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A Novel Close Loop Analysis of Gamma Prototype Stirling Engine †

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Abstract: Air pollution is greatly influenced by the emissions generated by automotive engines, making it a pressing concern. To address this issue, a considerable amount of research is currently devoted to recovering waste heat from these engines. A gamma-type Stirling engine has been meticulously chosen to achieve this specific objective. This study elucidates a new isothermal method that effectively analyses Stirling engines. A set of differential equations is proficiently solved by employing the powerful MATLAB software. Remarkably, the simulation results obtained from this computational approach closely align with the experimental data, indicating the accuracy and reliability of the methodology. Furthermore, this research delves into the feasibility of employing the Stirling engine as a Combined Cooling, Heating and Power (CCHP) system, shedding light on its potential applications in various domains.

Keywords: Stirling engine; thermodynamics; power; efficiency

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1. Introduction

In the pursuit of preserving fossil fuels and mitigating greenhouse effects, there is a growing interest in exploring alternative energy sources. Among these alternatives, renewable energy resources (such as biomass, solar, geothermal, and wind energy) are considered highly promising due to their clean, efficient, and sustainable nature [1].

An example of an externally heated engine is the Stirling engine, which boasts various advantageous characteristics. It is thermally regenerative, exhibits a straightforward design, operates with minimal noise, ensures safety during its use, and can easily adapt to a wide range of heat sources, including solar, biomass, geothermal energy, and even industrial waste [2,3].

In an ideal scenario, Stirling engines operate on an exceptionally efficient thermodynamic cycle. The engine's internal gas undergoes four distinct processes: two isothermal heat-exchange processes (expansion and compression) and two isochoric heat-exchange processes (heating and cooling). However, the actual cycle experiences significant drawbacks attributed to the irreversibility and non-ideality of transport mechanisms within the engine's various components. These factors negatively impact the engine's overall performance [4,5].

The regenerator holds significant importance within the engine as a crucial component. Functioning as an internal heat exchanger, it operates like a thermal sponge, absorbing and releasing heat throughout the cycle, consequently bolstering the engine's power and efficiency. Remarkably, the heat absorbed and restored to the gas in the regenerator during one cycle is typically four times greater than the heat that passes through the heater in the same cycle [6]. Schmidt [7] originally introduced the first-order analysis, presenting a closed-form analytical method that relies on mass and energy conservation algebraic equations. This approach predicts engine power and efficiency when sinusoidal volume variation occurs. Schmidt's theory [7] operates under the assumption of isothermal working spaces for compression and expansion. However, it is essential to note that this analysis is highly idealized, as it does not account for heat transfer and internal pumping losses.

In this research paper, a new isothermal model is formulated, and results are compared with classical isothermal, adiabatic and experimental results. The new isothermal model is created using the assumption that expansion and compression spaces are isothermal, and the regenerator is divided into two parts, the one attached to heater as R1 and the cooler as R2. The advantage of this model is that it is easy to investigate the temperature of the regenerator area and implement and solve it.

2. Engine Data

The engine under study is a gamma-type double-piston Stirling engine. In this engine, the exhaust gas of the diesel engine is used as the heat source. To heat the heater tubes of the Stirling engine, especially the cap is made for the Stirling engine. Further details of Engine data is given below.

Table 1. Engine data of gamma type double piston Stirling engine [1].

Engine Data	Values	Engine Data	Values
Cooler temperature	294	Pressure (bar)	3.58
Heater temperature	424	Rotational speed (rpm)	882
Cooler void volume	223×10^{-6}	Phase angle (degree)	88°
Heater void volume	87.28×10^{-6}	Regenerator void volume (m^3)	308.93
Expansion swept volume (m^3)	221×10^{-6}	Compression swept volume (m^3)	194×10^{-6}
Expansion clearance volume (m^3)	24×10^{-6}	Compression clearance volume (m^3)	35×10^{-6}

3. Isothermal Method

For the development of the isothermal model following are the assumption made.

