24th COBEM - 2017

24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-2516
PORTRAYAL OF DISPLACER AND TEMPERATURE IN BETA-TYPE STIRLING ENGINE

Bryan C Caetano
Oscar R Sandoval Rodríguez
Ramon M Valle
Programa de Pós-Graduação em Engenharia Mecânica, Universidade Federal de Minas Gerais, Belo Horizonte, Brasil
email: bcastrocaetano@gmail.com
ramon@demec.ufmg.br
osc.sandoval@hotmail.com

Isadora F Lara
Giovanni Fialho del Giudice
Programa de Graduação em Engenharia Mecânica, Universidade Federal de Minas Gerais, Belo Horizonte, Brasil
email: isadoraflara@gmail.com
giovannigiudice@gmail.com

Abstract. *In this paper, a Beta-type Stirling engine was designed, manufactured and optimized, having its performance tested with air at atmospheric pressure and with a gas burner as the heat source. On the prototype tests the clearance between the cylinder and the displacer, and the hot-source temperature were modified for the purpose of correlate the impact on the performance. The trials were performed with cylinder-displacer gaps of 1.5, 1.0 and 0.5 mm, as well as temperature differentials of 580°C and 480°C. Deviations of engine torque, power and rotation are presented using an electrical test bench. The results indicate that a maximum power of 0.24 W and a mean gauge effective pressure of 17.4 kPa was attained at 680 rpm with a displacer-cylinder of 22 mm and a differential temperature between hot and cold chamber of 580°C.*

Keywords: *Renewable energy, Beta Stirling engine, Piston- Cylinder arrangement.*

1. INTRODUCTION

Since the quality of the environment and the levels of consumption of energy have more rigorous laws, the recent works (Cinar *et al.*, 2005; Kagawa, 2000; Darlington and Strong, 2005) appoint that Stirling engines are coming to be increasingly popular, for being simple and easy to operate, running with almost any combustible fuel, especially renewable energy, and having this energy transformation at high efficiency. It is a relatively new technology, since it was not until engineers at Holland's N.V. Philips Company rediscovered the Stirling engine that researches really began; they needed an energy source for their remote regions. Only by 1946, had Phillips made vast enhancements on its performance as shows Ross (1981) and Martini (1978), with velocities such as 5000 rpm and efficiencies up to 38%, as an engine detailed by Walker (1980).

Before manufacture, it is a necessity to be able to predict the output power of a projected Stirling engine. Studies by Angkee and Srikam (2011) show that it is not by simply by increasing size or temperature that the engine will increase its power in the same proportion, as shown on the Table 1. In this table, P is the fluid pressure in static conditions, V_{SE} is the swept volume of the expansion space, n is the engine speed, T_E and T_C are the expansion and compression space temperature respectively and finally, W_n is the dimensionless West number.

Table 1- Stirling output examples by Angkee and Srikam (2011).

	Power (W)	P (bar)	V_{SE} (cc)	n (rpm)	T_E (K)	T_C (K)	W_n
Iwamoto	150	1	40200	150	403	313	0.119
Kongtragool	1.169	1	894	52.1	436	307	0.125
Kongtragool	6.1	1	7391	20	439	307	0.140
Kongtragool	11.8	1	894	133	589	307	0.189
Ishiki	91	1	1767	500	635	305	0.176
Ecoboy (N_2)	60	8	81.4	1155	703	323	0.129
Ecoboy (He)	102	9	81.4	1103	721	313	0.192
Ecoboy (N_2)	71	8	81.4	954	737	313	0.170
Kongtragool	32.7	1	7391	42.1	771	310	0.148
Angkee, 350°C	26.63	7	165	267	623	308	0.153
Angkee, 500°C	95.4	7	165	360	773	308	0.320
Basic 400 hp	291000	110	17400	452	967	313	0.395
NS-03M	3810	62	161	1401	971	313	0.319
MP1002CA	250	15	59.4	1500	973	313	0.219
NS-03T	4140	64	268.7	1299	991	313	0.214
GPU-3	8950	69	120	3600	1019	313	0.340
Batmaz	118	2	440	1200	1223	318	0.114
Cinar	128	4	276	891	1273	293	0.125
Cinar	5.98	1	192	208	1273	303	0.146
Karabulut	65	2.5	183	555	1373	293	0.237

According to the literature presented, is observed that one parameter involved in the calculus of the engine output power is the area between cylinder and displacer. Aksoy (2013) developed a study on the effects of increasing the heat transfer surface area and regenerator volume by grooving axial slots on the displacer. The results showed an increase on the maximum output power, 172.63 W for the smooth displacer and 265.87 W for the grooved, both with same heat transfer coefficients.

