

Simulation and testing of a solar reciprocating steam engine

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ABSTRACT

A solar thermal system for steam generation was built in the city of San Carlos, Salta province, Argentina. This system is a Linear Fresnel Collector (LFC) used for producing steam for many applications, such as vegetable drying, water desalination, and power generation. A steam engine was coupled to this LFC for power generation. This paper shows the steam engine simulation, where thermal and dynamic equations were developed and written on the Simusol program. Mechanical power output and other parameters of the engine were also simulated as well as steam consumption, in order to establish laws in relation with mechanical load on axis and regulate the rpm regime and inlet pressure for optimal steam engine operation. The results of the simulation of power output are compared with experimental measurements. The steam engine was tested, and the obtained experimental results demonstrate the feasibility of generating power (i.e. $2 \text{ kW} \pm 5\%$ (288 rpm) or $8 \text{ kW} \pm 2\%$ (400 rpm)). Furthermore, it is found that the experimental measurements are in an acceptable agreement with the simulation outcomes of the analytical model. Finally, knowing the steam engine behavior allows to optimize the solar energy resource use for power generation.

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1. Introduction

According to the British Petroleum Company [1], renewable energy sources used in power generation grew by 15.2% in 2016, slightly below the 10-year average growth of 15.9%. Globally, solar power generation grew by 32.6%. At the same time, solar steam generation projects have been growing significantly, with currently more than 300 solar plants around the world at different stages of development [2]. All of them include turbines in their power block. In 1981, a steam engine had been used to convert steam into energy at Australia's first ever solar farm in the remote town of White Cliffs in New South Wales [3], supplying the local hospital, school, post office and twelve homes [4,5]. Steam engine technology is mainly used in small thermal and electrical applications [6] because of their lower initial cost and better efficiency than that of certain turbines [7]. For this reason, the Institute of Non-Conventional Energy (INENCO) built a small-scale Linear Fresnel Reflector (LFR) in Argentina producing steam at pressures above 15 bars in which a double effect reciprocating steam engine with 7 HP output was installed to produce electricity [8]. The Fresnel generator produces

both steam to heat a drier and electricity to drive the fan in the drier [9]. San Carlos, Salta province, where the prototype was installed, receives high amounts of solar radiation, and direct normal irradiance (DNI) can reach maximum around 1000 W/m^2 .

Fig. 1 shows the steam engine and the solar Fresnel prototype. This Fresnel prototype was tested and analyzed using a real time numerical simulation with the Simusol program, developed by Saravia et al. [10].

In this paper, we explain the numerical simulation of the steam engine and discuss the main results. The output variables, such as temperature, pressure, steam fluxes, steam quality, rpm regime, among others, are showed. Real-time evolution of the thermodynamic state variables within the engine cylinder is described, and engine head thermal losses by convection and conduction are considered.

2. Background

The search for optimal technologies for thermoenergetic conversion dates back several decades. Badr et al. [11] studied differently combined heat-and-power (CHP) technologies, such as turbines, screw, rotary vane and Vankel expanders. They analyzed parameters such as power output, rotational speed, specific diameter and specific speed. As a result of their research, turbines

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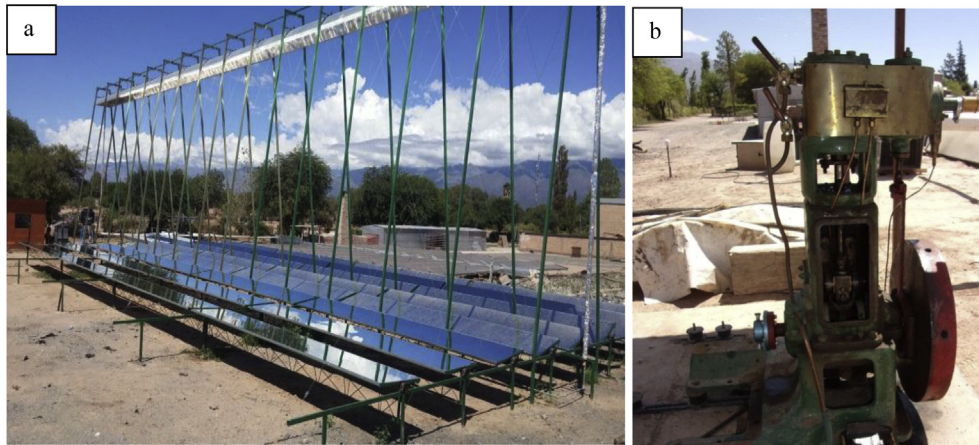


Fig. 1. a) Linear Fresnel system where steam engine is mounted. b) Lateral view of the steam engine.

showed better performance for use with steam, but high losses at the speed reduction boxes take place. This caused further losses in the net energy available as well as extra costs. For low rotational speeds, reciprocating engines need to be built with larger dimensions than the other technologies and are less sensitive to wet steam than turbines.

Turbine should not be used as expander for small-scale power generation due to the complexity and costs [12]. Therefore, most of the researchers have been scoped on finding alternative expanders to solve efficiency problems [13]. The most alternative volumetric expanders are of piston, screw, scroll and hermetic turbo generator types [14–18].

In small-scale renewable systems (power output < 50 kW), reciprocating steam engines have begun to gain importance for power generation because they simplify the engine architecture and the valve distribution system. Badami and Mura [19] preliminarily designed a small-scale wood waste Rankine Cycle with a reciprocating steam engine and studied its main characteristics and operative performance. They concluded that variable speed strategy would be the best solution considering the trade-off between efficiency and complexity.

Chan et al. [20] reviewed and compared different low-temperature technologies that were suitable for waste heat energy. They focused on low-grade heat, including chemical heat pumps, such as adsorption and absorption cycles for cooling and heating; and thermodynamic cycles, like organic Rankine cycle, supercritical Rankine (SRC) cycle, and trilateral cycle (TLC). They realized that both SRCs and TLCs have the potential for better energy and exergy performances in order to produce electricity.

Zhang et al. [21] studied a single-valve reciprocating air engine and proposed its simulation. Its engine performance, flow loss mechanism, and internal flow structure were studied in detail. In the cylinder, mass-averaged pressure and temperature were analyzed through the simulation, where the flow structure and characteristics were shown. Results of this work show that the flow speed is very high due to the pressure difference during the air intake period and that it causes strong vortex flow inside the cylinder, leading to a big loss of total pressure.

A recent work presented by Tenissara et al. [22] was performed about the simulation, construction, and evaluation of cheap piston expander whose working fluid is compressed air for low-pressure power generation. This research presents a piston expander by modifying diesel engine which can work at pressures around 1–10 bars. This work has 3 contribution parts: mathematical modeling, designed expander, and the evaluation which shows that

at pressure ratios 3, 4 and 5 at average speed of 683 rpm (optimal speed), break powers are 160 W, 257 W, and 431 W.

Other experimental studies were carried out by Kanno et al. [23] on a steam engine in a single microtube where piston was a portion of water contained in the cylinder. It worked at temperature around 230 °C (expansion) and the fluid was cooled at 80 °C (contraction). The microtube steam engine had a diameter of 1 mm and achieved indicated work of 1 W with a cycle efficiency of 5%. Pressures and volume changes are shown in their studies, where pressure profile presents almost the same pattern for each cycle. These parameters were measured in the experimental prototype of the steam engine. The liquid-engine efficiency was calculated by defining experimental parameter conditions, geometrical configuration, and energy balance.

Willich et al. [24] presented a detailed computational-fluid-dynamic study of the thermodynamic losses associated with heat transfer in gas springs. These calculations were performed on OpenFOAM software and took into account convective and wall conductive mechanism for thermal losses.

Most articles found in the literature involve the simulation and testing of Stirling engines with different energy sources for their operation [25]. Moreover, studies and numerical simulations on double-effect steam engines connected to small-scale LFR were not found in the available literature. We propose a contribution that makes possible to know the behavior of this steam engine, both computationally and experimentally. This proposal will allow us to optimize the solar energy resource for power generation using our small-scale Linear Fresnel technology.

