CONCEPTUAL AND BASIC DESIGN OF A STIRLING ENGINE PROTOTYPE FOR ELECTRICAL POWER GENERATION USING SOLAR ENERGY

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ABSTRACT

The research consisted in a conceptual and basic design of a prototype Stirling engine with the purpose of taking advantage of the solar radiation to produce electric energy. The work began with a bibliography review covering aspects as history, basic functioning, design configurations, applications and analysis methods, just to continue with the conceptual design, where the prototype specifications were determined. Finally, a basic dimensioning of the important components as heat exchangers (heater, cooler, and regenerator), piston, displacer and solar collector was elaborated. The principal conclusions were that the different analysis methods had dissimilitude among their results; in this sense, a construction of the prototype is necessary for the understanding of the complex phenomena occurring inside the engine.

INTRODUCTION

As a safer alternative to the steam machines of XIX century, Robert Stirling invented the “Stirling Engine”. The problem was that the steam boilers tend to explode due to the high temperatures and pressures in addition to the deficiency in the metallurgy at the time. The Stirling engines are capable of reaching good efficiencies, unfortunately it is necessary very specific materials to put up with the extreme conditions and also very precise manufacturing is needed, which increases the overall cost of the machines.

The growing energetic crisis and the global warning claim for ecological solutions and even more efficient machines than those we use in today´s modern industry. In this sense, if the Stirling engine is used to generate electrical power from the solar radiation, it results in an environmental clean and efficient choice.

NOMENCLATURE

Variables:

Pow	Power
Q	Heat
S	Stroke
T	Temperature
V	Volume
W	work
f	Frequency
k	Gas thermal conductivity
m	Mass
p	Pressure

Subscripts

C	Compression
E	Expansion
DC	Displacement compression
DE	Displacement expansion
DP	Displacement piston
MC	Dead compression
ME	Dead expansion

Greek symbols

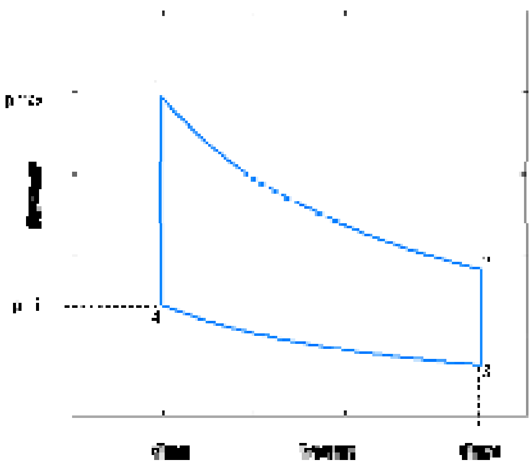
α	Crankshaft turn angle
β	Lag angle between displacer and piston in crankshaft turn angle
ø	Diameter
η	Efficiency

STIRLING CYCLE

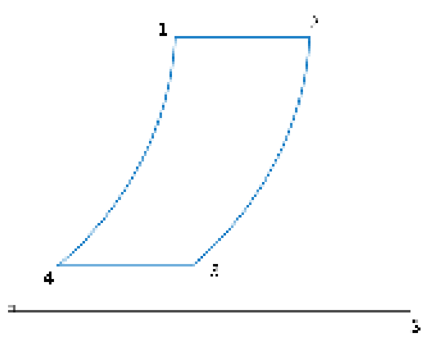
Ideally, the Stirling cycle, have four fundamental processes, considering an ideal gas with constant caloric coefficients. (See fig. 1).

- Isothermal Expansion (1-2): the enclosed gas is heated; meanwhile it increases its volume. Effective work is done during this stroke.

- Isochoric Displacement (2-3): after the expansion, the gas is moved from the heat zone to the cold zone, removing heat from it.
- Isothermal Compression (3-4): in this stage the gas is then compressed at a lower temperature constant during this process.
- Isochoric Displacement (4-1): finally, the gas is then moved again to the heat zone where the cycle starts again, taking the heat up from the previously isochoric displacement on its way back.



(a)



(b)

Figure 1. (a) p-v Diagram of the Stirling cycle. (b) T-S Diagram of the Stirling cycle.

The net work done by the gas through all the cycle is:

$$W = -mRLn\left(\frac{V_2}{V_1}\right)(T_E - T_C) \qquad (1)$$

And the total heat supplied is:

$$Q_E = mRT_E Ln\left(\frac{V_2}{V_1}\right) \qquad (2)$$

(Note: the deduction of the equation 1 and 2 is beyond the present work scope)

STIRLING CYCLE THEORIC EFFICIENCY

From the previous equations we can determine the Stirling engine efficiency as the following:

$$\eta = \frac{W}{Q_E} \qquad (3)$$

Replacing the expression in the above formula we have that the Stirling engine efficiency is:

$$\eta = \frac{\left|-mRLn\left(\frac{V_2}{V_1}\right)(T_E - T_C)\right|}{mRT_E Ln\left(\frac{V_2}{V_1}\right)}$$

Simplifying:

$$\eta = 1 - \frac{T_C}{T_E} \qquad (4)$$

Note that the theoretical Stirling engine efficiency it´s the same as the Carnot efficiency, which is the highest possible efficiency that any thermal engine can achieve.

