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Thermodynamic Analysis of a Gamma-configuration Stirling Engine

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ABSTRACT

Stirling engine is seen as a viable alternative to the conventional power generator due to its special capability to work with externally supplied heat sources which make the renewable and waste energy can be applied directly for a green energy production. Researchers from institutions and industries around the world have put their efforts in developing Stirling engine from numerical analysis to prototype development. In this paper, a simple thermodynamic analysis has been done on a proposed slider-crank gamma-configuration Stirling engine. The purpose of this analysis is to investigate the engine working capability of the proposed design prior to its prototype development. The analysis is done based on modified Schmidt ideal adiabatic model. The expansion and compression space temperature were set as 573K and 300K, respectively and air was used as the working fluid. With the assumed speed of 300rpm and running at atmospheric pressure, from the analysis, the proposed engine is predicted to produce 33.41W of power output. Additional to that, the selection of 90° phase angle also has been verified and the relationship between the temperature difference and produced power output also has been investigated. It is found that with 400K increment of the temperature difference between expansion and compression space, the produced power output can be increased by 94.93%. From the analysis, it showed that the proposed design can be proceeded for the prototype development.

1. Introduction

Today, with the growing awareness of the shortage of fossil fuel resources, researchers tend to focus their research on developing one of the most effective energy conversion systems, the Stirling engine [1]. Stirling engine is an engine that operates on a closed regenerative thermodynamic cycle, in which the working fluid inside the engine undergoes cyclical expansion and compression process at hot and cold ends, respectively at different temperatures [2]. One of the special characteristics of this engine is, it is an externally heated engine that uses any external heat sources in converting heat energy such as solar energy, biomass and geothermal into useful mechanical energy [3-7]. This special characteristic of the Stirling engine has made it a potential ‘future green engine’ [8]. Other than that, Stirling engine also possesses other advantages like low noise production, high efficiency, less air pollutant, easy maintenance and low vibration [9-11]. Due to its promising alternative solution in

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producing clean energy, efforts have been done worldwide in developing Stirling engine for the future application including numerical analysis and prototype development [12-21].

Typically, kinematic Stirling engines are categorised into three main configurations namely alpha, beta and gamma. All of them have similar thermodynamic cycle but only differ in their mechanical characteristics [22]. In alpha Stirling engine, the hot and cold power pistons are arranged separately in expansion and compression spaces, respectively and connected through a regenerator or connecting pipe. Unlike other Stirling engines, alpha type Stirling engine does not have a displacer piston. This property allows the alpha engine to generate power from both hot and cold pistons thus making it advantageous than the others. However, the pistons have to be properly sealed to hold the working fluid [23,24]. For beta engine, the power piston and displacer are arranged concentrically in a similar cylinder. This arrangement gives the engine a minimum dead volume thus a higher efficiency and power can be produced. However, this engine commonly adopts a rhombic drive mechanism which makes the design more complex than other engine types. For gamma-configured Stirling engine, like the beta type engine, this engine also consists of a power piston and a displacer, but they are arranged in separate cylinders, connected by a connecting pipe. This engine commonly uses a simple slider crank drive as its driving mechanism, making it the simplest Stirling engine to design. Researchers have observed that gamma-configured Stirling engines potentially have the highest mechanical efficiency among the other Stirling engines [11]. Adding to that, researchers also found that only gamma-type Stirling engines are suitable for low temperature difference applications which may include low concentration ratio solar power, low temperature geothermal energy and waste energy recovery [25]. According to Thombare and Verma [10], the development of Stirling engine that works with low temperature difference has the potential to be more attractive in the future since the application is wider. Therefore, researchers have focused more on developing gamma-configured Stirling engine, including the authors.

With limited number of prototype development of Stirling engines, the authors had decided to develop a prototype model of gamma-configuration Stirling engine with the specific geometrical measurements for the use of low temperature differential application. However, prior to the prototype development, a numerical analysis needs to be done in predicting and verifying the working condition of the particular proposed engine design. Therefore, in this paper, a simple thermodynamic analysis has been conducted on a proposed design of a slider-crank drive gamma-configuration Stirling engine. The engine reciprocating displacement and volumetric variations are determined. Their effect on the working fluid mass distribution, engine cycle pressure and power output produced are presented and discussed. Other than that, the selection of 90° phase angle has been verified for the proposed design.

2. Methodology

The thermodynamic analysis is done by adopting a modified model of Schmidt theory done by Berchowitz and Urieli [26]. This model is based on first-order analysis and it has been widely used among researchers since the process is fast and the results obtain are within 20% of deviation of its accuracy [22]. Comparing this model to other models, like the third-order analysis proposed by Finkelstein and fourth-order analysis namely computational fluid dynamic (CFD), these models are more complex, time consuming and expensive to be applied. Thus, the first-order analysis is more preferable in estimating the Stirling engine performance at their initial stage of the engine development. The analysis model was based on an ideal adiabatic model in which the engine was split into three different components which are expansion space (hot-end space), regenerator and compression space (cold-end space).