3. The model

The steam flows from the LFC, alternately to each cylinder, so that while in one of them the admission occurs, in the other the exhaust takes place. This alternative steam inlet and outlet is automatically achieved by the same machine through the distribution valve. This distribution mechanism is arranged in a way that the steam processes are repeated periodically. So, timing is controlled by a mechanical piston valve that moves by a cam and camshaft, as was seen in Fig. 1b. Both movements of the camshaft and the main rod are 90° crankshaft out of phase. It allows closing the port in advance while the exhaust stroke occurs and begins the compression stroke.

Thermodynamic equations involve concepts of enthalpy, internal energy, pressure and density of the steam, which depend on the temperature and the steam fraction. This dependence is established

by sixth-degree polynomials obtained from the steam thermodynamic tables. Thermodynamic and mechanical equations also depend on the position of the crankshaft. There are four strokes for steam cycle, known as admission, expansion, exhaust, and compression. The position of the crankshaft is identified by an angle given by the mechanical Newton equations for both crankshaft and the piston. The thermodynamic and mechanical equations interact with one another while the steam volumes depend on the position of the piston.

3.1. Thermal model

For this modeling, saturated steam (steam fraction close to one) is considered. The used thermodynamic equations are explained separately in the following sub-sections.

3.1.1. Admission stroke

The purpose of this phase is to calculate the unknown steam variables: temperature θ and title x , as functions of time. Initially, the cylinder contains a steam mass m_0 corresponding to the volume V_0 at the position y_0 .

Fig. 2 shows a schematic model of the engine head and the main parameters used to describe the admission process. The steam mass and volume at time τ are $m_{(\tau)}$ and $V_{(\tau)}$, respectively.

The admission steam at temperature θ_2 , steam fraction x_2 , and velocity v_2 are introduced using an intake manifold with an area A_2 . The piston has an active area A_p . The admission steam density δ_2 can be calculated as:

$$\delta_2 = \delta_{l(\theta)}(1 - x) + \delta_{g(\theta)}x \quad (1)$$

where $\delta_{l(\theta)}$ and $\delta_{g(\theta)}$ are liquid and gaseous densities, respectively. The input steam mass is calculated as:

$$m_{(\tau)} = \delta_2 v_2 A_2 \tau + m_0 \quad (2)$$

and the steam volume is:

$$V_{(\tau)} = A_p (y_0 - y_{(\tau)}) \quad (3)$$

The steam density δ can also be calculated as a function of mass and volume or alternatively using the title as:

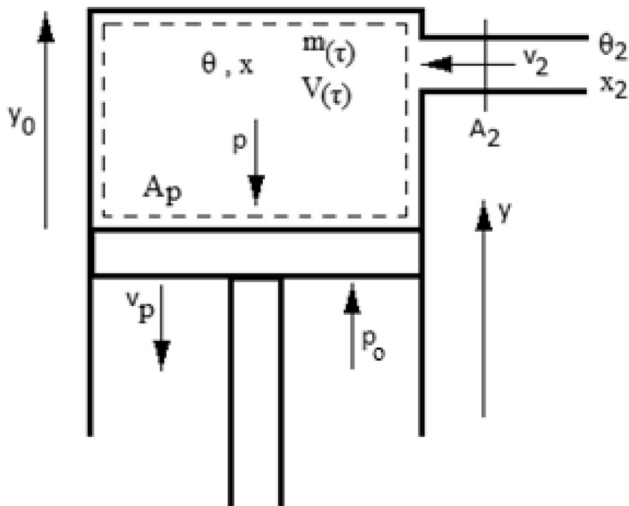


Fig. 2. Engine head schematic model.

$$\delta = \frac{m_{(\tau)}}{V_{(\tau)}} = \frac{\delta_2 v_2 A_2 \tau + m_0}{A_p (y_0 - y_{(\tau)})} = \delta_{l(\theta)}(1 - x) + \delta_{g(\theta)}x \quad (4)$$

This is the first equation expressing a relation between the two unknown variables θ and x .

The second equation is given by the energy balance, and the energy terms to be considered are:

- Steam's internal energy:

$$E_u = \frac{d}{d\tau} (m_{(\tau)} u_{(\theta, x)}) \quad (5)$$

- Admission steam enthalpy:

$$E_h = \delta_2 v_2 A_2 h_{(\theta_2, x_2)} \quad (6)$$

- Piston work output:

$$W_p = -p_{(\theta)} A_p v_p \quad (7)$$

- Heat losses:

$$Q_{l-adm} = (\theta - \theta_e) \left[\frac{2\pi(y_0 - y)}{\left(\frac{4}{h_c} + \frac{1}{\ln\left(\frac{d_o}{d_i}\right)} + \frac{1}{\ln\left(\frac{D_o}{D_i}\right)} \right)} + \frac{k_i A_p}{e} \right] \quad (8)$$

Heat losses through the piston were neglected. However, in Ref. [26], the thermal state of the entire engine was studied through a three-dimensional network of thermal nodes in which thermal transfer through the piston as well as piston's axle are taken into account. As input parameter, temperatures measured inside the cylinder (Fig. 15a–c) were taken into account. As result, the heat losses through the pistons represent approximately 7% of the total heat losses.

Then, the conservation energy law is:

$$E_u = E_h + W_p - Q_{l-adm} \quad (9)$$

This is the second equation expressing a relation between the two unknown variables θ and x .

Heat losses are determined by Q_{l-adm} where d_i is the inner diameter of the cylinder, d_o is the outer diameter and D_o is the outer diameter of head insulation, h_c is the convective coefficient of steam, k_i is the thermal conductivity coefficient for iron and k_{is} for insulation, e is the thickness of the cylinder lid, and θ_e is the environment temperature.

$d\theta/d\tau$ and x , can be obtained from the previous balance equations by algebraic manipulations:

$$\frac{d\theta}{d\tau} = \frac{E_h + W_p - Q_{l-adm} - E_u - m \frac{\Delta u_{gl}}{\Delta \delta_{gl}} \frac{d}{d\tau} \left(\frac{m}{V} \right)}{m \left[\frac{du_l}{d\theta} + \frac{d\Delta u_{gl}}{d\theta} x - \frac{\Delta u_{gl}}{\Delta \delta_{gl}} \frac{d}{d\theta} \left(\frac{m}{V} \right) \right]} \quad (10)$$

$$x = \frac{\delta_2 v_2 A_2 \tau + m_0 - \delta_{l(\theta)} (y_0 - y_{(\tau)}) A_p}{\Delta \delta_{gl}(\theta) (y_0 - y_{(\tau)}) A_p} \quad (11)$$

where:

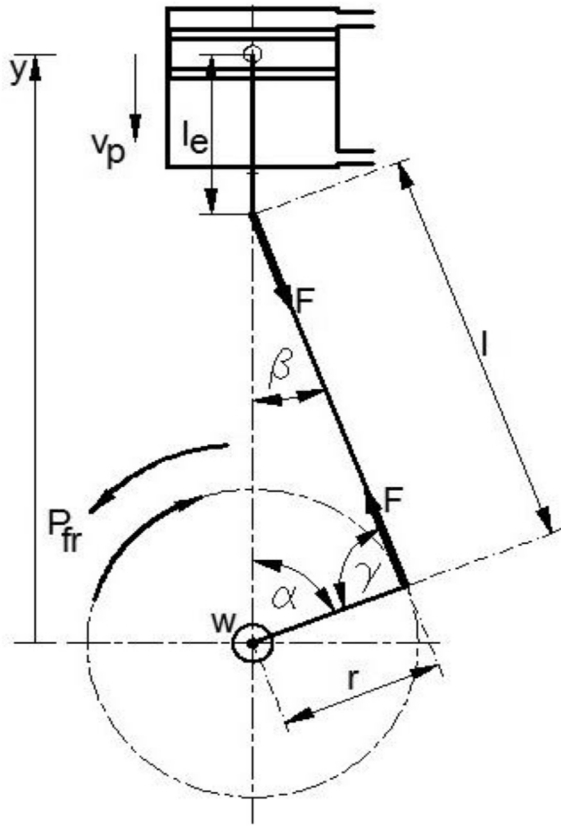


Fig. 3. Piston, rod, and crankshaft scheme system.

$$\Delta u_{gl} = u_g - u_l \quad (12)$$

$$\frac{d\Delta u_{gl}}{d\theta} = \frac{du_g}{d\theta} - \frac{du_l}{d\theta} \quad (13)$$

$$\Delta \delta_{gl} = \delta_g - \delta_l \quad (14)$$

u_l and u_g are liquid and gaseous internal energies.