BASIC ENGINE CONFIGURATIONS

There are many possible configurations for the Stirling engines, most of them are simple variations of the basic ones, which are:

- Alpha: it´s a piston-piston configuration. Essentially two pistons are responsible of the various processes of the thermodynamic cycle (see fig. 2)

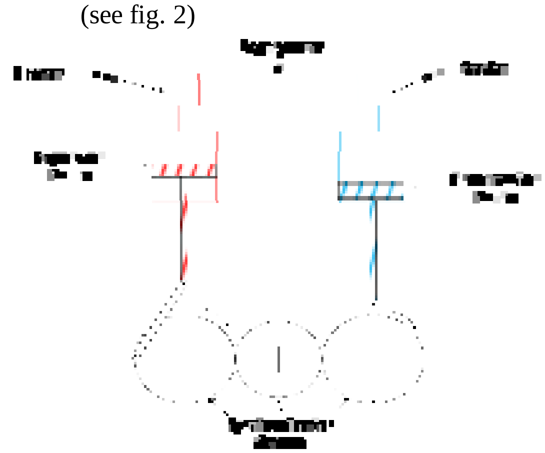


Figure 2. Alfa configuration

- Beta: unlike the previous, the beta is a piston-displacer configuration. The displacer is in charge of only moving the gas between the heating and cooling zone, while the piston does the compression and the expansion of the gas. A remarkable quality of this configuration is that the piston and displacer are in the same cylinder (see fig. 3), which brings two important advantages:

- Higher compression ratios
- Permits a more compact design

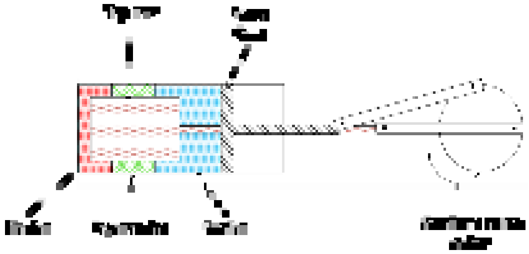


Figure 3. Beta configuration

- Gamma: it consists of a displacer and a power piston as the beta configuration, with the difference that these two elements have their own cylinder (see fig. 4)

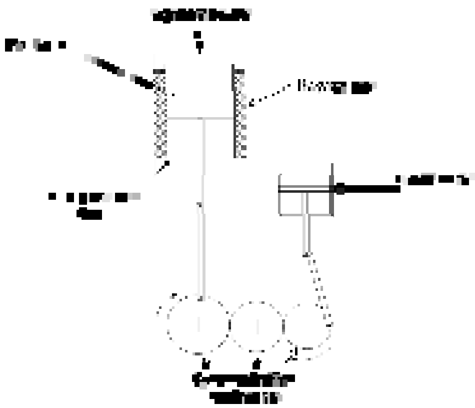


Figure 4. Gamma configuration

- Free Piston: It´s a piston-displacer configuration. The difference is that there is not any mechanical link between these main elements. This allows a completely sealed design without gas leakage, compact design and lower power loss from friction in the mechanical parts. It´s commonly used with a linear electrical generator for the mechanical power conversion into electricity.

ANALYSIS AND DESIGN METHODS

Schmidt Analysis:

The Schmidt analysis is a thermodynamical analysis considering the sinusoidal variations of the various spaces inside the engine due to the mechanical linkage (i.e. crankshaft). It´s a more accurate analysis but still continues to be ideal because the irreversibilities are not considered. It was first deduced by Gustav Schmidt in 1871. The principal assumptions made are [2]:

1. The gas used fallows the ideal gas law.
2. The gas´s mass is constant.
3. The instantaneous pressure is the same through the entire engine.
4. The compression and expansion processes are made at constant temperature.
5. Sinusoidal volumes variations.
6. Perfect regeneration (100% effective regenerator).
7. Non temperature gradients heat exchangers.
8. The temperature of each space is constant.
9. Constant velocity engine.

Based on a Beta configuration the equations are the following:

- Expansion Volume (V_E):

$$V_E = \frac{V_{DE}}{2} \cdot (1 - \cos\alpha) + V_{ME} \quad (5)$$

- Compression Volume (V_C):

$$V_C = \frac{V_{DC}}{2} \cdot (1 + \cos\alpha) + V_{MC} + \frac{V_{DP}}{2} \cdot (1 - \cos(\alpha - \beta)) - V_{SOL} \quad (6)$$

- Total Volume (V_T):

$$V_T = V_E + V_C + V_R \quad (7)$$

V_R is the regenerator volume.