The assumptions made in applying Schmidt analysis are listed as follows [21,27,28]:

i. The working fluid obeys ideal gas law, PV=mRT.
ii. The engine is perfectly sealed and there is no working fluid leakage. Thus, the total mass of working fluid is constant.
iii. Temperature in each working space (expansion, regenerator and compression space) is known and there are no temperature gradients in each working space.
iv. The engine is running at a constant speed.
v. Uniform instantaneous pressure in the working spaces (expansion, regenerator and compression spaces).
vi. Displacer and power pistons are moving in a pure sinusoidal motion.
vii. The mechanical and heat losses of the engine are neglected.

The proposed design of slider-crank drive gamma configuration Stirling engine is illustrated as in Figure 1 and the geometrical variables of the engine is defined as in Table 1.

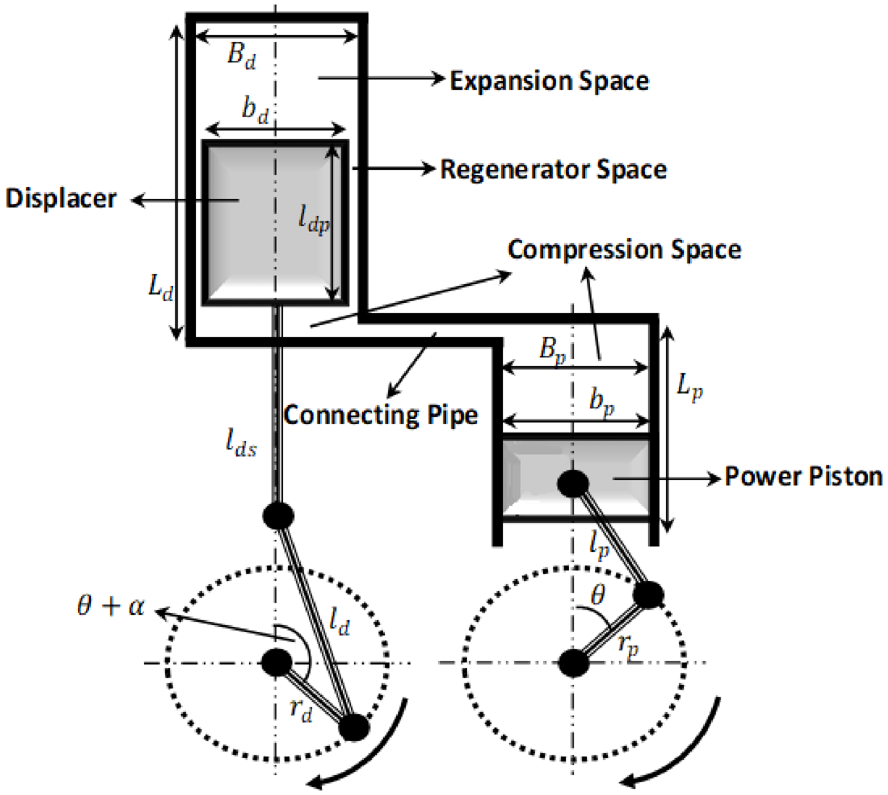

Fig. 1. Illustration of a gamma-configuration Stirling engine

By referring to Figure 1, it can be seen that the displacer is moving with a phase angle, α ahead the power piston. The analysis is done with the initial condition of power piston at the bottom dead centre (BDC) of the power piston cylinder and it is presumed that the engine is running at a constant speed of 300rpm with reference to the typical minimum speed of kinematic Stirling engine [18]. It is also set that the engine is running at atmospheric pressure with air as the working fluid and the expansion and compression temperature is 573K and 300K, respectively. Figure 2 below shows the overall process of the thermodynamic analysis of the proposed gamma-configuration Stirling engine.

Table 1
Geometrical variables of the gamma-configuration Stirling engine

Components	Label	Dimensions
Displacer cylinder bore	B_d	85 mm
Displacer cylinder length	L_d	177 mm
Displacer piston bore	b_d	84 mm
Displacer piston length	l_{dp}	115.5 mm
Displacer shaft	l_{ds}	89 mm
Regenerator length	L_r	115.5 mm
Connecting pipe length	L_{cp}	113 mm
Connecting pipe surface area	A_{cp}	272.27 mm^2
Power piston cylinder bore	B_p	54.5 mm
Power piston cylinder length	L_p	61.1mm
Power piston bore	b_p	54 mm
Connecting rod length	l_d=l_p=l	100 mm
Crank offset radius	r_d=r_p=r	30.5 mm
Crank angle	θ	0° $\leq$ θ $\leq$360°
Phase angle	α	90°