Another parameter researched to understand the operation of Stirling engine is the clearance between cylinder and displacer, the "shuttle heat transfer" and the sum of enthalpy pumping is known as "Dispalcer gap losses", Jens (2013) . In these works, was shown that the engine output power was also dependent of this parameter. An example is the analytical model developed by Mabrouk et al. (2014) that shows the displacer gap losses are minimum in gaps between 50 and 70 microns. Now, Chen (2017) for Gamma-Type Stirling engines, shows that by reducing the displacer-cylinder gap from 1.5 mm to 0.875 mm, the power was increased 32.03% and efficiency 5.87%. However, a too small gap can create too much friction and reduce the engine power.

The present study is concerned with the manufacture of Beta-Type engines, which are showed as for having the greatest shaft power values between the Alpha- and Gamma-Types, although it is not the most suitable when work in low temperature differentials, Cheng and Yang (2012).

In this paper, the Beta-Type Stirling engine assembled works with varied high temperature differentials and displacer diameters, having the power piston diameter, cylinder, phase angle and swept volume as constants. This paper has the purpose to experimentally understand how the gap between the displacer and cylinder affects the power output of the engine. For that, experimental and theoretical results are presented.

2. THERMODYNAMIC MODELS

Two forms of engine analysis were executed. The first was a theoretical analysis known as the Schimidt cycle, proposed by Gustaf Schmidt and is exposed by Martini (1983). The calculation of the power output is wrought by using the calculated pressure as the pressure on top of the power piston as in the Eq.(1), and neglecting the pressure variation on the displacer.

$$P = \frac{M.R}{\frac{H(N)}{TH}+\frac{C(N)}{TC}+\frac{RD}{TR}} \tag{1}$$

The variables *M* and *R* denote the gas molar mass and the universal gas constant respectively. The temperatures *TH, TC* and *TR* are measured in the expansion space, compression space and in the regenerator correspondingly. Volume variations for the expansion space are represented by *H(N)* and for the compression space *C(N)*. The dead volume in the regenerator is symbolized by *RD*.

The power was calculated by the J.R. Senf method, presented below in the Eq.(2) and used by Martini (1978).

$$W1 = \frac{\omega(1-AU)\cdot PX\cdot(XY)\cdot sen(AL)}{2\cdot(Y+(Y^2-X^2)^{\frac{1}{2}})}\cdot(\frac{Y-X}{Y+X})^{\frac{1}{2}} \quad (2)$$

Where *W1* is the work per cycle in Joules, *PX* is the maximum pressure reached during the cycle in Pascal. *AU* is the ratio between *TC* and *TH*, *VL* is the maximum volume in the expansion space, *AL* is the phase angle between the pistons. The variables *XY*, *Y* and *X* can be found in the work done by Moura (2012).

The second engine analysis was a prediction method developed by William Beale, the creator of the free-piston Stirling engine (Walker, 1980). He developed a non-dimensional number, the non-dimensional power, and created a graph based on a constant known as the Beale number. Therefore, the power could be obtained using the Eq. (3). This equation was used by Adaileh and Alahmer (2015).

$$P_o = B_n P V_E f \quad (3)$$

Where *Bn* is the Beale number (attained by the Figure 1), *P* is the mean pressure, V_E is the expansion volume, *f* is the frequency and *Po* the output power. Figure 1 shows the Beale number linked to the expansion space temperature.