- Pressure:

$$P = \frac{m \cdot \bar{R}}{\frac{V_E}{T_E} + \frac{V_C}{T_C} + \frac{V_R}{T_R}} \quad (8)$$

- Overlap Volume [2]:

$$V_{SOL} = \frac{V_{DE} + V_{DC}}{2} - \sqrt{\frac{(V_{DE})^2 + (V_{DC})^2}{4} - \frac{V_{DE} \cdot V_{DC}}{2} \cdot \cos\beta} \quad (9)$$

Since the displacer and piston are contained in the same cylinder their strokes overlap, for this reason there is an overlap volume.

Due to the complexity in the integration of the pressure in terms of the volume variation, a synthesized expression by J.R. Senft for beta configurations is used [3].

- Work:

$$W = \frac{\pi(1-\tau)P_{max}V_{DD}k\sin\beta}{Y + \sqrt{(Y^2 - X^2)}} \sqrt{\left[\frac{Y - X}{Y + X}\right]} \quad (10)$$

Where:

$$X = \sqrt{[(\tau - 1)^2 + 2(\tau - 1)k\cos\beta + k^2]} \quad (11)$$

$$Y = \tau + 4x\frac{\tau}{1+\tau} + D \quad (12)$$

$$D = \sqrt{(1 + k^2 - 2k\cos\beta)} \quad (13)$$

$$\tau = \frac{T_C}{T_E} \quad (14)$$

$$x = \frac{V_{mt}}{V_d} \quad (15)$$

$$k = \frac{V_{DP}}{V_d} \quad (16)$$

In the figure below, it can be seen the difference between the ideal Stirling Cycle and the cycle with the Schmidt analysis.

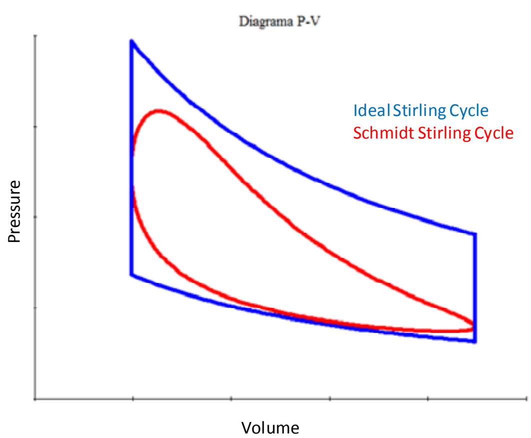


Figure 5. Stirling cycle p-V Diagram.

Second order Analysis:

This analysis brings the Schmidt analysis one step forward. Upon calculation of the estimated work done by the engine and the heat input required, the energy losses due the different irreversibilities are then calculated. This analysis can be found with greater detail in "Stirling Design Manual" [3].

Third order Analysis:

A numerical analysis it´s done to solve the mass, energy and momentum conservation equations, which are too complex to solve analytically.

Non-dimensional Analysis [4]:

At the initial stage of design, power and velocity of the engine estimation is needed. With this method a non-dimensional analysis considering the basic design variables is done. [4].

The principal parameters of which the power and this procedure are based are:

- Mean gas pressure (*Pm.)*.
- Cinematic gas viscosity (v).
- Expansion displacement volume (V_{de}).
- Expansion temperature (T_E).
- Compression temperature (T_C).

Three non dimensional numbers are deduced:

- Non-dimensional engine speed:

$$n^* \equiv \frac{nV_{de}{}^{2/3}}{v} \quad (17)$$

- Non-dimensional work output:

$$w_s{}^* \equiv \frac{T_E R V_{de}{}^{2/3}}{v^2} \quad (18)$$

- Non-dimensional specification number:

$$S^* \equiv \frac{T_E R V_{de}{}^{2/3}}{v^2} \quad (19)$$

Beside the previous non-dimensional numbers, there are three more numbers that will be used in junction.

- Non-dimensional pressure:

$$P^* \equiv \frac{P_m}{P_{lim}} \quad (20)$$

- Non-dimensional temperature:

$$T^* \equiv \frac{T_E - T_C}{T_E + T_C} \quad (21)$$

- Non-dimensional power:

$$Pow^* \equiv W_s^* n^* \quad (22)$$

Prediction procedure:

A new non-dimensional number is defined, which relates the work output of the engine with it fundamental parameters

$$W_s{}^* \equiv \frac{W}{P_m V_{de} P^* T^*} \quad (23)$$

The non-dimensional specification number is redefined to:

$$S^* \equiv \frac{T_{lim} R V_{de}{}^{2/3}}{v_{lim}{}^2} \quad (24)$$

Where T_{lim} is the highest temperature and the v_{lim} is the cinematic viscosity at highest temperature and pressure.

