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*DOI:*
10.1016/j.egypro.2017.12.547

Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 142 (2017) 1668–1673

Energy Procedia

www.elsevier.com/locate/procedia

9th International Conference on Applied Energy, ICAE2017, 21-24 August 2017, Cardiff, UK

# CFD modelling and parametric study of small scale Alpha type Stirling Cryocooler

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**Abstract**

In this paper, a small-scale Stirling Cryocooler is developed using computational fluid dynamics through COMSOL Multiphysics 5.3. The model has been validated using experimental data from an Alpha type Stirling engine reported in the literature. Extensive parametric study was carried out to investigate the effect of various parameters such as operating speed and phase angle with the aim of optimizing the Cryocooler design parameters and operating conditions. Results showed that there exists an optimal value for the regenerator porosity 50%, phase angle $90^0$ and heat exchanger lengths 142mm where 455W cooling power was obtained at heat sink temperature of 193K.Whereas increasing the speed and pressure resulted in an increase in cooling power at the heat sink.


Peer-review under responsibility of the scientific committee of the 9th International Conference on Applied Energy.

*Keywords:* Stirling Cryocooler, CFD, Heat transfer, Parametric Study, Regenerator

## 1. Introduction:

Miniature Cryocoolers have remained a popular area of interest in recent decades due to their wide range of commercial and military applications, such as providing cooling for space surveillance, infrared detector systems, thermal imaging cameras and superconducting quantum interference devices [1]. While several types of Cryocoolers exist, Stirling Cryocoolers in particular are attractive due to their ability to operate at high speeds, their high coefficient of performance (COP) and quiet operation [2]. The design of Stirling Cryocoolers is quite difficult as their successful operation relies on the involvement of multiple physical phenomena, hence theoretical

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Peer-review under responsibility of the scientific committee of the 9th International Conference on Applied Energy .
10.1016/j.egypro.2017.12.547

thermodynamic models used in literature can prove to be too simplified and unable to capture the inherent complexity of these machines. To overcome this, tests can be performed using a prototype intended to measure the performance and optimize the Cryocooler [3], but there is a disadvantage in that this can add unnecessary cost to the manufacturing process. Computational fluid dynamics (CFD) modelling software can prove beneficial as they have the capability to predict the cooling power at various design and operating conditions. Caughley et al. [4] used CFD as a tool to help understand the underlying fluid dynamics and heat transfer mechanisms that occur inside a Stirling Cryocooler with the aim of improving performance. The results of Caughley's CFD analysis was not only close to experimental values but also highlighted possible areas of improvement such as increasing the length of the warm heat exchanger to achieve relatively better cooling. Alfarawi et al [5] developed and validated a CFD model of the STG-05 Stirling Engine designed by Dieter Viebach and investigated the effect of phase angles and dead volumes and using the results deduced that for the STG-05, the optimum phase angle was at 105° rather the 90° used in its present configuration. Almajri et al [6] also developed a CFD model for the seldom studied alpha type Stirling engine, using COMSOL Multiphysics [7] to perform a parametric study on the engine operating parameters and showed the possibility of using CFD as a method to enhance Stirling engine designs. There is very limited work on using alpha Stirling as Cryocooler, therefore this work aims to develop an efficient miniature alpha type Stirling cryocooling system capable of cooling to temperatures lower than -40°C for a wide range of medical applications including the destruction of cancerous tissues.

**Nomenclature**

| | | | |
|---|---|---|---|
| $CFD$ | computational fluid dynamics | Sc | compression space swept volume ($m^3$) |
| $COP$ | coefficient of Performance | $Se$ | expansion space swept volume ($m^3$) |
| $Cp$ | heat capacity of the gas at constant pressure (J/Kg K) | T | temperature (K) |
| $eff$ | effective volume heat capacity (J/Kg K) | $Xc$ | compression space piston displacement (m) |
| I | identity matrix | $Xdc$ | height of dead volume (compression space)(m) |
| K | permeability | $Xde$ | height of dead volume (expansion space)(m) |
| k | thermal conductivity of fluid ($W/m^2K$) | $\epsilon p$ | Porosity |
| $keff$ | effective thermal conductivity (W/m K) | $\alpha$ | Phase angle (rad) |
| $p$ | instantaneous gas pressure (Pa) | $\mu$ | dynamic viscosity of fluid (Pas) |
| $q$ | heat flux | $\rho$ | density of fluid ($Kg/m^3$) |
| $Re$ | Reynolds Number | βF | Forcheimer coefficient |
| $Rs$ | gas constant (J/Kg K) | θp | volume fraction |

## 2. CFD modelling of the cryocooler:

COMSOL Multiphysics software was used to model the Stirling Engine. The geometry was first developed using the Computer aided design software, SOLIDWORKS and was then subsequently imported into COMSOL 5.3. Equations detailing the piston displacement and velocities alongside the volume variations in the cylinders were uploaded and saved as local variables, whereas regenerator porosity, charge pressure and engine RPM were defined as parameters. The materials section was used to define the materials and properties of the different domains, with the working fluid defined as air, the connecting pipe as copper and all other solid parts as steel.

