Computer Assisted Mechanics and Engineering Sciences, 2: 129–139, 1995.

Computer simulation of a new type heat engine operation[1]

Lech Brzeski and Zbyszko Kazimierski
Technical University of Łódź (I-10), ul. Stefanowskiego 1/15, 90-924 Łódź, Poland

(Received September 3, 1994)

A new type of the externally heated engine is the subject of the present paper. Air can be used as a working medium of the engine. Heat delivered to the working air may come from a combustion chamber or another heat generator of an arbitrary type. The engine construction and the thermodynamic cycle performed by the engine are original ones. The engine operation has been investigated basing on the presented computer simulation. As a result, the time-dependent pressures and temperatures in each part of the engine have been determined. The engine power and efficiency have also been calculated. When the lowest basic pressure of the engine cycle is equal to 1 MPa, the power of 30 kW per 1 liter of the cylinder volume and the efficiency 0.38 at 1500 rev/min can be achieved. The main aim of this paper is to present a numerical investigation of the engine operation for higher values of the basic cycle pressures. It has been shown that for the pressure equal to 3 MPa, the power of about 100 kW per 1 liter and the efficiency 0.40 at 1500 rev/min can be theoretically obtained. The mechanical losses of the engine are not taken into account during the power and efficiency calculations.

1. Introduction

The properties and advantages of the external combustion engines have been quoted in [1] and [2]. The construction of the presented engine is an original one and it is subject of the patent [3].

The main advantages of the discussed engine if compared to the known Stirling engine [1, 2] are: the working medium is air, the conventional, simple crankshaft system may be used, and the oil lubrication is possible.

The engine is composed of four essential parts: expander, compressor, heater and cooler. This is a two-stroke engine (Fig. 1). The volume over piston 'pt' (Fig. 1) is called expander 'e' and the volume under the piston is called compressor 'c'.

The principle of the engine operation is shortly described below:

If the piston 'pt' starts to move down from its upper position, the governed valve 1 opens and high pressure air flows from the heater 'h' to the expander 'e' (Fig. 1) expanding during the piston movement. At the same time, the air contained in the volume 'c' (below the piston, Fig. 1) is compressed. When the pressure in the compressor reaches the level of the pressure of the discharging heater 'h', the self-acting valve 3 opens and the air exchange in the heater starts. Then, the governed valve 1 closes and further compression of the air in the heater is caused by the piston movement down until it reaches its lowest position. Valve 3 closes in this position and the isochoric heating begins in the heater. The piston starts to move up, the governed valve 2 opens and the air contained in the expander is pushed out to the cooler 'cl' (Fig. 1), simultaneously the self-acting valve 4 opens and the cold air flows into the compressor 'c' from the cooler 'cl'. Finally, the piston reaches its upper position, all valves are closed and the engine cycle ends.

The working air heating period may be prolonged if more than one heater is used. Two heaters, working commutatively, are employed in the presented engine.

The computational model presented in this paper is developed for the engine described in [4]. Especially, a more detailed description of the heater operation has been given.

[1] The paper has been presented at Japan–Central Europe Joint Workshop on Advanced Computing in Engineering, Pułtusk, Poland, September 26–29, 1994.

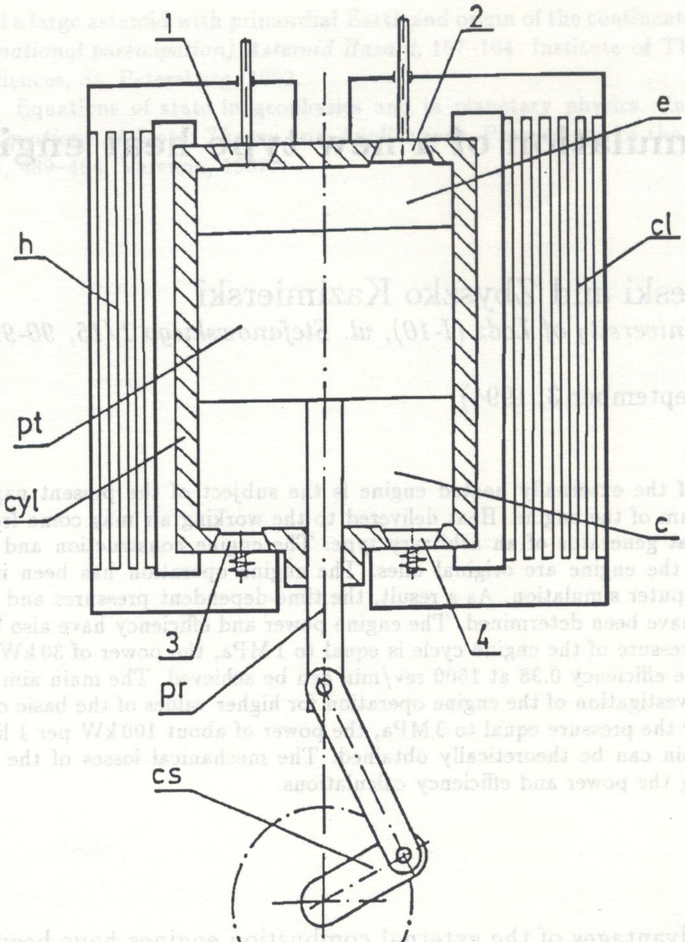

Fig. 1. Schematic diagram of the engine, 'e' — expander, 'c' — compressor, 'h' — heater, 'cl' — cooler, 'pt' — piston, 'cyl' — cylinder, 'cs' — crankshaft, 'pr' — piston rod, 1, 2 — governed valves, 3, 4 — self-acting valves

The previously published results, [4], were obtained for only one, relatively low basic pressure of the engine cycle equal to 1 MPa. The aim of this paper is to present a numerical investigation of the engine output parameters for higher values of the basic cycle pressure.