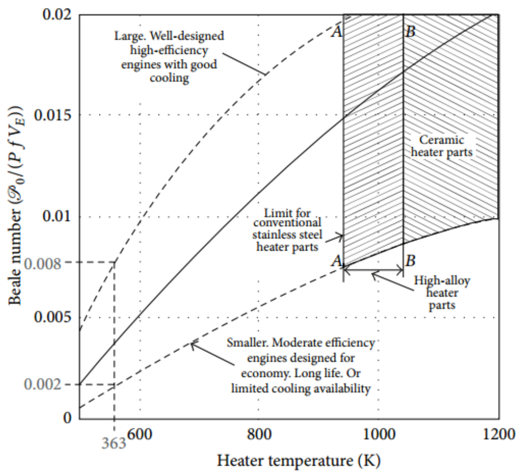

Figure 1- Beale number for heater temperature by Adaileh and Alahmer (2015).

However, it is important know that nether of both methods is able to portray the engine exactness, since they do not take in direct consideration the displacer-cylinder gap.

3. EXPERIMENTAL PROTOTYPE

The engine manufactured was a Beta-Type Stirling engine, with one cylinder where both the piston and displacer move, having also a heater and cooler, in addition to a crankshaft. The phase angle is 90° between the displacer and piston. The piston is made of a self-lubricant material, graphite plugged bronze, which reduces its friction. A secondary lubrication method was also used, using oil SAE 15W-40 to lower the mechanism's resistance; it was possible since the cooling system operated at set point temperature of 28°C, so the oil only suffered small deterioration when it was close to the heater. The displacers and the cylinder were made with SAE 1020 steel. The cooling system was a cylinder with cooling fins, made with aluminum and with water as assistance, since the temperatures were greater than the atmospheric air unaided was capable of cooling. The crankshaft was made out of steel, as well as the flywheel. Two flywheels were needed to balance the inertia of the system. The piston was connected to the crankshaft by two sets of paired rods, and the displacer by one set. The rods were made out of Aluminum SAE 6061. The Table 2 shows the engine parameters and some conditions of working.

3.1 Experimental setup

The engine instrumentation is presented in the Figure 2. A electric dynamometer was constructed for measure torque and power using the tension of the DC motor C4557-60003 as reference. The torque increments were obtained using electrical resistors. Eight resistor values were used for each displacer, from 45 Ω to 0 Ω.

Table 2 - Engine proprieties

Proprieties	Values
Engine type	Beta
Bore	23 mm
Displacer stroke	36 mm
Displacer diameter 1	20 mm
Displacer diameter 2	21 mm
Displacer diameter 3	22 mm
Power piston stroke	20 mm
Working fluid	Air
Compression ratio	1.50
Heating temperature	876 K and 676 K
Maximum engine power	0,2469 W
Maximum engine torque	3.44 mN.m

