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DESIGN AND MANUFACTURE OF BETA STIRLING ENGINE

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ABSTRACT: This paper explains the design and fabrication of a beta- type stirling engine. Stirling engines are external combustion engines which run on Stirling cycle. This property of the engine enable it to use heat sources including solar, agricultural residue like paddy straw, sugarcane leaves, wheat stalk, groundnut shell, coconut husk, etc and geothermal, or even industrial waste heat. These engines can be used for stationary power generation applications as the efficiency is comparable with theoretical Carnot efficiency. They can be used for domestic power generation in rural areas where the energy requirement is not attained by conventional energy sources. This application of Stirling engine can match the rural energy demand with the availability of bio energy and energy resources. A 200 W beta type Stirling engine with rhombic drive mechanism is designed and fabricated for fulfilling the rural house hold energy requirements.

KEYWORDS: Stirling engine, Stirling Cycle, Rural Energy

I. INTRODUCTION

Stirling Engine was invented and patented by a Scottish engineer named Sir Robert Stirling in the year 1816. Stirling engine is theoretically one of the most efficient thermodynamic engines for heat to work conversion. They are external combustion engines that can use any form of heat source such as exhaust gas from I.C. Engines, Boilers, Solar thermal energy, biomass combustion to name a few. The operating temperature of the stirling engine is between 650°C to 800°C, which makes it ideal for waste heat recovery and solar thermal applications.[1].

II. STIRLING CYCLE ANALYSIS

Stirling engine is one of the high efficiency thermodynamic engine which has efficiency close to Carnot efficiency. The engine analysis can be performed after considering various assumptions like reversible process of all cycles, no change in temperature inside engine, perfect gas as working fluid, reversible cyclic process, no change in mass of working fluid, etc[2]. Stirling engines are heat engines working on theoretical Stirling cycle which undergoes isothermal compression and expansion process and constant volume heating and cooling process [1].

The figure 1 shows stirling cycle with isothermal and constant volume process. The working fluid is compressed isothermally in the process 1. In the constant volume process 2 heat is added. Expansion takes place isothermally in the process 3 and constant volume heat rejection takes place in the process 4.[3]

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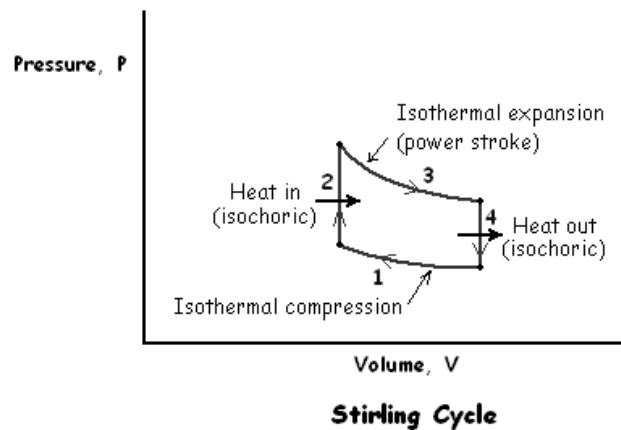


Figure 1: PV Diagram of Stirling Cycle

The ideal thermodynamic stirling cycle has the following advantages over theoretical Carnot cycle.

- 1.The efficiency of an ideal stirling cycle with regeneration is equal to Carnot efficiency ,which has maximum possible theoretical efficiency of all heat engines operating between same temperature limits.
- 2.The two constant volume process of stirling engines increase the work output with reasonable increase in pressure and volume unlike Carnot cycle where pressure and volume increase are very high [4]
3. Of all heat engines operating between same pressure, temperature and volume ratios stirling engine has high mechanical efficiency [5]

In actual stirling engines the mass of gas in the dead volume is significant which undergoes isothermal compression and expansion process .This resulted in minimum loss of work per cycle compared with theoretical results [3]