- The assumption is that the compression and expansion process occurs under isothermal conditions. This means that the working spaces experience no temperature change during these processes, and it is presumed that heat exchangers operate with perfect effectiveness as a consequence of this isothermal assumption.
- The engine maintains a constant speed, and motion is sinusoidal.
- The temperature profile of the regenerator is linear and constant.
- There are no fluid losses and dissipation effects over the cycle.
- The regenerator part is divided into two parts, i.e., the first is connected to the heater (R_1) and the second is connected to the cooler part (R_2). This is the new assumption made in this isothermal method. For the formulation of the isothermal method above assumptions are considered.

The equation for the expansion and compression space volume calculation is given below. Where as V_{DE}, V_{DC}, V_{SE}, V_{SC}, $\varnothing$ are the expansion dead volume, compression dead volume, expansion swept volume, compression swept volume and phase angle respectively.

$$V_E = \frac{V_{SE}}{2}\left[1-\cos\left(\alpha\right)\right]+V_{DE} \tag{1}$$

$$V_C = \frac{V_{SE}}{2}\left[1+\cos(\alpha)\right]+\frac{V_{SC}}{2}\left[1-\cos(\alpha-\phi)\right]+V_{DC} \quad (2)$$

The total mass of the engine is given by

$$M = \sum m_i = m_{R_1} + m_{R_2} + m_E + m_C \quad (3)$$

$$M = \frac{P}{R}\left[\frac{V_{R_1}}{T_{R_1}}+\frac{V_{R_2}}{T_{R_2}}+\frac{V_E}{T_E}+\frac{V_C}{T_C}\right] \quad (4)$$

The temperature ratio, swept volume ratio, expansion and compression dead volume ratio is given by [8]

$$t = \frac{T_C}{T_E} \quad \text{Temperature ratio} \quad (5)$$

$$v = \frac{V_{SC}}{V_{SE}} \quad \text{Swept volume ratio} \quad (6)$$

$$X_{DE} = \frac{V_{DE}}{V_{SE}} \quad \text{Expansion space dead volume ratio} \quad (7)$$

$$X_{DC} = \frac{V_{DC}}{V_{SE}} \quad \text{Compression space dead volume ration} \quad (8)$$

Here, regenerator dead volume ratio is assumed to be

$$X_{R_1} = \frac{0.5V_R}{V_{SE}} \quad \text{Regenerator dead volume ratio (connected to heater part)} \quad (9)$$

$$X_{R_2} = \frac{0.5V_R}{V_{SE}} \quad \text{Regenerator dead volume ratio (connected to cooler part)} \quad (10)$$

The regenerator temperature of each part is given by the following equations [8].

$$T_{R_1} = \frac{T_E+3T_C}{2} \quad \text{Regenerator temperature (connected to heater part)} \quad (11)$$

$$T_{R_2} = \frac{3T_E+T_C}{2} \quad \text{Regenerator temperature (connected to cooler part)} \quad (12)$$

Now put the expansion, compression space volume, regenerators temperature, cooler and heater temperature in Equation (4), and we will get the following Equation (13).

$$m = \frac{P}{R}\left[\frac{V_{SE}}{2T_E}-\frac{V_{SE}}{2T_E}\cos(\alpha)+\frac{V_{DE}}{T_E}+\frac{4V_{R_1}}{T_E+3T_C}+\frac{4V_{R_2}}{T_C+3T_E}+\frac{V_{SE}}{2T_C}+\frac{V_{SE}}{2T_C}\cos(\alpha)+\frac{V_{SC}}{2T_C}-\frac{V_{SC}}{2T_C}\cos(\alpha-\phi)+\frac{V_{DC}}{T_C}\right] \quad (13)$$