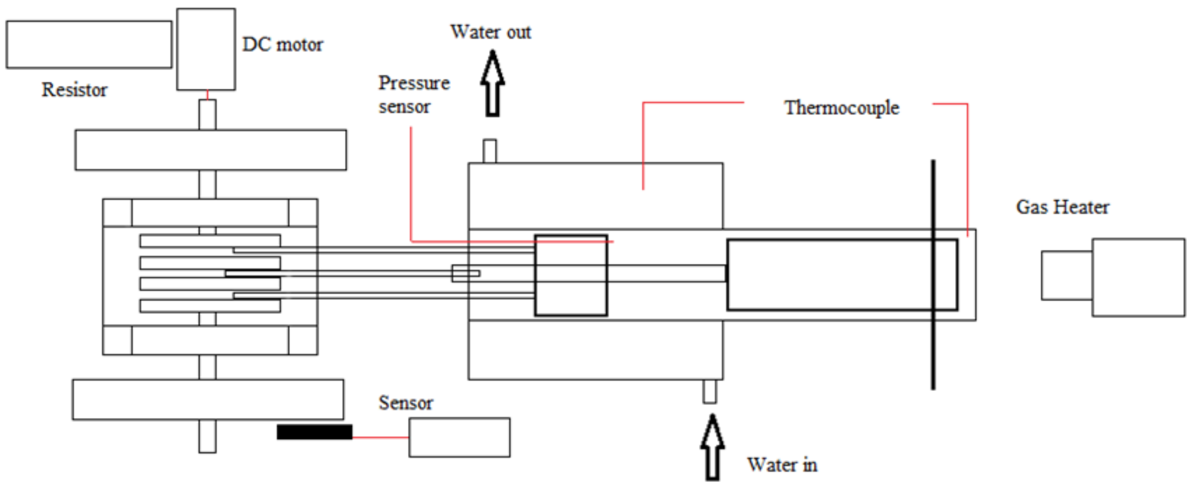

Figure 2 - Engine scheme drawing

The pressure measurement was done using two-gauge pressure sensors type MPX5050DP. One measures the pressures above the atmospheric and other the ones below, both of them in the piston head. An optical sensor, TCRT5000, obtained the position and rotation with a scale placed into the flywheel, with 15° of distance between black and white sections. The temperature was measured by three type K thermocouples, which work from -125°C to 1250°C. More information about the system calibration and details for engine instrumentation is presented in Caetano (2016).

4. RESULTS AND DISCUSSION

The engine prediction methods show a power output between 0,42 W and 0,12 W for the Beale number, and one of 23,8 W for the Schmidt theory. It displays that the most recent methods have a more decent approximation of the reality. However, the error is still too ample; for the Beale number the same engine can be 350% more potent, screening crude inaccuracies.

The Figure 3 shows the power and speed graph, pointing that the engine had maximum torques with the smallest clearance (22 mm displacer and 0.5 mm clearance) and for each displacer a different speed level was attained.

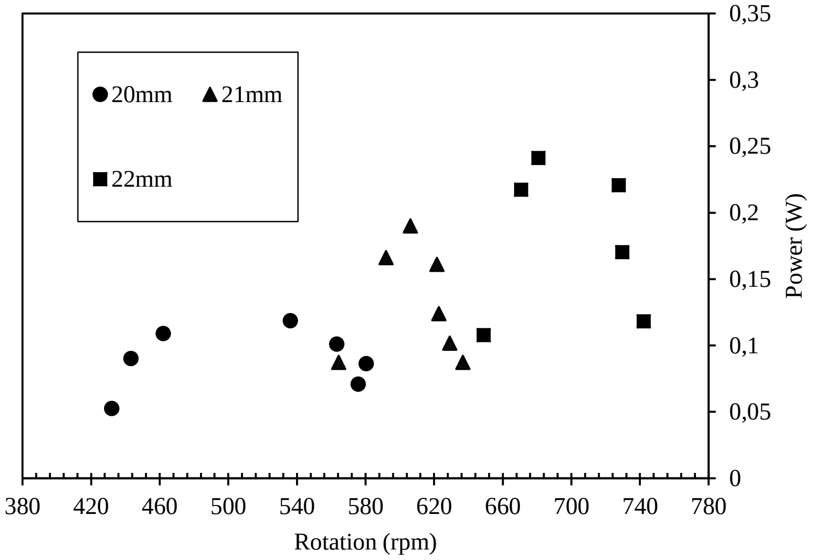

Figure 3 - Power vs Rotation for three displacers tested.

Results conveying two different temperature differentials are also presented in Figure 4, they show a smaller output power for the lower temperature. Observing the graph, the engine reach a higher power with a $\Delta T = 575$ ˚C between the cold and hot chamber. So, the engine works with a different range of speed in the tests, where the higher range is obtained with the higher ΔT.