Experimentally, a relationship between maximum non-dimensional power and optimum non-dimensional velocity was established by the authors. Also a relationship between the non-dimensional velocity and the non-dimensional engine specification number was made.

$$Pow^*_{max} = 0{,}24 n^*_{opt} \quad (25)$$

$$n_{opt}^{*} = 6{,}8 \times 10^{-5}(S^{*})^{0{,}6} \qquad (26)$$

Based on the above, a design procedure diagram is presented by the authors [4]:

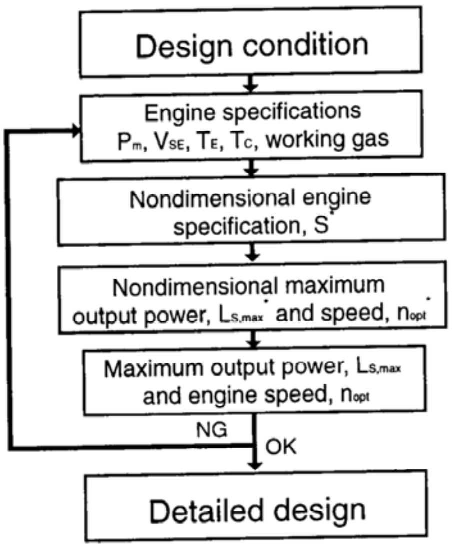


Figure 6. Design procedure for the non-dimensional analysis [4].

NON-DIMENSIONAL NUMBERS FOR POWER ESTIMATION

Beale Number (B_N): is number used to estimate the power output of a Stirling engine or to establish a comparison between engines. Typically Beale numbers are between 0.11 and 0.15.

$$B_N = \frac{Pow}{Pm \cdot V \cdot f} \qquad (27)$$

West Number (W_N): is another non-dimensional number used to estimate the power output of a Stirling engine. The difference is that it has a temperature correction. West numbers typically are between 0.25 and 0.35.

$$W_N = \frac{Pow}{Pm \cdot V \cdot f} \cdot \frac{(T_E + T_C)}{(T_E - T_C)} \qquad (28)$$

PROTOTYPE DESIGN

Design Objectives:

- The prototype must be designed to generate at least 50 watts.
- The design has to be simple.
- Establish a basic design of the prototype for further development with much more detail.

Methodology:

For the prototype design a methodology is presented based on the Schmidt and non-dimensional analysis. (See figure 7)

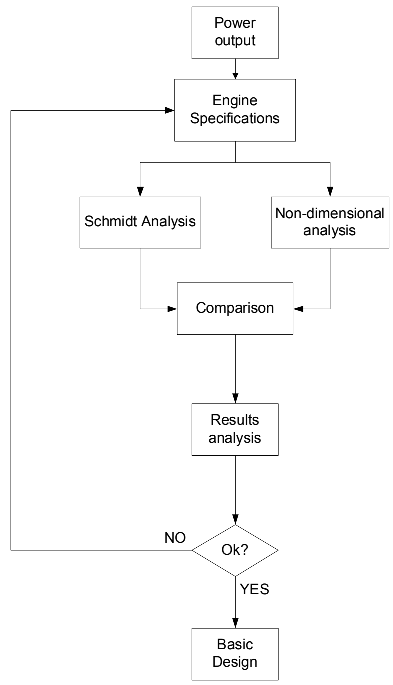


Figure 7. Proposed design methodology

CONCEPTUAL DESIGN

The prototype design is based on a beta configuration, for giving higher power capabilities because it is possible to achieve higher compression ratio, also because both the displacer and the piston are located in the same cylinder, it is possible to design a more compact prototype, producing less shadow over the collector, increasing the efficiency of the latter.

The future used gas is helium, it is the preferable choice among several prototypes because it has good thermal conductivity and it doesn´t react with the engines materials, besides it doesn´t has diffusion leakage through the materials as the hydrogen does. Another commonly used gas is air, because of its availability.

Table 1. Gases thermal conductivity

	Helium	Hidrogen	Air
k [W/m K]	0,33	0,412	0,062

Engine specifications

Using the non-dimensional analysis, a study was continued to see how the principal variables (expansion volume, mean pressure, expansion temperature) affect the engine performance.

