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PHYSICO-MATHEMATICAL MODEL OF A HOT AIR ENGINE USING HEAT FROM LOW-TEMPERATURE RENEWABLE SOURCES OF ENERGY

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The paper provides a novel physico-mathematical model able to estimate the maximum performances of a hot air Ericsson-type engine - which is capable to run on renewable energy, such as solar energy, geothermal energy or biomass. The computing program based on this new model allows to study the influences of the main parameters over the engine performances and to optimize their values.

Keywords: Ericsson-type engine, renewable energy, theoretical model, maximum performances.

1. Introduction

In the last decades we witnessed an extraordinary development of numerous engines and devices capable of transforming heat into work [1]. The field of the “unconventional” engines has risen again into the attention of the researchers and of the investors. Several former abandoned or marginal technical solutions of hot air engines were reanalyzed by researchers - as the Ericsson-type reciprocating engines [2], Stirling engines or Stirling-related engines [1], [3] and the Vuilleumier machine.

The Ericsson-type hot air engine is an external combustion heat engine with pistons (usually, with pistons with reciprocating movement) and with valves. An Ericsson-type engine functional unit comprises two pistons. The compression process takes place inside a compression cylinder, and the expansion process takes place inside an expansion cylinder. Heat is added to the cycle inside a heater. Sometimes a heat recuperator or regenerator is used to recover energy from the exhaust gases, fig. 1. A closed-cycle Ericsson-type engine can be

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pressurized in order to increase the work yield per cycle. The Ericsson engine can use heat from any sort of primary energy - from combustion of fossil fuels or, much more interesting, from renewable sources: wood chips, waste heat disposed by industrial processes, solar [4], [5], [6], [7] or geothermal energy. In the last years the Ericsson-type engine is assumed as a realistic future solution for small-scale solar units that are able to provide electrical energy and hot water for domestic use [8]. Such unit can use solar energy during the day and fossil fuels in the rest of the time.

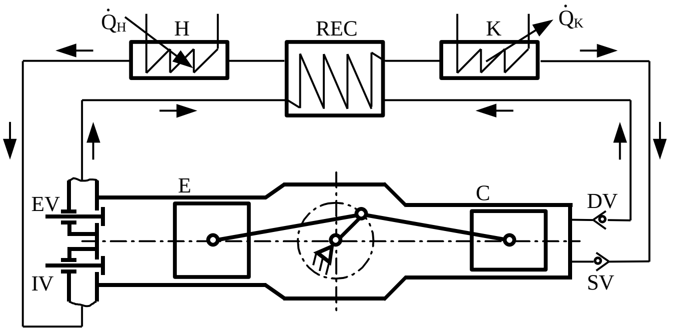

Fig. 1. Typical layout of an Ericsson-type engine: H - heater; REC – recuperator; K – cooler; DV – discharge valve; SV – suction valve; C – compressor; E – engine; IV – intake valve; EV – exhaust valve; $\dot{Q}_H$ - heat transfer rate inside heater; $\dot{Q}_K$ - heat transfer rate inside cooler

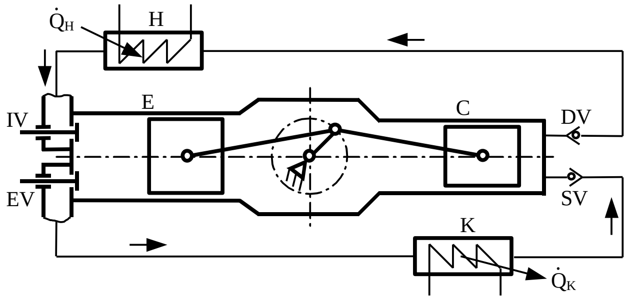

Fig. 2. Proposed layout for low-temperature Ericsson engine

The theoretical models of the hot air Ericsson-type engines are very important - especially for the first stages of the design process - because they state the maximum performances of the engine. The present paper introduces a new theoretical model for the Ericsson-type engines, developed especially for engines running on low temperature differences.

An Ericsson-type hot air engine using heat from low-temperature renewable sources of energy will limit the maximum temperature of the air inside the heater to a certain value. It is also compulsory to limit the discharge pressure given by the compressor because the temperature of the discharged air must be lower than the maximum temperature. So, the recuperator is not needed, and will be eliminated from the layout, as in fig. 2.

2. Physical model

The model considers each component of the installation separately. Inside the compression cylinder a theoretical compression cycle takes place (with suction, compression and discharge processes), and inside the expansion cylinder we have the cycle of a pneumatic motor (with intake, expansion and exhaust processes). Inside the heater the working agent undergoes its own cycle, based on the assumption of a constant-volume heating.

