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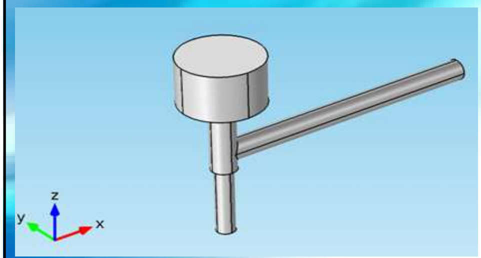
A non-conventional five piston double-acting Stirling Engine filling system CFD simulation and experimental validation



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Contents

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1. Introduction

- What is new in this engine?
- Goal: Analysis of the working cycle of the new design
- Means: Real engine working in laboratory and theoretical model for its analysis (COMSOL Multiphysics).

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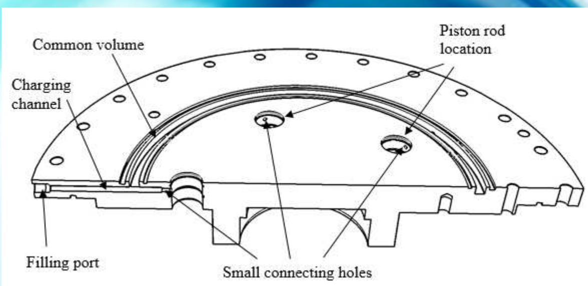
2. Analysis Base Model

- Common volume joining all filling channels.
- Small holes connecting filling to pistons.
- Pressure analysed.



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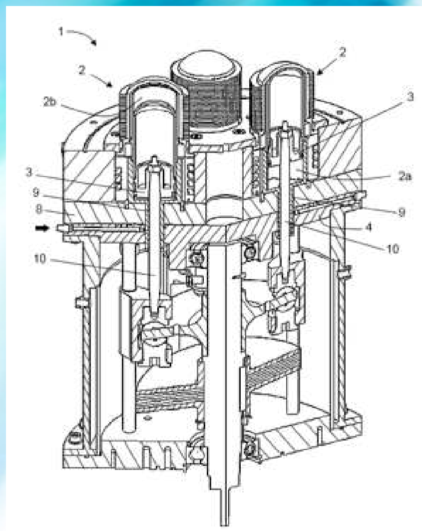
2. Analysis Base Model



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2.1. Theoretical analysis applied

- Navier-Stokes equation including heat transfer

$$\rho\frac{\partial u}{\partial t} + \rho(u\cdot\nabla)u = \nabla\left[-pl + \mu(\nabla u + (\nabla u)^T) - \frac{2}{3}\mu(\nabla u)l\right] + F$$

$$\frac{\partial \rho}{\partial t} + \nabla(\rho u) = 0$$

$$\rho C_P\frac{\partial T}{\partial t} + \rho C_P u\cdot\nabla T = \nabla\cdot(k\nabla T) + Q$$

2.1. Theoretical analysis applied

- When viscous heating and pressure work is considered:

$$\rho C_P \frac{\partial T}{\partial t} + \rho C_P u \cdot \nabla T = \nabla \cdot (k \nabla T) + Q + Q_{vh} + W_p$$

$$W_p = \frac{T}{\rho}\left(\frac{\partial \rho}{\partial T}\right)|_p\left(\frac{\partial p_A}{\partial t} + u + \nabla p_A\right)$$

$$Q_{vh} = \mu\left(\nabla u + (\nabla u)^T - \frac{2}{3}(\nabla u) l\right) : \nabla u$$

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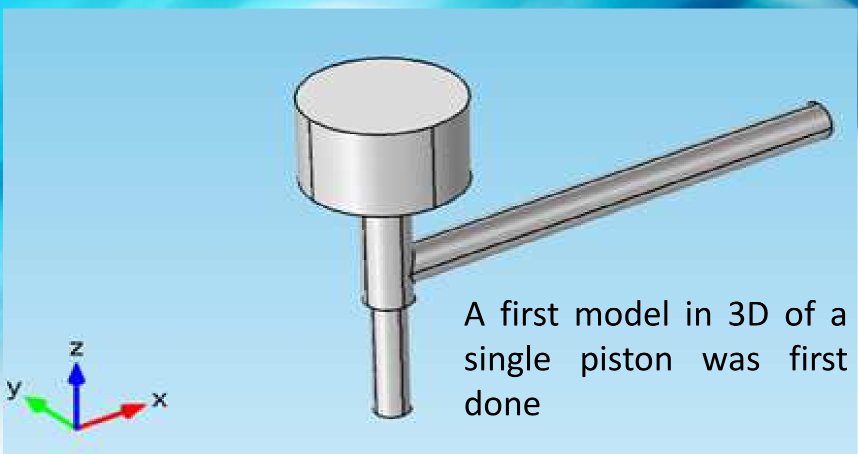
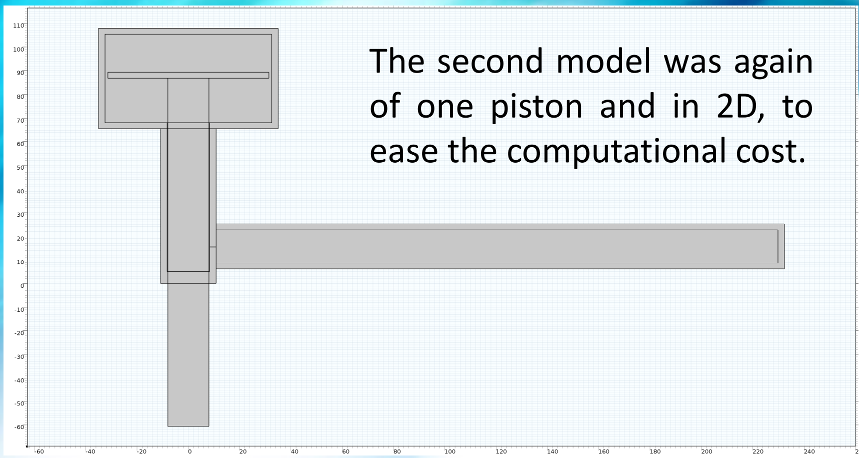
3. Theoretical analysis



