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Flow in a New Type Externally Heated Air Engine with Two Blowers

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The experimental investigation of the 2-stroke Externally Heated Valve Engine (EHVE), different from the known Stirling engine, were carried out during 1999–2000. The main disadvantages of the engine were well known. The following changes are proposed in the improved version of the engine, in the case when natural fuels are exhausted (gas and oil):

1. in order to enlarge heat transfer in this air engine heater and cooler (a little mass flow rate is employed in each piston engine), a working air circulation forced by two blowers, one in a large volume (arbitrary) heater and the second in a similar volume cooler, are applied;
2. a 4-stroke engine cycle has been introduced;
3. self-acting valves are eliminated and only two governed are used instead.

Numerical solutions in the form of time-dependent pressure and temperature diagrams for each volume of the engine (i.e. for a cylinder, a heater and a cooler) and mass flow rates through valves, as well as $p-V$ diagrams will be presented. Finally, the essential results, i.e. the power and efficiency, are given.

Keywords: externally heated 4-stroke air engine.

1. Introduction and the principle of the engine operation

When the petrol and natural gas are exhausted, a small piston engine with external heating will be needed.

The experimental investigation of the 2-stroke Externally Heated Valve Engine (EHVE) was carried out during 1999-2000 and reported in [1]. In order to enlarge heat transfer in this air engine heater and cooler, a working air circulation is forced by two blowers: one in a large-volume (arbitrary) heater and the second one in a similar volume cooler. This is a different engine from that one described in [2] where only one blower is installed in the closed air loop, working in a large-volume heater. A theoretical model of the 4-stroke engine is presented in the paper. This engine cycle has been introduced to eliminate the so called ,,hot cylinder” of the expander being a part of the 2-stroke engine – model [1].

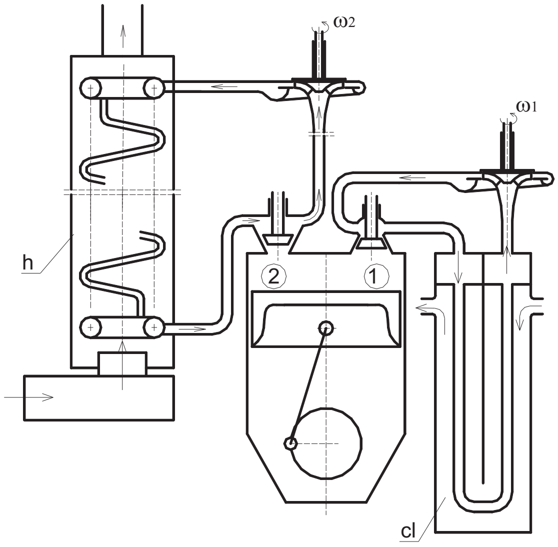

Figure 1 Schematic diagram of the engine elements

The essential part of the engine is only one cylinder and two valves: valve No.1 and No.2 – see Fig. 1. Besides, the 4-stroke closed cycle engine is composed of three elements, namely: a cylinder, a heater and a cooler. The theoretical model, quoted and described in [2], is based on the time-dependent equations of mass and energy conservation, formulated for each elements separately. The calculation is developed in order to take into account the crank-shaft dynamics. The heater and cooler blower are installed only to enlarge heat transfer in these two elements of the engine. A very little mass flow rate is employed in each piston engine and this case it causes too little heat transfer. The operation of the 4-stroke cycle is as follows:

- The first stroke: starting from its lower position, the piston pushes out the expanded air from the cylinder to the cooler through valve No.1 (air stream $\dot{m}_4$, Fig.1);
- The second stroke: the piston moves down (see Fig.1) from its upper position and air from cooler flows into the cylinder through valve No.1 (air stream $\dot{m}_1$);
- The third stroke: the piston starts from its lower position and compresses the air in the cylinder. At about 70^o of the crank-shaft angle, before the piston achieves its upper position, valve No.2 opens and the compressed air is delivered to the heater (air stream $\dot{m}_2$);
- The fourth stroke: the piston moves down from its upper positions and sucks the hot air from the heater into the cylinder through open valve No.2, when the crank-shaft angle is equal to 90^o with respect to the piston upper position,

valve No.2 closes and then the hot air in the cylinder expands until the piston reaches its lower position (air stream $\dot{m}_3$).