Steam fraction is important for engine performance. It should be between 0.8 and 1 to avoid poor engine performance and component damage.

Through the time lapse of admission stroke, temperature and steam fraction are determined by Eqs. (10) and (11), respectively.

3.1.2. Expansion stroke

This phase starts immediately after the admission stroke; the steam produces work due to its high initial pressure and internal energy. Thermal losses are also considered at this stage. The energy balance equation is proposed as follows:

$$E_u = W_{exp} - Q_{l-exp} \quad (15)$$

where:

$$E_u = \frac{d}{d\tau} (m_{(\tau)} u_{(\theta, x)}) \quad (16)$$

$$W_{exp} = (p - p_o) A_p v_p \quad (17)$$

$$Q_{l-exp} = (\theta - \theta_e) \left[\frac{2\pi(y_0 - y)}{\left(\frac{\frac{4}{d_i}}{h_c} + \frac{\frac{1}{k_i}}{\ln\left(\frac{d_o}{d_i}\right)} + \frac{\frac{1}{k_{is}}}{\ln\left(\frac{d_o}{d_i}\right)} \right)} + \frac{k_i A_p}{e} \right] \quad (18)$$

Similarly to Section 3.1.1, the differential equation for temperature is given by the following expression:

$$\frac{d\theta}{d\tau} = \frac{-m \left(\frac{\Delta u_{gl}}{\Delta \delta_{gl}} \right) \left(\frac{m v_p}{A_p (y_0 - y)^2} \right) + W_{exp} - Q_{l-exp}}{m \left[\left(\frac{du_l}{d\theta} + x \frac{d\Delta u_{gl}}{d\theta} \right) - \left(\frac{\Delta u_{gl}}{\Delta \delta_{gl}} \right) \delta_l + x \frac{d\delta_{gl}}{d\theta} \right]} \quad (19)$$

The steam fraction can be calculated by Eq. (11), similarly to the admission stroke. Pressures p and p_o depend on the steam temperature of upper and bottom cylinders, respectively.

3.1.3. Exhaust and compression strokes

In the exhaust stroke, the steam flows to the atmosphere, leaving the cylinder. The piston moves from bottom dead center (BDC) to upper dead center (TDC), or vice versa, driving the steam through the exhaust pipe. Eqs. (4), (10) and (11) describe the steam evolution in the exhaust stroke but with a difference: a negative steam flow. Before the exhaust stroke ends, the exhaust port is closed and residual steam compression starts. The compression work modifies the internal energy, pressure, and temperature of the steam. This stage progresses until the intake port is opened again allowing steam to enter the cylinder and the cycle to start again. Eqs. (4), (10) and (11) can also be applied to describe the steam behavior as in the expansion stroke, but with a negative steam flow.

All equations are valid to describe the behavior of the steam in both upper and bottom cylinders. The subscript “o” (not to be confused with subscript “0” corresponding to the initial instant) was added to the parameters that belong to the bottom cylinder (e.g. p_o , y_o and θ_o) in the simulation.

Table 1 shows the numerical values for crankshaft angles for the beginning of a stage and the end of another in the engine operating cycle. Timing values given by crank angle variation, for the steam engine under study, are fixed from factory.

The values shown in Table 1 were determined in practice by turning the flywheel manually and observing the steam output through the purge valves of each cylinder. A significant difference can be observed in timing angles of expansion and exhaust strokes between both cylinders. These values were incorporated into the simulation by means of tables that depend on the angular position of the crankshaft. Therefore, eight tables were defined; one for each stroke and for each cylinder.

3.2. Mechanical model

To describe the dynamic movements of the piston, rod, and crankshaft, dynamic equations based on Newton's law are necessary. The pressure is obtained from thermal simulation and included in the dynamic formulation of the balance equations. Friction losses have not been considered so far, but they will be considered as a function of mechanical power output by mechanical efficiency. Dynamic effects of the components such as the connecting rod, which generates harmonics, are also disregarded. A scheme of the piston, rod and crankshaft system is shown in Fig. 3.

The steam pushes the upper face of the piston, lowering it from TDC to BDC. The backpressure under the piston is the steam pressure in the bottom cylinder p_o .

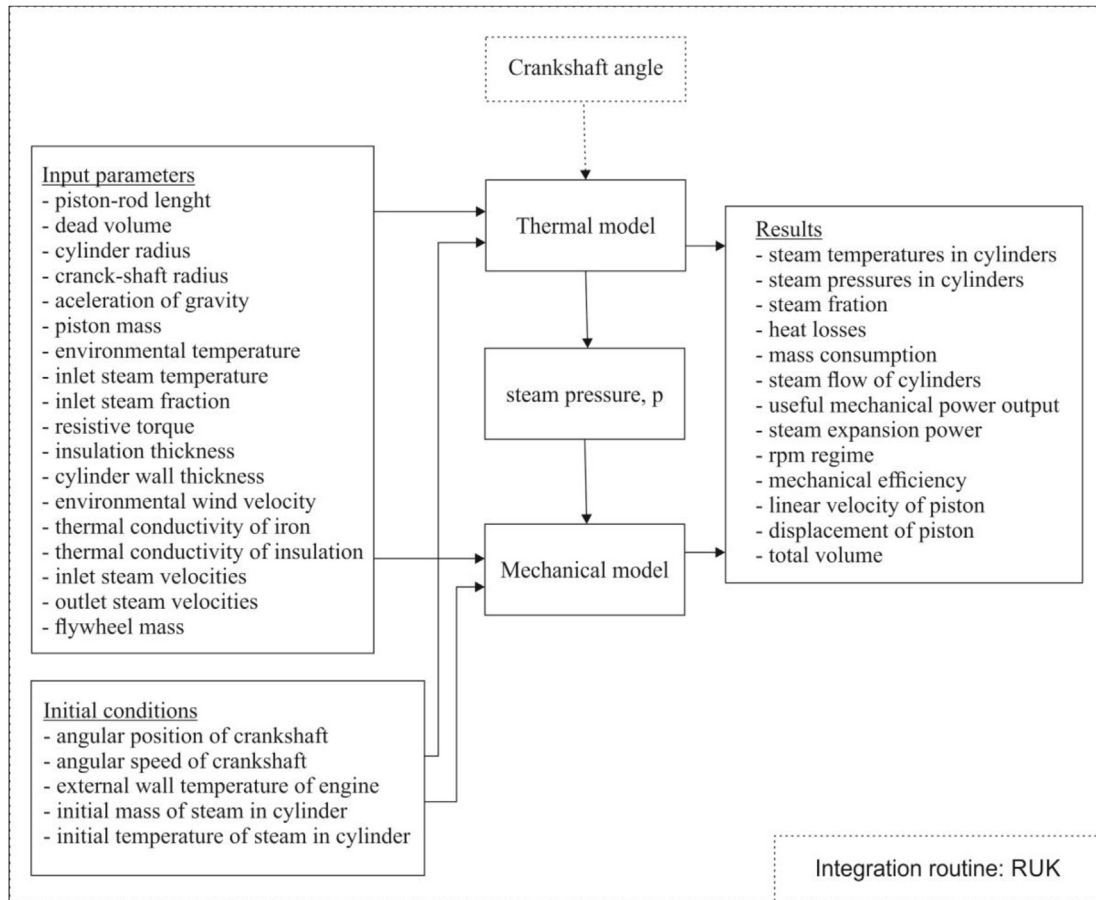


Fig. 4. Diagram of simulation and interaction between the different parts of it. An integration routine RUK type was used.

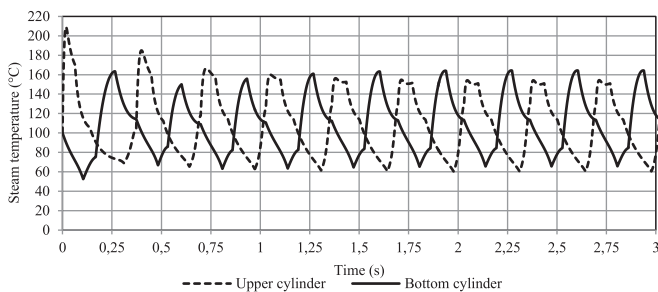


Fig. 5. Steam temperatures for upper and bottom cylinders.