The main assumptions used to build the CFD Model were the working fluid being ideal, no leakage of working fluid throughout the Cryocooler, pistons were considered to move in a sinusoidal manner with a constant phase angle difference and the flow of the working fluid assumed to be laminar.

The physics modules selected to create the model were fluid flow, heat transfer and the 'Arbitrary Lagrange-Eulerian' (ALE) method to model the deformation of the air due to piston operation. For the fluid flow interface; as the flow was assumed to be laminar (Reynolds number <2000) and compressible, it was described using the Navier-Stokes equation for the conservation of momentum shown in equation (1) and the continuity equation for the conservation of mass, equation (2)

$$\rho\frac{\partial u}{\partial t}+\rho(u.\nabla)u = \nabla.\left[-pI+\mu(\nabla u+(\nabla u)^T)-\frac{2}{3}\mu(u.\nabla)I\right] \quad (1)$$

The interface settings were configured to enable porous media domains in the model to account for the flow in the regenerator, which was modelled using the Brinkman variant of the conservation equations (3, 4);

$$\frac{1}{\epsilon_p}\rho\frac{\partial u}{\partial t}+\frac{1}{\varepsilon_p}\rho(u.\nabla)u\frac{1}{\epsilon_p} = \nabla.\left[-pI+\mu\frac{1}{\epsilon_p}(\nabla u+(\nabla u)^T)-\frac{2}{3}\mu\frac{1}{\epsilon_p}(u.\nabla)I\right]-\left(\mu K^{-1}+\beta_F|u|\frac{Q_{br}}{\epsilon_p^2}\right)u \quad (2)$$

$$\rho\frac{\partial\varepsilon_p\rho}{\partial t}+\nabla.(\rho u) = Q_{br} \quad (3)$$

In the heat transfer module, constant temperature walls were applied across the hot and cold cylinders to simulate the effect of the hot and cold sides of the cryocooler with the other walls left as being thermally insulated. The following equations (4), (5) and (6) were used to model heat transfer in the fluid equations and solid domains.

$$\rho C_p\frac{\partial T}{\partial t} + \rho C_p u.\nabla T+\nabla.q = Qtot \quad (4)$$

$$q = -k\nabla T \quad and \quad \rho = \frac{p_A}{R_s T} \quad (5 \text{ and } 6)$$

To model heat transfer in the regenerator, the heat transfer in porous media module was selected using equations (7, 8 and 9) to model the process. The thermal properties of Chrome Nickel were used to define the solid section with the thermal properties of air, treated as an 'ideal gas' used to describe the fluid section of the regenerator.

$$q = -k_{eff}\nabla T \ , \quad k_{eff} = \theta_p k_p + (1-\theta_p)k \quad and\ \theta_p = 1-\epsilon p \quad (7, 8 \text{ and } 9)$$

To effectively describe the movement of the pistons, the ALE method was used to define the moving boundaries using the prescribed displacement interface to set the respective piston displacement and velocities. The displacement of the pistons in the expansion and compression space with respect to the crank angle, equations (10) and (11) respectively, were defined according to Schmidt's ideal analysis [8].

$$X_e = X_{de}+\frac{S_e}{2}[1+cos(\theta)] \ , \ X_c = X_{dc}+\frac{S_c}{2}[1+cos(\theta-\alpha)] \quad (10 \text{ and } 11)$$

Where $X_{de}$ and $X_{dc}$ refer to the heights of the clearance volumes, $\theta$ being the crank angle, $S_e$ and $S_c$ the respective piston strokes and $\alpha$ being the phase angle difference between the pistons.

The dimensions and operating conditions of the Stirling cryocooler as reported by Karabulut et al [9] are shown in table (1) with engine speed 900 RPM. For the boundary conditions, constant temperature walls of 233 K and 300 K were applied on the expansion and compression space respectively with the other walls considered adiabatic.