The blower working in the heater volume is a complex and technologically advanced element of this engine. The blower working in the cooler volume is, however, very common and usual element the engine. Expecting that it will be possible to manufacture the above mentioned elements, the theoretical model of the engine operation and the results of calculation are valuable.

2. Theoretical model

The geometrical values of the valve flow section $\mathrm{A}_j(\alpha)$ were α is the crank-shaft angle, are determined for each valve as follows:

Valve No.: j=1,2 ; where $\dot{m}_4$ and $\dot{m}_1$ are the mass flow rates through valve No.1 but $\dot{m}_2$ and $\dot{m}_3$ are the mass flow rates through valve No.2.

The following nomenclature are involved:

- for $\alpha_{j,0} \le \alpha \le \alpha_{j,0} + \Delta\alpha_j$

$$A_j\left(\alpha\right) = 0.5 A_{j,\max}\left[1 - \cos\pi\frac{\left(\alpha - \alpha_{j,0}\right)}{\Delta\alpha_j}\right] \tag{1}$$

- for $\alpha_{j,0} + \Delta\alpha_j \le \alpha \le \alpha_{j,c} - \Delta\alpha_j$

$$A_j\left(\alpha\right) = A_{j,\max} \tag{2}$$

- for $\alpha_{j,0} - \Delta\alpha_j \le \alpha \le \alpha_{j,c}$

$$A_j\left(\alpha\right) = 0.5 A_{j,\max}\left[1 - \cos\pi\frac{\left(\alpha - \alpha_{j,c}\right)}{\Delta\alpha_j}\right] \tag{3}$$

The volume of the cylinder are described by simplified formulae:

$$V_C = A_C\left[h_0 + \frac{s}{2}\left(1 + \cos\alpha\right)\right] \tag{4}$$

where: $A_C = \pi d_C^2/4$, d_c is the diameter of the piston.

The first stroke begins at α=0, s is the engine stroke, h_0 determines the dead piston volume. The characteristic of the electric starter is following:

$$M_{st} = 80 - 1.7\omega \tag{5}$$

where angular velocity:

$$\omega = \frac{d\alpha}{dt} \tag{6}$$

The unsteady engine operation is taken into account. The simplified equation of the crank-shaft motion takes the form:

$$I_{cs}\frac{d\omega}{dt} = M_{en}(t) - M_{rc}(t) + M_{st}(t) \tag{7}$$

where: I_{cs} -moment of the crank-shaft inertia, $M_{rc}(t)-$receiver torque and frictional resistance torque. $M_{st}(t)$ – torque of the electric starter.

The engine torque depends on the pressure p_C in the cylinder (α=0 when the piston starts from its lower position):

$$M_{en}(t) = -A_C p_C(t) \frac{s}{2} \sin\alpha \tag{8}$$

The pressure in the cylinder p_C, the heater p_H and the cooler p_{CL} are determined on the basis of the equations used.

2.1. *Basic equations for the cylinder*

The equation for the mass conservation is as follows:

$$\frac{d\rho_C}{dt} = \frac{1}{V_C}\left(\pm\dot{m}_4 \pm \dot{m}_1 \pm \dot{m}_2 \pm \dot{m}_3 - \rho_C \frac{dV_C}{dt}\right) \tag{9}$$