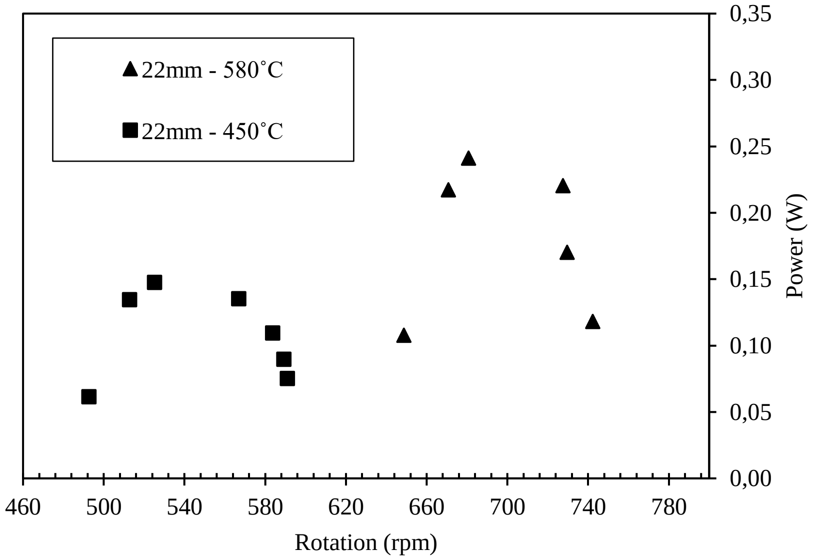

Figure 4 - Power vs Rotation for the 22 mm displacer and two different temperatures.

Figure 5 show the smallest peaks of minimum gauge pressure were obtained using the displacement piston with the smallest gap. And with it, the highest MEP (Mean Effective Pressure) values and consequently the highest power values were obtained. A lower peak of pressure when the piston reaches BDC (Bottom Dead Centre) decreases the pumping work at the time of compression of the power piston, which can increase the effective power of the motor shaft. This effect can be confirmed by Figure 3, where the smaller displacement piston has the largest power range.

Also by Figure 5, the peak of maximum pressure remained practically constant in the three displacers tested. That is, the gap between the displacing piston and the cylinder influences little or almost no relevant impact to change the maximum pressure peak at the moment of expansion when the power piston is in TDC (Top Dead Centre).

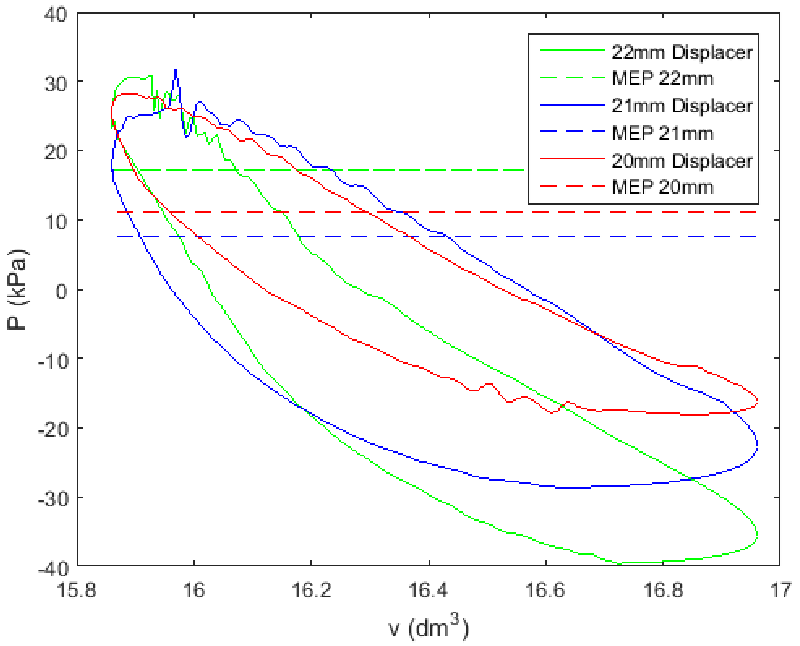

Figure 5 – P vs V for the three displacers tested.

The Figure 6 present the characteristic P vs V for the 22 mm displacer in the same torque request, but with variation on the differential temperature (ΔT). In the tests, the variations were 570 °C and 450 °C, both with temperature of cold region controlled in 35 °C. Thereby, it is shown that the increase in the heat rate change the peak of positive pressure and consequently the MEP and the output power in the Stirling engine. This characteristic is also present in the Figure 4, where it is observed the direct proportion between the pressure and the heat (increase of hot temperature).