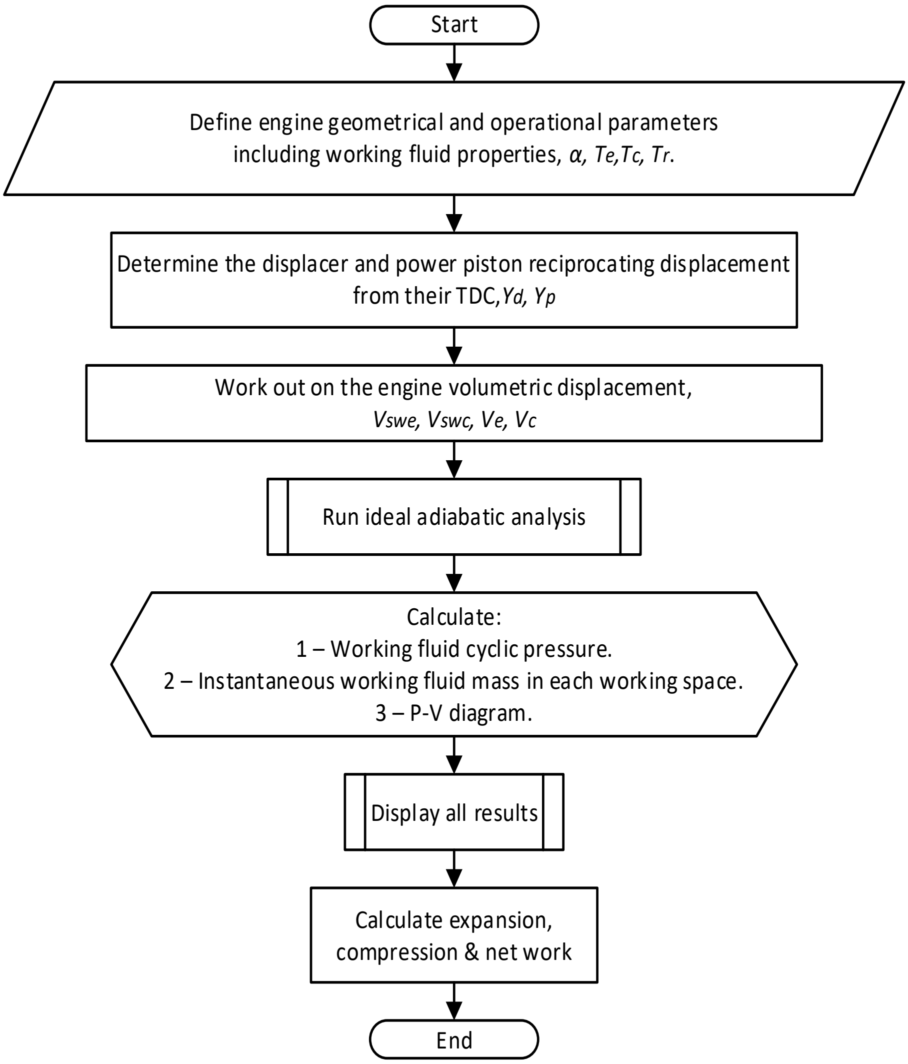

Fig. 2. Flowchart of the analysis process

2.1 Reciprocating Displacement from Top Dead Centre (TDC)

The reciprocating displacement of displacer, Y_d and power piston, Y_p can be calculated by using Eq. (1) and Eq. (2), respectively. Both Eq. (1) and Eq. (2) were derived based on the engine geometry and trigonometry rules.

$$Y_d = (r_d + l_d) - [r_d \cos(\theta + \alpha + \pi)] + [({l_d}^2) - (r_d \sin(\theta + \alpha + \pi))^2]^{1/2} \quad (1)$$

$$Y_p = (r_p + l_p) - [r_p \cos(\theta + \pi)] + [({l_p}^2) - (r_p \sin(\theta + \pi))^2]^{1/2} \quad (2)$$

where θ is defined as the crank angle measured from the engine vertical direction and rotating in clockwise direction. Meanwhile, α is the phase angle difference between displacer and power piston.