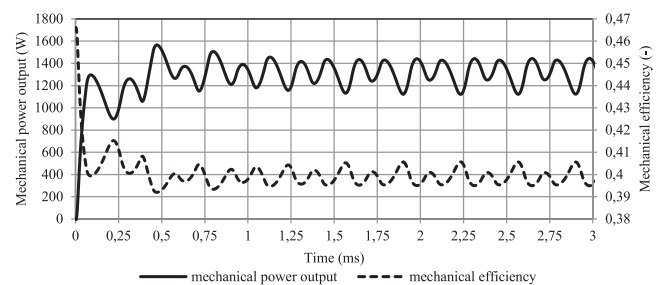


Fig. 7. Mechanical power output and mechanical efficiency.

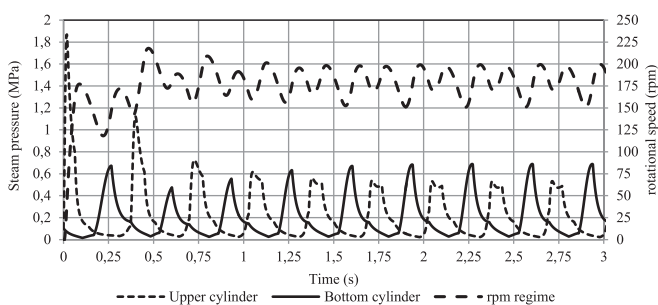


Fig. 6. Steam pressures for upper and bottom cylinders and rotational speed of crankshaft.

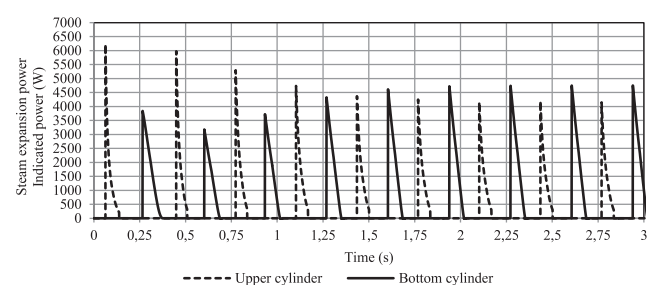


Fig. 8. Indicated power for expansion strokes in upper and bottom cylinders.

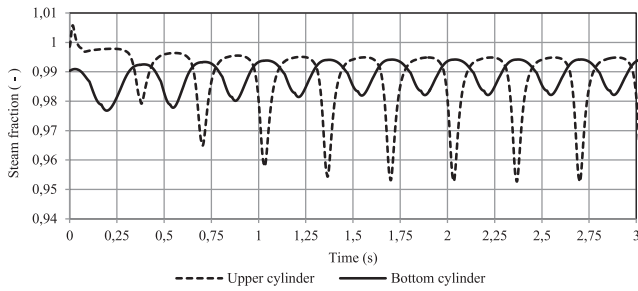


Fig. 9. Steam fraction.

The driving force F is due to pressure, and it is transmitted through the rod producing the torque which rotates the crankshaft.

The piston moves with a velocity v_p and the rod-crankshaft system transform the linear motion into a circular one with angular velocity w . The y variable represents the instantaneous position of the piston from the crankshaft axis. The α and β angles are the angular position of the crankshaft and rod, respectively. The γ angle allows us to calculate the torque on the axis of the steam engine.

The rod has a length l and the crankshaft r radius. The quotient of those two parameters, $\lambda=r/l$, has a value of 0.69 for our steam engine. The piston axis has a length l_e and it transmits the vertical movement to the rod. Regarding to the expansion ratio, many authors have defined this parameter in different ways. The first manuals [27] of steam engines define it as a relation between the stroke of the piston from the start of steam expansion and the total stroke or at the end of the expansion. Prasad [3] and Fu et al. [28] defined the expansion ratio based on steam volumes before and after expansion stroke takes place. Other more recent authors [29,30] preferred to define the expansion ratio taking into account pressures at those points of the expansion stroke. All of them conclude that engine parameters, for example valve lifts, engine speed, cylinder dimensions including valve timings affect power output and efficiency in different ways; high engine speed, high expansion ratio and cylinder insulation enhance performance. For a particular engine speed and cylinder dimensions, there may be an optimum timing, giving rise to an optimum expansion ratio. The expansion ratios for the steam engine under study are fixed at 2.3 and 2 for both upper and bottom cylinders, respectively.

The brake torque P_{fr} represents the opposite torque due to the mechanical resistance of the load on the mechanical axis.

According to Fig. 3, we can obtain a geometric relationship

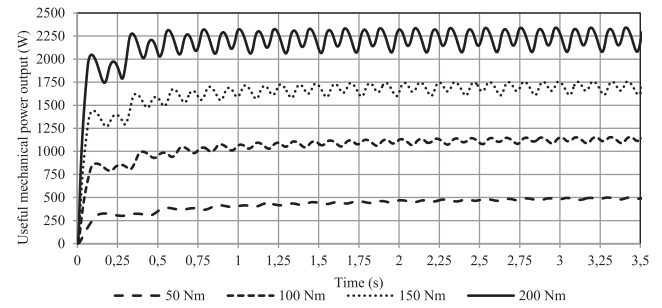


Fig. 11. Mechanical power output for different values of mechanical load, at 290 rpm regime.

between the various angles and lengths, as shown in Eq. (20):

$$\frac{y}{\sin \gamma} = \frac{r}{\sin \beta} = \frac{l}{\sin \alpha} \quad (20)$$

Position and linear velocity of the piston can be calculated with Eqs. (21) and (22):

$$y = l_e + l \sqrt{1 - \lambda^2 \sin^2 \alpha} + r \cos \alpha \quad (21)$$

$$v_p = \left(-l \frac{\lambda^2 \sin 2\alpha}{\sqrt{1 - \lambda^2 \sin^2 \alpha}} - r \sin \alpha \right) w \quad (22)$$

The piston linear acceleration can be calculated by differentiation from Eq. (22). However, in order to take into account the dynamic properties of the components that are the piston mass M and the crankshaft and flywheel inertia I , dynamic equations are proposed as follows:

$$M \frac{dv_p}{dt} = -Mg - (p - p_o)Ap - F \cos \beta \quad (23)$$

$$I \frac{dw}{dt} = -y \sin \beta F - P_{fr} \quad (24)$$

$$\sin \beta = \frac{r}{l} \sin \alpha = \lambda \sin \alpha \quad (25)$$

The force F can be removed by combining Eqs. (23) and (24):

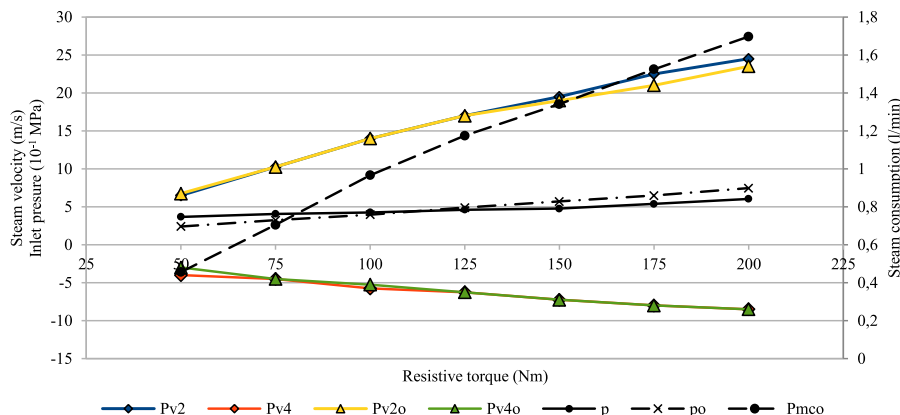


Fig. 10. Performance of steam engine for variable resistive torque at 290 rpm.