The mass flow rates $\dot{m}_j$ (mass streams $\dot{m}_4$ and $\dot{m}_1$ for valve No.1 and $\dot{m}_2$ and $\dot{m}_3$ for valve No.2) are calculated from the well-known gas dynamics formulae. The mass flow rates flowing towards the cylinder are positive (marked by "+" in Eq.(9)) but those flowing out of the cylinder are negative (marked by "–" in Eq.(9)). The direction of the mass streams depend on the pressure relations determined in the calculations performed. The equation of energy conservation for the cylinder is:

$$\begin{aligned}\frac{dT_C}{dt} \quad = \quad & \frac{1}{\rho_C V_C}\left[-T_C\left(\pm\dot{m}_4 \pm \dot{m}_1 \pm \dot{m}_2 \pm \dot{m}_3\right) + \right. \\ & \kappa\left(\left\{\begin{array}{c} +\dot{m}_4 T_{CL} \\ -\dot{m}_4 T_C \end{array}\right\}\left\{\begin{array}{c} +\dot{m}_1 T_{CL} \\ -\dot{m}_1 T_C \end{array}\right\}\left\{\begin{array}{c} +\dot{m}_2 T_H \\ -\dot{m}_2 T_C \end{array}\right\}\left\{\begin{array}{c} +\dot{m}_3 T_{CL} \\ -\dot{m}_3 T_C \end{array}\right\}\right) - \\ & \left.\frac{p_C}{C_V}\frac{dV_C}{dt}\right]\end{aligned} \tag{10}$$

The terms in the braces are selected depending on the signs of the mass flow rates determined in Eq.(9). The equation of state:

$$p_C = \rho_C R T_C, \tag{11}$$

allows one to calculate the pressure p_C in the cylinder.

2.2. *Basic equations for the heater*

The volume V_C of the heater is much greater then the cylinder volume V_C so that the surface of the heat exchange of the heater A_H can be designed to assure the demanded heat stream. The heat blower (Figure 1), especially its ω_2, allows one to obtain the Reynolds number inside the heater tubes higher than 10^5 and the heat transfer coefficient from the inner tube walls to the working air about 10^3. The temperature of the tube walls, denoted by T_{wH} , is assumed to be constant in this model of calculation.

The equation of mass conservation for the unknown assumed the form:

$$\frac{d\rho_H}{dt} = \frac{1}{V_H}\left(\pm\dot{m}_2 \pm \dot{m}_3\right). \tag{12}$$

The mass flow rates directed towards the heater are positive (marked by "+" in Eq.(11)), and those flowing out of the heater are negative. The sign of the mass flow rates is checked in the process of calculation.

The equation of energy conservation for the heater (the unknown value is T_H) is given below:

$$\begin{aligned}\frac{dT_H}{dt} &= \frac{1}{\rho_H V_H}\left[-T_H\left(\pm\dot{m}_2 \pm \dot{m}_3\right)+\right. \\ &\left.\kappa\left(\begin{Bmatrix} +\dot{m}_2 T_C \\ -\dot{m}_2 T_H \end{Bmatrix}\begin{Bmatrix} +\dot{m}_3 T_C \\ -\dot{m}_3 T_H \end{Bmatrix}\right)+\frac{\dot{Q}_{AH}}{C_V}\right]\end{aligned} \tag{13}$$

The terms in the braces are selected depending on the signs of the mass flow rates determined for Eq.(11). The heat stream $\dot{Q}_{AH}$ is calculated from the formula:

$$\dot{Q}_{AH} = \alpha_{AH} A_H \left(T_{wH} - T_H\right) \tag{14}$$

where α_{AH} is the heat transfer coefficient discussed above.

The equation of state:

$$p_H = \rho_H R T_H, \tag{15}$$

allows for calculation of the pressure in the heater.

2.3. *Basic equations for the cooler*

The volume V_{CL} of the cooler is much greater than the cylinder volume V_C. All remarks connected with the heater refer to the cooler as well. We select the angular velocity ω_1 of the cooler blower to achieve proper values of Re_{CL} and α_{ACL}.