III.STIRLING ENGINE CONFIGURATION

Stirling engine configuration can be of three types namely Alpha, Beta and Gamma [4].Of these Alpha configuration is simple with two pistons housed in two separate cylinders of inline, parallel or V configuration. Beta configuration has single cylinder for displacer and power piston with relatively complex arrangement. Gamma configuration has separate piston and displacer in different cylinders with simple arrangement [6]

IV BETA STIRLING ENGINE WITH RHOMBIC MECHANISM

Beta Stirling engine has one displacer and one power piston arranged in a single cylinder. The Beta configuration is complex to design and manufacture.[6] The designed Beta Stirling engine uses rhombic drive mechanism for converting reciprocating motion of piston and displacer into rotary motion of the shaft. The rhombic drive was invented by Meijer of Philips for the application in Stirling engine in the year 1959.[7] This mechanism has advantage over crank mechanism with minimum force on the piston cylinder side wall. This mechanism also has the ability of using flexible seal between piston and cylinder wall.[8].Beta configuration was compact and small dead volume compared with other configurations.

V.WORKING PRINCIPLE

Rhombic drive mechanism uses power piston and one displacer piston which are housed in a single cylinder.These two are linked by a separate piston yoke and displacer yoke. These yokes also connects the piston and displacer connecting rods with the crank mechanism. These crank also connects with two meshed gears which rotates in opposite direction. The entire mechanism is symmetric to vertical axis and provides smooth motion of power piston and displacer piston within the cylinder and provides minimum side thrust. [9].The unique feature of this mechanism is the change in

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volume of compression space above the piston and expansion space above the displacer which pushes the working fluid between the hot and cold region of the Stirling engine.

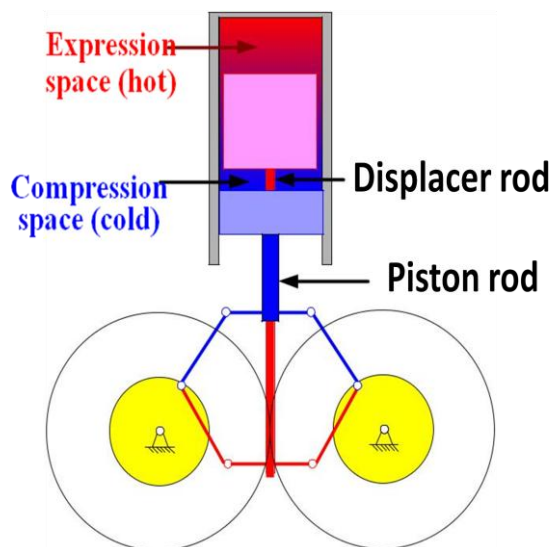


Figure 2: Rhombic mechanism

VLDESIGN ANALYSIS OF STRILING ENGINE

In this paper stirling engine analysis is done using Schmidt analysis. Schmidt analysis is an ideal thermodynamic analysis used to predict the performance of Stirling engines of alpha, beta and gamma configuration. This analysis is carried out for performance prediction in the design stage to know the possibility of working of the intended engine design. It takes into account the sinusoidal volume variations of both compression and expansion cylinder space and thus provides close approximation of actual performance compared to ideal Stirling cycle[2].

Schmidt analysis is based on the following assumptions

1. Gas mass in the engine cylinder is constant.
2. Engine temperature in all parts remains unchanged.
3. Perfect gas equation is applied to the working fluid.
4. Constant pressure is maintained throughout the gas in a cycle.
5. Crank motion is sinusoidal.