2. Computational Model

2.1. Basic equations and general assumptions

The air flow in the closed cycle of the engine is governed by the energy and mass conservation equations and by the equation of state. The equations of energy and mass are formulated for the migrating volume V_u surrounded by the moving closed surface A_u. The local velocity of the surface $\mathbf{u}$ may vary from zero (control surface) to the velocity of fluid elements $\mathbf{v}$ (fluid surface). The relative velocity of the air flowing through the moving surface is $\mathbf{w} = \mathbf{v} - \mathbf{u}$.

The energy equation takes the form

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\iiint\limits_{V_u} e_T \rho \,\mathrm{d}V_u\right) = -\iint\limits_{A_u} i_T \rho\, \mathbf{w}\cdot\mathbf{n}\,\mathrm{d}A_u - \iint\limits_{A_u} \mathbf{n}\cdot\mathbf{u}\, p\,\mathrm{d}A_u + \overbrace{\iint\limits_{A_u} \dot{q}_A\,\mathrm{d}A_u}^{Q_A} + \overbrace{\iiint\limits_{V_u} \dot{q}_V\,\mathrm{d}V_u}^{Q_V} \tag{1}$$

where: e_T — the total internal energy, i_T — the total entalpy, $\mathbf{n}$ — the unit normal vector, $\dot{q}_A$, $\dot{q}_V$ — the surface and volume heat flux densities, respectively.

The equation of mass conservation has the form

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\iiint\limits_{V_u} \rho \,\mathrm{d}V_u\right) = -\iint\limits_{A_u} \rho\, \mathbf{w}\cdot\mathbf{n}\, \mathrm{d}A_u\,. \tag{2}$$

The equation of state:

$$p = \rho_T(c_p - c_v)\,. \tag{3}$$

It is assumed that kinetic energy of the air inside the volumes of the engine elements is very small compared to its internal energy, and Eqs. (1), (2) and (3) will be solved taking mean values of the specific heats c_p, c_v in the range of the temperatures and pressures of the engine cycle. The surface heat flux $\dot{Q}_A$ is determined using the local heat transfer coefficient α_A. The details of the α_A determination are given in further part of the present paper devoted to the heater operation modelling.

Additionally, is it assumed that the gas parameters are only time-dependent. An exception to this assumption will be introduced separately for the heater. The mass and energy streams specified in the RHS of Eqs. (1) and (2) take place on several surfaces A_{uj}, $j = 1, 2, \ldots, J$.

Taking into account the above assumptions, Eqs. (1) and (2) may be rewritten in the form

$$c_v\frac{\mathrm{d}}{\mathrm{d}t}(T\rho V_u) = -\sum_{j=1}^{J} c_pT_j \underbrace{(\rho\, \mathbf{v}\cdot\mathbf{n}\, A_u)_j}_{\dot{m}_j} - p\frac{\mathrm{d}V_u}{\mathrm{d}t} + \underbrace{\sum_{l=1}^{L} (\dot{q}_A A_u)_l}_{\dot{Q}_A} + \dot{Q}_V\,, \tag{4}$$

$$\frac{\mathrm{d}}{\mathrm{d}t}(\rho V_u) = -\sum_{j=1}^{J} \underbrace{(\rho\, \mathbf{v}\cdot\mathbf{n}\, A_u)_j}_{\dot{m}_j} \tag{5}$$

where $\dot{m}_j$ are the mass flow rates through valves ($j = 1, 2$ for the expander, $j = 3, 4$ for the compressor and $j = 1, 3$ for the heater).

The mass flow rates $\dot{m}_j$, $(j = 1, 2, 3, 4)$ are calculated according to the known gas dynamics formulae supplemented by the coefficient which defines an effective valve cross-section flow area determined empirically for a given valve geometry and flow conditions [4].

The specified set of equations is employed for the simulation of the working air flow through the engine elements. The numerical models of the expander, compressor, heater and cooler are given below.

2.2. Modeling of the engine parts operation

2.2.1. Expander

The unknowns for the expander are: pressure p_e, temperature T_e and density ρ_e. The time-dependent volume of the expander is given by the formula:

$$V_e(t) = A_{ce}\left[h_{eo} + \frac{s}{2}(1 - \cos\omega t)\right] \tag{6}$$

where $A_{ce} = \frac{\pi}{4}d_c^2$, d_c — cylinder diameter, h_{eo} — clearance over the piston in its upper position, s — stroke, ω — angular velocity of the crankshaft.

The crankshaft angle is denoted by $\theta = \omega t$ and the cross-section of the governed valve area A_j ($j = 1,2$) is described by the formula

$$A_j(\theta) = \frac{A_{j\,\max}}{2}\left[1 - \cos\pi\frac{\theta - \theta_{j,k}}{\Delta\theta_{j,k}}\right]. \tag{7}$$

The subscript $k = 0,2$ determines the moment of j-th valve opening ($k = 0$) and closing ($k = 2$). $A_{j\,\max}$ is the maximum value of the j-th valve cross-section area and $\Delta\theta$ is the angle determining the time of A_j variation from 0 to $A_{j\,\max}$.

