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Potential of Increasing the Performance of V-Type Alpha Stirling Engine Using 3D CFD Modelling

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Abstract With the current global levels of fossil fuel consumption and CO2 emissions, the use of Stirling engines for electrical power generation offers the potential of exploiting the vast amount of waste heat available from industrial processes, thus reducing the use of fossil fuel and carbon emissions. Stirling engines have the advantages of high thermal efficiency, flexibility of using various energy types including renewable sources. Currently, there are few Stirling engine manufacturers worldwide and limited research work for its development. Using 3D CFD modelling, this paper investigates the routes to develop V-type alpha Stirling engine from upgrading commercially available V-type compressor, in order to reduce the time and cost for engine development. Also, a new concept for the engine cooler design was modelled and integrated in the overall engine model, showing improved performance for the engine due to lower coolant temperature. The maximum indicated power from the developed engine increase from 110.98 W to 132 W by using the finned cooler.
Keywords: CFD,Cooler Design,V-type compressor, V-type Stirling Engine.

إمكانية زيادة الأداء لمحرك ستيرلينج نوع ألفا(V) باستخدام نمذجة 3D CFD

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الملخص مع المستويات العالمية الحالية لاستهلاك الوقود الأحفوري وانبعاثات ثاني أكسيد الكربون ، يوفر استخدام محركات ستيرلينغ لتوليد الطاقة الكهربائية إمكانية استغلال الكم الهائل من حرارة النفايات المتاحة من العمليات الصناعية، وبالتالي تقليل استخدام الوقود الأحفوري وانبعاثات الكربون. تتميز محركات ستيرلينغ بالكفاءة الحرارية العالية، ومرونتها في إمكانية استخدام أنواع الطاقة المختلفة بما في ذلك المصادر المتجددة. حاليا ، يوجد عدد قليل من الشركات المصنعة لمحركات ستيرلينغ في جميع أنحاء العالم وأعمال بحث محدودة لتطويرها. باستخدام النمذجة ثلاثية الأبعاد(3D CFD Modelling)، تبحث هذه الورقة في طرق تطوير محرك ستيرلينغ نوع ألفا(V)وذلك بترقية ضاغط نوع Vمتوفر تجاريا ، من اجل تقليل الوقت والتكلفة لتطوير المحرك. أيضا ، تم تصميم مفهوم جديد لتصميم مبرد المحرك ودمجه في طراز المحرك الكلي ، مما أدى إلى تحسين أداء المحرك نظرا لانخفاض درجة حرارة سائل التبريد. أمكن الحصول على زيادة في قدرة المحرك المطور من 110.98 واط إلى 132 واط باستخدام المبرد المصمم الجديد ذو الزعانف.

الكلمات المفتاحية:CFD, تصميم المبرد, محرك ستيرلينغ نوع (V), ضاغط نوع V.

Abbreviations
*CFD*Computational Fluid Dynamics
TDCTop Dead Centre
ρ Density of Fluid (Kg/m^3)
p Instantaneous Gas Pressure (Pa)
T Temperature (K)
Rs Gas Constant (J/Kg K)
V Volume (m^3)

Introduction

Stirling engines offer the potential of exploiting the vast amount of waste heat available from industrial processes and renewable thermal energy sources to generate electrical power. The majority of this waste heat sources comes from fossil fuel burning processes that produce CO2 emissions [1-2]. Stirling engine is an externally heated engine which can generate power using the temperature differential between high and low temperature heat sources. It was invented in 1816 by Robert Stirling, operating based on a closed regenerative thermodynamic cycle with repeated compression and expansion of working fluid at different temperature levels [3-4]. Computational Fluid Dynamics (CFD) modelling of the thermal performance of the Stirling engine is an effective tool to improve its design, efficiency and power output. Various work has been carried out to develop thermal models of the Stirling engine.