2.2 Swept Volume

The swept volume for displacer, V_{swe} and power piston, V_{swc} can be determined in terms of Y_d and Y_p and their respective cross-sectional area of displacer and power piston. The expressions of the swept volume are as follow,

$$V_{swe} = \frac{\pi {b_d}^2}{4}(Y_d) \quad (3)$$

$$V_{swc} = \frac{\pi {b_p}^2}{4}(Y_p) \quad (4)$$

2.3 Expansion and Compression Space Volume

The total expansion and compression space of the Stirling engine can be calculated by summing together the working space swept volume with their clearance or dead volume. In expansion space, the clearance volume is defined as the dead volume on top its TDC and the volume parallel to the top regenerator space. However, for compression space, the clearance volume is including the bottom space of the displacer, connecting pipe volume and dead space on top of piston TDC. The equations are as follow [28],

$$V_e = V_{swe} + V_{cle} \quad (5)$$

$$V_c = V_{swc} + V_{clc} \quad (6)$$

2.4 Ideal Gas Equation of State

By applying the ideal gas equation of state, the initial total maximum working fluid mass, M can be calculated by taking the initial condition of pressure, volume and temperature [29,30].

$$M = (P_{atm} V_{total,max})/(RT_i) \quad (7)$$

As stated in the assumption above, the total mass of the working fluid is constant, which gives the expression as [28],

$$M = m_e + m_r + m_c \tag{8}$$

From the ideal gas equation of state, the change in mass due to the change in pressure of expansion, regenerator and compression space are found as [28],

$$m_x = (PV_x)/(RT_x) \textit{ where } x = e, r, c \textit{ (i.e. expansion, regenerator and compression)} \tag{9}$$

Substituting Eq. (9) into Eq. (8) gives [29,30],

$$M = (P/R)\left(\frac{V_e}{T_e} + \frac{V_r}{T_r} + \frac{V_c}{T_c}\right) \text{ where } T_r = (T_e - T_c)/\ln\left(\frac{T_e}{T_c}\right) \tag{10}$$

The regenerator temperature, T_r is assumed to be constant throughout the cycle with the value of $T_r = 421.88$ K. Since it is assumed that the pressure to be uniform throughout the working spaces, rearranging Eq. (10) for P gives,

$$P = M.R/\left(\frac{V_e}{T_e} + \frac{V_r}{T_r} + \frac{V_c}{T_c}\right) \tag{11}$$

2.5 Instantaneous Mass of Working Fluid

As stated in Eq. (9), the equation of instantaneous working fluid mass in each working space can be written as [28],

$$m_e = (P.V_e)/(R.T_e) \tag{12}$$

$$m_r = (P.V_r)/(R.T_r) \tag{13}$$

$$m_c = (P.V_c)/(R.T_c) \tag{14}$$

2.6 Cyclic Energy Flow

Since the analysis is conducted using ideal adiabatic model and only considering three working spaces (expansion, regenerator and compression space), the energy evaluation will only consider the total indicated work of the engine. Therefore [22,31],

$$W = W_e + W_c \tag{15}$$

$$dW = dW_e + dW_c \text{ where,} \tag{16}$$

$$dW_e = P.dV_e \quad \text{and} \quad dW_c = P.dV_c \tag{17}$$

The thermodynamic analysis model used in this study has been validated with previous study by Parlak *et al.,* [21] by comparing the patterns of the plotted data for the gamma-configured Stirling engine.

3. Results and Discussions

Based on given engine geometrical variables as in Table 1 and by using Eq. (1) and Eq. (2), the reciprocating displacement of displacer and power piston from their TDC position can be plotted as shown in Figure 3. The initial positioning of the power piston is set to be at bottom dead centre (BDC) of the cylinder, gives the maximum displacement of 0.061m which is equivalent to its stroke length. The maximum displacement from TDC of the displacer also shares the same value with power piston since both of them are having the same value of crank offset radius and connecting rod length, which are the only components that determined the stroke length. During the engine cycle, it can be seen that the displacer is leading the power piston with 90° phase angle. This is to allow the displacer to displace the working fluid traversing back and forth between the expansion and compression spaces during the engine running. The selection of phase angle also is very important in determining the volume variation in the engine cycle that at the end, affects the engine performance.