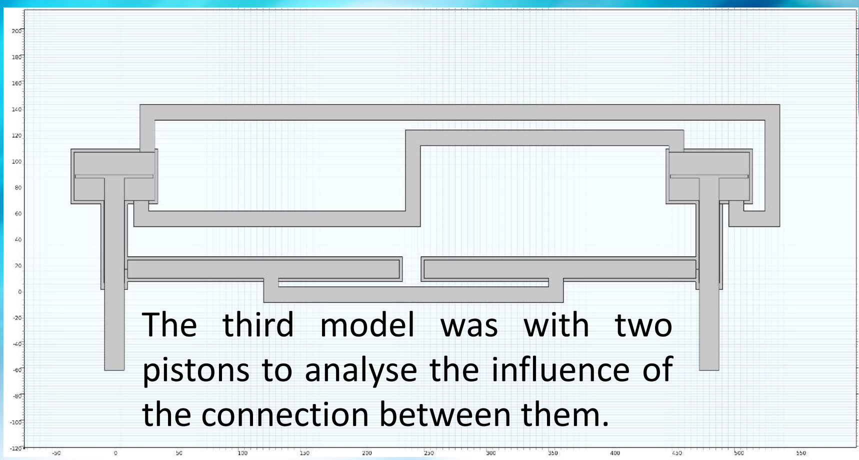
A first model in 3D of a single piston was first done

3. Theoretical analysis



The second model was again of one piston and in 2D, to ease the computational cost.

3. Theoretical analysis



The third model was with two pistons to analyse the influence of the connection between them.

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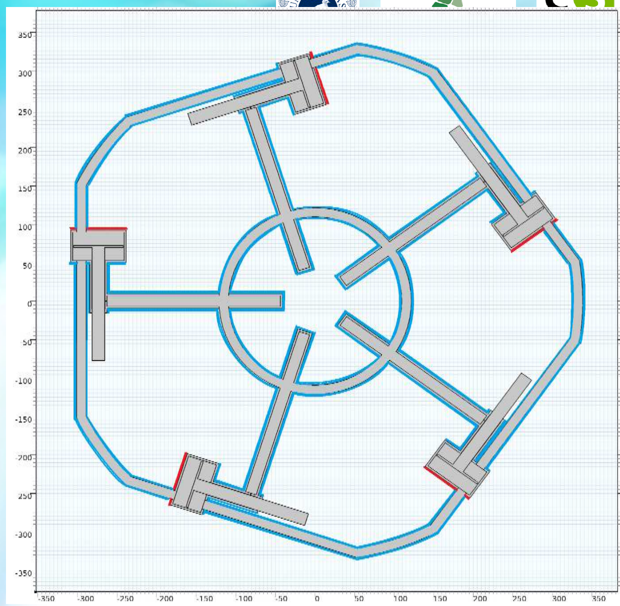
3. Theoretical analysis

Fourth and last model has 5 pistons to fully analyse the connection and its effect in the pressure.



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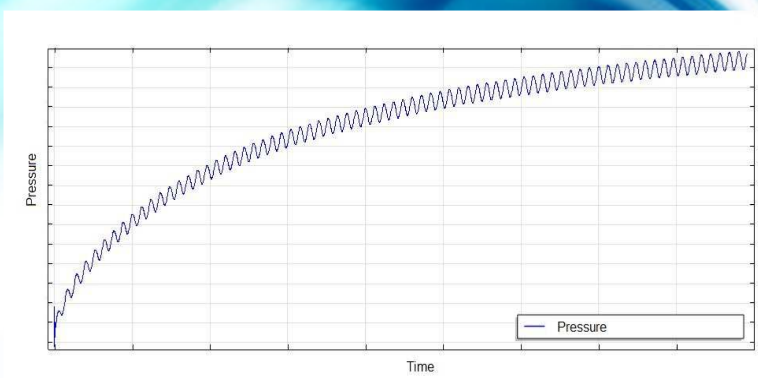
3.1. Calculation and results

- Results obtained in the different models analysed confirmed the theory.
- Representation of the variation of the pressure in the connecting channel in one of the models.

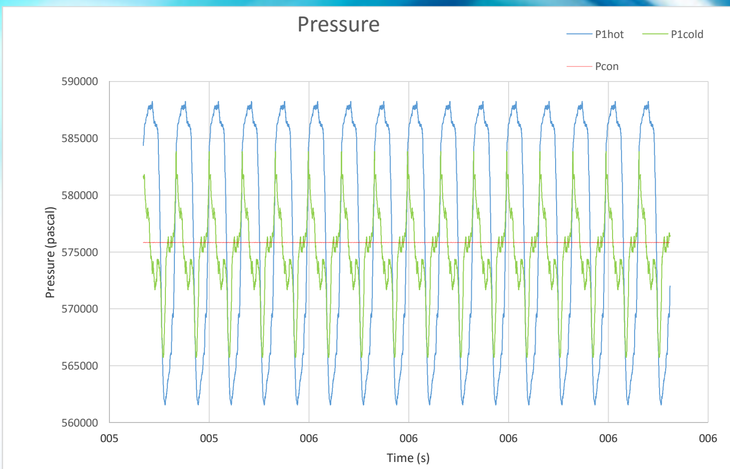


3.1. Calculation and results

Pi	Photmax	Pcoldmax	Pconnec
0.2	0.58	0.55	0.58
0.55	1.12	0.81	0.93
1.07	1.56	0.90	1.13
1.3	2.02	1.11	1.45

Megapascals

- Final results in the 5 pistons model with different initial pressures.