It is assumed that for the expander $\dot{Q}_A + \dot{Q}_V = 0$, which means that the heat produced inside the cylinder (by friction forces) is transferred out through the cooled walls of this cylinder.

The set of Eqs. (3), (4) and (5) for the expander may be presented in the form

$$p_e = \rho_e T_e (c_p - c_v), \tag{8}$$

$$\frac{dT_e}{d\theta} = \frac{1}{\omega\rho_e V_e}\left[\dot{m}_1(\kappa T_h - T_e) - \dot{m}_2(\kappa - 1)T_e - \frac{\omega p_e}{c_v}\frac{dV_e}{d\theta}\right], \tag{9}$$

$$\frac{d\rho_e}{d\theta} = \frac{1}{\omega V_e}\left(\dot{m}_1 - \dot{m}_2 - \rho_e\omega\frac{dV_e}{d\theta}\right). \tag{10}$$

The algebraic formulae for $\dot{m}_1$ and $\dot{m}_2$ are given explicitly in [4].

2.2.2. Compressor

The unknowns for the compressor are: pressure p_c, temperature T_c and density ρ_c. The time-dependent compressor cylinder volume is

$$V_c(t) = A_{cc}\left[h_{co} + \frac{s}{2}(1 + \cos\omega t)\right] \tag{11}$$

where $A_{cc} = \frac{\pi}{4}(d_c^2 - d_{pr}^2)$, d_{pr} — piston rod diameter (Fig. 1), h_{co} — clearance under the piston at its lowest position.

The mass flow rates $\dot{m}_3$ and $\dot{m}_4$ through the self-acting valves no. 3 and 4, Fig. 1, are calculated similarly as for the governed valves [4]. However, the cross-section areas for these valves are calculated simultaneously with the whole engine operation process, because A_3 and A_4 depend on the pressures in the compressor p_c, heater p_h and cooler p_{cl}.

The self-acting valve movement is determined using additional differential equations, taking into account the inertia of their movable parts. The mass is supported by a spring and the viscotic damping is introduced into the model.

The angle of the valve 3 opening $\theta_{3,0}$ is determined from the condition $p_c = p_h$ and $\theta_{4,0}$ from $p_c = p_{cl}$. Also for the compressor, it was assumed that $\dot{Q}_A + \dot{Q}_V = 0$.

The set of Eqs. (3), (4) and (5) for the compressor takes the following form:

$$p_c = \rho_c T_c (c_p - c_v), \tag{12}$$

$$\frac{dT_c}{d\theta} = \frac{1}{\omega\rho_c V_c}\left[\dot{m}_3(\kappa T_{cl} - T_c) - \dot{m}_4(\kappa - 1)T_c - \frac{\omega p_c}{c_v}\frac{dV_c}{d\theta}\right], \tag{13}$$

$$\frac{d\rho_c}{d\theta} = \frac{1}{\omega V_c}\left(\dot{m}_3 - \dot{m}_4 - \rho_c\omega\frac{dV_c}{d\theta}\right). \tag{14}$$

The algebraic formulae for $\dot{m}_3$ and $\dot{m}_4$ are given explicitly in [4].

2.2.3. Cooler

The cooler volume is much greater than the volumes of other engine parts. Taking it into account, the isobaric model of air cooling with constant pressure p_{cl} is assumed. The air mass contained in the cooler is almost constant. The pressure p_{cl} is the lowest, basic pressure of the engine cycle. There exits a link between p_{cl} and the total air mass enclosed in the engine volumes $V_e + V_c + V_h + V_{cl}$ for a given rotational frequency. The level of the basic pressure p_{cl} is controlled by introducing the proper total mass of the working air into the engine volumes before the engine starts. The influence of different values of p_{cl} on the engine power and efficiency is discussed in this paper. The cooler works as a stationary heat exchanger which causes the working air temperature drop from $T_{e\,min}$ to T_{cl}, where $T_{e\,min}$ is the lowest temperature of the air leaving the expander, and T_{cl} is the temperature of the air at the compressor inlet.

2.2.4. Heater

The unknowns for the heater are: pressure p_h, temperature T_h and density ρ_h.

The heater volume V_h is not a function of time. The surfaces surrounding V_h are not mobile. The heat production connected with the friction forces acting inside the heater $\dot{Q}_{V_h}$ is neglected as it is very small when compared to the heat flux $\dot{Q}_{A_h}$ delivered by the heating medium.

In the range $\theta \geq \theta_{3,0}$ (after opening of valve 3) the working air exchange occurs. The hot air flowing into the expander is pushed out by the comparatively cool air coming from the compressor.

Two heater operational models will be considered. In the first model, the temperature $T_h(\theta)$ represents the whole internal energy of the air in the heater, and the density $\rho_h(\theta) = M_h(\theta)/V_h$ represents the varying mass $M_h(\theta)$ of the air contained in the heater. This model assumes immediate mixing of the hot and cool air in the heater after opening of the valve 3. The second model admits mixing of the hot and cool air in the finite period of time and takes into consideration the air temperature differences along the heater tube length.