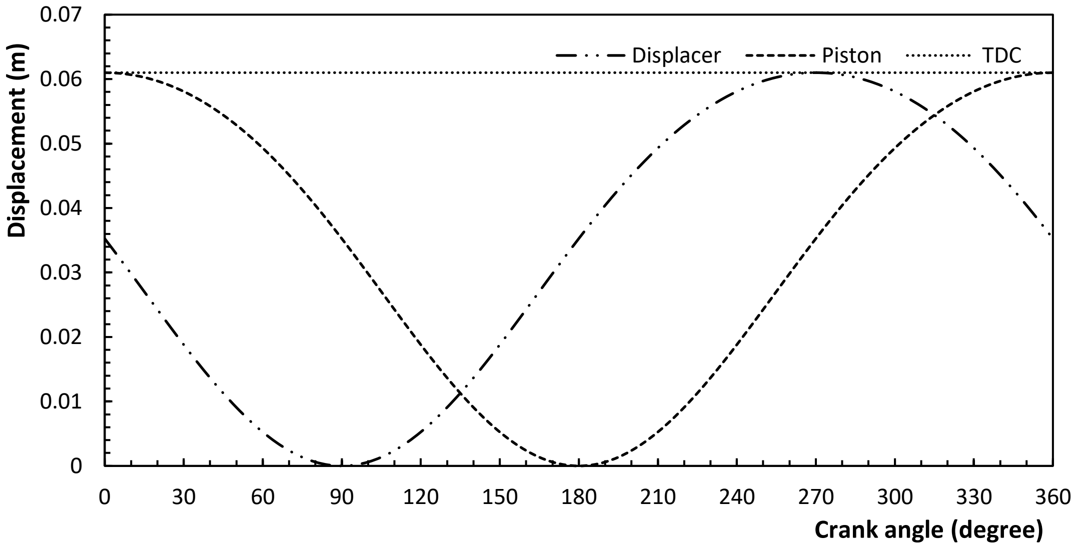

Fig. 3. Displacer and power piston reciprocating displacement from TDC

Figure 4 represents the engine volumetric displacements of expansion and compression space of gamma configuration Stirling engine and its cyclic pressure. As the power piston and displacer moving upwards toward their TDC, the volume in compression space is increased while the volume in expansion space is decreased. The increased volume of compression space is due to the increasing clearance volume at the bottom of the displacer is more rapid than the reduction volume in the power piston cylinder. After a short while, the compression space reaches its maximum volume at 75° crank angle while the expansion space reaches its minimum volume at 90° crank angle. The alternate increasing and decreasing between both working spaces is ongoing until the compression space reaches its minimum volume at 235° crank angle and the expansion space reach its maximum volume at 270° crank angle. The total maximum working volume is seen to be at the beginning and the end of the cycle, while the minimum total working volume is in the middle of the engine cycle which is at 180° crank angle.

By referring to Figure 4 again, the working fluid cyclic pressure is also plotted on the same graph in determining its relation towards the working spaces volume variation. Based on the set expansion space's temperature of 573K and compression space's temperature of 300K, the cyclic pressure is varying from 109.16 kPa to 199.28 kPa throughout the cycle. The pressure is uniform across the working spaces as per stated in assumption at the beginning of the analysis. As plotted in Figure 4, the cyclic pressure and the working space total volume is closely related to each other. As can be seen, the maximum pressure is achieved when the total engine volume is at its minimum value and vice versa. Therefore, the total engine volume variation, which is derived from the reciprocating displacement of the displacer and power piston plays an important role in determining the engine cycle pressure.

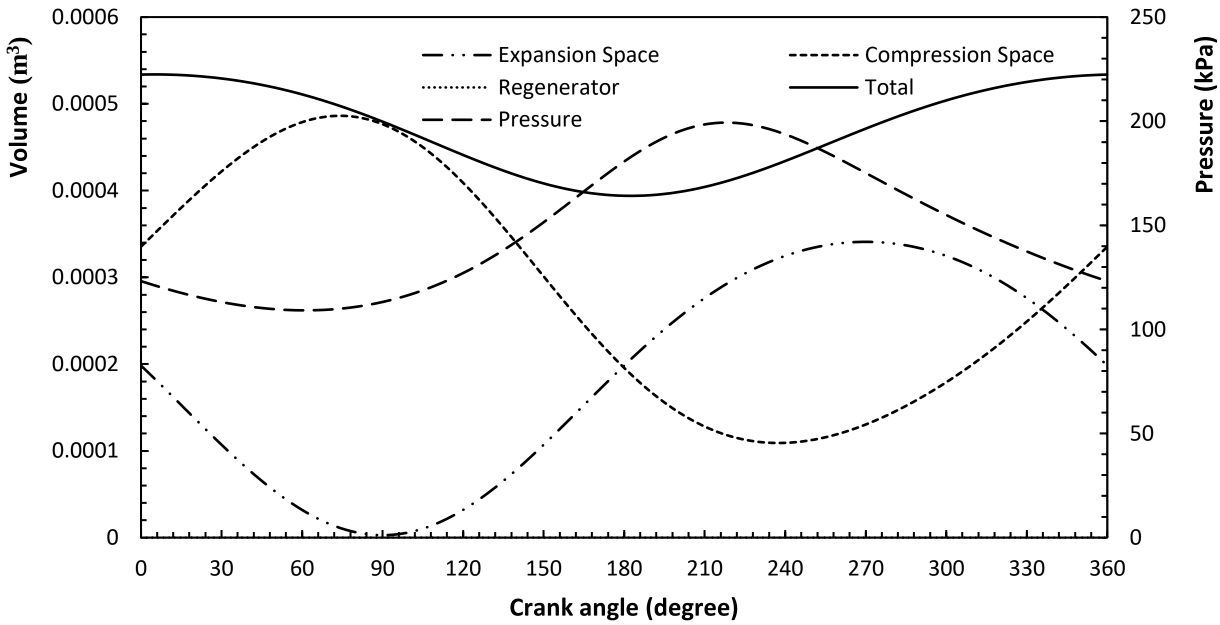

Fig. 4. Engine volumetric displacements in each component and its cyclic pressure