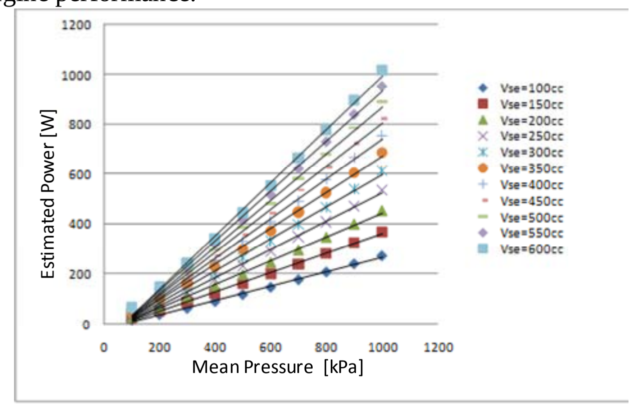


Figure 8. Estimated power function of mean pressure for several displacement volumes

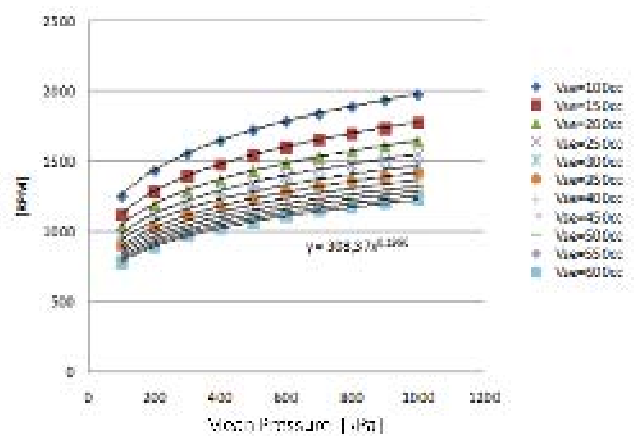


Figure 9. Estimated engine velocity function of mean pressure for several displacement volumes.

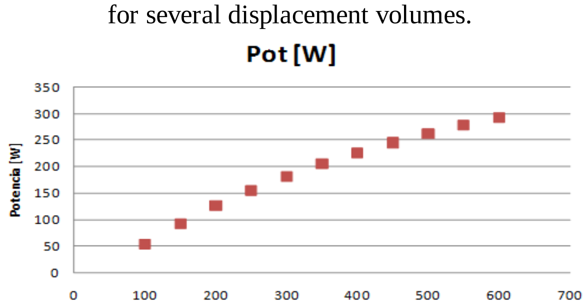


Figure 10. Estimated power as a function of expansion space temperature.

From the previous analysis a power goal was set to 200 W as main design parameter. A set of five possibilities were found to meet this goal

Table 2. Engines specifications for a 200W estimated power.

Engine	V_{DE} [cm^3]	Pm [kPa]	RPM (Estimated)
A	150	600	1550
B	200	500	1480
C	250	420	1400
D	300	400	1300
E	350	370	1200

Note: These results were found using:

- T_E = 900K
- T_C = 313.15 K

Engine "B" was selected with the following specifications:

- Expansion volume (V_E) = 200 cm^3.
- Compression volume (V_c) = 200 cm^3.
- Expansion temperature (T_E) = 600 °C = 900K
- Compression temperature (T_C) = 40 °C = 313,15 K
- Gas: Helium. R = 2077 J/ kg K
- Gas mass (m) = 2,34 x 10^{-4} kg.= 0,0584 mol (He)
- Charge pressure: P_{charge} = 370 kPa
- Charge volume: V_{charge} = total volume = 391,421 cm^3
- Charge temperature: T_{charge} = 25 °C = 298,15 K

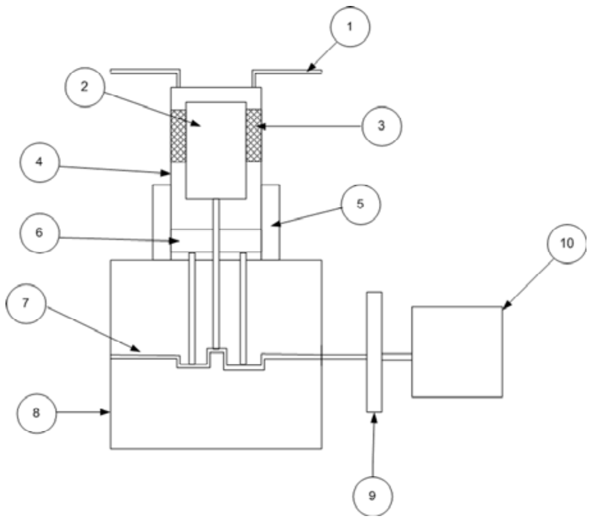
A study for the estimation of the selected prototype performance was made using the Schmidt analysis, Beale and West number.

Table 3. Estimated power from Schmidt analysis, Beale and West number.

		Beale		West	
	Schmidt	0,11	0,15	0,25	0,35
Power [W]@1480 rpm	2467	272	371	298	417

BASIC DESIGN

The principal elements to design are:



1. Heater
2. Displacer
3. Regenerator
4. Main Cylinder
5. Water jacket
6. Piston
7. Crankshaft
8. Engine Block
9. Flywheel
10. Generator

Figure 11. Prototype scheme and its parts.