For the first model in the range $\theta_{1,0} \leq \theta \leq \theta_{3,2}$ (when the valves 1 and 3 are open), the heat flux $\dot{Q}_{A_h}$ delivered through heater surface area $A_h = z_h \pi d_h l_h$ (z_h — number of tubes of diameter d_h and length l_h), is described by the equation

$$\dot{Q}_{A_h} = \alpha_A A_h (T_{wh} - T_h) \tag{15}$$

where T_{wh} is the heater tube wall temperature. The heat transfer coefficient α_A is determined as the function of the Nusselt number Nu, i.e. $\alpha_A = \frac{Nu\lambda}{d_h}$, $Nu = f(Re, Pr)$. λ, Re and Pr are the air heat conductivity, the Reynolds and Prandtl numbers for the air flowing inside the heater tubes, respectively. The values of the specific heat c_p, viscosity μ and quoted above λ, used in the calculations for Re and Pr numbers, were taken as averaged values of these parameters in the range of temperatures and pressures occuring for $\theta_{1,0} \leq \theta \leq \theta_{3,2}$. The velocity of the air inside the tubes necessary for the determination of Reynolds number has also been taken as the mass averaged value for $\theta_{1,0} \leq \theta \leq \theta_{3,2}$. The unsteadiness of the air flow inside the tubes, which may cause an increase in α_A, was ignored in this calculation procedure.

The basic equations of state, energy and mass for the first model of the heater operation, derived from (3), (4) and (5), take for $\theta_{1,0} \leq \theta \leq \theta_{3,2}$ the following form,

$$p_h = \rho_h T_h (c_p - c_v), \tag{16}$$

$$\frac{dT_h}{d\theta} = \frac{1}{\omega \rho_h V_h} \left[\dot{m}_3(\kappa T_c - T_h) - \dot{m}_1(\kappa - 1) T_h + \frac{\dot{Q}_{A_h}}{c_v} \right], \tag{17}$$

$$\frac{d\rho_h}{d\theta} = \frac{1}{\omega V_h} (\dot{m}_3 - \dot{m}_1). \tag{18}$$

For $\theta \geq \theta_{3,2}$ the heater volume is closed. The density of the air inside the heater is constant and the heat transfer is isochoric. This period of heating lasts for the first heater till the end of the running cycle and during the whole next cycle, it means for $\theta_{3,2} \le \theta \le 4\pi$. At $\theta = 2\pi$ the heating period of the second heater is finished and this heater is switched into the engine operation. The changes of $T_{\rm h}(\theta)$ and $p_{\rm h}(\theta)$ for this period of the engine work are calculated from the known isochoric process relations.

It has been assumed that during isochoric period of heating a kind of whirl movement of the air inside the heater tubes takes place. The local values of the Reynolds number connected with this movement have been estimated and the heat transfer coefficients α_A were determined as the functions of the Nusselt, Prandtl and Reynolds numbers.

The second model of the heater constitutes only a certain extension of the first one. It has been assumed that the pressure is the same in the whole volume of the heater and the solution for $p_{\rm h}(\theta)$ is taken from the first model. The amount of heat delivered to the working air is the same in both models.

The characteristic feature of the second model is that the whole space of the heater $V_{\rm h}(\theta)$ has been divided into n elementary control volumes and the temperature differences, not only in time, but also along the heater tube length, have been determined.

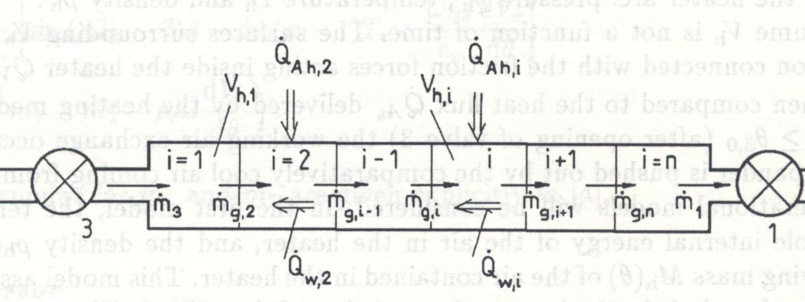

Fig. 2. Schematic diagram of the heater divided into n elementary control volumes

The whole heater volume $V_{\rm h}(\theta)$ is divided into $i = 1, 2, \ldots, n$ parts so that the elementary control volumes and elementary heating surface areas are:

$$V_{{\rm h},i} = \frac{V_{\rm h}}{n}, \qquad A_{{\rm h},i} = \frac{A_{\rm h}}{n}. \tag{19}$$

The scheme of the heater division and the nomenclature used is explained in Fig. 2. The solution for the temperatures $T_{{\rm h},i}(\theta)$ $(i = 1, 2, \ldots, n)$ are received on the base of the following set of energy conservation equations:

$$\frac{{\rm d}T_{\rm h,1}}{{\rm d}\theta} = \frac{1}{\omega\rho_{\rm h,1}V_{\rm h,1}}\left[\dot m_3(\kappa T_{\rm c} - T_{\rm h,1}) - \dot m_{\rm g,2}(\kappa-1)T_{\rm h,1} + \frac{\dot Q_{\rm w,1}}{c_v} + \frac{\dot Q_{A_{\rm h},1}}{c_v}\frac{\rho_{\rm h,1}}{\rho_{\rm h}}\right]; \quad \text{for the first cell } (i=1),$$

$$\frac{{\rm d}T_{{\rm h},i}}{{\rm d}\theta} = \frac{1}{\omega\rho_{{\rm h},i}V_{{\rm h},i}}\left[\dot m_{{\rm g},i}(\kappa T_{{\rm h},i-1} - T_{{\rm h},i}) - \dot m_{{\rm g},i+1}(\kappa-1)T_{{\rm h},i} + \frac{\dot Q_{{\rm w},i}}{c_v} + \frac{\dot Q_{A_{\rm h},i}}{c_v}\frac{\rho_{{\rm h},i}}{\rho_{\rm h}}\right]; \quad \text{for the } i\text{-th cell, where } i = 2, 3, \ldots, n-1, \tag{20}$$

$$\frac{{\rm d}T_{{\rm h},n}}{{\rm d}\theta} = \frac{1}{\omega\rho_{{\rm h},n}V_{{\rm h},n}}\left[\dot m_{{\rm g},n}(\kappa T_{{\rm h},n-1} - T_{{\rm h},n}) - \dot m_1(\kappa-1)T_{{\rm h},n} + \frac{\dot Q_{{\rm w},n}}{c_v} + \frac{\dot Q_{A_{\rm h},n}}{c_v}\frac{\rho_{{\rm h},n}}{\rho_{\rm h}}\right]; \quad \text{for the } n\text{-th cell } (i=n).$$