The mass of the working fluid as a function of the crank angle is depicted in the Figure 5. The total amount of mass is remained constant since the assumption made is that the engine is perfectly sealed and there is no fluid leakage. Based on this figure, it can be seen that the mass curve of working fluid in compression space almost mirrors the expansion curve. This indicated that the working fluid alternately enters and leaves compression and expansion spaces response to the reciprocating movement of the displacer and power piston. The compression space exhibits a maximum amount of working fluid in early cycle. This means that the engine is experiencing a compression process during this phase. The same situation also occurs in expansion space. The maximum working fluid in expansion space is at the half-end of the cycle, which means that the expansion process takes place at the half end of the cycle.

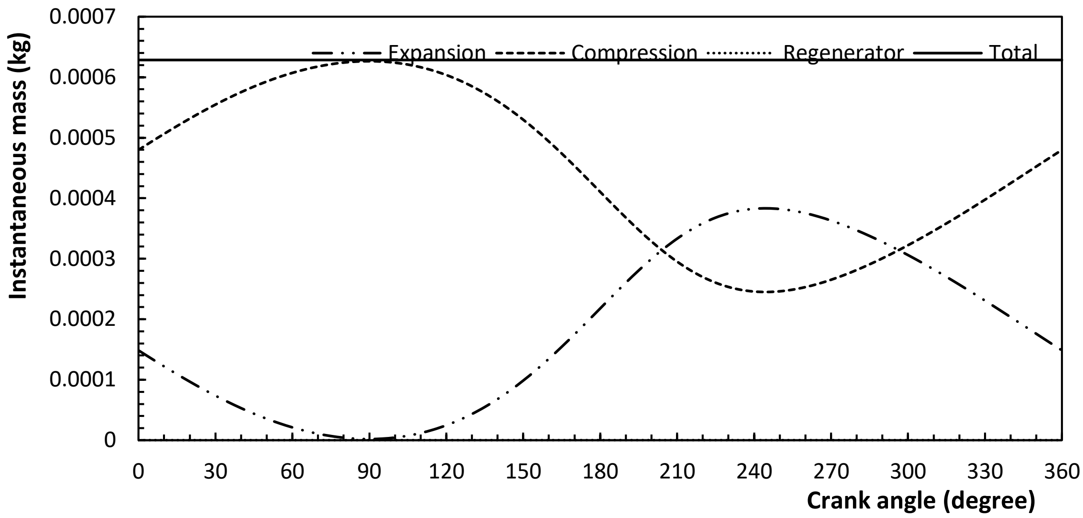

Fig. 5. Instantaneous working fluid mass in each working space

Meanwhile, the P-V diagram for the expansion and compression space of the proposed gamma-configuration Stirling engine is presented in Figure 6. The area enclosed by the expansion curve and compression curve illustrated the positive work done and negative work done produced, respectively. However, the area enclosed by the total P-V diagram represents the net work done of the system. With the assumed speed of the engine is 300rpm and the net work done produced, it is calculated that the net indicated power produced is 33.41W.