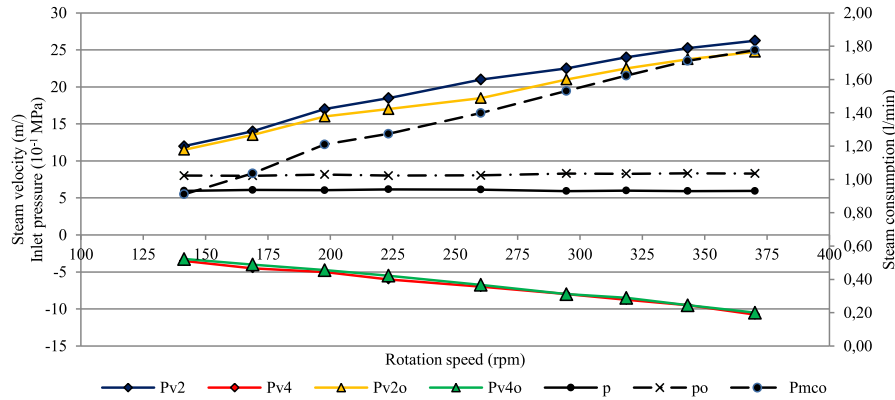


Fig. 12. Performance of steam engine for variable rpm regime at 175 Nm.

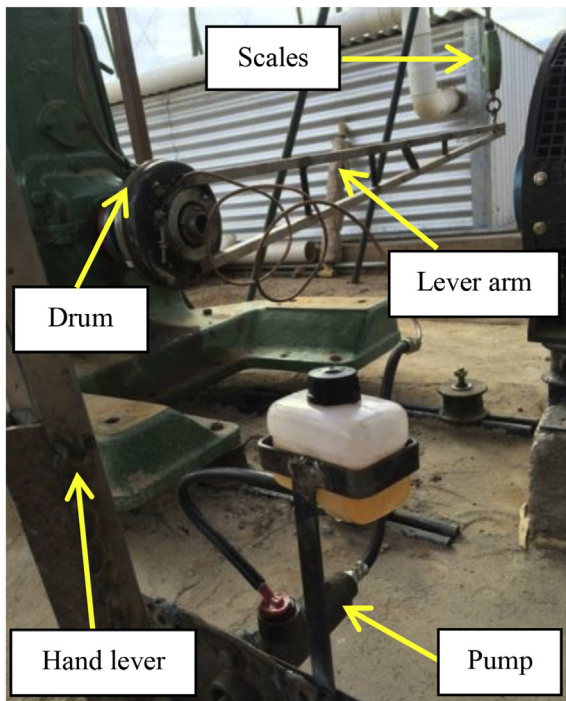


Fig. 13. Equipment for power output measuring.

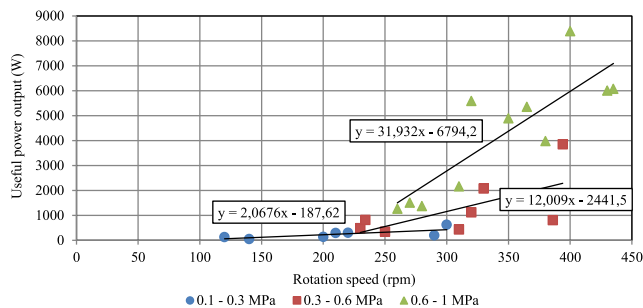


Fig. 14. Mechanical power output measures.

Eq. (26) concatenates linear parameters with rotatory ones. The angular acceleration of the crankshaft can be calculated with Eq. (27):

$$\frac{dw}{d\tau} = \frac{\frac{\cos \beta}{y \sin \beta} P_{fr} - M Pri3 w^2 - M g - (p - p_o) A p}{M Pri1 - \frac{\cos \beta}{y \sin \beta} I} \quad (27)$$

where:

$$Pri1 = -\frac{l \lambda^2 \sin 2 \alpha}{\sqrt{1 - \lambda^2 \sin^2 \alpha}} - r \sin \alpha \quad (28)$$

$$Pri3 = r \cos \alpha + r \frac{2 \lambda \cos 2 \alpha \sqrt{1 - \lambda^2 \sin^2 \alpha}}{1 - \lambda^2 \sin^2 \alpha} + \frac{\lambda^3 \sin 2 \alpha \frac{\sin 2 \alpha}{\sqrt{1 - \lambda^2 \sin^2 \alpha}}}{1 - \lambda^2 \sin^2 \alpha} \quad (29)$$

The useful power output of engine is calculated as follows:

$$N_u = \eta_{mec} P_{fr} w \quad (30)$$

$$\eta_{mec} = 9 \times 10^{-7} n^2 - 5.31 \times 10^{-4} n + 0.46 \quad (31)$$

η_{mec} is the mechanical efficiency that considers frictional losses due to superficial contact between mobile parts. Its value depends on rpm regime and was determined by experimental tests in laboratory [8]. For practical calculation, an average value of 40% can be considered for mechanical efficiency. Furthermore, the mechanical efficiency depends on lubrication and temperature of engine components. Frictional losses increase with poorer lubrication and higher temperature than in nominal conditions of operation.

4. The simusol software

Simusol is an open-access software developed by researchers of Instituto de Investigaciones en Energía No Convencional (INENCO). It runs on Linux Operative System as well as Windows and can analyze thermal, mechanical, and electrical systems. A graphical interface, Dia, is used to assemble diagrams and write equations. The output of simulation is function of time, which can be shown as graphs and a spreadsheet. Additional programming in FORTRAN can be added to the main program if a more advanced calculation is needed. For further information visit www.simusol.org.

$$\frac{M \frac{dv_p}{dt} + M g + (p - p_o) A p}{\cos \beta} = \frac{I \frac{dw}{dt} + P_{fr}}{y \sin \beta} \quad (26)$$

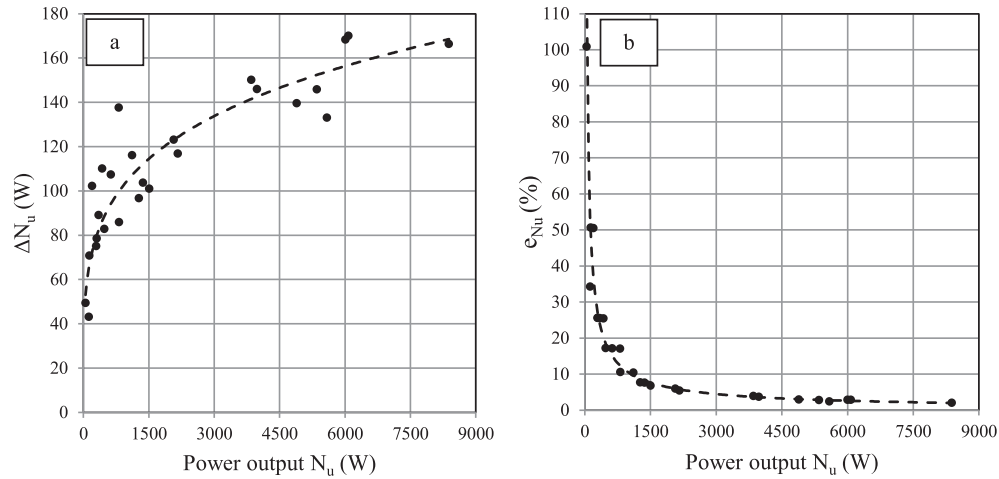


Fig. 15. Uncertainty in measurement for the entire experimental test of power output. a) Absolute error, ΔN_u ; b) relative error, e_{Nu} .

Table 1
Numerical values for the crank angle.

Cylinder	Admission	Expansion	Exhaust	Compression
Upper	76°	77°	146°	61°
Bottom	80°	95°	124°	61°

5. Results of simulation

Eqs. (1)–(33) were written on *Dia* graphical interface. Fig. 4 shows a diagram of interaction between mechanical and thermal model for both upper and bottom cylinders. Both models are related by steam pressure. It is necessary to define the initial conditions for all parameters such as steam temperature, angular velocity, steam mass, outside temperature of engine wall, steam fraction at admission pipe, etc.

Differential equations are solved by fourth-order Runge-Kutta method (RUK) [31] which is a simple, robust and good general candidate for numerical solution of differential equations when combined with an intelligent adaptive step-size routine.

Curves from Figs. 5–9, and 11 are obtained with a resistive torque of 175 Nm and a 186 rpm regime. Temperatures are given by Eqs. (10) and (19) and their variation are shown in Fig. 5.

The steam temperature changes with time and cyclic process; decreasing slopes correspond to the admission, expansion and exhaust strokes while increasing slopes are caused by the compression stage. Compression is produced by the thrust of the piston and closing of the exhaust valve in advance and it implies an increase in temperature.