We try to achieve α_{ACL}= O(10^3), and we calculate:

$$\dot{Q}_{ACL} = \alpha_{ACL} A_{CL} \left(T_{wCL} - T_{CL}\right) < 0\,. \tag{16}$$

The cooler wall temperature T_{wCL} is assumed constant in this model of the engine operation. The equation of mass conservation is:

$$\frac{d\rho_{CL}}{dt} = \frac{1}{V_{CL}}\left(\pm\dot{m}_1 \pm \dot{m}_4\right). \tag{17}$$

The signs of mass flow rates are positive if they flow towards the cooler. The equation of energy conservation for the cooler is as follows:

$$\begin{aligned}\frac{dT_{CL}}{dt} &= \frac{1}{\rho_{CL} V_{CL}}\left[-T_{CL}\left(\pm\dot{m}_1 \pm \dot{m}_4\right)+\right. \\ &\left.\kappa\left(\begin{Bmatrix} +\dot{m}_1 T_C \\ -\dot{m}_1 T_{CL} \end{Bmatrix}\begin{Bmatrix} +\dot{m}_4 T_C \\ -\dot{m}_4 T_{CL} \end{Bmatrix}\right)+\frac{\dot{Q}_{ACL}}{C_V}\right]\end{aligned} \tag{18}$$

The pressure in the cooler is calculated from the equation of state:

$$p_{CL} = \rho_{CL} R T_{CL}. \tag{19}$$

2.4. *Engine internal power and efficiency*

The internal work produced by 4-stroke engine cycle is:

$$L_i = \int\limits_{t=0}^{t=2\tau} p_C(t) \frac{dV}{d\alpha} \omega dt \tag{20}$$

where: τ is the period of the crank-shaft full turn, in the 4-stroke engine cycle is therefore 2τ.

The internal power of the engine is:

$$P_i = \frac{\omega}{2 \cdot 2\pi} L_i. \tag{21}$$

The heat delivered to the working air can be determined as:

$$Q_d = \int\limits_{t=0}^{t=2\tau} \dot{Q}_{AH} dt. \tag{22}$$

The internal efficiency of the engine is defined in the following way:

$$\eta_i = \frac{L_i}{Q_d}. \tag{23}$$

Besides, the loop of the engine operation has an internal accuracy checking system.

3. Results of the computer simulations

The numerical task is reduced to a solution of the set of 6 non-linear, ordinary differential equations. The Runge-Kutta procedure of the four order is employed in this solution. This set is supplemented with a set of state equations for pressure. A periodic, asymptotic solution for the nearly steady engine operation at a given angular velocity is presented in the paper.

A model of the engine operation, very similar to that one discussed above, presented in several papers, quoted in [2], was tested and verified experimentally during the investigation of the 2-stroke version of this engine. The results of the verification are to be found in [1].

The essential data for the engine dimension and outer parameters of its operation are as follows:

a) cylinder: diameter d_C =0.07 m, stroke s =0.06 m (V_C=230.9 10^{-6} m^3);

b) heater: tube diameter d_H =0.006 m, length l_H =3.0 m, number of tubes n_H = 20;

c) cooler: d_{CL}=0.008 m, l_{CL} =3.0 m, n_{CL} = 15.

The heater wall temperature T_{wH}=1073 K, the cooler wall temperature T_{wCL}=300 K. The angles of the valve opening and closing are:

$\alpha_{1,o}$= 0^o, $\alpha_{1,c}$=370^o, $\alpha_{2,o}$=465^o, $\alpha_{2,c}$=635^o.

Unfortunately, at the second valve it creates opening at 170^o what makes the cam slopes in the four stroke engine to be done of $170^o/2=85^o$ only (while cam shaft rotates two times slower).

A change of the stream from $\dot{m}_4$ to $\dot{m}_1$ (for valve No.1) is switched at $\alpha = 180^o$. For valve No.2, a change of stream $\dot{m}_2$ to $\dot{m}_3$ occurs at $\alpha = 540^o$. The amount of air mass enclosed in V_C, V_H and V_{CL} decides about the levels of higher (in the heater) and lower (in the cooler) pressures which occurs during the engine operation.