The following parameters are required for Schmidt analysis[10]

1. Mean operating pressure
2. Power piston swept volume
3. Displacement piston swept volume
4. Clearance volume
5. Hot side Temperature
6. Cold side Temperature
7. Phase angle

Piston Displacer engine(Beta type engine)

Nomenclature

V_l =Volume hot live (cm³)

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V_p =Volume cold live for power piston (cm^3)

V_k =Volume cold live for displacer (cm^3)

$2(RC)$ = Stroke of displacer (cm)

DB = Diameter of displacer (cm)

DD = Diameter of displacer rod (cm)

$2(R2)$ = Stroke of power piston (cm)

DC=Diameter inside engine cylinder(cm)

HD=Volume hot dead(cm^3)

RD=Volume dead regenerator(cm^3)

CD= Volume cold dead(cm^3)

T_h =Effective hot gas temperature(K)

T_r =Effective regenerator gas temperature(K)

T_c =Effective cold gas temperature(K)

M=Engine gas inventory(g mol)

R=Universal gas constant (8.314 J/g mol.K)

P(N)=Common gas pressure at particular point in cycle (MPa)

F=Angle of crank (degrees)

The maximum hot live volume (V_l)

$$V_l = 2 (RC)(DB)^2 \frac{\pi}{4} \quad (1)$$

The maximum cold live volume(associated with displacer)(V_k)

$$V_k = 2 (RC) [(DB)^2 - (DD)^2] \frac{\pi}{4} \quad (2)$$

The maximum cold live volume(associated with power piston)(V_p)

$$V_p = 2 (R2) [(DC)^2 - (DD)^2] \frac{\pi}{4} \quad (3)$$

For any angle F,the array of hot volume $H(N)$

$$H(N) = \frac{V_l}{2}[1-\cos(F)]+HD \quad (4)$$

For any angle F,the array of cold volume $C(N)$

$$C(N) = \frac{V_k}{2}[1+\cos(F)] + CD + \frac{V_p}{2}[1-\cos(-F-AL)] \quad (5)$$

Volume CD includes the dead volume in the cooler as well as the dead volume between the strokes of the displacer and the power piston. In the Beta type engines the strokes of displacer and the power piston should overlap so that they almost touch at one point in the cycle. This over lap volume is subtracted from the dead volume in the cold heat exchanger.[11]

For a beta type engine with this type of stroke over lap and $AL=90^\circ$ and $V_p=V_k$ then

$$CD=VM - \frac{V_p}{2} (2 - \sqrt{2}) \quad (6)$$

Where VM is cold dead volume in heat exchanger and clearance and ducts

Total gas volume at any crank angle is

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$$V(N) = H(N) + C(N) + RD \tag{7}$$

The Pressure at any crank angle is given by the perfect gas law

$$PV = MRT \tag{8}$$

$$P = \frac{MRT}{V}$$

$$P = \frac{MR}{\bar{T}}$$

There fore for the beta Stirling engine the pressure at any crank angle is given by

$$P(N) = \frac{M(R)}{\frac{H(N)}{T_{\square}} + \frac{C(N)}{T_c} + \frac{RD}{T_r}} \tag{9}$$

J.R.Sneft presents a Schmidt equation for finding the energy generated per cycle. In this analysis it is assumed that the temperature of dead space gas has the arithmetic mean between the hot and cold gas spaces in a beta type engine where the displacer and power piston essentially touching at one point during the cycle.[11]

$$W1 = \frac{\pi(1-AU)PX(V_l)XY \sin(AL)}{Y + \sqrt{(Y^2 - X^2)}} \tag{10}$$

$$X = \sqrt{[(AU-1)^2 + 2(AU-1)(XY) \cos(AL) + (XY)^2]} \tag{11}$$

$$Y = \frac{AU + 4(XX)(AU)}{(1+AU)} + Z \tag{12}$$

$$Z = \sqrt{(1+(XY)^2 - 2(XY) \cos(AL))} \tag{13}$$

$$AU = \frac{T_c}{T_{\square}} \tag{14}$$

$$XX = \frac{RD + HD + CD}{V_l} \tag{15}$$

$$V_l = V_k \tag{16}$$

$$XY = \frac{V_p}{V_l} \tag{17}$$

W = Work per cycle(Joule)