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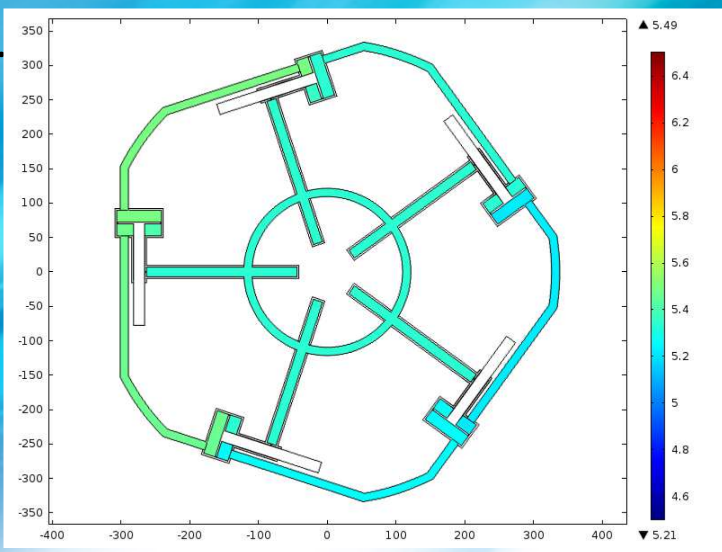

- Representation of the pressure in one specific time for the 5 pistons.

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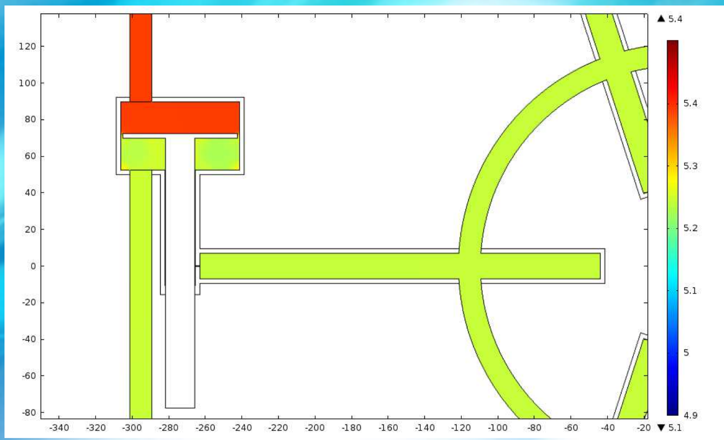
3.1. Calculation and results



- Representation of the pressure in one specific time for the 1 piston.

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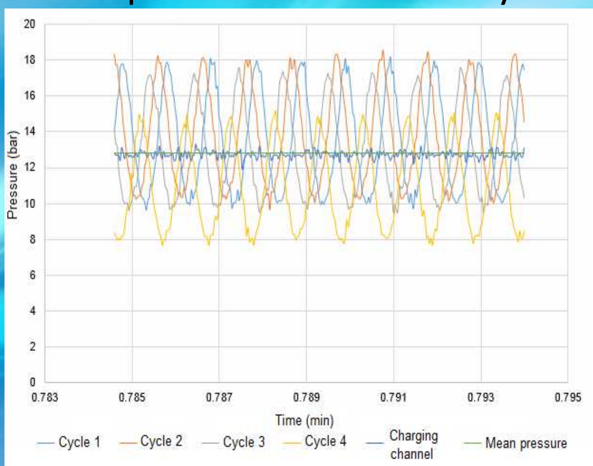
4. Experimental analysis and discussion



- Pressure values obtained in the real engine while functioning.

5. Conclusions

- A non-conventional five piston double-acting Stirling engine filling system is presented. Innovation introduced: no moving mechanical parts which allows identical pressure in all pistons easily.
- A one-cylinder model has been considered studying the influence of the hole geometry in the resulting pressure drop. The model has been modified to adjust to real engine until five cylinders where considered.

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5. Conclusions

- Main conclusion: each working volume work independently from the others and from the common volume of the filling system, which remains at constant pressure. Thus, this common volume does not actuate as a dead volume with its dramatic influence in the engine performance.

6. Future proposals

- Analyse different geometries of the connecting hole of the filling channel and the piston.
- Analyse more different initial pressures.
- Analyse different geometries of the filling system.

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7. Acknowledgements

- Universidad de Málaga. Programa de Becas de Iniciación a la Investigación 2015.
- Centro Stirling of the Mondragón Corporation.

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Thermo- & CFD Modelling of Stirling Machines and Components

A NON-CONVENTIONAL FIVE PISTON DOUBLE-ACTING STIRLING ENGINE FILLING SYSTEM CFD SIMULATION AND EXPERIMENTAL VALIDATION

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Keywords: Stirling, filling system, pressure drop, CFD.

ABSTRACT

An even filling pressure for different working fluid spaces in multiple piston double-acting Stirling engines is a key factor from the point of view of mechanical parts durability and homogeneity of the generated electrical power.

In this paper, a non-conventional five piston double-acting Stirling engine filling system is presented. This filling system has no moving mechanical parts, such as electrical valves or non-return valves, and it allows an even filling process for the different working volumes in the engine from just one point. This filling system has been designed so that each thermodynamic cycle volume is connected through a small hole to a common volume for all of them, which is connected at the same time to the mentioned filling port of the engine.

During runtime, these five connecting small holes, make the working volumes independent from each other and from the common volume of the filling system. Thus, this common volume does not actuate as a dead volume with its dramatic influence in the engine performance.

The sizing of this connecting hole has been analysed numerically. Firstly, a one-cylinder model has been considered studying the influence of the hole geometry in the resulting pressure drop. Once this model has been fully studied, a more elaborated model involving two cylinders has been studied to analyse the influence of the common filling volume. And last, another model including the five cylinders has been simulated and validated by means of experimental tests in a laboratory prototype.

This study has resulted in a patent application (P201530956).

INTRODUCTION

The durability of mechanical parts of any multi-cylinder Stirling engine is highly influenced by the smoothness level of the resulting torque in the shaft, which depends at the same time on how well the inertial forces are balanced and how homogenous the mean pressure for different working fluid spaces is.