The physico-mathematical model was developed based on the following principal hypotheses: the working agent is an ideal gas; dead spaces inside the cylinders are zero; the volumes of the connecting pipes are neglected; the valves open instantaneously, at null pressure differences, in such a manner that they have no influence over the pressure.

Usually, compression and expansion are assumed as isentropic processes [8] while cooling and heating are assumed as isobaric processes (heater and cooler having infinite volumes). In order to get a more realistic model, polytropic compression and expansion are assumed in this study. In order to get better performances by producing more work, the compression cylinder can be cooled (polytropic index of the cooling process - n_c - is lower than adiabatic index - k) and the expansion cylinder can be heated.

The suction process inside compressor and the exhaust process from the engine are assumed to be isobaric. This implies that the process inside the cooler is also isobaric. It is possible to admit such hypothesis, especially because these three processes take place simultaneously. We assume that during the exhaust / suction stroke the mass inside the cooler is constant, the air leaving the cooler toward compressor being replaced by the air coming from the engine.

The model takes into account a heater having finite volume. During the discharge process the air is moving from the compression space to the heater; the air pressure and temperature rise. So, the discharge process can be modeled as isentropic compression, with the volume of the heater added to the compression space as dead space. It is also possible to model the discharge process as polytropic process, but such hypothesis requires some information about the heat exchange during the process; the model ceases to be a theoretical one in this case. In order to avoid differential equations and to keep the model simple, the compression during discharge is considered to be performed separately by the air initially placed inside the heater and by the air initially placed inside the compressor. At the end of the process the gas discharged by compressor inside the heater is mixing with the gas resident to the heater, and the process is considered to be isobaric and instantaneous.

During the intake process a mass of air (the same mass as the one sucked in by the compressor) flows from the heater and enters inside the engine. The process can be modeled as an isentropic expansion of the air from the heater. At the end of the intake process the heater will be separated from the expansion space by closing the intake valve. The parameters at the end of the intake are calculated by assuming that pressure at the end of the polytropic expansion inside the engine is equal to the suction pressure.

The model allows to achieve the values of the thermodynamic variables for the entire cycle, and also to calculate the work and heat exchanged by any process. The theoretical efficiency of the entire engine can be estimated.

3. Mathematical model

The model uses the following initial data: adiabatic index k and individual gas constant R for air, diameters D_c and D_e for compression and expansion cylinders, strokes S_c and S_e, volume V_h of the heater, polytropic indexes n_c and n_e, pressure p_s and temperature T_s at the beginning of the suction process and the maximum temperature T_{max} allowed for the heater.

The variations of the pressure and of other important parameters inside the compressor are:

- for suction, process 0-1 (according to fig. 3)

$$p = p_s = ct. \tag{1}$$

- for compression, process 1-2

$$p(V) = p_s \left(\frac{V_1}{V}\right)^{n_c},\ p \in [p_1 = p_s, p_2 = p_{d\,\min}], \tag{2}$$

$$V \in [V_{c\max}, V_2(p_{d\,\min})],\ V_2(p_{d\,\min}) = V_{c\max}\left(\frac{p_s}{p_{d\,\min}}\right)^{1/n_c}, \tag{3}$$

- for discharge, process 2-3

$$p(V, p_{d\,\min}) = p_{d\,\min}\left(\frac{V_2(p_{d\,\min}) + V_h}{V}\right)^k,\ p \in [p_2, p_{d\,\max}], \tag{4}$$

$$p_{d\,\max} = p_{d\,\min}\left(\frac{V_2(p_{d\,\min}) + V_h}{V_h}\right)^k,\ V \in [V_h + V_2(p_{d\,\min}), V_h]. \tag{5}$$

Variation of pressure during the constant volume heating inside the heater (process 3-4) is

$$p(p_{d\,\min}, T_{2h}) = p_{d\,\max}\frac{T}{T_3(p_{d\,\min}, T_{2h})},\ p \in [p_{d\,\max}, p_{\max}]. \tag{6}$$

Variations of the pressure and of other important parameters inside the engine are:

- for intake, process 4-5

$$p(V, p_{d\,\min}, T_{2h}) = p_4(V, p_{d\,\min}, T_{2h})\,(V_h / V)^k, \tag{7}$$

$$V \in [V_h, V_h + V_{5e}(p_{d\,\min}, T_{2h})], \tag{8}$$

$$V_{5e}(p_{d\,\min}, T_{2h}) = V_h \left(\frac{p_4(p_{d\,\min}, T_{2h})}{p_5(p_{d\,\min}, T_{2h})} \right)^{1/k} - V_3, \tag{9}$$

$$p_5(p_{d\,\min}, T_{2h}) = \left[\frac{T_4(p_{d\,\min}, T_{2h})\, R\, m_{2h}(p_{d\,\min}, T_{2h})}{V_h \left[p_4(p_{d\,\min}, T_{2h}) \right]^{(k-1)/k}} \right]^k, \tag{10}$$

$$T_5(p_{d\,\min}, T_{2h}) = T_4(p_{d\,\min}, T_{2h}) \left[\frac{p_5(p_{d\,\min}, T_{2h})}{p_4(p_{d\,\min}, T_{2h})} \right]^{\frac{k-1}{k}} = T_{2h}, \tag{11}$$

- for expansion, process 5-6

$$p(V, p_{d\,\min}, T_{2h}) = p_5(V, p_{d\,\min}, T_{2h}) \left(\frac{V_{5e}(V, p_{d\,\min}, T_{2h})}{V} \right)^{n_e}, \tag{12}$$

$$V \in [V_{5e}(p_{d\,\min}, T_{2h}), V_{e\max}], \; p_6(V_{e\max}, p_{d\,\min}, T_{2h}) = p_s, \tag{13}$$

- for exhaust, process 6-7

$$p = p_s = ct. \tag{14}$$

For closing the cycle, we impose two requirements that must be fulfilled simultaneously: $T_5 = T_{2h}$ and $p_6 = p_s$. This goal can be achieved through iteration.

The pressure during the cooling process 7-8 is constant,

$$p = p_s. \tag{15}$$

Inside the relations above, p_{dmin} and p_{dmax} are the minimum and maximum discharge pressure, V_{cmax} and V_{emax} are the maximum volumes of the compression and expansion spaces, T_{2h} and m_{2h} are the initial temperature and mass of the air inside heater at the beginning of the discharge process, T_3 is the medium temperature inside the heater at the end of the discharge process, p_{max} is the maximum pressure inside the installation during the cycle, V_{5e} is the volume of the air inside the expansion cylinder at the end of the intake process.

Once determined the variation laws of the pressure with the volume, it is easy to calculate the work exchanged during each process, by defining

$$L = \int p\, dV. \tag{16}$$

Heat and internal energy variations can be calculated applying to each process the first law of thermodynamics.

Heat received, work yield and thermal efficiency of the cycle are:

$$Q_{received} = Q_{34} + Q_{56}, \tag{17}$$

$$W_{cycle} = W_{01} + W_{12} + W_{23} + W_{45} + W_{56} + W_{67}\,, \tag{18}$$

$$\eta = W_{cycle} / Q_{received}\,. \tag{19}$$

4. Numerical example

An instantiation of a low-temperature Ericsson engine with the layout presented in Fig. 2 is considered. The main dimensional features of the installation are $D_c = D_e = 0.36$ m, strokes $S_c = 0.342$ m and $S_e = 0.36$ m, heater volume $V_h = 0.084$ m^3. The functional parameters of the simulated working regime are: suction pressure and temperature $p_s = 0.4$ MPa and $T_s = 298$ K, heater maximum temperature $T_{max} = 388$ K, polytropic indexes $n_c = 1.39$ and $n_e = 1.39$. The working agent is air, with R = 287 J / (kg K) and k = 1.4.

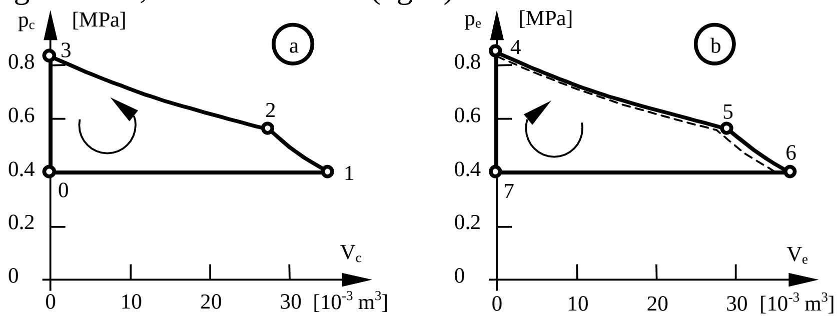

Fig. 3. Thermodynamic cycle inside compressor (a) and inside the engine (b); the dashed curves in (b) represent the compression and discharge processes inside compressor

The results of the calculations show that the analyzed low-temperature installation produces 604.3 J/cycle, with an efficiency of 18.55%. This value is not so small as it looks at the first glance, because is quite close to the efficiency of the correspondent Carnot cycle (described by the same two temperatures), namely 23.2%. It must be noted that this efficiency is similar to the efficiency of the nowadays commercially available solar cells (between 14…19 %). But, unlike the solar cells, the analyzed system can operate with any (renewable) energy source, which is a great advantage. The proposed model allows to establish the maximum performances of this unusual machine. Some results of the numerical simulation are graphically presented in fig. 3, fig. 4, fig. 5 and fig. 6.