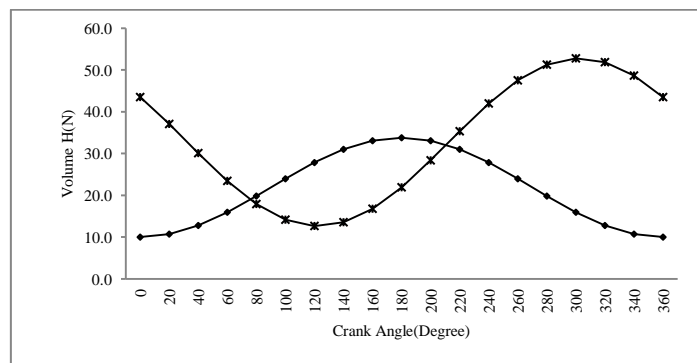


Figure 7: Displacer and Piston cylinder volume change with crank angle

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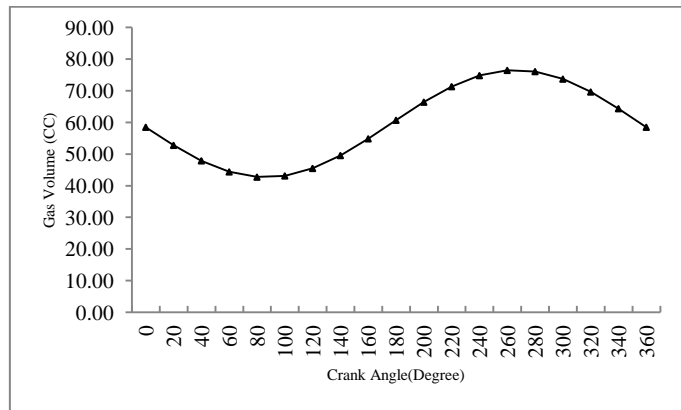


Figure 7: Working fluid volume change with respect to Crank angle

The figure 6 indicates the volume changes of power cylinder and displacer cylinder compression and expansion volume. As indicated the expansion volume above the displacer cylinder undergoes expansion initially. At the same time the compression volume above the power piston undergoes compression. But gradually the working fluid pressure starts to move the power piston and work is done on the power piston. Figure 7 indicates the working fluid volume changes with crank as the motion of the piston and displacer are sinusoidal. This ensures the change in volume is considered for analysis of the Stirling cycle and make the result as close as the actual output from the engine.

VII. NOMENCLATURE AND MANUFACTURING

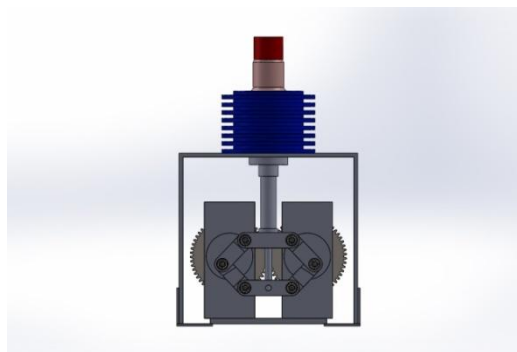


Figure 3: Front view

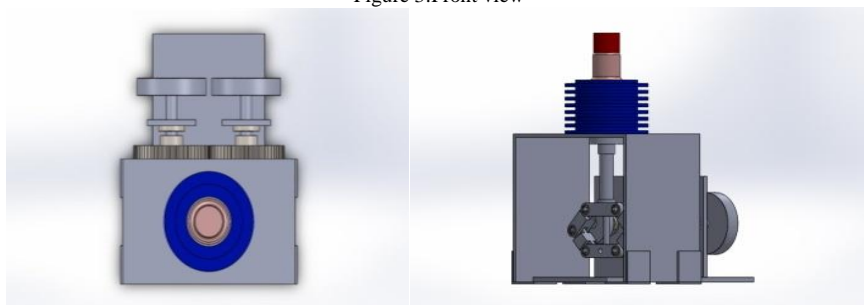


Figure 4: Top view

Figure 5: Isometric view

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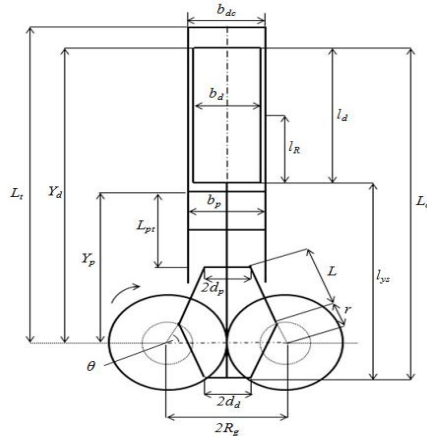