Now arrange the above equation using Equations (5)–(12), and the mass equation is given by

$$m = \frac{PV_{SE}}{2RT_C}\left[B - A\cos(\alpha-\delta)\right] \quad (14)$$

Whereas A, B and δ is given by

$$A = \sqrt{t^2 - 2t + 1 + 2(t-1)v\cos(\phi) + v^2} \tag{15}$$

$$\delta = \arctan\left(\frac{V\sin(\phi)}{t - 1 + V\cos(\phi)}\right) \tag{16}$$

$$B = t + 2X_{DE}t + \frac{16X_{R_1}t}{1+3t} + \frac{16X_{R_2}t}{t+3} + 1 + V + 2X_{DC} \tag{17}$$

If c = A/B, then from Equation (14), the equation for the mass in the Stirling engine is given by following equation

$$P = \frac{2mRT_C}{V_{SE}B\left[1 - c\cos(\alpha - \delta)\right]} \tag{18}$$

$$W_C = \int P.dV_C \tag{19}$$

$$W_E = \int P.dV_E \tag{20}$$

4. Results and Discussion

4.1. Model Validation

For the model validation of the new isothermal model, the PV diagram is drawn below, and the PV curve is compared with the experimental, adiabatic, isothermal and ideal cycle. The curve of the new isothermal model is within the range of the experimental curve, as shown in Figure.

Figure 1a represents the pV-diagram of the new isothermal model and other thermodynamic models. The dotted line depicts the experimental curve, and the dotted-dash curve represents the new isothermal model curve. The new isothermal model curve is superimposed on the experimental curve. The advantage of this new model is estimating regenerator temperature on the hot and cold sides. The temperature to the cooler side of the regenerator is estimated at 326 K, and the temperature to the heater side is 391 K. The work calculated by the new isothermal model is 9.80 J, and the power is 145.78 W. The thermal efficiency is 26.90%. The experimental determined thermal efficiency is 24.7%, closely related to the new thermodynamic model efficiency [1]. The new isothermal model calculates a −2.8% error, which is very close to the agreement. Figure 1b represents the derivative of work done through the expansion, compression and total work done. It is important to note that the Stirling engine under consideration is gamma type double piston. The results are presented in Figure 1b as a single cylinder to understand. In Figure 1b, the heat flow through the expansion space is 18.40 J and rejected through the compression space is −13.45 J.