The engine starts using the electric starter with a given characteristics. When the mass air flows in the engine elements fulfil the condition pressure equal to 8 bar the electric starter works only during 3 turns of the engine shaft then the pressure rises in the cylinder and the asymptotic solution is reached. The typical result of the asymptotic solutions at n=2100 rpm is shown in Figure 2a)-d).

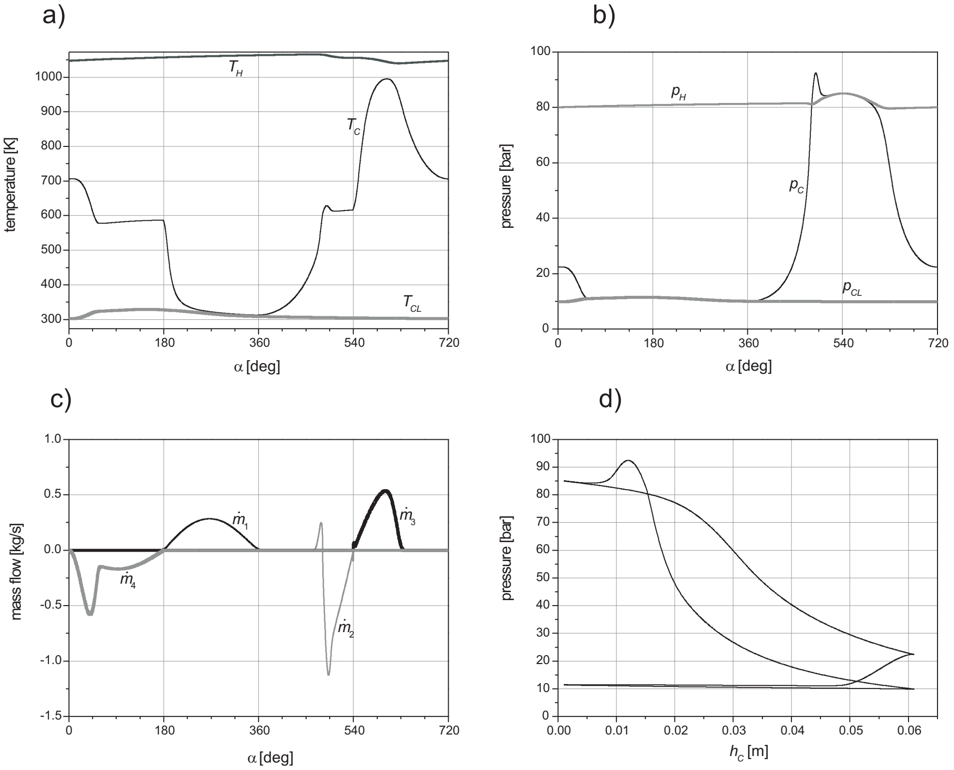

Figure 2 a) temperatures, b) pressures, c) air flows and d) $p-V_C$ diagram at asymptotic solution with engine rotation speed about 2100 rpm. Indexes: $_H$ – for heater, $_C$ for cylinder and $_{CL}$ for cooler, $_{1-4}$ for respective mass flows

The part a) presents the one cycle of the four-stroke engine temperatures diagram, here T_H marks the heater, T_C cylinder and T_{CL} cooler values. In Fig.2b) the pressure changes are shown while part c) contains air flows values. The part d) is a diagram of the $p - V_c$ dependence, here the area within the pressure loop is the work calculated accordingly to Eq.(20). The satisfying working capacity of the engine is characterized by the given geometrical parameters. Such results were obtained for the heater wall temperature of 1073 K and cooler wall of 300 K. Simulated asymptotic solutions were found stable. The cylinder volume V_C is not of highest importance due to fact it is small with respect to both heater and cooler volumes – V_H and V_{CL}, respectively. The amount of mass of air enclosed in those volumes has to be changed by way of numerical experiments.