It is important that the total mass of the working fluid contained in each enclosed volume of the engine is maintained to have closely equal masses of working fluid [1]. This is necessary to prevent average pressure differences between the enclosed volumes that will produce force imbalances on the transmission mechanism and therefore a non-smooth resulting torque and non-homogeneous electrical power. In order to prevent that, it is mandatory to assure an even filling pressure for different working fluid spaces and a proper pressure equalization system during runtime.

For that reason, it is possible to affirm that an even filling pressure for different working fluid spaces is a key factor from the point of view of mechanical parts durability and homogeneity of the resulting torque and therefore, of the generated electrical power.

Although this matter is not of little importance in Stirling applications, not much information is available in the literature about it.

Most of the small power scale Stirling engines, which work at relatively low pressures, have a buffer space connected to the workspaces through piston rod seal´s clearances. In these cases, the filling port is located in the buffer space and the overall system is filled through it, as the crankcase is totally enclosed. However, higher power scale Stirling engines, which usually work at much higher pressures and have the crankcase open to the atmosphere, have more complex filling systems comprising electrical valves or non-return valves.

The WhisperGen unit is a 1kWe micro-CHP system based on Stirling technology, whose engine is completely enclosed together with the transmission mechanism and the electrical generator. The filling system of this engine is quite simple, as it is composed of just a non-return valve set on the crankcase, which is used to fill the buffer space and the workspaces of the Stirling engine.

The Stirling engine manufactured by Stirling Biopower, however, have a more complex valve system that allows, on one hand, to fill the engine and on the other hand, to balance the pressure of the working fluid between the cycle volumes. This system allows an internally defined volume within the engine to be held at a low pressure near the minimum pressure of the cycle volumes. The valve may further be actuated to unload the engine to provide low starting torque and to unload the engine in case of an engine or load malfunction. The control valve further provides a controlled leakage path for working fluid flowing between the cycle volumes and the minimum pressure volume, which acts as part of a pressure balancing system for the Stirling engine which maintains the volume or mass of a working fluid in a balanced condition between the cycle volumes [2].

In this paper, a non-conventional five piston double-acting Stirling engine filling system is presented. This filling system has no moving mechanical parts, such as electrical valves or non-return valves, and it allows an even filling process for the different working volumes in the engine from just one point. The filling system presented in this paper has been designed so that each thermodynamic cycle volume is connected through a small hole to a common volume for all of them, which is connected at the same time to the mentioned filling port of the engine. This new design has resulted in a patent application (P201530956).

Experimental results show that the pressure of this common volume remains constant and equal to the mean pressure of the five thermodynamic cycles during engine operation. Thus, this common volume does not actuate as a dead volume with its dramatic influence in the engine performance.

In order to analyse gist of this fact, several CFD simulations have been carried out. Firstly, a one-cylinder model has been considered studying the influence of the hole geometry in the resulting pressure drop. Once this model has been fully studied, another model including two cylinders has been studied. Then, a more elaborated model including the five cylinders has been simulated.

COMSOL Multiphysics 4.4 software has been used to perform the simulations. Several references can be found in the literature where this simulation software has been used in order to simulate Stirling engine systems and subsystems [3] [4].

ANALYSIS BASE MODEL

The studied engine is a five piston double-acting Stirling engine. One of the features that makes this engine special is the filling system. It has been designed so that each thermodynamic cycle volume is connected through a small hole to a common volume for all of them, which is connected at the same time to the filling port of the engine, see Figure 1. The diameter of these small holes is 0.4mm and the charging channel has a length of approximately 210mm. The common volume is a circular channel, which connect the five small holes of 0.4mm of diameter and the filling port [5].

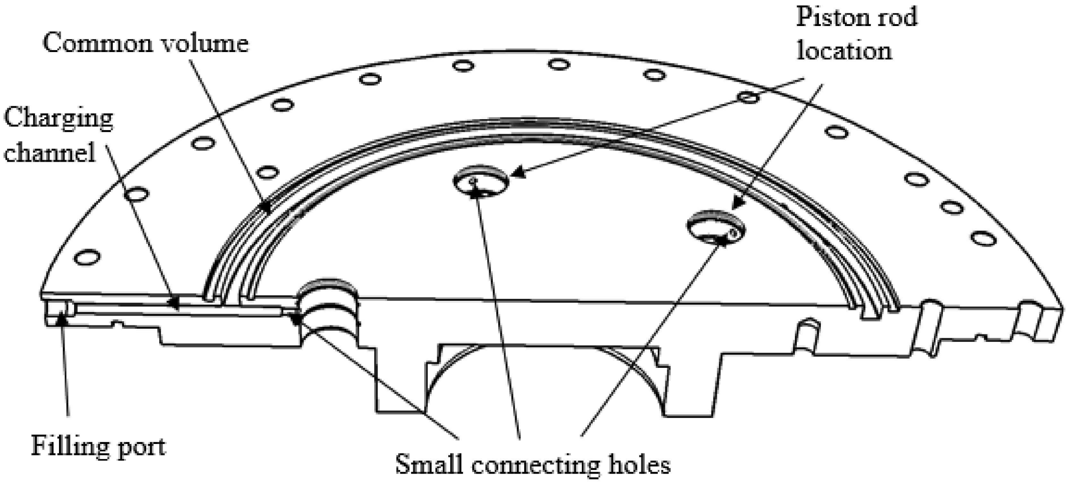

Figure 1. Scheme of the particular piece of the new engine presented.

Physics in the theoretical model

For the analysis of the theoretical model, different physics have been used in the software COMSOL Multiphysics.