The density in each cell is calculated from the formula

$$\rho_{{\rm h},i} = \frac{p_{\rm h}(\theta)}{T_{{\rm h},i}(c_p - c_v)}. \tag{21}$$

The density ρ_h used in (20) is the same as the one defined in the first model, additionally, it is determined by the formula

$$\rho_h = \sum_{i=1}^{n} \frac{\rho_{h,i} V_{h,i}}{V_h} .$$

The internal mass flow rates between the cells are determined from the equations of the mass conservation:

$$\dot{m}_{g,i} = \dot{m}_{g,i-1} - \omega V_{h,i-1} \frac{d\rho_{h,i-1}}{d\theta} , \quad \text{for } i = 1, 2, \ldots, n, \tag{22}$$

where $\dot{m}_{g,i-1} = \dot{m}_3$ for $i = 1$.

The heat fluxes $Q_{A_h,i}$ are calculated from the formula analogous with (15), i.e.:

$$\dot{Q}_{A_h,i} = \alpha_A A_{h,i} (T_{wh} - T_{h,i}) \tag{23}$$

where α_A has been calculated according to the first model procedure.

For the isochoric process, i.e. in the range $\theta_{3,2} \leq \theta \leq 4\pi$, additional internal heat fluxes $Q_{w,i}$ between cells, caused by the mentioned whirl motion, are introduced. These heat streams are proportional to the assumed local mass flow rates $\dot{m}_T(\theta)$ of the air circulating between cells so that:

$$\begin{aligned} \dot{Q}_{w,1} &= \dot{m}_T(\theta) c_p (T_{h,2} - T_{h,1}) , && \text{for } i = 1 , \\ \dot{Q}_{w,i} &= \dot{m}_T(\theta) c_p (T_{h,i-1} - 2T_{h,i} + T_{h,i+1}) , && \text{for } i = 2, 3, \ldots, n-1 , \\ \dot{Q}_{w,n} &= \dot{m}_T(\theta) c_p (T_{h,n-1} - T_{h,n}) , && \text{for } i = n . \end{aligned} \tag{24}$$

The mixing air streams $\dot{m}_T(\theta)$ (same for all the cells) diminish in time according to the formula:

$$\dot{m}_T(\theta) = \dot{m}_{T0} e^{-\frac{\Delta_T}{\omega}(\theta - \theta_{3,2})} . \tag{25}$$

The coefficient Δ_T determines the intensity of $\dot{m}_T(\theta)$ diminishing in time. The details connected with the internal heat streams $\dot{Q}_{w,i}$ will be explained in the future papers.

2.3. Engine power and cycle efficiency

The mechanical work produced or consumed in the range of $\theta_A \leq \theta \leq \theta_B$ by the expander is given by the formula

$$L_{eA-B} = \int_{\theta_A}^{\theta_B} p_e(\theta) \frac{dV_e}{dt} \frac{d\theta}{\omega} . \tag{26}$$

The derivative $\frac{dV_e}{dt}$ may be calculated from (6).

The compressor work in the range of $\theta_A \leq \theta \leq \theta_B$ is

$$L_{cA-B} = \int_{\theta_A}^{\theta_B} p_c(\theta) \frac{dV_c}{dt} \frac{d\theta}{\omega} . \tag{27}$$

The derivative $\frac{dV_c}{dt}$ can be calculated from (11).

The sum of these two parts of the work for the whole cycle is

$$\sum L_{e+c} = L_{e0-2\pi} + L_{c0-2\pi} . \tag{28}$$

The engine power is determined as follows

$$P = \frac{\omega}{2\pi} \sum L_{e+c} . \tag{29}$$

The heat introduced into the heater for the range $\theta_{1,0} \le \theta \le \theta_{3,2}$ is calculated from the formula

$$Q_{\mathrm{h\,I}} = A_{\mathrm{h}} \int_{\theta_{1,0}}^{\theta_{3,2}} \alpha_A (T_{\mathrm{wh}} - T_{\mathrm{h}}) \frac{\mathrm{d}\theta}{\omega} . \tag{30}$$

For the range $\theta_{3,2} \le \theta \le 4\pi$, when the isochoric heating takes place, the amount of the delivered heat is

$$Q_{\mathrm{h\,II}} = c_v M_{\mathrm{h}}(\theta_{3,2}) [T_{\mathrm{h}}(4\pi) - T_{\mathrm{h}}(\theta_{3,2})] \tag{31}$$

where $M_{\mathrm{h}}(\theta_{3,2})$ is the constant amount of the air mass closed in the heater during the isochoric heating period.

