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Modeling of the Ericsson engine

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A B S T R A C T

An ERICSSON engine is a reciprocating thermal motor with external heat supply and separate compression and expansion spaces. It uses a monophasic gaseous working fluid. Unlike the Stirling engine, the ERICSSON engine is equipped with valves around the cylinders to isolate the cylinders from the heat exchangers during the expansion and the compression processes. The ERICSSON engine can be provided with a heat recovery exchanger and it can operate according to a closed or an open cycle. This engine is suitable for low power (up to some kW) thermal energy conversion from renewable energy sources like biomass or solar energy. Dimensionless quantities are defined such as the pressure ratio β, the temperature ratio θ, the cylinder capacity ratio φ, the relative dead volumes μ_E and μ_C, the thermal efficiency ηth and the net dimensionless indicated power Π. The relationships between these quantities are established. The modeling is based on the assumptions of a Joule cycle with internal heat recovery exchanger realized by a perfect gas with constant heat capacity. These relationships allow to determine the pressure in the heater as a function of the temperature ratio and the engine geometrical data. It is shown that there is a well defined operating range for which the engine can produce mechanical energy as a function of the quantities β, φ, μ_E, μ_C, θ, and irrespective of the fact that the expansion space dead volume is recompressed or not.

1. Introduction

Amongst the wide diversity of thermal engines, a special family of engines can be identified from the following features: separate reciprocating compression and expansion machines, external heat supply, regenerator or recuperator, monophasic gaseous working fluid. These engines are sometimes called 'hot air engines' [1], even if the air used in the XIXth century engines has been replaced by high pressure hydrogen or helium in a lot of modern engines. The family of hot air engines is divided into two subgroups: the Stirling engines, invented in 1816, have no valves (Fig. 1a), whereas Ericsson engines, invented in 1833 (Fig. 1b) have valves in order to isolate the cylinders.

The valves give some advantages to the Ericsson engine [2]. Amongst them, the most important one is that the heat exchangers are not to be considered as unswept dead volumes whereas the Stirling engine designer is faced with the difficult compromise between heat exchanger transfer area maximization and heat exchanger volume minimization. Other important advantages are the replacement of the Stirling regenerator by a simple counter-flow heat exchanger, the removal of the so-called "Stirling thermal aberration" [2], and the possibility to use a simplified kinematic mechanism. The main disadvantage of the Ericsson engine compared to the Stirling engine is due to the presence of the valves which increase the mechanical complexity of the engine and can reduce its reliability. The theoretical Ericsson cycle with two isobaric processes and two isothermal processes is not suited to describe the Ericsson engine. Indeed, the lack of heat transfer area in the cylinders leads to equip the engine with heat exchangers outside the cylinders in order to exchange heat with the hot and the cold sources. So the Joule cycle with two isentropic and two isobaric processes is better suited to describe the Ericsson engine. This cycle is also often used for gas turbines. In the large power range a lot of studies have been devoted to turbo machinery Joule cycle for solar energy conversion [3]. There are so far, few works in the field of low power solar reciprocating engines [4]. Some studies are devoted to the reverse Joule cycle thermodynamic analysis [5,6] or to Joule cycle engines with "scroll" compression and expansion machines [7,8]. There are also some references to internal combustion Ericsson engine [9,10] instead of external heat supply. The studies dealing specifically with external heat supply Joule cycle reciprocating engines are limited [11–13]. The studies reporting experimental results are rare [14,15].

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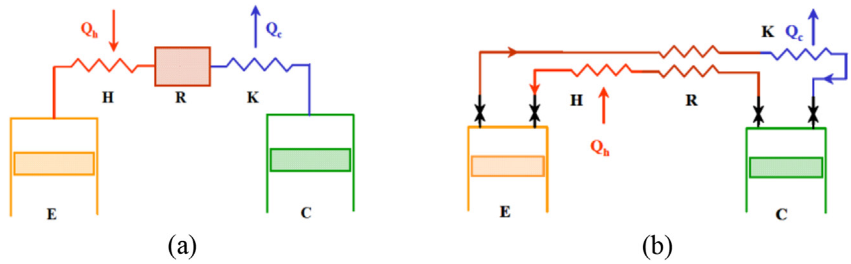

Fig. 1. Stirling engine (a) and Ericsson engine (b) principle.

Regarding the Ericsson engine modeling, a comparison of the respective advantages and disadvantages of the Joule cycle Ericsson engine and the Rankine cycle steam engine has been drawn [16] and it has been shown that the operation of Ericsson engine according to the so-called "humid cycles" is not interesting as soon as heat recovery Joule cycles are considered [17]. An energy, exergy and cost analysis has demonstrated that the Ericsson engine is suitable for micro-cogeneration applications [18]. A dynamic simulation model shows how the Ericsson engine responds to perturbations and transients [19] allowing to design an appropriate control system. Finally studies have been devoted to the global effect of the in–cylinder heat transfer on the energy performance of the Ericsson engine and the interest or not to promote these transfers [20] or to the modeling of the instantaneous fluid to wall in–cylinder heat transfer [21].