Temperatures reach values of approximately 160 °C when the engine is in steady state. The minimum steam temperature for the upper and bottom cylinders is about 60 °C. This is because the expansions are completed along the piston stroke. The initial peak of steam temperature inside the upper cylinder is due to initial conditions for both angular position and angular velocity of the crankshaft, 45° and 10 rad/s, respectively. The simulation starts with the piston near TDC, descending towards the BDC. Consequently, the initial high effort decreases until the permanent regime is reached. This effort results in higher pressure inside the cylinder (Fig. 6).

There is a timing issue of the upper cylinder. The admission port opens prematurely when the compression stroke has not finished yet and the admission stroke in the upper cylinder begins. Meanwhile, in the bottom cylinder, this upper admission pressure

behaves as a back-pressure and it leads to a slight increase on pressure in the lower cylinder while exhaust is taking place. The variation in the pressure inside the cylinders produces an irregularity in the speed regime that is absorbed by the flywheel. The forces acting on the active faces of the piston have a periodic behavior, which implies that the rotational movement of the crankshaft and moving parts has variable acceleration. This behavior is common in this type of machine. The steam engine reaches the steady state in a few seconds; this is observed in practice when the steam engine is operating.

Eq. (30) gives the power output based on mechanical efficiency η_{mec} , brake torque P_{br} and angular velocity w . The waveform of power output is similar to rpm regime or angular velocity because brake torque (which represents the mechanical load on the engine axis) is fixed as input parameter in the simulation, and the mechanical efficiency is almost constant with the engine in permanent rotation. Both power output and efficiency are shown in Fig. 7.

The mechanical power developed by the expansion of the steam depends on the difference in pressures between the piston faces and the instantaneous velocity of the piston while expansion takes place in one of cylinders and exhaust occurs in the other. This mechanical power is known as *indicated mechanical power* and it is a representative parameter of steam engines. It can be calculated by Eqs. (32) and (33) for both cylinders.

$$N_i = -10^5(p - p_o) A_p v_p \quad (32)$$

$$N_{io} = 10^5(p_o - p) A_p v_p \quad (33)$$

Fig. 8 shows that indicated power reaches peak values between 4.2 kW and 4.7 kW. This is in accordance with the indicated power measured in laboratory tests [8] for a regime close to 200 rpm and working pressures between 0.62 MPa and 0.67 MPa (Table 2), where a regime of 186 rpm corresponds to an indicated power of 5.9 CV (4.35 kW).

The LFR mounted in San Carlos, Salta generates dry saturated steam that allows us to put the steam engine into operation. The title of the steam used to supply the engine must be optimum, close to the unity. Fig. 9 shows the variation of steam fraction x , which is calculated by Eq. (11) for both cylinders. For the proposed simulation, the steam fraction remains very close to unity. This is because the initial temperature of the outer wall of the engine is set up at 100 °C. In practice, the engine head is preheated by circulation of the steam through the cylinders and the preheating steam is

conducted to the outer environment through the purge valves.

The initial peak of the steam fraction for the upper cylinder takes place due to the high pressure (1.8 MPa) and temperature (210 °C) that correspond to the initial position of the piston. It is well-known that steam fraction cannot be higher than unity but the reason why steam fraction is just above 1 is that the simulation is not able to determine whether the engine is working with dried steam or superheated steam at the beginning. However, results of the simulation of the steam fraction are considered satisfactory.

Lower values for steam fraction are achieved at the end of the expansion stroke when the steam develops mechanical work and condensates. Fig. 9 shows that the steam fraction at the end of the expansion stroke is smaller in the upper cylinder than in the bottom cylinder. This is because the expansion period in the upper cylinder is longer than in the bottom cylinder. However, the difference between the two curves is insignificant.

5.1. Resistive torque dependence

Equations were run on Simusol by changing the resistive torque in each simulation from 50 Nm to 200 Nm. It was necessary to adjust the inlet and outlet speed of the steam for both cylinders due to this variation, to reach a regime close to 290 rpm. Mechanical power output depends mainly on axial load regime and mechanical efficiency. Steam velocities are shown in Fig. 10, where all the curves have a rising trend while the load increases; a higher load needs higher steam velocity (or steam flow). Curves in Fig. 10 represent the following parameters:

- Pv2: Inlet steam velocity in upper cylinder.
- Pv4: Outlet steam velocity in upper cylinder.
- Pv2o: Inlet steam velocity in bottom cylinder.
- Pv4o: Outlet steam velocity in bottom cylinder.

Steam consumption can also be calculated taking into account the steam that enters the cylinders in admission strokes and it does not consider steam leaks through seals. Its variation related to resistive torque is also shown in Fig. 10.

Mechanical power output also depends on resistive torque, as shown in Fig. 11, where an increasing trend with increasing load can be noted. It is useful mechanical power that the engine develops at 290 rpm. Temporal variation of useful mechanical power output is shown for different values of resistive torque. It can be noticed that for higher loads on the mechanical axis, higher oscillations take place on rpm waves because the force that drives the piston is

higher.

Table 3 summarizes results from simulation for variable resistive torque and constant rpm regime. It can be noticed that pressures as well as temperatures must increase to supply mechanical power output.

5.2. Rpm regime dependence

In this case, the simulation was carried out for a variable rpm regime with a constant resistive torque (175 Nm). Fig. 12 shows that there is no practically differences in steam consumption, as well as steam velocities, with a slight increasing in the low rpm zone. Inlet pressures in both cylinders remain constant and it is in accordance with the fact that the torque in engine's axis is related to the transmitted force by the connecting rod, and consequently to the pressure inside the cylinders.

As seen in Table 4, power output varies with rpm and steam flow, for a given axis load. This could represent a problem for power electric generation due to rpm regime could be different from the nominal one (50 Hz).

Knowing the variation of steam velocities required by the engine allows us to determine the calibration curves of the steam consumption for different axial load regimes. Equations for regulation of steam velocities and flow are shown in Table 5a–b.

The analysis of steam consumption and power output for variable load and constant rotation regime (or vice versa) is carried out. Curves in Figs. 10 and 12, as well as their regression equations (Table 5) and reference values of pressures and temperatures (Tables 3 and 4), are useful to manipulate such variables and perform the dispatch of power electric in the power block.

6. Experimental setup

Two experimental tests were carried out; the first one was to measure the useful power output by a torque-meter and the second one was to measure temperature of the steam inside of the upper cylinder. The design details of the engine and main results of the test engine are given in Table 6.

6.1. Useful power output

The steam engine was mounted in the LFR's installation and tested. The resistive torque was produced by a mechanical brake (Fig. 13). For the measurement of mechanical power, a brake was made and a lever arm was used to conduct the braking force, which

Table 2

Operation values obtained from laboratory tests of engine characterization.

Regime	rpm	45	65	96	116	136	200	206	230	246	270	288
Indicated power	CV	0.59	1.55	3.06	2.81	4.33	6.37	6.89	7.69	5.88	7.82	6.88
Working pressure	MPa	0.22	0.41	0.62	0.48	0.62	0.67	0.72	0.72	0.52	0.63	0.57

Table 3

Summary of simulation result for variable torque resistive and constant rpm.

P_{fr}	n	N_u	p	p_o	θ	θ_o	P_{mco}	$Pv2$	$Pv4$	$Pv2o$	$Pv4o$
Nm	rpm	W	MPa		°C		l/min	m/s			
50	261	530	0.37	0.24	135.24	126.40	0.46	6.50	−4.00	6.75	−3.00
75	281	850	0.41	0.32	139.03	135.39	0.70	10.25	−4.50	10.25	−4.50
100	291	1176	0.43	0.39	146.10	143.55	0.97	14.00	−5.75	14.00	−5.25
125	291	1412	0.46	0.48	147.37	157.05	1.17	17.00	−6.25	17.00	−6.25
150	290	1673	0.48	0.57	150.29	151.23	1.34	19.50	−7.25	19.00	−7.25
175	291	2057	0.54	0.65	154.77	162.02	1.52	22.50	−8.00	21.00	−8.00
200	291	2349	0.60	0.75	159.20	167.57	1.69	24.50	−8.5	23.50	−8.50

Table 4

Summary of simulation result for variable rpm resistive and constant torque.