Basic dimensioning of the solar collector:

This element is in charge of focus the solar radiation onto the engine heater. For the basic design, a irradiance of 100 W/m^2 was supposed. With the latter, and the estimated ideal power output of the prototype and its efficiency the total frontal area of the solar collector is estimated. This was found using the equation below [6].

$$A_a = \frac{Pow}{\eta_G} \times \frac{1}{irrad} \quad (29)$$

Considering the ideal power output estimation calculated with the Schmidt analysis (2,467 W) with an efficiency of 65.2%, plus the collector efficiency of 60% [6], the total front area is:

$$A_a = 6{,}3\, m^2 \quad (30)$$

For reaching the expansion temperature (900 K), a solar collector concentration ratio of approximately 10^3 is needed. (See figure below, [6]).

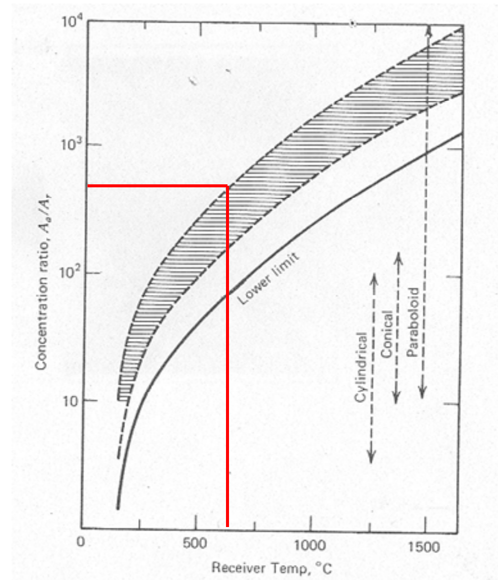


Figure 12. Concentration ratio vs Receiver temperature for a solar collector [6].

Piston Basic Design

The design of the piston is considering a super-squared configuration, this means that diameter is larger than the stroke of the piston (S<Ø).

A suggestion found in the bibliography [7] is that the piston diameter should be two times bigger than the stroke, for this reason and to meet with the 200cm^3 of the engine displacement, the piston dimensions are:

- Piston diameter = 80mm
- Piston stroke = 40 mm

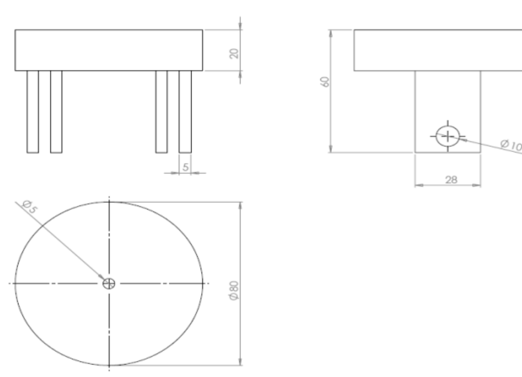


Figure 13. Piston basic dimensions (mm)

Main Cylinder Basic Design:

The main cylinder is the engine part where the principal elements of a Stirling engine are. These elements are:

- Piston
- Displacer
- Regenerator

Besides this, the main cylinder should meet with the following requirements:

- Allow free movement for the displacer and piston.
- Resist the materials efforts.
- Enough space for the regenerator
- Maintain dead volume as minimum as possible.
- As the engine is water cooled, it should have a water jacket for this purpose.

Based on the dimensions of the piston the main cylinder is as follows:

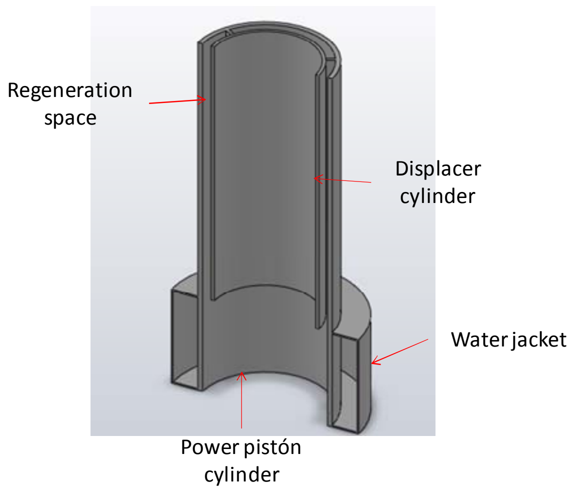


Figure 14. Main cylinder parts

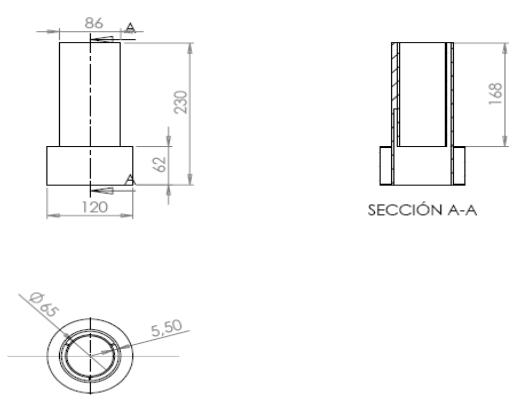


Figure 15. Main cylinder basic dimensions (mm).