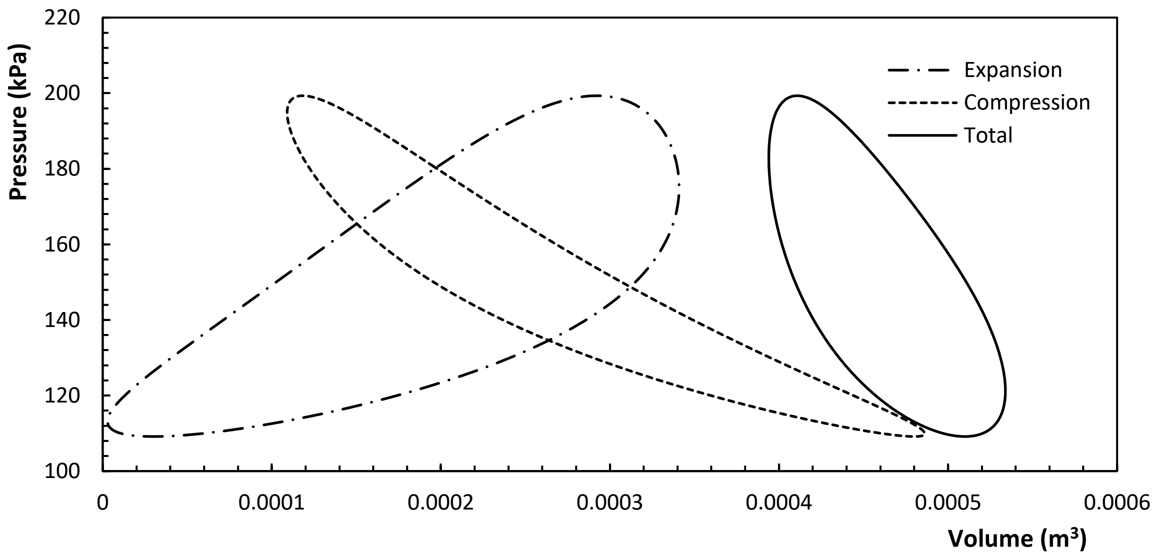

Fig. 6. P-V diagram for proposed gamma-configuration Stirling engine

It is mentioned earlier that the variation of phase angle was acknowledged as one of the methods in controlling Stirling engine performance since it can influence the total volumetric change in the cylinder, pressure amplitude and heat transfer of the engine [32]. Generally, a 90° phase angle is applied in an engine design because it gives a better engine performance. However, this is not necessary an optimum value because it will depend on the engine actual design itself and the gas flow path. It can be either less or more than 90° [33]. Alfarawi *et al.,* [34] in their Gamma Stirling engine studies found that, the power output increased as the phase angle increased up until an optimum value, then it falls down. They showed that their optimum angle phase value was 105° rather than 90°. Therefore, for this analysis, range of phase angles, from 70° to 120° are selected in determining the most effective phase angle. The P-V diagrams of the results of different phase angle are plotted as shown in Figure 7. It can be seen that different phase angle does affect directly the total volumetric change in the cylinder and the pressure amplitude. As the phase angle increased, the pressure amplitude also increased. By referring to Figure 8, with the same engine speed of 300rpm, the highest power output produced by the engine is when the phase angle at 90° with 33.41W. The power outputs produced by 70°, 80°, 100°, 110°, and 120° are 31.42W, 32.93W, 32.83W, 31.17W and 28.47W, respectively. These results showed an agreement with the statement stated before, which the phase angle of an engine would have its optimum value. The optimum power output produced is likely a result of balance between the engine pressure drops and the increase in heat transfer rate caused by the increased in volumetric gas exchange, compression and pressure ratios [34]. Therefore, for this proposed engine design, the selection of 90° phase angle is a right decision.