Several studies have been reported for using CFD modelling to investigate the performance of Stirling engines [5-13]. Mahkamov [5] modelled an axisymmetric engine geometry with k-e turbulence model to study the performance of a solar Stirling engine. Results predicted from this approach were more accurate than those from the "lower-order" methods. Mahkamov [6] developed a 3D CFD model for Biomass Stirling Engine to identify the parameters that cause reduction in power output. Salazar et al. [7] and Chen et al. [8] developed a comprehensive study on β and γ Stirling engine using an axisymmetric CFD model and 3D CFD modelling respectively to study the working cycle. Both studies help to understand the basic process of the engine cycle and predicts engine power by define the heat input. Results showed the heat transfer rates were much higher than those gained by second-order model. Costa et al. [9] conducted a 3D CFD study on different regenerator parameters under oscillating flow to simulate the conditions of Stirling engine's flow through wound-woven wire matrix regenerator. Results of this study defined the pressure drop phenomena in the regenerator and the effect of heat transfer characteristics of Stirling engine regenerators. The adoption of 3D CFD modelling was used Chen et al. [10, 11] to investigate the flow through the regenerator of a twin-power-piston γ Stirling engine. The study concludes some important information for regenerator design requirements of γ Stirling engine such as regenerator matrix material and matrices arrangement, matrix wire diameter). (Chen et al. [12] carried out a parametric study for a low-temperature-differential γ-type Stirling engine, 3D CFD modelling to determine the best design parameters that enhance the engine performance in terms of power output and efficiency.

Engine development from air compressor

Fig. 1 (a) shows a the 3D drawing for the lower part of the V type compressor used in this work, which is going to be the lower part of the developed engine. It has 88.39 cc swept volume for both pistons (50 mm Stroke and 45 mm Piston Diameter) at a phase angle of 90°. Fig. 1 (b) shows a 3D drawing for the full-developed Stirling engine, which consists of the Lower part (pistons assembly), Heater, Regenerator, Cooler and Connecting pipes.

The design was based on the annulus approach to keep the flow with high Reynolds number so that a high heat transfer coefficient can be achieved during cycle operation. The regenerator is a micro-channels matrix placed in a cylindrical housing. The heater and the cooler were placed on the top of the piston assembly of the compressor, with water jacket and heating coil then added around the cooler and heater volumes respectively. Fig. 2 shows a schematic diagram for the developed engine with its major geometrical parameters.

Developed engine CFD modelling

The geometrical parts of the developed engine were shaped as a 3D CAD model using SolidWorks and then imported to COMSOL-Multiphysics software (version 5.3).The operating parameters of the developed engine including properties of regenerator, charge pressure, cold and hot temperature values, engine speed, phase angle and other constant parameters were defined in the built-in parameters feature called (Parameter table). The table also contains the geometrical parameters which are used to calculate the dead volumes of the engine. Another built- in table called (Variable table) were filled with the engine variables such as crank angle, pistons' displacements and velocities, expansion and compression volumes.

The Regenerator was modelled using "Heat Transfer in Porous Media" module, with all parameters and properties were specified in tabular form. The different materials of all engine parts were chosen by using the built-in materials from the software; Air for the working fluid, Copper for the connecting pipes and Steel for the other parts of the engine. Following that is applying physics into the different boundaries and domains of the geometry with the boundary condition defined and time dependent equations selected.

Assuming the gas to be ideal, the state equation is,

$$\rho = p_v/(R_s T) \qquad (1)$$

Navier-Stokes, Brinkman and continuity equations were used to model the compressible laminar flow and Reynolds number considered to be less than 2000.

The working gas inside the engine is Air where its density is calculated using the state equation (See equation 1) while its other thermal properties depend on the pressures and temperatures using in the CFD simulation. These properties are predefined in COMSOL-Multiphysics using piecewise polynomials covering the temperature variation in the range of 200 to 1600 K [13]. More details about developing and validating the engine CFD model can be found in another published work by the author in [14].