P_{fr}	n	N_u	p	p_o	θ	θ_o	P_{mco}	P_{v2}	P_{v4}	P_{v2o}	P_{v4o}
Nm	rpm	W	MPa		°C		l/min	m/s			
175	141	1058	0.59	0.80	158.71	170.51	0.91	12.00	−3.50	11.50	−3.25
175	169	1238	0.61	0.80	159.47	170.48	1.04	14.00	−4.50	13.50	−4.00
175	198	1433	0.60	0.82	159.18	171.31	1.21	17.00	−5.00	16.00	−4.75
175	223	1601	0.62	0.80	159.95	170.57	1.27	18.50	−6.00	17.00	−5.50
175	260	1849	0.61	0.80	159.74	170.65	1.40	21.00	−7.00	18.50	−6.75
175	295	2085	0.59	0.83	158.45	172.01	1.53	22.50	−8.00	21.00	−8.00
175	319	2257	0.59	0.83	158.88	171.81	1.62	24.00	−8.75	22.50	−8.50
175	343	2440	0.59	0.83	158.45	172.09	1.71	25.25	−9.50	23.75	−9.50
175	370	2650	0.59	0.83	158.64	171.99	1.78	26.25	−10.75	24.75	−10.50

Table 5a

Adjustment curves for control of steam engine for variable torque resistive and constant rpm.

Parameter	Description	Equation of regulation	Determination coefficient R^2
P_{v2}	Inlet steam velocity in upper cylinder	$P_{v2} = -0.0001 P_{fr}^2 + 0.0978 P_{fr}$	0.9959
P_{v4}	Outlet steam velocity in upper cylinder	$P_{v4} = -0.0002 P_{fr}^2 + 0.1554 P_{fr}$	0.9943
P_{v2o}	Inlet steam velocity in bottom cylinder	$P_{v2o} = 0.0002 P_{fr}^2 - 0.073 P_{fr}$	0.9542
P_{v4o}	Outlet steam velocity in bottom cylinder	$P_{v4o} = -0.0001 P_{fr}^2 - 0.0654 P_{fr}$	0.9957
P_{mco}	Steam mass consumption	$P_{mco} = -0.00009 P_{fr}^2 + 0.0103 P_{fr}$	0.9974

Table 5b

Adjustment curves for control of steam engine for variable rpm and constant torque resistive.

Parameter	Description	Equation of regulation	Determination coefficient R^2
P_{v2}	Inlet steam velocity in upper cylinder	$P_{v2} = -0.00007 n^2 + 0.1493 n$	0.9967
P_{v4}	Outlet steam velocity in upper cylinder	$P_{v4} = -0.00001 n^2 - 0.023 n$	0.9970
P_{v2o}	Inlet steam velocity in bottom cylinder	$P_{v2o} = 0.00005 n^2 + 0.0811 n$	0.9547
P_{v4o}	Outlet steam velocity in bottom cylinder	$P_{v4o} = -0.00002 n^2 - 0.0196 n$	0.9989
P_{mco}	Steam mass consumption	$P_{mco} = -0.000007 n^2 + 0.0073 n$	0.9945

Table 6

Design details of the engine and main results of the test.

Parameters	Specifications
Engine type	Double effect reciprocating engine
Diameter of piston (mm)	60
Piston stroke (mm)	70
Rod – crankshaft ratio (dimensionless)	0.69
Expansion ratio, upper cylinder (dimensionless)	2.3
Expansion ratio, bottom cylinder (dimensionless)	2
Working fluid	Water steam
Useful power output	1896 W @ 290 rpm @ 0.70 MPa
Indicated power	4350 W @ 288 rpm @ 0.63 MPa
Peak steam expansion power	4102 W @ 225 rpm @ 1.10 MPa
Average engine mechanical performance	40% @ 290 rpm

was measured on a scale.

The oil pressure from the pump opens the brake pads and the drum tries to drag the pads and the plate. The braking force P is measured on a dynamometer. Eq. (34) relates useful mechanical power N_u with the braking force and rpm, while Eq. (34) gives the mechanical torque.

$$N_u = 0.69869 n P \quad (34)$$

$$T = 6.67457 P \quad (35)$$

N_u is given in Watt, n in rpm, P in pounds, and T in Newton-meters. The admission pressure of the steam engine was also measured by a manometer installed in the steam pipes of the LFR, making it possible to group the power measurements according to the working pressure range (Fig. 14).

Three linear correlation equations are obtained from the measurements to determine the mechanical power developed by the steam engine as a function of rpm regime and working pressure. Measurements above 300 rpm show that the steam engine can develop higher power, but the maximum electric power generation will be possible if a mechanical transmission with variable ratio is available. Actually, the nominal regime of the steam engine is 290 rpm and the mechanical transmission ratio is 5.5. This is because the electric generator connected to the system works in a regimen of 1500 rpm. For 290 rpm and 0.7 MPa working pressure, the steam engine develops a mechanical power of 2.4 kW.

The uncertainty of the entire experimental test is shown in Fig. 15a–b, where ΔN_u is the absolute error and e_{N_u} is the percentage relative error, respectively. Horizontal axes show the measured power output.

It can be observed that the relative error is lower the greater the

power output developed by the steam engine, which means that it is convenient to perform the measurements in high rpm regimes as well as high mechanical power. For power outputs greater than 800 W relative errors are less than 10%.

Fig. 16a–b shows how the useful power output and the rpm regime vary as the pressure varies, respectively. For both figures, an increase is observed as inlet pressure increases. As seen in Fig. 14, higher power outputs can be achieved with high rpm regimes and higher pressures.

6.2. Steam expansion power

To determine the power developed by the expansion of the steam inside the cylinder, temperature of the steam was measured. It was not possible to introduce a pressure probe to determine the indicated power of the engine based on the indicated diagram. The internal temperature measurement was performed in the upper cylinder using a very thin k-type thermocouple of 0.127 mm diameter (Fig. 17). In addition, it was only possible to access the upper cylinder by the lid of the steam engine so only the steam temperature in the upper cylinder was measured, for different speed regimes and with the engine shaft without mechanical load.

The thermocouple was previously calibrated using a Dual Block 9009 from Hart Scientific. An Arduino HX711 board was used for A/D conversion and an Arduino Mega 2560 R3 datalogger was used to download the dataset. With this thermocouple-converter-data logger set, the experiment was performed at a frequency of 80

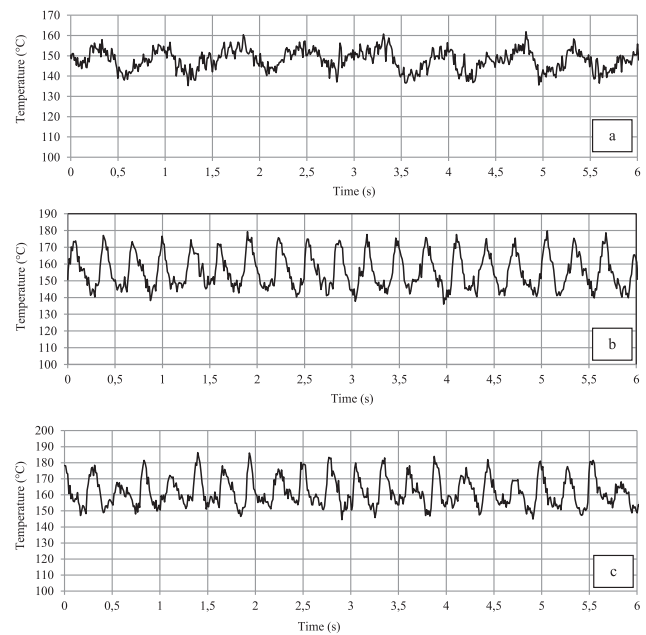


Fig. 18. Steam temperatures in upper cylinder. a) 88 rpm, b) 186 rpm, c) 225 rpm.

samples per second (80 Hz). This corresponds to a set of approximately 20 samples per engine cycle, depending on the rpm regime.