The total heat delivered during the cycle to the working air is

$$\sum Q_{\mathrm{h}} = Q_{\mathrm{h\,I}} + Q_{\mathrm{h\,II}} . \tag{32}$$

The engine cycle efficiency is calculated from the formula

$$\eta_c = \frac{\sum L_{\mathrm{e+c}}}{\sum Q_{\mathrm{h}}} . \tag{33}$$

The engine cycle efficiency, calculated according to (33), does not take into account mechanical losses caused mainly by piston rings friction. The mechanical losses may consume up to 10% of the engine power. Therefore, the work and power calculated according to (28) and (29), respectively, can be determined as the indicated ones.

3. Results of computer simulation

An example of the engine being subject of the presented computer simulation has the cylinder volume of 1 liter ($d_c = 0.13\,\mathrm{m}$, $s = 0.0753\,\mathrm{m}$).

Additionally, the following data have been accepted: $V_{\mathrm{h}} = 0.33 \times 10^{-3}\,\mathrm{m}^3$, $V_{\mathrm{cl}} = 5 \times 10^3\,\mathrm{m}^3$, $A_{j\,\max} = 7 \times 10^{-4}\,\mathrm{m}^2$ $(j = 1, 2, 3, 4)$, $\omega = 157\,\mathrm{rad/s}$, $\delta_{ME} = \theta_{1,2} - \theta_{3,0} = 12°$.

The wide discussion of the engine power and efficiency η_c, determined according to the formulae (29) and (33) as the functions of the rotational frequency ω, the heater volume V_{h} and the heater wall temperature T_{wh} for one value of the cooler pressure $p_c = 1\,\mathrm{MPa}$, has been presented in [4]. The preliminary optimization of the engine cycle for $p_{\mathrm{cl}} = 1\,\mathrm{MPa}$ shows that the power of 30 kW per 1 liter of cylinder volume at 1500 rev/min can be obtained.

The extension of the heater model introduced in this paper allows for estimation of the spatial differences of the heater temperature during the engine cycle. A very simple example of such calculations has been performed. The heater volume has been divided into four ($n = 4$) parts. The results of temperature distribution are presented in Fig. 3a. It appears that $\dot{m}_{gi}$ are very small for $\theta \ge \theta_{3,2}$ (Fig. 3c), so the mixing of the cool and hot air inside the closed heater may be caused only by streams $\dot{m}_T(\theta)$. If one assumes that $\dot{m}_{T0} \approx \dot{m}_{g1}(\theta_{3,2} - 10°)$ (i.e. $\dot{m}_{T0} \approx 0.1$), the internal heat fluxes $\theta_{\mathrm{w},i}$ cause a reduction of the temperature differences along the heater tube length in the range $\theta_{3,2} \le \theta \le 4\pi$. An illustration of this process is shown in Fig. 3a. For lower values of $\dot{m}_{T0}$, e.g. $\dot{m}_{T0} = 0.01$, the difference reduction $T_{\mathrm{h},i}$ is caused only by $\theta_{A_{\mathrm{h}},i}$ and it is very slow, so that at the end of the isochoric heating period, i.e. for $\theta = 4\pi$, the differences $(T_{\mathrm{h},4} - T_{\mathrm{h},1})$ may reach about 240 K. The problem of the spatial distribution of the air temperature inside the heater volume needs further investigations and will be the topic of future papers. The main aim of this paper is determination of the engine power and efficiency (according to (29) and (33)) for a higher value of the basic pressure of the cycle, i.e. the cooler pressure p_{cl}. This pressure is changed in the range from 1 up to 3 MPa.

Figure 4 presents the periodic solution of the problem in the range of one engine cycle for $p_{\mathrm{cl}} = 3\,\mathrm{MPa}$. The first model of the heater operation has been employed in this calculation (without the temperature T_{h} being differentiated in the heater space). The diagrams of the temperatures,