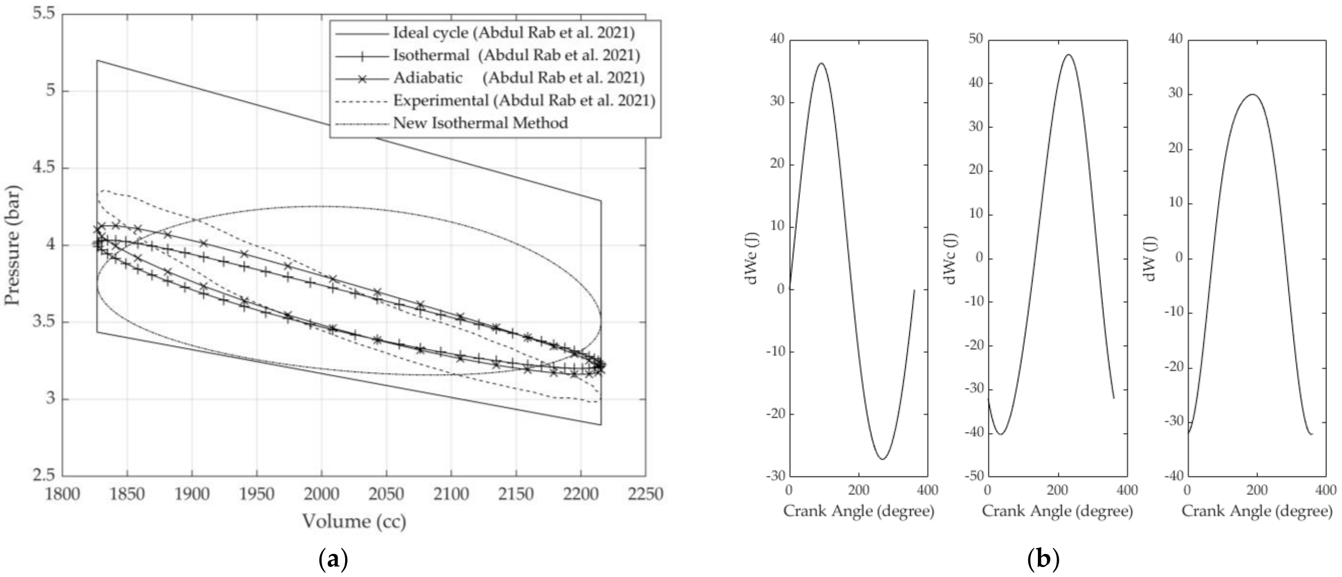

Figure 1. (**a**) Comparison of pV-diagram of new isothermal model and other models (**b**) derivative of work done.

Table 2. Summary of thermodynamic models.

Parameters	Isothermal [1]	Adiabatic [1]	Experimental [1]	New Isothermal
Work (J)	9.08	9.70	7.50	9.80
Power (W)	133.8	143.75	111.43	145.78
Efficiency (%)	30.70	30.90	24.70	26.90

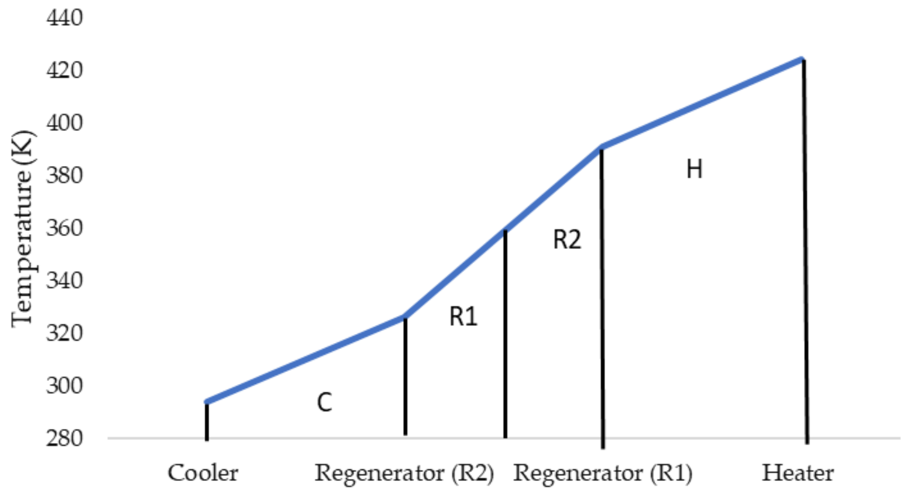

Figure 2. Temperature diagram of the heat exchanger of Stirling engine.

Cryogenic fluids, such as liquid air/nitrogen, have been recognized as efficient energy storage mediums, boasting a high storage density of 0.77 (MJ/kg). They can be produced by liquefying air/N2 using surplus electricity from off-peak periods or renewable energy sources. The well-established cryogenic industry infrastructure facilitates the storage and transportation of these fluids. In this research work, the feasibility of using cryogenic fluid as coolant is used, such as liquid nitrogen. The temperature range of liquid nitrogen is −10 to −150 °C. If we used −140 °C as liquid nitrogen as a coolant, the maximum power is achieved as 300 W by using this new isothermal model. This power can be utilized as combined cooling, heating, and power system (CCHP) on a microscale for one room. So, the Stirling engine is also one of the best options for alternate energy production for the CCHP system.

5. Conclusions

In this research, a new isothermal model is made using the concept of working space experiencing no temperature change. Moreover, the regenerator is further divided into two parts; it is easy to understand the regenerator for the performance of the Stirling engine. The new isothermal model's results are very close to other thermodynamic models. The power and efficiency investigated by the new isothermal model are 145.78 W and 26.90%, respectively. The experimental power and efficiency are 111.43 W and 24.70%, respectively [1]. The temperature of the regenerator connected to the heater part is 391 K, and that to connected to the cooler part is 321 K.

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