Figure 6.Engine Nomenclature

Table 1: Dimension and Nomenclature of Rhombic drive

Components Label	Nomenclature	Dimension (mm)
Total length	L_t	283
Displacer cylinder bore	b_{dc}	37.5
Displacer piston length	l_d	65
Displacer piston bore	b_d	31.20
Regenerator length	l_R	32.5
Power piston length	L_{pt}	150
Power piston cylinder bore	b_p	31.25
Displacer piston yoke shaft	l_{ys}	250
Power piston yoke	$2d_p$	60
Displacer piston yoke5	$2d_d$	60
Connecting rod length	$L_p=L_d=L$	42.5
Crank offset radius	r	13.75
Spur gear pitch diameter (PCD) 130	$2R_g$	100
Displacer yoke distance	L_{dt}	315
Piston displacement	Y_p	170.66
Displacer displacement	Y_d	274.20
Displacer displacement	Y_d	287.15 (for $\theta=90$) & 278.94 (for $\theta=0$)
Piston displacement	Y_p	205.34 (for $\theta=90$) & 186.05 (for $\theta=0$)

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Table 3. Fabricated engine parts and materials

S.No	Parts	Qty	Material
1	Displacer	1	Stainless steel
2	Displacer cylinder	1	Mild steel
3	Piston	1	Aluminium
4	Power cylinder	1	Brass
5	Cooler	1	Aluminium
6	Piston connector	1	Brass
7	Piston stem	1	Aluminium
8	Displacer rod	1	Mild steel
9	Displacer yoke	1	Aluminium
10	Piston yoke	1	Aluminium
11	Pins	2	Aluminium
12	Gears	2	Mild steel
13	Flywheel	2	Mild steel



Figure 7. Power Cylinder with fins



Figure 8. Displacer Piston



Figure 9. Displacer Cylinder

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Figure 10.Links



Figure 11.Axis and bearings

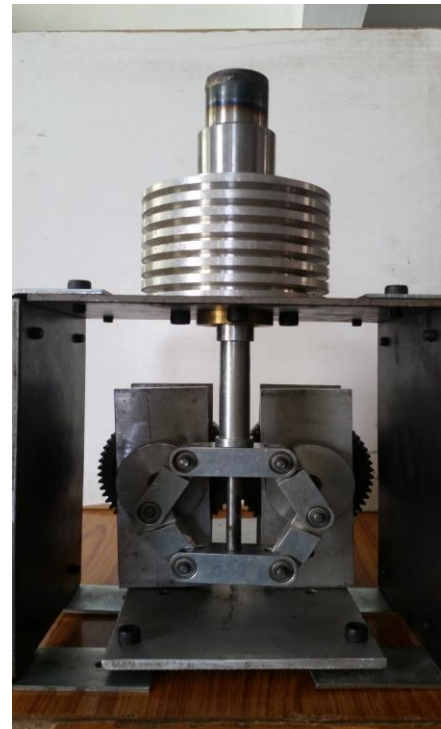


Figure 12.Assembled Engine

VIII.CONCLUSION

In this work the design and manufacture of Beta type Stirling engine with rhombic drive was analysed. The design of rhombic drive in a beta Stirling engine is limited due to the drive complexity and tight tolerances. One of the main challenge in the design is the proper assumption of many unknown parameters which proved to be a challenge. The accuracy of results and difficulty of design calculation depends on number of assumed unknowns. The mechanism proved to be smooth and efficient when running and superior in tackling the side thrust associated with the conventional crank and shaft mechanism.

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