<u>Heater Basic Design:</u>

The heater should provide 153J in each half revolution of the engine. Supposing that the engine velocity will be 1,480 rpm, the heat exchange rate will be:

$$\dot{Q}_E \approx 7567\ W$$

The present design was base on the following assumptions:

- Stable conditions
- Constant gas properties at 900K and 1.8 MPa (maximum pressure)
- Laminar flow, completely developed.
- A set o 20 tubes heat exchanger.
- A ΔT of 5 K between the external tube wall of the heat exchanger and the gas.

Design procedure of the heat exchanger

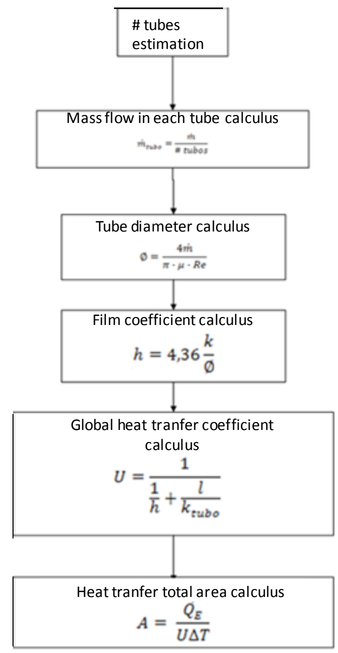


Figure 16. Heater design procedure.

Mass flow is the total gas engine passing through the tubes in half revolution of the engine:

$$\dot{m} = 1{,}1544 \times 10^{-2}\frac{kg}{s}$$

$$\dot{m}_{tube} = 5{,}772 \times 10^{-4}\frac{kg}{s}$$

For calculation of the tubes diameter, the following equation was used [8] based on a Reynolds number of 2000.

$$\emptyset = \frac{4\dot{m}_{tube}}{\pi \cdot \mu \cdot Re} \approx 8{,}6mm \qquad (31)$$

Where:

- $\mu = 4{,}2867 \times 10^{-5} Pa \cdot s$

With the tubes diameter found and with the gas conductivity ($k = 0{,}336 \frac{W}{m \cdot K}$) at the condictions above mentioned, the film coefficient is searched using the Nusselt Number correlation (Nu = 4.36) [8].

$$h = 4{,}36 \frac{k}{\emptyset} = 170{,}344 \frac{W}{m^2 \cdot K} \qquad (32)$$

With the thickness and conductivity of the tubes, in this case, stainless steel, the global heat transfer coefficient is found.

$$U = \frac{1}{\frac{1}{h} + \frac{e}{k_{tubo}}} = 167{,}68 \frac{W}{m \cdot K} \qquad (33)$$

Where:

- $k_{tube} = 24 \frac{W}{m \cdot K} @900K$
- Thickness $e = 2{,}24\ mm$

The total heat transfer area needed is then:

$$A = \frac{\dot{Q}_E}{U \cdot \Delta T} = 9{,}02 m^2 \qquad (34)$$

Water Jacket basic design

In the present work, the basic design for the water jacket limited to the estimation of water needed by the prototype to perform at the parameters above established.

The cooling happens in half revolution of the engine. Supposing that the prototype velocity is 1480 rpm, the heat transfer rate is:

$$\dot{Q}_c = \frac{Q_C}{t} = 2632{,}86\ W \qquad (35)$$

$$t = 2{,}027 \times 10^{-2} s$$

Assuming a ΔT of 10 K between the water inlet and outlet, the total water mass flow is:

$$\dot{m} = \frac{\dot{Q}_C}{c_p \Delta T} = 0{,}063 \frac{kg}{s} \qquad (36)$$

Considering:

- Inlet water temperature = 25 °C
- Water pressure = 101 kPa
- Water specific heat $c_p = 4178{,}37 \frac{J}{kg \cdot K}$

Regenerator Basic Design

The most important element of a Stirling engine is the regenerator. It is the responsible of its high efficiency and thus its viability.

For the design, a simple methodology based on turbines regenerator was used [5].