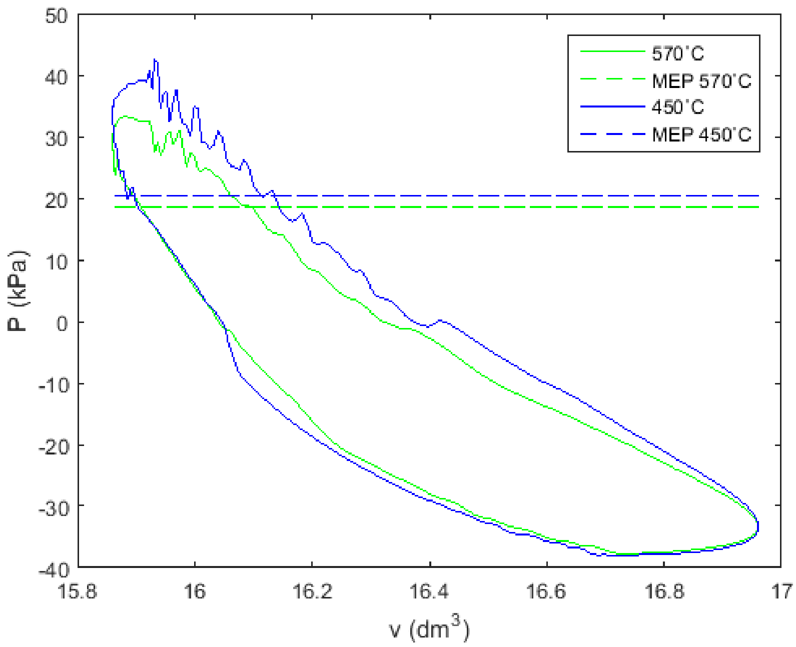

Figure 6 - P vs V for the 22 mm displacer tested, same torque and variation on ΔT.

Another study was made in Figure 7, the displacer, torque and differential temperatures are kept constant, only the rotation has been altered. In both cases, the MEP did not present significant variation and attended 20 kPa. This result was expected because there was no change in torque. Thus, another study was done on the pressure sensor and it was observed that due to sensor uncertainty is 2.5% of the full scale, the variation of the velocity presented did not show significant changes in the cycle.

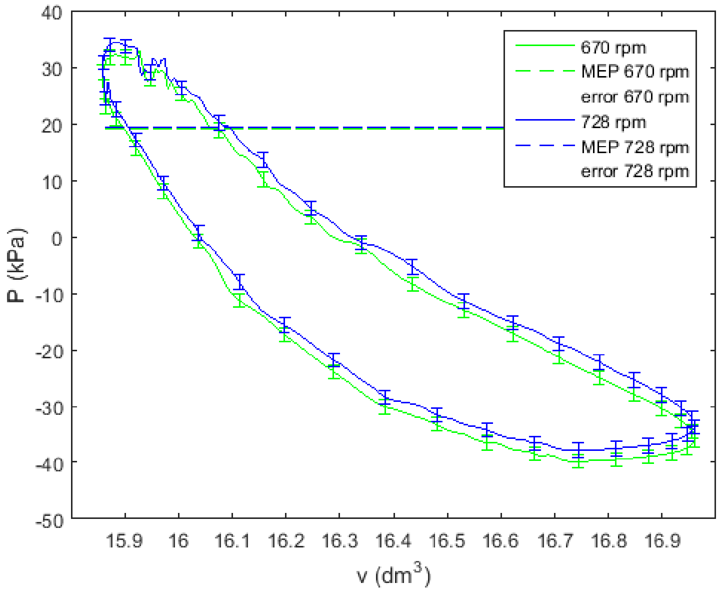

Figure 7 - P vs V for the 22 mm displacer tested, same torque and ΔT but a small variation on rotation.

Lastly, was detected that the lower pressures were achieved during the expansion process, which was facilitated with the lowest gap between cylinder and displacer. The Table 3 regarding the maximum power output for each displacer is illustrated below.

Table 3. Power, rotation and pressures for each displacer.

Displacer diameter - ΔT	Max. Power (W)	Rotation (rpm)	MEP (kPa)
20 mm – 575 ˚C	0,12	536	12.0
21 mm – 575 ˚C	0,19	606	8.9
22 mm – 575 ˚C	0,24	680	17.4
22 mm – 450 ˚C	0,15	525	19.9

It is possible to notice a maximum power and torque of 0.24 mW and 3.44 mN/m respectively, which obtained with the 22 mm displacer for the temperature disparity of 581 °C at 683 rpm. A bigger regeneration area is the reason to why the 22 mm displacer has shown the biggest power, torque and rotation values, as well as a smaller dead volume.