Figure 3 shows the change of angular velocity of the engine shaft, and, Fig. 4 change of engine torque during one cycle of work of the 4 stroke engine at the asymptotic solution mentioned earlier. At Figure 5 final values reached at stable solutions of the engine efficiency, power and average rotation speed for different values of initial pressure in the engine are shown, but for T_{wH}=1073 K and T_{wCL}=300 K. Numerous iterations until stable state was reached (about 500 simulated cycles of engines work) convinced about proper behavior of the system.

At the higher state of the heater wall temperature $T_{wH} = 1273$ K we found higher values of the power P and efficiency η delivered – about 70 kW per 1 liter of volume and 31 % at rotation speed of 3000 rpm.

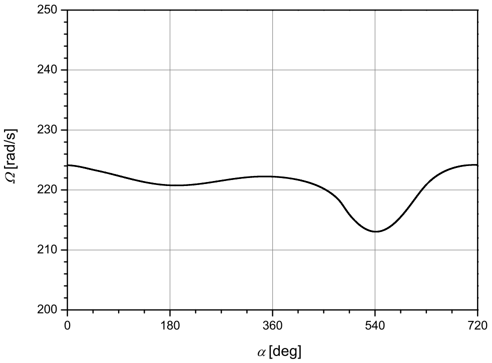

Figure 3 Change of engine rotation speed Ω at asymptotic solution at about 2100 rpm

4. Conclusions

The 4-stroke, externally heated engine that operates according to presented principle has been investigated. The computer simulation of its work proofs on the base of Fig. 5 that the engine can deliver of about 50 kW per 1 liter at the rotation speed of about 2580 rpm when the initial pressure is about 36 bar. For the presented set

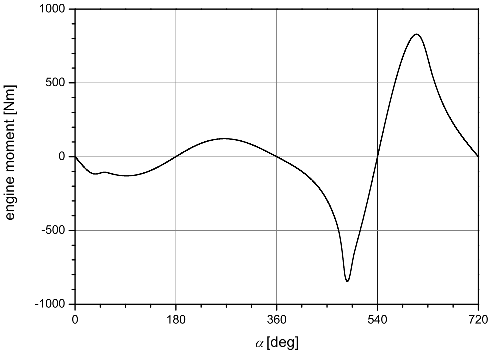

Figure 4 Engine moment at one cycle of work of the 4 stroke engine

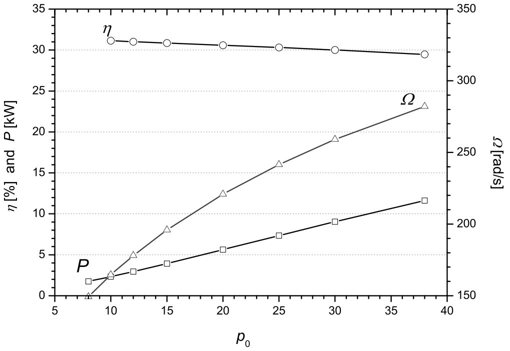

Figure 5 Final values of the engine efficiency ηg[%], power P [kW] and average rotation speed Ω [rad/s] for different values of initial pressure in the engine

of wall temperatures and compression ratio the internal efficiency reaches about 30 %.

The air engine proposed is an improved version of the EHVE (Externally Heated Valve Engine) investigated within 1999-2000 [1]. The elimination of the EHVE disadvantages has been done by replacing technologically complex elements in this version of engine. Here it is a blower working in the heater volume. In the near future it will be possible to manufacture such blower, and then proposed engine becomes a prospective solution of the externally heated engine.

The important parameter that decides about level of the engine power and efficiency are timings of valve No.2, which operates between cylinder and heater. An extensive analysis has led to conclusion that shortening of its opening time causes the increase of efficiency of the engine. Unfortunately, values of $\alpha_{2,o}$=465^o and $\alpha_{2,c}$=635^o is for the time being the optimal for the valve control. Only for such values of the valve timing for the 4-stroke engine it is possible to well made the valve cam.

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