As the Stirling engine works due to a temperature difference between the hot and cold ends, the heat transfer has been considered. The physics module in COMSOL analyses the heat transfer in solids (conduction through the wall) and heat transfer in fluids (convection in the working fluid). The COMSOL module uses the following equations (including Navier-Stokes equation):

$$\rho\frac{\partial u}{\partial t}+\rho(u\cdot\nabla)u=\nabla\left[-pl+\mu(\nabla u+(\nabla u)^T)-\frac{2}{3}\mu(\nabla u)l\right]+F \tag{1}$$

$$\frac{\partial\rho}{\partial t}+\nabla(\rho u)=0 \tag{2}$$

$$\rho C_P\frac{\partial T}{\partial t}+\rho C_P u\cdot\nabla T=\nabla\cdot(k\nabla T)+Q \tag{3}$$

Beside those equations, there are other equations to be applied for the fluid. These equations are related to the pressure work and the viscous heating. The *pressure work* is a way COMSOL Multiphysics consider the pressure of the fluid when it has an important impact in the performance of the model. On the other hand, the *viscous heating* considers the heat produced due to a viscous fluid moving. The equations related to this two “sub-physics” are:

$$\rho C_P \frac{\partial T}{\partial t} + \rho C_P u \cdot \nabla T = \nabla \cdot (k \nabla T) + Q + Q_{vh} + W_p \tag{4}$$

$$W_p = \frac{T}{\rho} (\frac{\partial \rho}{\partial T})|_p (\frac{\partial p_A}{\partial t} + u + \nabla p_A) \tag{5}$$

$$Q_{vh} = \mu \left(\nabla u + (\nabla u)^T - \frac{2}{3} (\nabla u) l \right) : \nabla u \tag{6}$$

It can be seen that the both physics are coupled in the Navier-Stokes equation, adding the specifics of pressure work and viscous heating in the fluid in the general study of heat transfer in the fluid.

The real performance of the model has been modelled completely considering these physics: the thermal exchanges (fluid and solid materials) and the pressure created while engine running. Thus, the results obtained in the modelling can be compared to the real results and therefore analyse the performance of the Stirling engine.

THEORETICAL ANALYSIS

The model used in COMSOL Multiphysics tries to reproduce the basis of the engine performance. As shown in the Figure 2, in a 3D model it has been reproduced the piston with its working volume with the charging channel drawn in scalable dimensions.

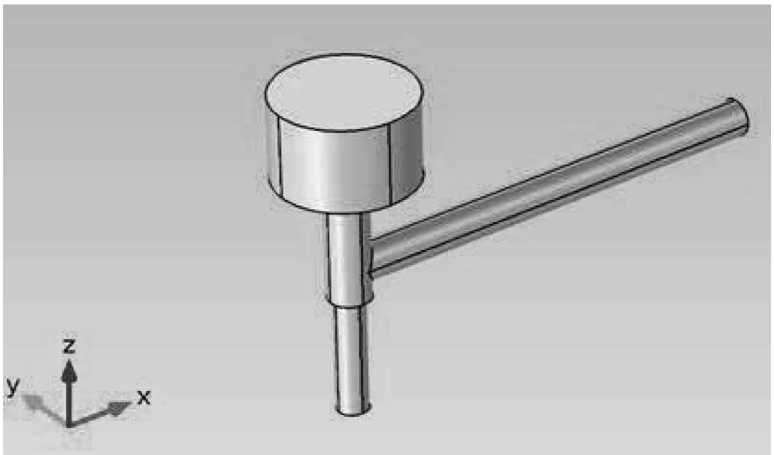

Figure 2. 3D model of one cylinder of the Stirling engine studied in COMSOL Multiphysics.

For faster analysis, a 2D model with one cylinder has been analysed, as mentioned in the introduction. The model is the one showed in Figure 3. This model includes the same features as the 3D model: a hot and a cold end, a moving piston and a charging channel. Every feature has been modelled, again, with scalable dimensions. After that model, another one with two cylinders has been modelled and is depicted in Figure 4. As can be seen, the connecting channels for simulating the working conditions have been added and, as before, the dimensions have been scaled.

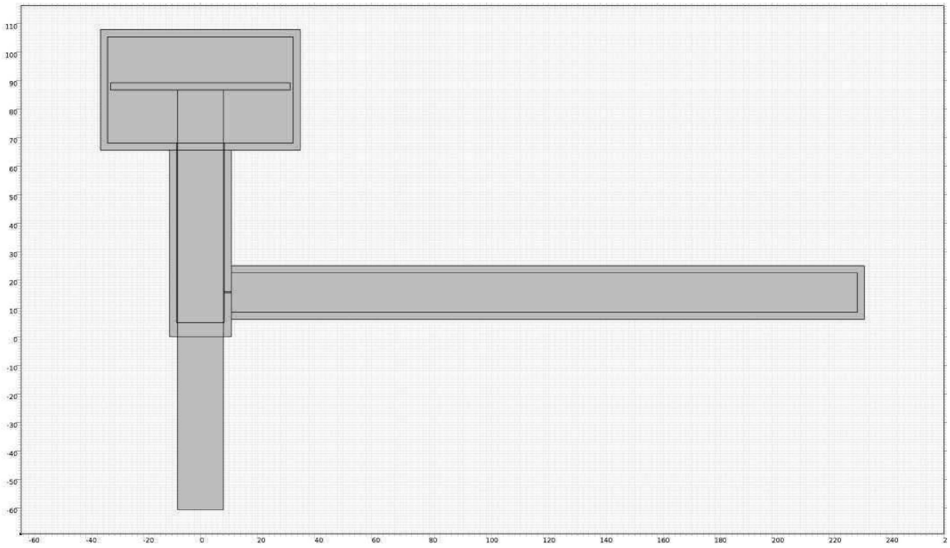

Figure 3. One cylinder 2D model in COMSOL Multiphysics.