5. CONCLUSION

In this work, a study on a Beta-Type Stirling engine showed as results a maximum power of 0.24 W and speeds up to 727 rpm. A great power loss is perceptible, due to friction and sealing issues. The maximum power obtained was 0.12 W, 0.19 W and 0.24 W for the 20 mm, 21 mm and 22 mm displacer respectively.

The heat exchange between the fluid and the cylinder walls can be improved by testing other working fluids with different pressures and different cylinder wall materials. It is thus possible to obtain a higher rate of heat exchange between the walls and the fluid, improving engine power and efficiency.

Whereas the hot and the cold chamber are mounted on the same cylinder, it was possible to observe a great loss of heat by conduction by the cylinder going from the hot chamber to the cold chamber. This type of loss is either entered into the cycle and consumes a greater input heat in the hot chamber, consequently overloading the cooling of the cold chamber. This is an important factor, since the insulation between the walls of the chambers can increase the overall thermal efficiency of the engine.

6. REFERENCES

Aladayleh W. and Alahmer A., 2015. “Recovery of Exhaust Waste Heat for ICE Using the Beta Type Stirling Engine”. In *Journal of Energy*, Volume 2015, p. 01.

Aksoy F. and Cinar C., 2013. "Thermodynamic analysis of a beta- type Stirling engine with rhombic drive mechanism". In *Energy Conversion and Management*, Vol. 75, p. 319-324.

Angkee S. and Srikam C., 2011. "Design and performance of a moderate temperature difference Stirling engine". In *Renewable Energy*, Vol. 36, p. 1728-1733

Caetano B. C., Rodriguez O. R. S., Maia T. A. C., Félix M. M. and Valle R. M., 2016. "Estudo da Influência das Dimensões do Pistão Deslocador nos Parâmetros de Desempenho de Um Motor Stirling do Tipo Beta". In *International Congress of Engines, fuels and Combustion*. Belo Horizonte, Brazil.

Cinar C., Yucesu S., Topgul T. and Okur M, 2005. "Beta- type Stirling engine operating at atmospheric pressure". In *Applied Energy*, Vol. 81, p. 351-357

Chen W. L., 2017. "A study on the effects of geometric parameters in a low-temperature differential c-type Stirling engine using CFD". In *International Journal of Heat and Mass Transfer*, Vol. 107, p.1002–1013.

Cheng C.-H. and Yang H.-S., 2012. "Optimization of geometrical parameters for Stirling engines based on theoretical analysis". In *Applied Energy*, Vol. 92, p. 395-405.

Darlington R. and Strong K., 2005. *Stirling and Hot Air Engines: Designing and Building Experimental Model Stirling Engines*, Crowood Press.

Jens P, Hans-Detlev K. A review of models for appendix gap losses in stirling cycle machines. In: 11th International Energy Conversion Engineering Conference; 2013.

Kagawa N., 2000. "An experimental study of a 3-kW Stirling engine". In *the Energy Conversion Engineering Conference and Exhibit. (IECEC) 35th Intersociety Energy*. Las Vegas, NV, USA.

Mabrouk M.T., Kheiri A. and Feidt M., 2014. "Displacer gap losses in beta and gamma Stirling engines". In *Energy*, Vol. 72, p.135 – 144.

Martini W. R., 1978. *Stirling Engine Design Manual*. U.S. Department of Energy, Office of Conservation and Solar Applications, Division of Transportation Energy Conservation, 1st edition.

Martini W. R., 1983. *Stirling Engine Design Manual*. U.S. Department of Energy, Office of Conservation and Solar Applications, Division of Transportation Energy Conservation, 2nd edition.

Moura M. M., 2012. *Proposta de Modificação de um Protótipo de Motor Stirling do tipo Beta*. Bachelor thesis, Universidade Federal de Minas Gerais, Belo Horizonte.

Ross A., 1981. *Stirling cycle engines: Solar Engines*, A Brief History of the Stirling Engine, pp. 13-16.

Walker G., 1980. *Stirling engines*. Oxford Press, 2nd edition.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.