 5. The COP increases with the increase in phase angle and the maximum was obtained at 70°. The COP of multi mesh regenerator is higher than the single mesh regenerator at all phase angles. Among the single mesh regenerators and the four multi mesh regenerators considered, #200-#300-#400 multi mesh regenerator gives the maximum COP. At 70^0 phase angle the maximum COP obtained was 0.156. Compared to an optimized single mesh regenerator, the maximum COP of the multi mesh regenerator was increased by 4 %.

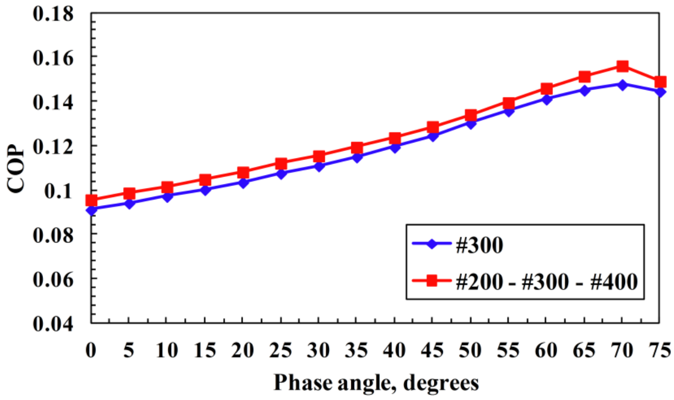

Fig. 5. COP of Single mesh and multi mesh regenerators (L1+L2+L3= 52 mm).

The variation of net refrigeration obtained for #300 and #200-#300-#400 regenerators with the increase in phase angle is shown in Fig. 6. The net cooling power increases with the increase in phase angle. The net refrigeration obtained is higher for multi mesh regenerator than that of a single mesh regenerator. At 70^0 phase angle, the #200-#300-#400 mesh regenerator gives the maximum cooling power (3.363 W).

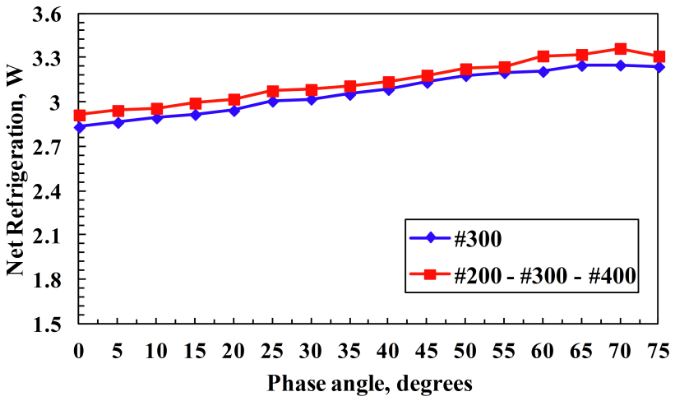

Fig. 6. Net refrigeration of single mesh and multi mesh regenerators (L1+L2+L3= 52 mm).

The hot end pV work required for #300 and #200-#300-#400 regenerators are shown in Fig. 7. The hot end pV work is minimum (21.56 W) for #200-#300-#400 regenerator.

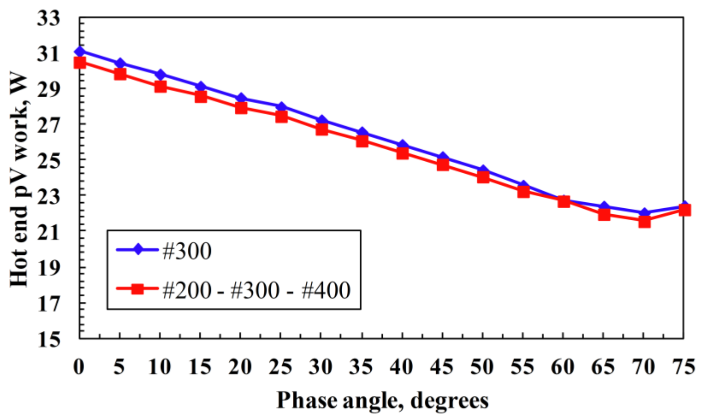

Fig. 7. Hot end pV work of single mesh and multi mesh regenerators (L1+L2+L3= 52 mm).

4. Conclusions

An investigation has been carried out to obtain the optimum parameters of regenerator of a Stirling cryocooler. The cooler is operating with a cold end temperature of 80 K and hot end temperature of 300 K at a frequency of 50 Hz. The NIST numerical model REGEN 3.3 was used for the study.

The significant observations from the study are given below.

- Among the four single mesh regenerators considered, the 300 mesh Stainless Steel wire matrix regenerator produces largest system COP. The optimum phase angle between mass flow and pressure at the cold end is 70^0.
- The multimesh regenerators with coarser matrix at the warm end and finer matrix at the cold end of the regenerator produce larger COP than a single mesh regenerator.
- Four multimesh regenerators composed of 3 different wire meshes were considered for investigation. The maximum COP was obtained for #200-#300-#400 multi mesh regenerator having lengths L1=5 mm, L2= 10 mm and L3=37 mm.
- Compared to an optimised single mesh regenerator, an optimised multimesh regenerator produces significantly larger COP. This may be because of the lower pressure loss through the regenerator due to the use of coarser mesh on the hot end and finer mesh on the cold end instead of using a mesh of uniform size.
- The optimization of a three mesh regenerator resulted in a design with 200 mesh for a length of 5 mm at the hot end, 300 mesh for a length of 10 mm in the mid section and 400 mesh for a length of 37 mm at the cold end of the regenerator.
- An optimized multimesh regenerator with coarse mesh on the hot side of the regenerator can be used for enhancing the performance of a Stirling cryocooler. The results obtained from the present study will help in the complete design of the miniature Stirling cryocooler.

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