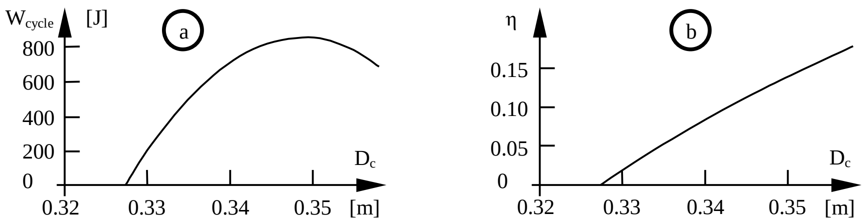

Fig. 4. Influence of the compressor’s diameter on produced work (a) and thermal efficiency (b)

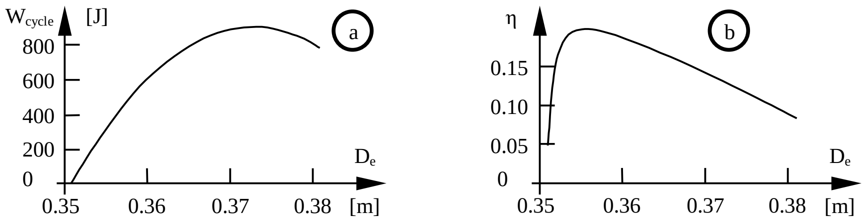

Fig. 5. Influence of the engine's diameter on produced work (a) and thermal efficiency (b)

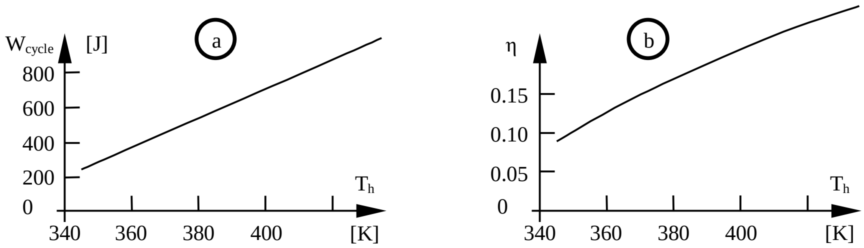

Fig. 6. Influence of the heater temperature on produced work (a) and thermal efficiency (b)

As it can be seen in fig. 3, the thermodynamic cycles inside the compression space and inside the expansion space emphasized the hypothesis of the theoretical model. It can be noted - from fig. 3-b - that the difference between the areas of the two cycles is quite small (about 6.8%, calculated value). The work spent for the compression cycle is –8225 J/cycle, and the work produced inside the engine is 8829 J/cycle. Such a large quantity of work must be cyclically provided for performing the compression cycle while the elementary work produced by the engine is not equal to the elementary work spent by the compressor. That is why installation must be fitted with a flywheel.

The curves in fig. 4 show that there is a minimum value of the compressor's diameter for which the work and efficiency are zero, and there is a value of the diameter for which the work reaches its maximum value.

The curves in fig. 5 indicate a minimum value of the engine diameter corresponding to zero work and efficiency. There are also certain values of the diameter for which work and efficiency reach their maximum values.

The curve in fig. 6-a shows that the value of work yield per cycle is increasing with temperature of the hot source – as expected. The efficiency is also increasing with the temperature.

5. Conclusions

The paper demonstrates the possibility of an efficient conversion of thermal energy into work when a low-temperature heat source (of less than 200 ºC), such as renewables, is used. In order to obtain this transformation in an engine with pistons like the Ericsson-type engine, with industrial usable quantities of work, it is compulsory to use large cylinder bores and large stokes of the pistons. This is because the system requires large quantities of hot air to be processed. Due to the low temperatures, the installation does not require a regenerator. The volume swept by the compressor piston is smaller than the one swept by the engine piston. The ratio between these two volumes is determined by the temperature of the heater.

The model considers a heater with finite volume, and consequently takes into account the pressure variations during discharge and intake processes.

The compatibility with a variety of heat sources, less exposure to the fossil fuels depletion, high thermal efficiency potential and low CO_2 emissions are the main benefits that produce renewed interest in Ericsson-type hot air engine.

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