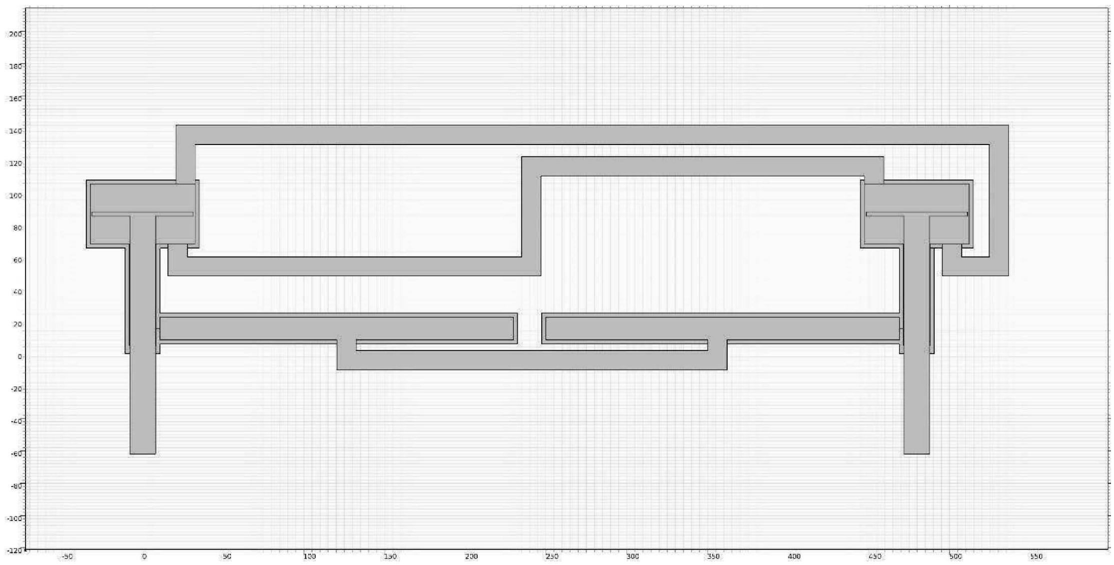

Figure 4. Two piston 2D Model (COMSOL Multiphysics).

The model has real dimensions for diameters and volumes, providing a good approximation of the real engine performance. In this two-cylinder model, a small channel has been created which joins the cold end of one piston with the hot end of the next one, recreating the working conditions in a single thermodynamic cycle.

Finally, a 5 piston double-acting Stirling engine has been modelled in 2D to fully represent the working cycles. As this model has been based on the two-cylinders one, again the real dimensions are considered and the phase angle for each cylinder with the following one is 72 degrees.

Boundary conditions for calculation

A heat flux has been set at the top part of the cylinder for the calculation and simulation. This heat flux corresponds to the one the working fluid gets from a heat source. It has been set a flux of *519 689.61W/m²* to take into consideration a 2000W per piston and a total of 10000W for the entire engine (5 pistons). As for the cold end, it has been set a heat flux release due to the convection effect of the cooling system water at 2m/s. The phase angle has been set to 72 degrees as mentioned before.

In Figure 5 both areas of heat flux have been indicated with different colours (red for heat influx and blue for heat outflow).

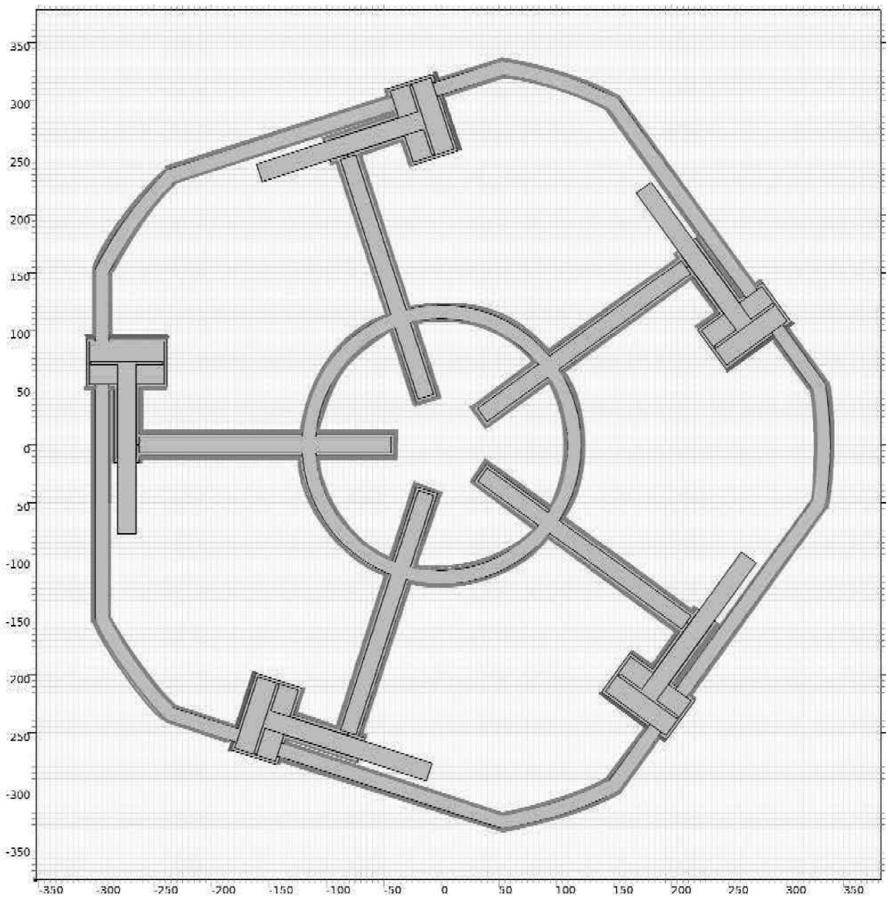

Figure 5. Model with two cylinders with the thermal boundary conditions applied.

The working rotational speed of the engine has been set to 1000 rpm, and the initial conditions of the working fluid to 0.2 MPa and 293 K. These values are equal to the charging values for the real engine. Other simulations have been carried out considering initial pressures of 0.55, 1.07 and 1.3 MPa as starting value of pressure in order to equal the mean pressure value achieved in the real engine.