[a]

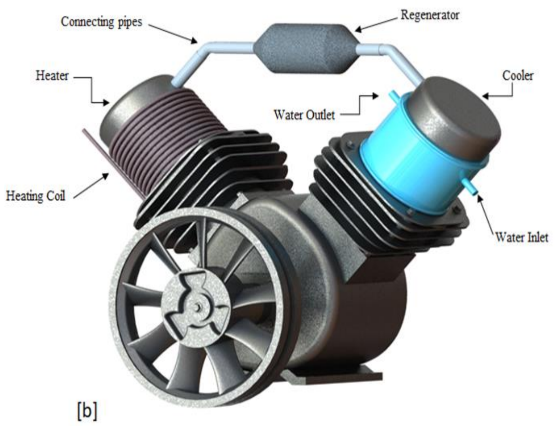


Fig. 1: [a] V-type compressor [b] Converted Stirling engine

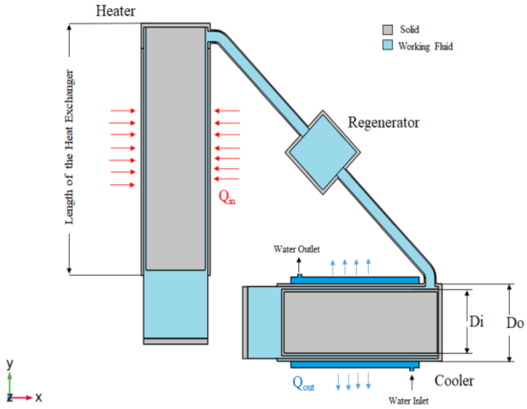
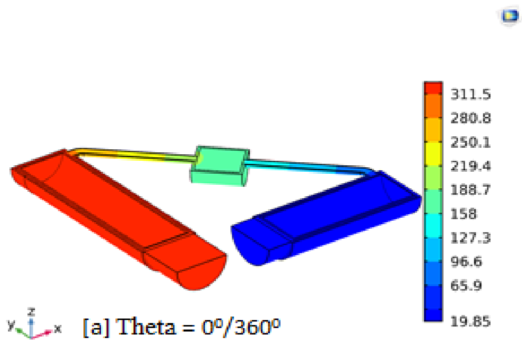
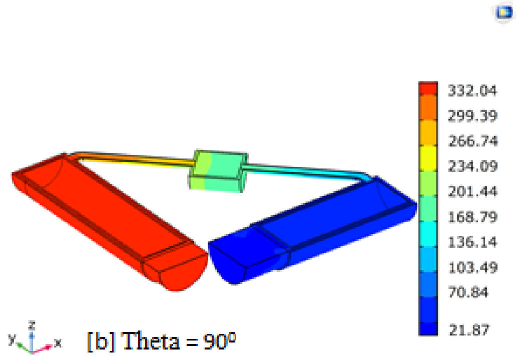


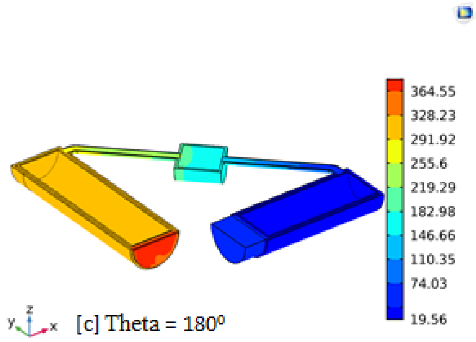
Fig. 2: Definition of engine geometrical parameters

Results and discussion

Fig. 3 (a) to (d) present the distribution of temperature inside the whole engine with different crank angle Theta 0^0, 90^0, 180^0 and 270^0 during the 5^{th} cycle. The temperature inside the cooler varies from 20 to 75 ^{0}C, with higher values near the cooler connecting pipe, as the working fluid comes from the hot space in the heater through the regenerator to the cooler. The temperature inside the heater varies from 327 to 367 ^{0}C. The working fluid moves from the cold space in the cooler via the regenerator towards the hot space. The higher temperature values appeared near the heater Top Dead Centre (TDC) position. The CFD model of the V-type alpha Stirling engine can predict a power output of 110.98 W. (See table in the Appendixshows all the dimensions for the engine parts, the engine operating conditions, regenerator properties and other constant parameters used.)