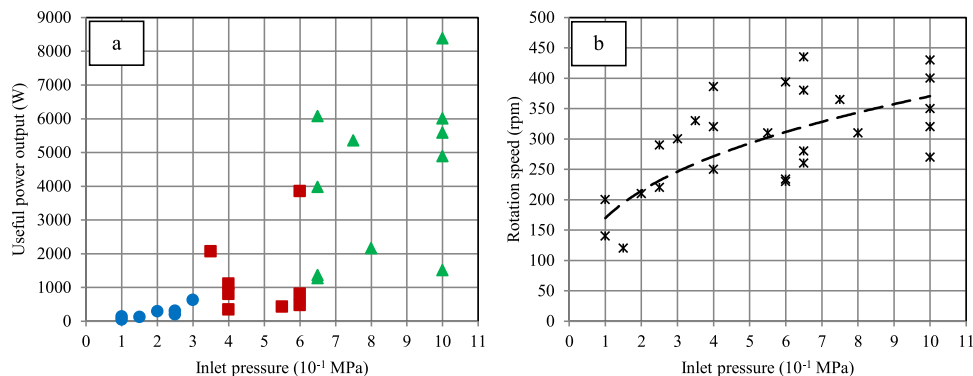


Fig. 16. a) Useful power output trend to increase by an increasing on inlet pressure. b) The rpm regime, as power output, also increases with inlet pressure.

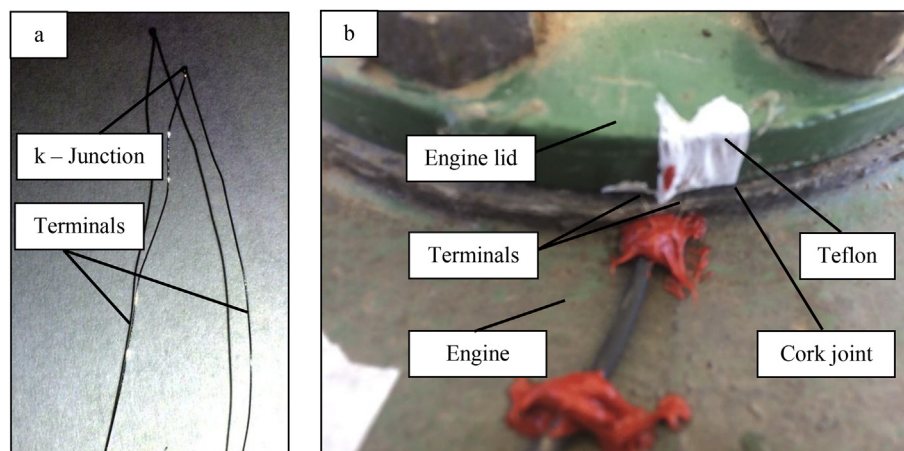


Fig. 17. a) k - type thermocouple, b) thermocouple inserted inside upper cylinder.

Table 7
Accuracies of the measuring devices.

Item	Accuracy
Thin thermocouple k-type	$\pm 2\text{ }^{\circ}\text{C}$ @ $100\text{ }^{\circ}\text{C}$ – $150\text{ }^{\circ}\text{C}$
Scale	$\pm 5\text{ }^{\circ}\text{C}$ @ $150\text{ }^{\circ}\text{C}$ – $200\text{ }^{\circ}\text{C}$
Speed sensor	$\pm 0.5\text{ lb}_f$ [32]
Pressure gauge	$\pm 1\text{ rpm}$
	$\pm 0.5\text{ bar}$

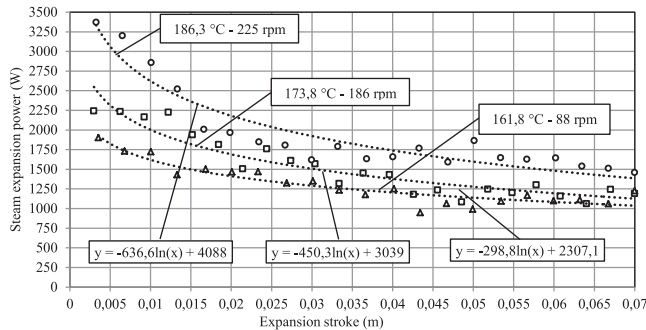


Fig. 19. Steam expansion power for different temperatures and rpm regimes.

Table 8
Adjustment between measured data and regression curves.

Temperature	186.3 °C	173.8 °C	161.8 °C
rpm	225	186	88
RMSE%	9.16	8.91	7.12
MBE%	0.21	0.19	0.77
R ²	0.89	0.88	0.87

Three measurements were performed for different rpm regimes: 88, 186 and 225 rpm (Fig. 18a–c).

The expansion power is calculated by Eqs. (32) and (33), under the supposition that pressure in the bottom cylinder is similar to the upper cylinder and a periodic wave offset of 180 geometric degrees with respect to the measured temperature wave. The accuracies of the measuring devices are listed in Table 7.

Taking a representative cycle as a reference, for each case, the power of expansion is calculated as the piston moves by performing its stroke. For the three cases, the expansion curves are presented in Fig. 19, which shows that the peak power reaches a value of approximately 3.5 kW, in the case of the steam working at 225 rpm. This power peak decreases with the working pressure (temperature) and rpm regime.

The regressions for the expansion curves are performed by logarithmic equations with a certain degree of adjustment given by coefficients such as RMSE% (root mean square error), MBE% (medium bias), and R² (determination coefficient) between measured data and regression curves (Table 8).

RMSE% coefficients are lower than 10% in all cases. Therefore, we

can say that the logarithmic correlations are helpful to estimate the expansion power developed by the steam. MBE% values are a little above zero, which means that the used regression slightly overestimates the measured data series. Consequently, a good precision is obtained in the measurement method.

7. Results comparison

The working parameters, experimental results and simulation results are presented in Table 9. One could see that the power peak expansion error is 27%, which is relatively large. That is because leaks by lamination in steam valve as well as thermal losses for condensation in the inner walls were not taken into account in the simulation.

Regarding power output error, a value of ± 3 – 4% can be considered as good for engineering applications and power electric generation with this kind of technologies. In our consideration, this error is acceptable taking into account the robustness, the years of usage of the engine and its date of construction.

8. Conclusions

Mathematical models and preliminary results from simulation and measurement were achieved. Engine models and simulations represent a useful tool for predicting engine performance and thermomechanical behavior.

The model presented here includes the dimensions design and thermodynamic input parameters and allows us to know the engine behavior for several mechanical load stages. It is possible to estimate the steam pressure to operate the engine when it is loaded as well as the steam consumption, which is important for designing the LFR. For example, to develop a power output of about 2.4 kW at 290 rpm, the steam pressure should be 0.7 MPa, so the LFR has to be able to produce water steam up to these conditions.

It is also possible to obtain different power outputs by steam speed regulation according to the load. Figs. 10 and 12 show that steam consumption varies essentially with rpm regime. The equations of regulation obtained from curves (Table 5) allow us to propose control strategies to act directly on the main steam valves that feed the steam engine, specifically, to control the steam flows at the intake of the engine. Flow regulation depends on the state of mechanical load and rpm regime of the engine.

The steam engine is designed to operate with low-temperature heat sources ($150\text{ }^{\circ}\text{C}$ – $250\text{ }^{\circ}\text{C}$). The results of the simulation are in agreement with the experimental results. For example, measurement of useful power output indicates that steam engine generates a power of 2.4 kW for a regimen of 290 rpm and working pressure of 0.7 MPa. This is consistent with the instantaneous power curve given by the simulation (Fig. 11).

This study provides an understanding of the behavior of the double-effect reciprocating steam engine and helps us to optimize the parameters of design and operation of our Linear Fresnel system in order to adjust pressure, flow and temperature of generated steam for electric power generation.

Table 9
Experimental and simulated results comparison.

Parameter	Experiment	Simulation	Error	Experiment	Simulation	Error
Rotation Speed (rpm)	186	186	—	290	290	—
Inlet pressure (MPa)	0.61	0.56	–9%	0.7	0.6	–17%
Inlet temperature ($^{\circ}\text{C}$)	162	156	–4%	171	159	–7%
Mass consumption (l/min)	1.46 [33]	1.52	4%	1.55 [33]	1.69	8%
Power output (W)	1369	1336	–3%	2440	2349	–4%
Peak power expansion (W)	3039	4187	27%	—	—	—

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Appendix A. Supplementary data

Supplementary data related to this article can be found at <https://doi.org/10.1016/j.energy.2018.03.110>.

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