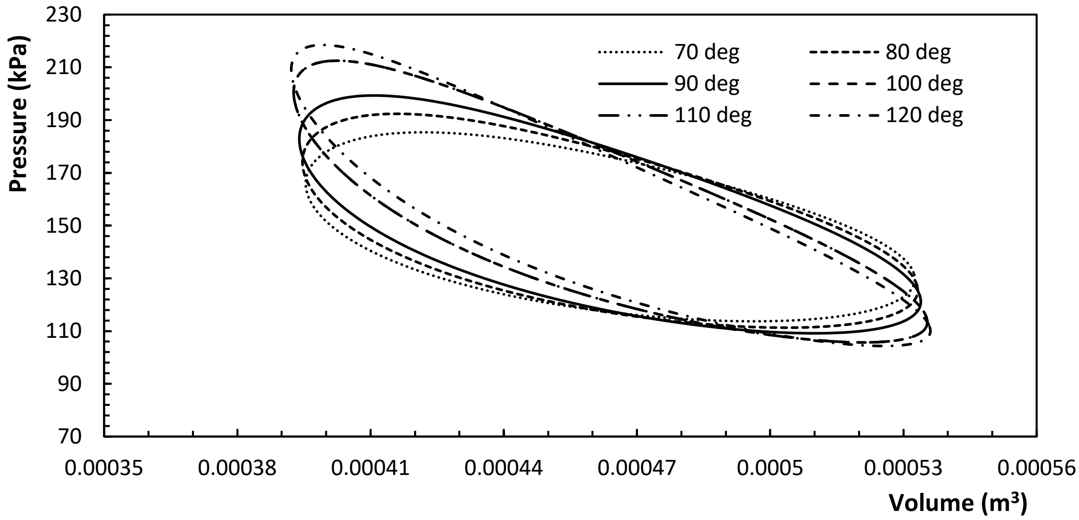

Fig. 7. P-V diagram for various phase angle, α setting

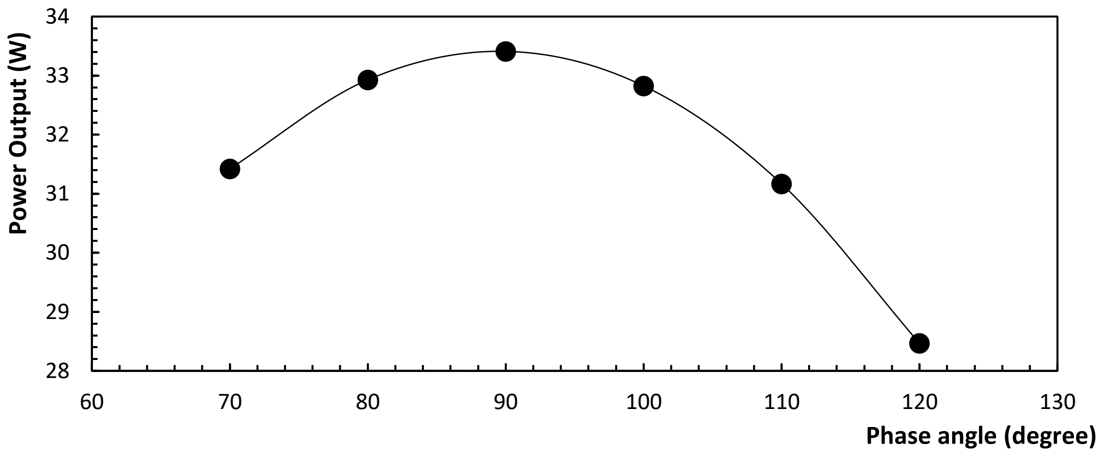

Fig. 8. Produced output power at different engine phase angle

Other than that, the relationship between working space temperature difference and power output is also investigated. In this analysis, the expansion space temperature is varying from 573K to 1073K while the compression space temperature is kept constant. Therefore, the increased expansion space temperature will increase the temperature difference between hot and cold working space. As illustrated in Figure 9, an increased in expansion space temperature resulting in directly increased the power output. This is because, with a constant volume and mass of the engine, an increased in expansion space temperature will increase the pressure in it. Therefore, it will shift up the upper line of the P-V diagram plotted, which will cover a bigger area under the curve, indicating that an increased in the total net work done, so do the power output. With the same running engine speed at 300rpm, by increasing the expansion temperature to 973K, the power output produced increased by 94.93% from the previous state, which was 573K. Therefore, increasing the temperature difference between expansion and compression working space would be an option in enhancing the engine power output production.

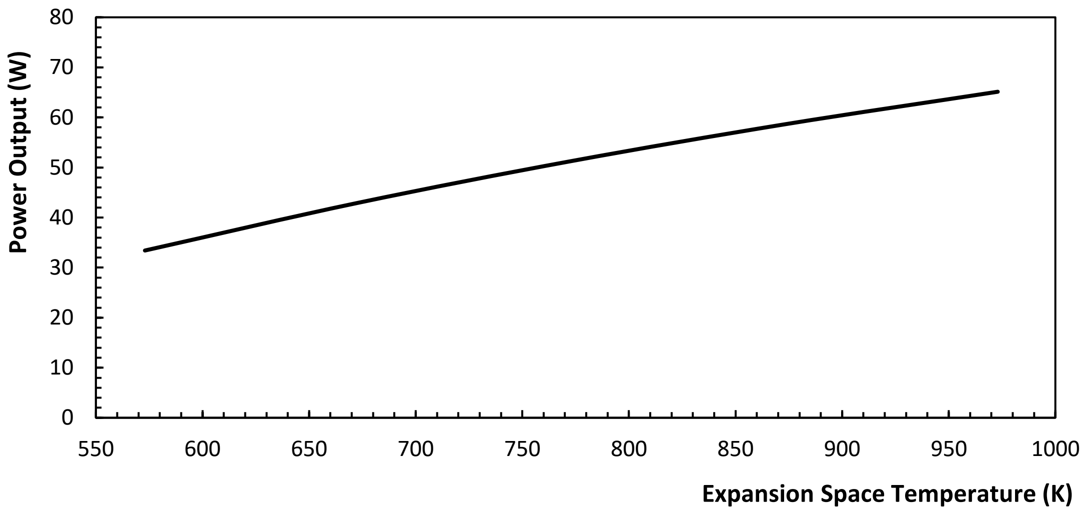

Fig. 9. Variation of power output against expansion space temperature

4. Conclusions

In this work, a simple thermodynamic analysis has been conducted on a proposed slider-crank gamma-configuration Stirling engine. By adopting a modified Schmidt ideal adiabatic model, the engine is predicted to produce 33.41W of power with the air as the working fluid, 573K and 300K of expansion and compression space temperatures, respectively, run at atmospheric pressure and at 300rpm of speed. The selection of 90° effective phase angle is also done in this work due to the direct influence of phase angle setting towards engine power output production. The performance of the engine can be enhanced by increasing the temperature difference between expansion and compression spaces. The study showed that the proposed design can be carried out for the development of the Stirling engine prototype.

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