Table1. Initial assumption of design parameters [9]

| | |
|---|---|
| Stroke | 60mm |
| Bore | 46mm |
| Ambient hx area | 200cm$^2$ |
| Heat sink hx area | 95cm$^2$ |
| Regenerator dead volume | 12cm$^3$ |
| Working fluid | Nitrogen |
| Charge pressure | 2 bar |
| Porosity | 70% |

Fine triangular elements were used to mesh the geometry of the engine on the recommendation of Almajri [6], resulting in 8400 elements total with an average element quality of approximately 0.95. The Cryocooler geometry is displayed in Fig. (1a) depicting the fluid (blue) and solid domains (grey) with Fig. (1b) shows the meshed model. Figure (2) shows the temperature distribution of the working fluid in the various domains of the Stirling Cryocooler when the expansion space is maintained at a temperature of 233 K operating at a frequency of 15Hz. Figure (3) depicts the variation of pressure and volume in the Cryocooler system (PV diagram) over the course of one cycle using the same operating conditions. The cooling power and input power can be found by integration of the expansion and compression space PV plots respectively. The corresponding cooling work per cycle for the figure below is 11.5 J (172W). The CFD model was validated using experimental results of the modelled Stirling engine reported by Karabulut et al [8] with maximum deviation of 14%.

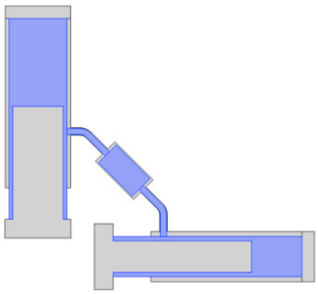


Fig. 1 (a). Cryocooler Geometry

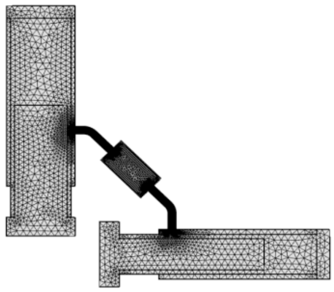


Fig. 1 (b). Meshed Geometry

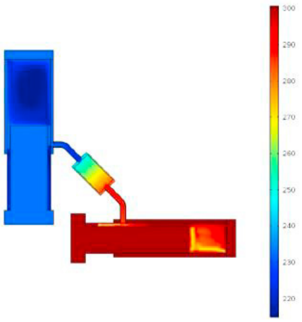


Fig. 2. Temperature Contour of Cryocooler at t=0.067s and 360 crank angle

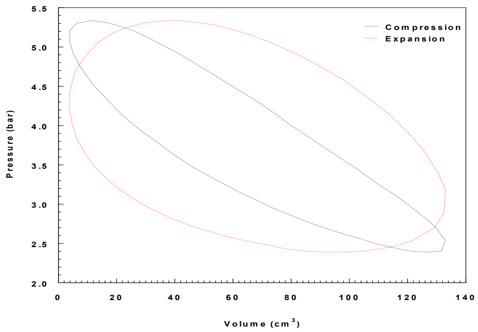



Fig. 3. PV diagram of the Stirling Cryocooler

## 3. **Results and Discussion:**

The CFD model was the used to find the minimum temperature that the cryocooler can achieve. Figure (4), shows the temperature versus time where initially there is almost no temperature decrease in the expansion space until just after 5 seconds of flow time and then the temperature decreases in an almost exponential manner up to approximately 34 seconds where the temperature no longer changes, signalling the system reaching steady state and therefore the minimum temperature (of approximately 170K) reached.

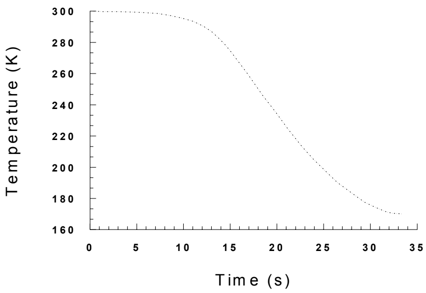
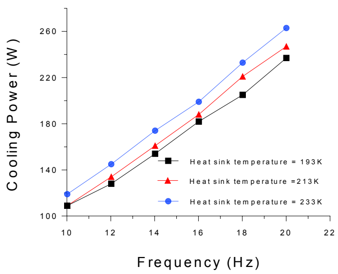


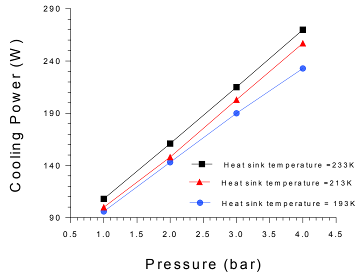

Fig.4. Cool down curve

According to the works of Walker [10], the performance of any Stirling Cryocooler is dependent on speed, pressure of working fluid, ratio of the temperatures between the expansion space and compression space, phase angle and volume of the heat exchangers. Figure 5 shows the effect of these parameters on the cooling power of the Stirling engine at -40C. Figure (5a) shows that the cooling power is proportional to the speed with greater heat lift attainable at higher speeds. Figure (5b) also details a linear relationship between the pressure and cooling power and this can be attributed to the assumption of the working fluid as an ideal gas. Fig (5c) shows the effect of the porosity is more pronounced at the highest expansion space temperature of 233K whereas when the heat sink has a temperature of 193K, changing the porosity has minimal effects on the cooling power. This may be due to other parameters having a greater influence at lower heat sink temperatures. The curve for each heat sink temperature is all in agreement however a porosity of 50% is required for better performance.