Calculation and results

The simulation has been calculated enough time for the model to get a stabilisation, both in temperature and in pressure. The consistent results obtained with the one and two-cylinder models lead the investigation for fully analysis of the five-cylinder model.

The first conclusion obtained is that the pressure not fluctuates in each filling conduct. The different measures of the pressure with the one-cylinder model show that it fluctuates during the working of the engine. This fluctuation was of ±5% around the mean pressure in the stabilization.

Once the five-cylinder model was simulated, different starting pressures have been simulated and the results given are very similar to experimental values, where the pressure in the filling conduct does not fluctuate. These results lead to the conclusion that the constant pressure in the filling conduct is the result of the sum of the fluctuating pressures of the 5 individual filling conducts. In Table 1 the maximum values of the final pressure in different points of the engine in the stabilization with the different starting pressure are shown. For example, when the initial pressure is 0.2 MPa, the final pressure in the filling channels is 0.76 whereas if the initial pressure is 1.3 MPa, it reaches 1.45 MPa. This values demonstrate the importance of the initial conditions in the final status while runtime.

Table 1. Initial pressure values and their corresponding final pressure values in MPa.

Pi	Photmax	Pcoldmax	Pconnec
0.2	0.58	0.55	0.58
0.55	1.12	0.81	0.93
1.07	1.56	0.90	1.13
1.3	2.02	1.11	1.45

In Figure 6 is depicted an example of the variation of the pressure in the hot end of a piston and it can be appreciated the stabilization of the measurements in pressure.

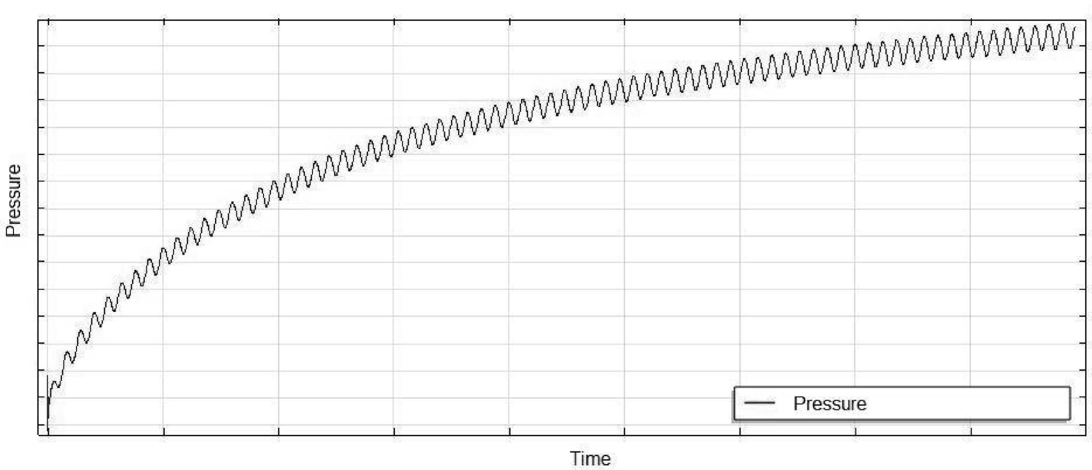

Figure 6. Example of the tendency of the pressure in the working fluid of the engine.

In Figure 7 an example of the stabilized results in one hot end, cold end and filling conduct is shown. The values of the pressure in the cold end fluctuates similar to a sinusoidal variation but with alterations in the value (green colour). This phenomenon is due to the connection with the hot end of the previous cylinder, which alters the pressure. The hot end pressure is a perfect sinusoidal variation as could have been expected due to the heat flux (blue colour) and, as previously commented, the pressure in the filling conduct is constant with small variations that are not appreciated in the scale of the graph (red colour).

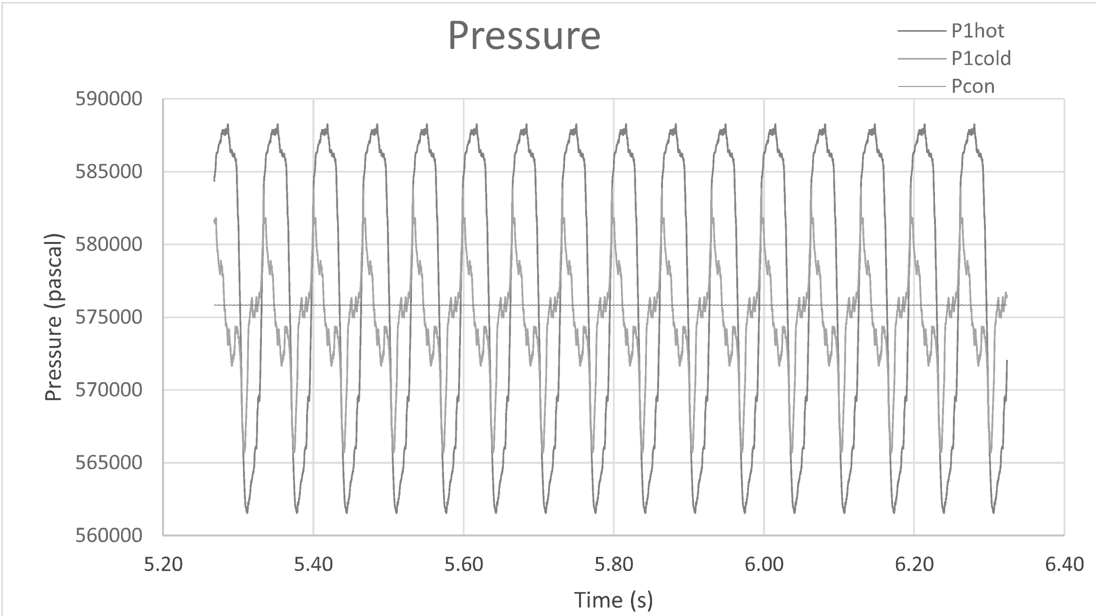

Figure 7. Representation of the final values in the stabilization in the hot and cold end and the filling channel for initial pressure of 0.2 MPa.