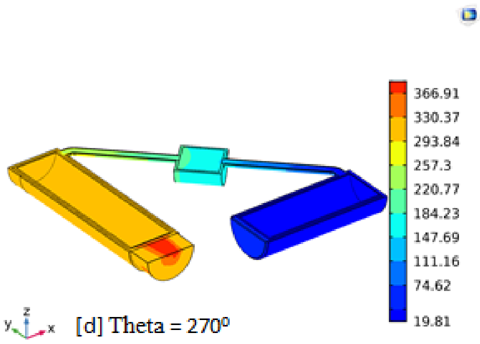
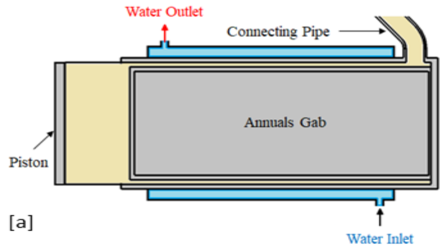
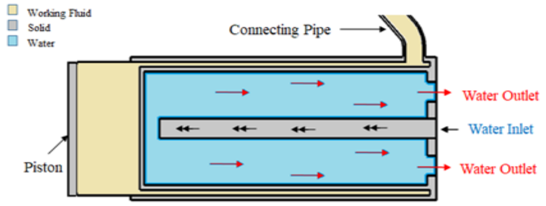


Fig. 3: [a] to [d] CFD Temperature contours at different [Theta]crank angels during 5^{th} cycle

Design improvement of the engine cooler

This section presents the CFD modelling for a new cooler design in order to improve the total engine performance. As shown inFig. 1 andFig. 2 and, a water-jacket was used to reduce the temperature of the cooler (Fig. 4(a)), but this was not very effective since it only reduced the outer surface of the cooler while the annuals gab was left at the ambient temperature. The new concept is based on inserting a pipe with certain length inside the annuals gab (Fig. 4 (b)), where water is injected as the cooling medium to replace the water-jacket cooling configuration. This will increase the heat output from the cooler as the surface area increases, where the surface area can be defined as the outer surface area of the cooler (covered by the water-jacket) plus the surface area of the annuals gap. (See Fig. 4 (a) and (b)). This design was chosen based on the use of the annular gap in the centre of the cooler. It is also advisable to insert the tube into the centre of the annulus in order to obtain symmetric and equal vortices along the entire circumference of the annulus gap.





[b]

Fig. 4: Cooler of Stirling engine cooling system[a]Water-jacket system and [b]New cooler configuration

A CFD model was developed to evaluate this new cooler configuration and to determine the best geometrical parameters including the inner pipe length and diameter, in addition to added holes and fins to the pipe to increase the turbulences inside the cooler gab as shown in Fig. 5 and Fig. 6.

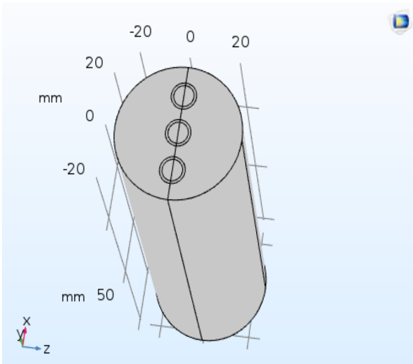


Fig. 5: Three dimensional view of the finned-

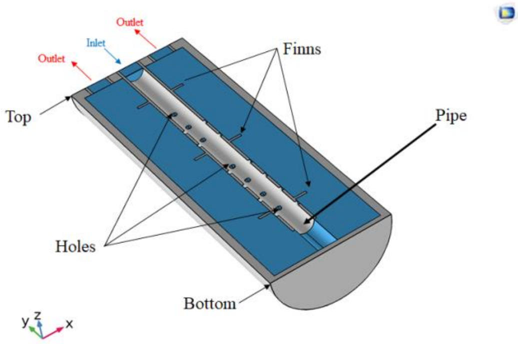
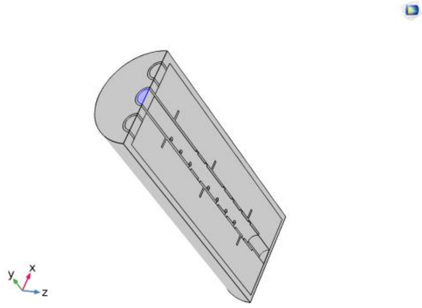
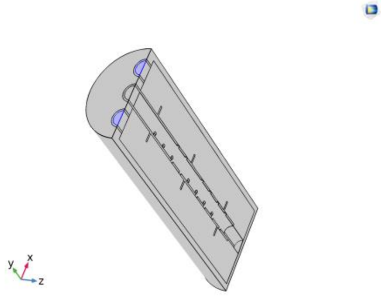