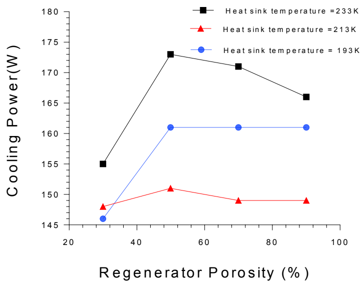
The effect of the phase angle difference between the two pistons was studied between a range of 60 degrees and 120 degrees as shown in figure (5d). The highest cooling power was observed when the phase angle between the pistons was 90 degrees which is usually the typical phase angle used when a Stirling device of the alpha type is used as an engine. Figure (5e) shows the peak cooling power occurs at different lengths for the heat exchangers but compared to the other parameters studied, the increase in power obtained by increasing the heat exchanger length is relatively small.



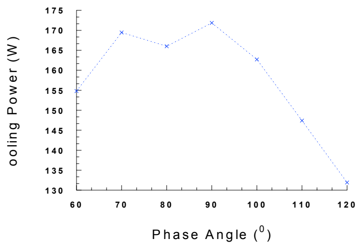

(a) Speed vs. Power



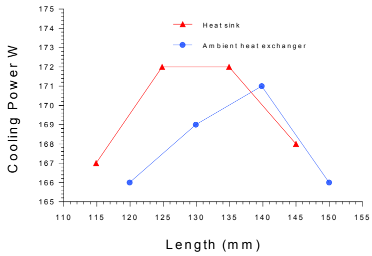

(b) Pressure vs. Power




(c) Porosity vs. Power



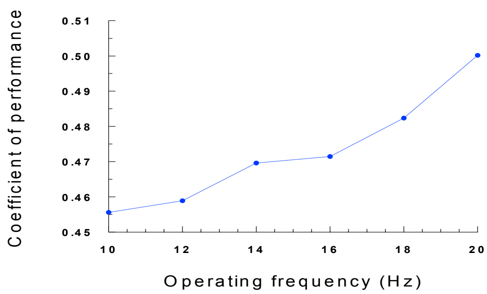

(d) Phase angle vs. Power




(e) Heat exchanger lengths vs. Power

Fig.5. Effect of operating parameters on the performance of the Cryocooler

The COP plot against frequency, figure (6a) shows that as the speed increases, so does the COP which suggests in addition to better performance, the Cryocooler is more efficient when run at higher speeds. Also, Figure (6b) shows that increasing the pressure of the fluid has no effect on the COP which suggests that while increasing the pressure allows more heat to be extracted at the cold expansion space, this comes at the cost of having to provide more work in the compressing the gas.



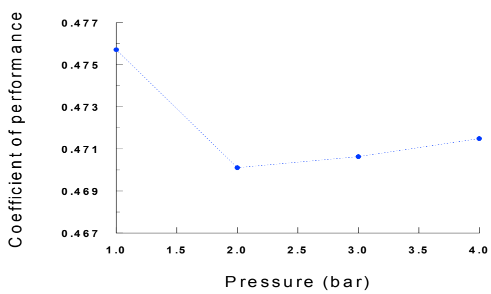




Fig.6. Effect of (a) Speed and (b) Pressure on COP

Using these new parameters (ambient heater area 235.6cm2, cooler dead area 116 $cm^2$, charge pressure 4 bar, porosity 50%, engine speed 1200RPM-20Hz ), a COP value of 0.57 was obtained with a cold head temperature of 233K with a maximum cooling power 455W.

## 4. Conclusion

In this paper, a CFD model was developed using COMSOL Multiphysics 5.3 computational fluid dynamic software. The model was validated using published experimental work and was consequently used to study the effect of the various parameters on the cooling capability of a small-scale Stirling Cryocooler. The parametric study confirmed that a phase angle of 90° was required for best performance, but also showed that a regenerator porosity of 50% would be desirable instead of the high porosity typically used in Stirling engines. The study also showed that increasing the lengths of the heat exchangers would improve performance up to a certain length 142mm after which further increases proved to hinder the amount of heat that could be absorbed by the heat sink. This work also highlights the potential of using CFD modelling to improve the design of Stirling Cryocoolers and achieve better performance.

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