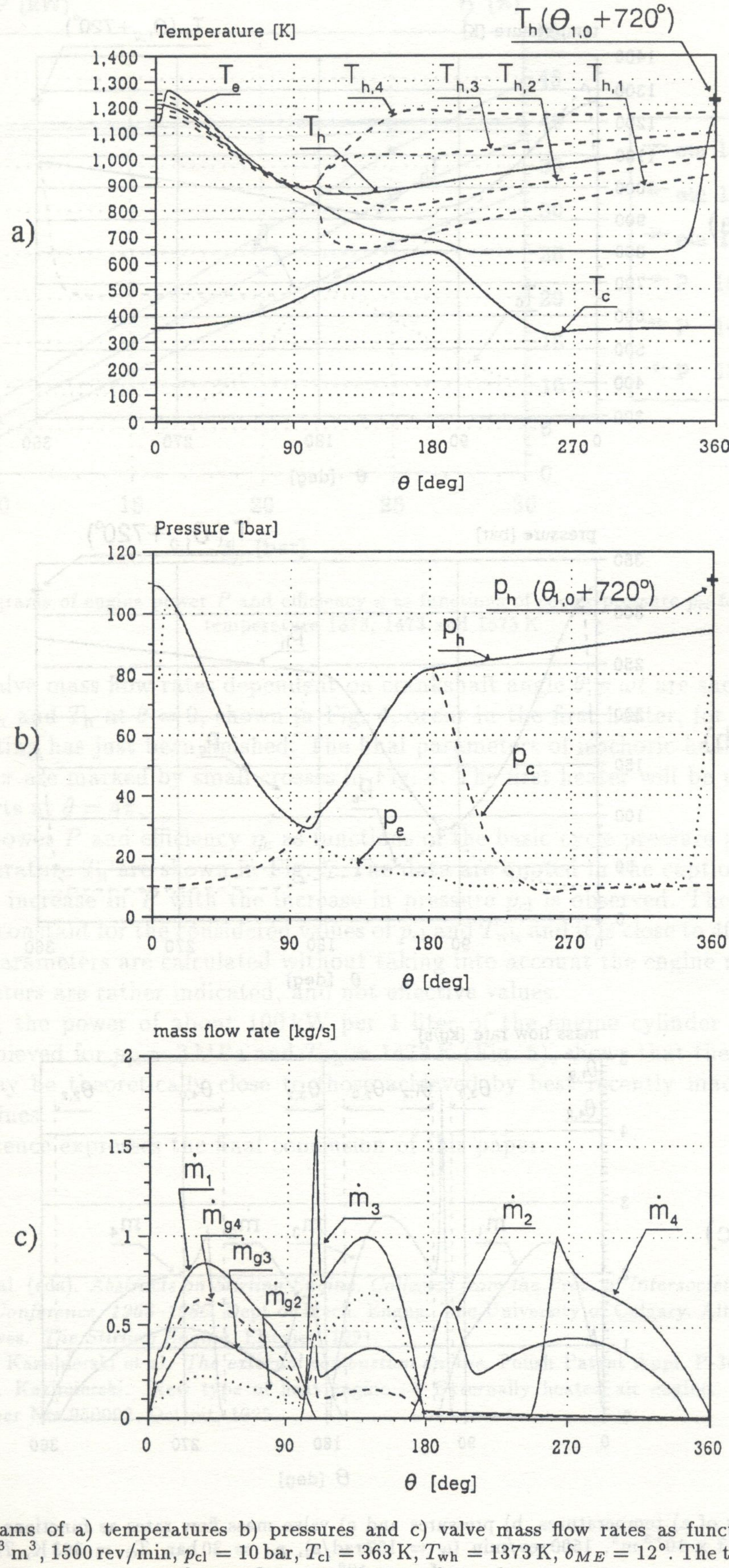

Fig. 3. Diagrams of a) temperatures b) pressures and c) valve mass flow rates as functions of angle θ: $V_h = 0.33\times10^{-3}$ m^3, 1500 rev/min, $p_{cl} = 10$ bar, $T_{cl} = 363$ K, $T_{wh} = 1373$ K, $\delta_{ME} = 12°$. The temperatures $T_{h,i}$ and mass flow rates $\dot{m}_{gi}$ are shown for $n = 4$