Fig. 6: Symmetrical view of the finned-cooler cooler

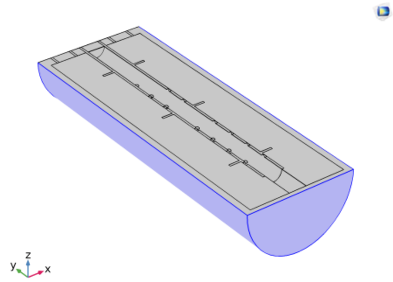
The boundaries of all faces that used in the CFD model of the finned-cooler can be seenFig. 7 and their values are located in Table 1.



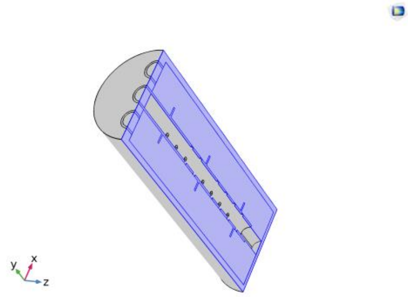
[a] Inlet mass flow rate and temperature



[b] Out flow



[c] Heat flux



[d] Symmetry

Fig. 7: Boundary conditions of the finned-cooler

Table 1:Value of the boundary conditions

Boundary	Value
ass flow rate	(0.01283/2) kg/s
Inlet temperature	(20) ^{0}C
Heat flux	(11836) W/m^2

Fig. 8 shows the temperature distribution along the cooler height (from top to the bottom) for five pipe lengths of 65, 108, 115, 120 and 124 mm, with inlet and outlet diameters of 10mm. With pipe lengths of 65 and 108 mm higher temperatures (around 50 ^{0}C) and non-homogenise temperature fluctuations were experienced, like temperature = 50 ^{0}C at the height of 40 mm while the temperature = 27 ^{0}C and 32 ^{0}C at 15 mm and 110 mm respectively. The pipe length of 120 mm produced smooth temperature fluctuations around 32 ^{0}C along the cooler's walls leading to cooler water and almost equal heat flux along the surface of the cooler.

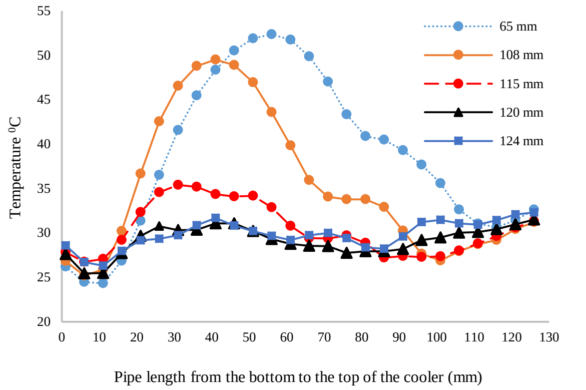


Fig. 8: Axial temperature for the finned cooler with different lengths

The CFD model for the pipe length of 120 mm was used to evaluate the best inlet and outlet diameters in terms of enhancing the cooler design. Fig. 9 shows the temperature distribution for inlet and outlet diameter of 6 and 8 mm, with the latter generating less temperature fluctuations and average temperature around 29.17 ^{0}C.