EXPERIMENTAL ANALYSIS AND DISCUSSION

5-piston double acting Stirling engine laboratory prototype has been used in order to carry out some experimental tests. The pressure variation of each working volume has been measured for 5 thermodynamic cycles during a certain running period, once the working temperatures have reached a stabilization. A sensor gauge has been placed in the cold end of each thermodynamic cycle and another one connected to the charging channel which is common for every piston.

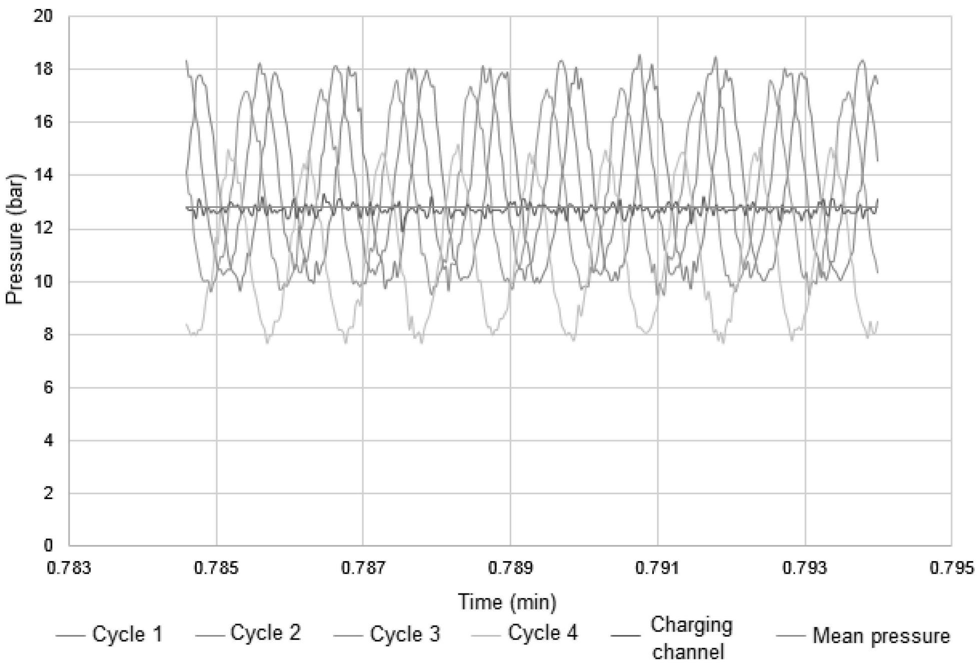

Figure 8. Pressure variaton of each thermodynamic cycle and charging channel.

The graph shown in Figure 8 is an example of the results obtained in the laboratory tests. These results correspond to a mean working pressure of 1.3 MPa approximately. Several tests have been performed for different charging pressures, but these have been selected as representative values in order to analyse the influence of the charging channel geometry.

On one hand, as shown in Figure 8, the pressure variation of Cycle 5 is missing. That is due to data acquisition problems during tests. Pressures of Cycle 1, Cycle 2 and Cycle 3 are similar; however, there is a significant difference compared with values measured for Cycle 4. It may be due to differences in the piston sealing performances or due to possible differences in reached temperatures in the hot and cold ends.

On the other hand, the phase angle of 72º for piston movement for 5 piston double-acting Stirling engine is also noticed in pressure curves, as it was expected.

Finally, it is important to highlight that the mean pressure of the five thermodynamic cycles is equal to the pressure measured in the charging channel. These results are consistent with the theoretical results where it is shown a constant pressure in the filling channel which is the result of the overlapping of the 5 oscillating pressures coming from the 5 thermodynamic cycles.

At first, it was thought that the reason that explained the fact that the pressure in the charging pressure remained constant was the high pressure drop due to the geometry of the small connecting holes of each working space; however, theoretical results of 2-piston model show that do exist a pressure oscillation in each filling conduct. This demonstrates that the constant mean pressure of the charging channel is not due to a high pressure drop in the channel, but due to the overlapping of the pressure oscillation of the five thermodynamic cycles.

CONCLUSIONS

In this paper, a non-conventional five piston double-acting Stirling engine filling system is presented. This filling system has no moving mechanical parts, such as electrical valves or non-return valves, and it allows an even filling process for the different working volumes in the engine from just one point. This filling system has been designed so that each thermodynamic cycle volume is connected through a small hole to a common volume for all of them, which is connected at the same time to the mentioned filling port of the engine.

A one-cylinder model has been considered studying the influence of the hole geometry in the resulting pressure drop. Once this model has been fully studied, a more elaborated model involving two cylinders has been studied to analyse the influence of the common filling volume. And last, another model including the five cylinders has been simulated and validated by means of experimental tests in a laboratory prototype.

The main conclusion of this study is that each working volume work independently from the others and from the common volume of the filling system, which remains at constant pressure. Thus, this common volume does not actuate as a dead volume with its dramatic influence in the engine performance. This effect has been shown by means of both theoretical CFD simulations and experimental tests in a laboratory prototype. The results demonstrate that the constant mean pressure of the charging channel is not due to a high pressure drop in the small connecting holes, but due to the overlapping of the pressure oscillation of the five thermodynamic cycles.

This study has resulted in a patent application (P201530956).

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NOMENCLATURE

C_p = heat capacity
F = any volumetric force applied to the fluid
k = kinetic turbulent energy
l = length
u = velocity field
p = pressure
p_A = pressure in a specific point
Q = heat transferred
Q_{vh} = heat from viscous heating
T = temperature
W_p = pressure work
ρ = density of the working fluid
μ = viscosity of the working fluid