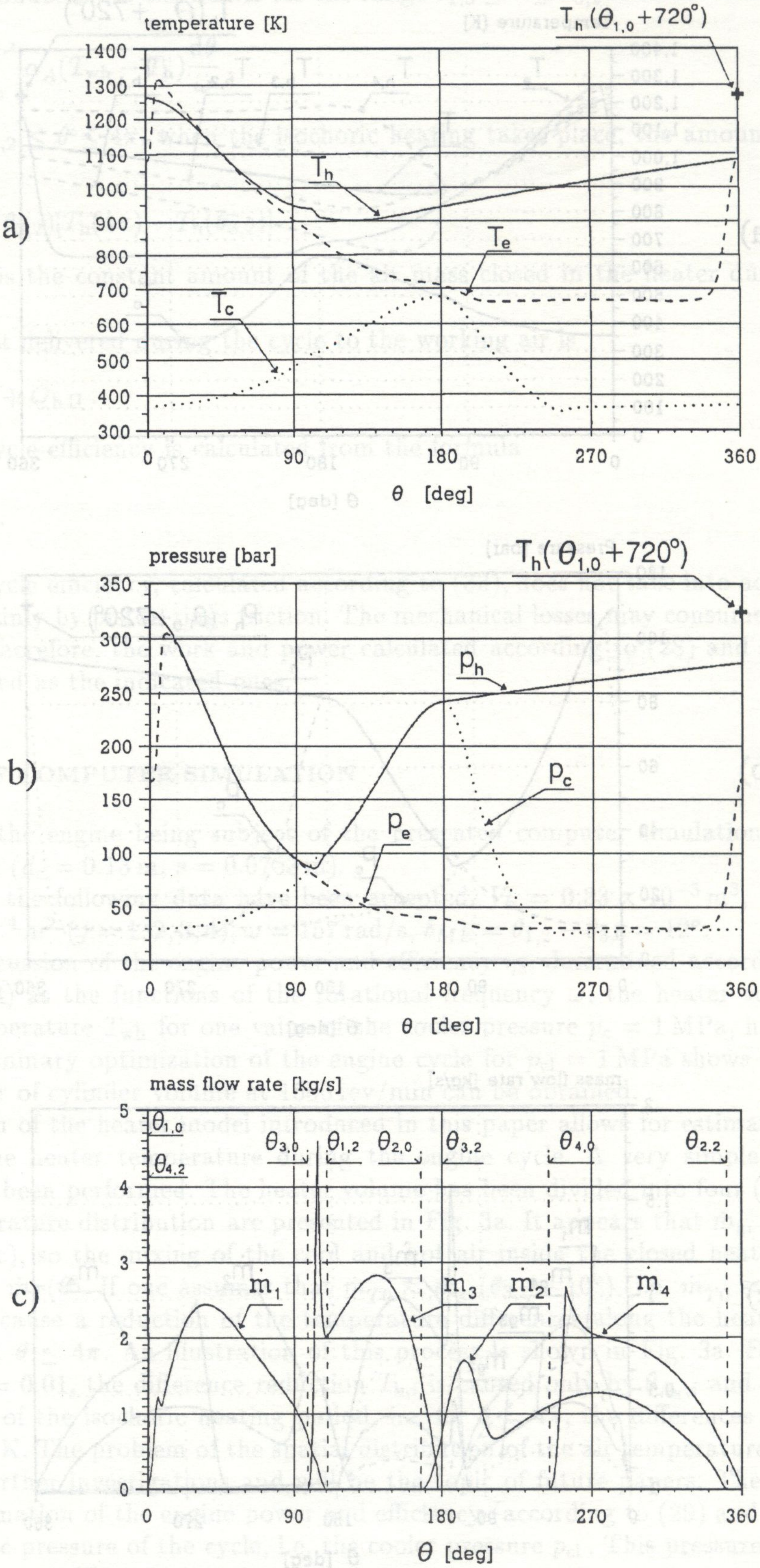

Fig. 4. Diagrams of a) temperatures, b) pressures and c) valve mass flow rates as functions of crankshaft angle θ: $V_h = 0.33 \times 10^{-3}\,\mathrm{m}^3$, 1500 rev/min ($\omega = 157$ rad/s), $p_{cl} = 30$ bar, $T_{cl} = 363$ K, $T_{wh} = 1473$ K, $\delta_{ME} = 12°$

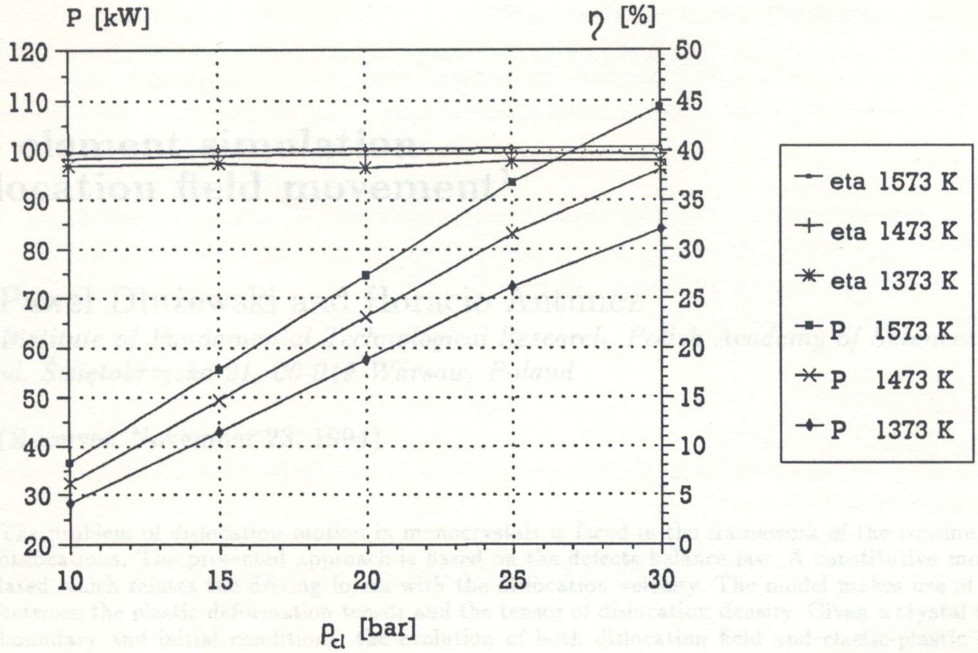

Fig. 5. Diagrams of engine power P and efficiency η as functions of cooler pressure p_{cl} for heating wall temperature 1373, 1473 and 1573 K

pressures and valve mass flow rates dependent on crankshaft angle $\theta = \omega t$ are shown in this figure. The values of p_h and T_h at $\theta = 0$, shown in Fig. 4, occur in the first heater, for which the period of isochoric heating has just been finished. The final parameters of isochoric heating in the second heater for $\theta = 2\pi$ are marked by small crosses in Fig. 4. The first heater will be used again for the cycle which starts at $\theta = 4\pi$.

The engine power P and efficiency η_c as functions of the basic cycle pressure p_{cl} and the heater tube wall temperature T_h are shown in Fig. 5. The data are quoted in the caption of this figure.

A significant increase in P with the increase in pressure p_{cl} is observed. The engine efficiency remains almost constant for the considered values of p_{cl} and T_{wh} and it is close to 40%. The discussed engine output parameters are calculated without taking into account the engine mechanical losses, so these parameters are rather indicated, and not effective values.

Nevertheless, the power of about 100 kW per 1 liter of the engine cylinder volume, which is theoretically achieved for $p_{cl} = 3$ MPa and $T_{wh} = 1473$ K (Fig. 5), shows that the presented engine performance may be theoretically close to those achieved by best recently made piston internal combustion engines.

The last sentence expresses the final conclusion of this paper.

References

[1] G. Walker et al. (eds). *Abstracts on Stirling Engine. Collected from the Proc. of Intersociety Energy Conversion Engineering Conference, 1966–1986.* Dept of Mech. Engng. The University of Calgary. Alberta, Canada, 1987.

[2] C.H. Hargreaves. *The Stirling Engine.* Elsevier, 1991.

[3] L. Brzeski, Z. Kazimierski et al. *The external combustion engine.* Polish Patent Appl. P-303619, 1994.

[4] L. Brzeski, Z. Kazimierski. New type of heat engine — Externally heated air engine. *SAE Int. Congress*, Technical Paper No. 950092. Detroit, 1995.