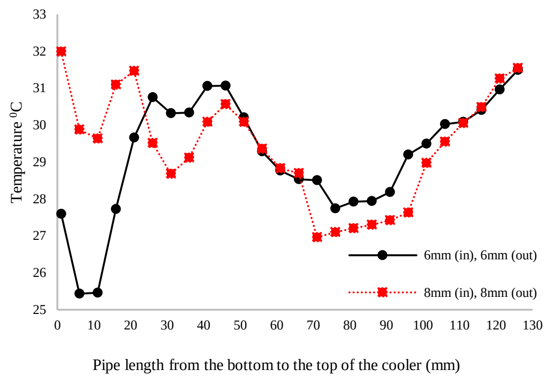
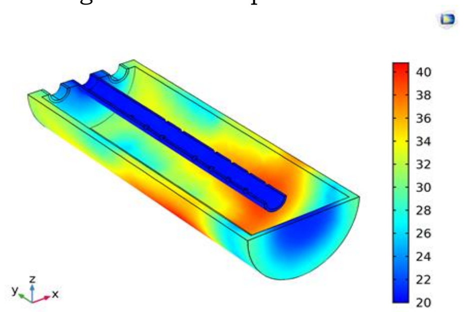
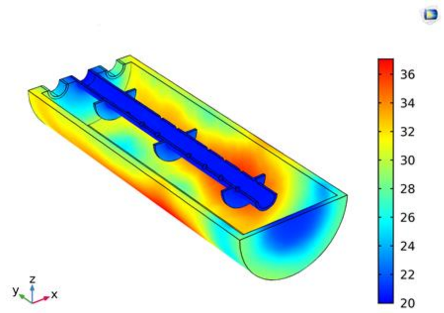


Fig. 9: Axial temperature for the finned cooler with different diameters

Fig. 10 (a) and (b) illustrate the temperature distribution. The effects of adding fines on the velocity and temperature distribution is clear where more streamlines are generated in the finned pipes leading to reduced average surface temperature.



[a] Non Finned



[b] Finned

Fig. 10:[a] and [b] Temperature distribution (^{0}C)

The CFD model of the V-type alpha Stirling engine predicted that by using finned cooler with a pipe length of 120 mm and diameter of 8mm the power output can increase from 110.98 W to 132 W. Fig. 11 shows the new developed Stirling engine with the finned cooler.

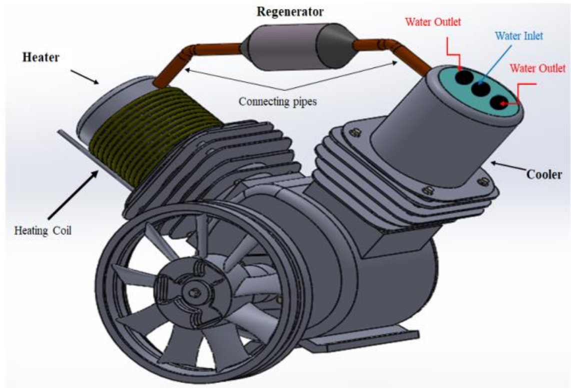


Fig. 11:V-type alpha Stirling engine with Finned-cooler

Conclusion

This work examined the methodof upgrading commercially available V-type compressor to develop V-type alpha Stirling engine to highlight the potential of reducing the cost and time for engine development. A detailed 3D CFD model was adopted in this study to simulate the whole engine.Moreover, aninnovativeperceptionfor the engine cooler design (called finned cooler) was modelled and combined in the overall engine model, showing enhanced performance for the engine due to lower coolant temperature. The resultsfrom the proposed engine show the maximum indicated power rise from 110.98 W to 132 W by using the new cooler.

Appendix

Table 2:Stirling engine specifications and operating conditions

Parameter	Value
Engine type	Alpha α (V)
Cylinder bore	54.0 mm
Cylinder Stroke	45.0 mm
Swept volume	88.39 cm3
Phase angle	90 ^{0}
Working gas	Air
Type of the regenerator's matrix	Micro-Channels
Channel diameter	0.50 mm
Regenerator diameter	40.0 mm
Regenerator length	40.0 mm

Diameter of the connecting pipe	5.00 mm
Length of the connecting pipe	220 mm
Heater/Cooler outer diameter	54.0 mm
Heater/Cooler inner diameter	45.0 mm
Length of the heater	186 mm
Length of the cooler	136 mm
Type of the heater	Heating coil
Type of the cooler	water jacket
Cooling	Water
Engine speed	500 rpm
Charge pressure	2.5 (bar)
Working fluid	Air
Porosity	0.3414%
Inlet water temperature	20 (°C)
Hot source temperature	327 (°C)

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