However none of these models permit the design of an Ericsson engine for a specific application nor the determination of the pressure to be established in the heater as a function of the thermal and geometrical data. The present study intends to address these needs [15].

2. Global analysis of the Joule cycle with internal heat recovery

2.1. Modeled system

One advantage of the Ericsson engine is modularity. It means that each part of the engine can be studied and optimized separately before being inserted in the whole engine. The Ericsson engine considered here operates in an open cycle (Fig. 2). The compressed air at temperature T_{cr} is preheated in the counter-flow heat recovery exchanger R up to T_{rh}. It flows then in the heater H where it is heated up to the temperature T_h. It is then admitted in the expansion cylinder E where it is expanded down to temperature T_{er} and ambient pressure. The expanded gas flows next in the heat recovery exchanger R in order to pre-heat the air entering the heater H. Finally the air flowing out of the heat recovery exchanger is delivered to the atmosphere at the temperature T_{rk} much lower than T_{er}.

2.2. General assumptions

The general assumptions of the model are as follows:

- The compression and expansion processes are isentropic.
- The working fluid is air considered as a perfect gas with constant specific heat capacity.
- Viscous losses of the working fluid are neglected. There are no pressure drops through the inlet and exhaust valves. Flows within the heat exchangers R and H are assumed to be isobaric. The valves opening and closures induce no pressure fluctuations in the heat exchangers. This implies that the heat exchangers volumes are large compared to the cylinders capacities. It is reminded that the possibility to use large heat exchangers is the main advantage of the Ericsson engine compared to the Stirling engine.

2.3. Dimensionless quantities

For each state i of the working fluid in the engine, the following dimensionless quantities are defined:

- the pressure ratio $\beta_i = p_i/p_k$; by extension, $\beta = p_h/p_k$
- the temperature ratio $\theta_i = T_i/_kT$; by extension, $\theta = T_h/T_k$
- the cylinder capacity ratio $\varphi_i = V_i/V_C$; by extension, $\varphi = V_E/V_C$
- the dimensionless mass or mass flow rate $\delta_i = m_i\, r\, T_k/(p_k\, V_C)$ $= \dot{m}_i\, r\, T_k/(n\, p_k\, V_C) = \beta_i\, \varphi_i/\theta_i$
- the dimensionless indicated work or power $\Pi_i = W_i/(p_k V_C)$ $= \dot{W}_i/(n p_k V_C) = \oint p_i \mathrm{d}V_i/(p_k V_C) = \oint \beta_i\, \mathrm{d}\varphi_i \quad i = \mathrm{E}$ or C
- the dimensionless heat exchanged in the heater $\Pi_{\mathrm{th}} = Q_{\mathrm{th}}/(p_k V_C)$
- the indicated thermal efficiency $\eta_{\mathrm{th}} = (\Pi_E - \Pi_C)/\Pi_{\mathrm{th}}$ $= \Pi_{\mathrm{net}}/\Pi_{\mathrm{th}}$

2.4. Dimensionless relationships

The net specific work is obtained by the balance of the specific enthalpy differences in the expansion and the expansion cylinders. The dimensionless indicated work (or power) writes:

$$\Pi = \frac{W}{p_k V_C} = \frac{w}{rT_k} = \frac{c_p}{r}\left(\left(\frac{T_h}{T_k} - \frac{T_{er}}{T_k}\right) - \left(\frac{T_{cr}}{T_k} - 1\right)\right) \quad (1)$$

According to the assumptions defined previously, the processes in the compression cylinder C and the expansion cylinder E (Fig. 2) are considered as isentropic processes of a perfect gas with constant specific heat so that the temperature ratios and the pressure ratio are linked by the following relations:

$$\frac{T_h}{T_k} = \theta \quad ; \quad \frac{T_{er}}{T_k} = \frac{\theta}{\beta^k} \quad ; \quad \frac{T_{cr}}{T_k} = \beta^k \quad (2)$$

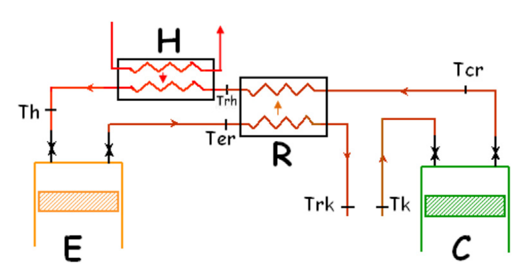

Fig. 2. The Ericsson engine considered.