First, with the regenerator effectiveness (E = 0.9), the number of transfer units are found with help of the following figure [5]

$$N_{tu} = 35$$

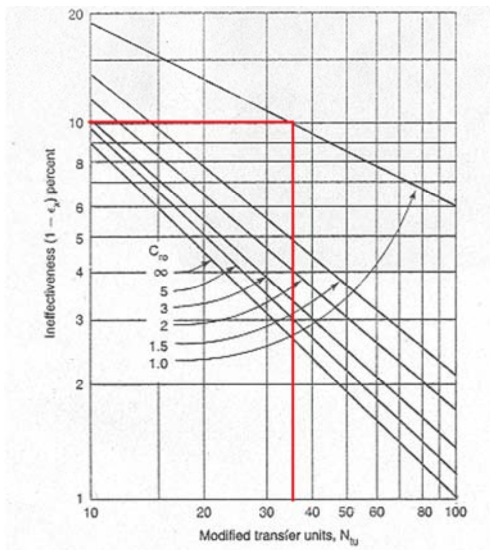


Figure 17. Ineffectiveness vs. number of transfer units [5]

Then, the film coefficient is calculated.

$$h_t \cdot A_h = (C_r + 1) \cdot (\dot{m} C_P) N_{tu} = 10.903 \frac{kW}{K} \qquad (37)$$

Where:

- $C_r = 4$
- $\dot{m} = 0.012 \frac{kg}{s}$
- $C_P = 5.1917 \frac{J}{g \cdot K}$
- $N_{tu} = 35$

Later, the gas mean speed is founded just to continue with the front flow area of the regenerator:

$$C_{HT} = \sqrt{R\overline{T_{ht}}\left[\frac{\dot{m}C_P}{h_t A_H}\right]\left[\frac{\Delta P}{P}\right]} = 18.973\ \frac{m}{s} \qquad (38)$$

- $R = 2077\ \frac{J}{kg\cdot K}$
- $\overline{T_{ht}} = 606.505\ K$
- $\frac{\Delta P}{P} = 0.05$

Regenerator front area:

$$A_{ff} = \frac{\dot{m}}{\overline{\rho_h} C_{HT}} = 4{,}414\ cm^2 \qquad (39)$$

- $\overline{\rho_h} = 1{,}433\ \frac{kg}{m^3}$

The calculation of the total heat transfer area is left, for this regenerator ducts geometry in which the gas passes has to be defined. For this purpose, the bibliography consulted [5] proposes s several commonly used geometries. In the present work a square geometry is used.

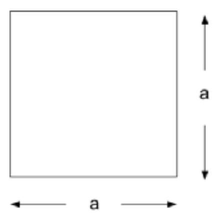


Figure 18. Frontal geometry of the regenerator gas ducts.

- a = 0,55 mm
- d_h= 0,55 mm
- N_u= 3,61

The total heat transfer area is:

$$A_{ht} = \frac{A_{ff}}{A_x} P_e L = 0.536 m^2 \qquad (40)$$

Displacer Basic Design

The displacer serves to move the gas from the expansion space to the cooling space and vice versa. First, It should avoid any parasitic heat transfer between these two spaces through it, and second, it should be as lighter as possible to reduce the engine inertia.

A void displacer design was selected without any pressurization, filled with a low weight isolation material, such as cotton

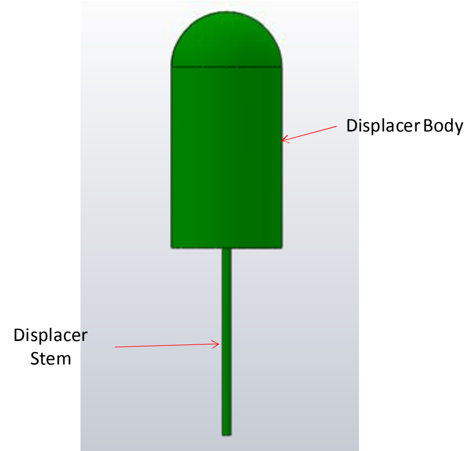


Figure 19. Displacer main parts.

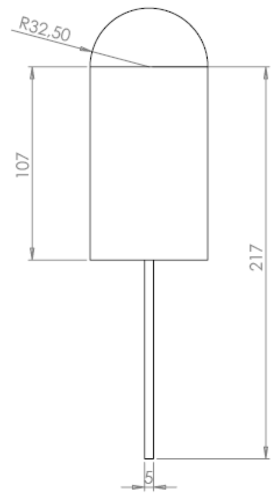


Figure 20. Displacer basic dimensions (mm).

Crankshaft Basic Dimensioning

In order for the prototype to work, the displacer should move in advance of the piston precisely 90° forward.

Piston elbow eccentricity = 20 mm
Displacer elbow eccentricity = 35 mm

CONCLUSIONS

- The present work consists of a first approach of a Stirling engine design and is only limited to a conceptual design and basic dimensioning, so a further work is needed to reach a fully and detailed design fallowed by actual prototype.
- There are dissimilarities among the various analysis methods used, so a real prototype is necessary for the understanding of the complex phenomena occurring inside a Stirling engine.
- A reliable analysis method is necessary for the design of future prototypes. For this an actual prototype becomes important.

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