Chapter 2
Early Design of the EHVE – Calculation Method and Thermodynamic Cycle

The EHVE operation has been explained in the Introduction, whereas the first publications devoted to this idea are [1, 2]. Besides, this design has been presented in [3, 4]. The experimental investigations published in [5] refer to a slightly altered early version of the EHVE, already mentioned in the Introduction, which will be presented in detail in Chap. 3.

2.1 Principle of the Early EHVE Operation

The early EHVE is shown in Fig. 2.1. It is composed of four essential parts: an expander, a compressor, a heater and a cooler. This is a 2-stroke engine. The volume over the piston is called an expander and that under it – a compressor.

A principle of the engine operation is shortly described below. If the piston starts to move downwards from its upper position, governed valve 1 opens and a high pressure working fluid flows from the heater to the expander, expanding during the further piston movement. At the same time, the fluid contained in the volume of the compressor (below the piston) is compressed. When the pressure in the compressor reaches the level of that in the discharged heater, self-acting valve 3 opens and the fluid exchange in the heater starts. Then, the governed valve closes and further compression of the working fluid in the heater is caused by the piston movement downwards until it reaches its lowest position. Here, valve 3 closes and the process of isochoric heating begins in the heater volume. The piston starts to move up, governed valve 2 opens and the fluid contained in the expander is pushed out to the cooler, simultaneously self-acting valve 4 opens and the cold fluid flows into the compressor from the cooler. Finally, the piston reaches its upper position, all valves are closed and the engine cycle is completed.

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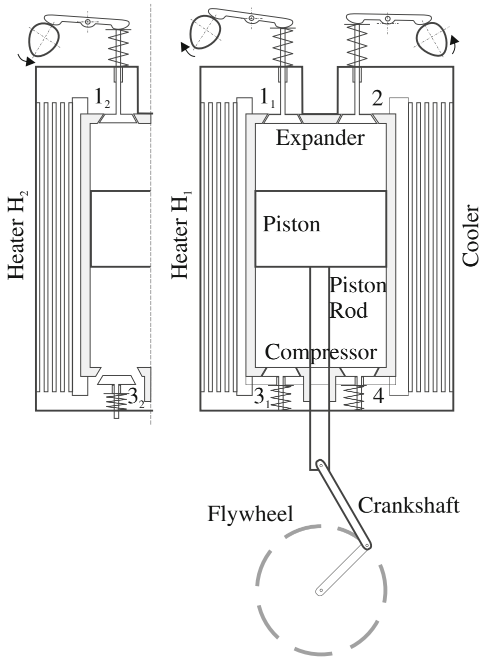

Fig. 2.1 Schematic diagram of the engine, *1*, *2* governed valves, *3*, *4* self-acting valves

The working air heating period can be prolonged if more than a single heater is used. Two such devices, working commutatively, are employed in the engine version under discussion. The heater H_1 is connected to the cylinders with a set of valves 1_1 and 3_1 and the heater H_2 with another set of valves 1_2 and 3_2. Besides, both cylinders use valves 2 and 4 connecting them to the cooler volume.

2.2 Basic Equations of the Theoretical Model

The flow of the working fluid in the closed cycle of the engine work is governed by the following set of equations – the energy and mass conservation equation and the equation of state. Here, this set is written for one of the heaters. 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 $\mathbf{u}$ can vary from zero, the assumed control surface, to the velocity of fluid elements $\mathbf{v}$, assumed as the fluid surface. The relative velocity of the fluid flowing through the moving surface is denoted as $\mathbf{w} = \mathbf{v} - \mathbf{u}$.

The energy equation takes the form

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\iiint_{V_U} e_T \rho \,\mathrm{d}V_U\right) = -\iint_{A_U} i_T \rho \mathbf{wn}\,\mathrm{d}A_U - \iint_{A_U} \mathbf{nu} p\,\mathrm{d}A_U + \overbrace{\iint_{A_U} \dot{q}_A \,\mathrm{d}A_U}^{\dot{Q}_A} + \overbrace{\iiint_{V_U} \dot{q}_V \,\mathrm{d}V_U}^{\dot{Q}_V}, \tag{2.1}$$

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

The equation of mass conservation has the form as follows

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\iiint_{V_U} \rho \,\mathrm{d}V_U\right) = -\iint_{A_U} \rho \mathbf{wn}\,\mathrm{d}A_U. \tag{2.2}$$

Finally, the equation of state is expressed as

$$p = \rho T (c_p - c_v). \tag{2.3}$$

It is assumed that the kinetic energy of the fluid inside the volumes of the engine elements is very low when compared to its internal energy, and Eqs. (2.1)–(2.3) will be solved taking mean values of the specific heats c_p and c_v in the range of temperatures and pressures of the engine cycle. The surface heat flux $\dot{Q}_A$ is determined with the local heat transfer coefficient α_A. The details of the α_A determination are given in the further part of this section devoted to the heater operation modelling.

Additionally, it is assumed that the fluid parameters are only time-dependent. As regards the cylinder, it is obvious but in the case of the heater and the cooler, it will be explained below. The heater and the cooler are assumed as counter-current heat exchangers, where temperature drops almost continuously along their lengths. We assume that all important changes in the heater and the cooler are the same along them. The collectors of heat exchangers are not taken into account as they are included in their volumes.

An exception to this assumption will be introduced separately for the heater further on. The mass and energy streams specified on the RHS of Eqs. (2.1) and (2.2) can be rewritten in the form

$$c_v \frac{\mathrm{d}\,(T \rho V_U)}{\mathrm{d}t} = -\sum_{j=1}^{J} c_p T_j \underbrace{(\rho \mathbf{vn} A_U)_j}_{\dot{m}_j} - p\frac{\mathrm{d}V_U}{\mathrm{d}t} + \sum_{l=1}^{L} \underbrace{(\dot{q}_A A_U)_l}_{\dot{Q}_A} + \dot{Q}_V, \tag{2.4}$$

$$\frac{\mathrm{d}\,(\rho V_U)}{\mathrm{d}t} = -\sum_{j=1}^{J} \underbrace{(\rho \mathbf{vn} A_U)_j}_{\dot{m}_j}, \tag{2.5}$$

where $\dot{m}_j$ are mass flow rates passing through valves ($j = 1, 2$ for the expander, $j = 3, 4$ for the compressor, and $j = 1, 3$ for the heater, correspondingly) and only one small heater is considered now.

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 the effective valve cross-section flow areas determined empirically for a given valve geometry and flow conditions [1–3]. The formulae for $\dot{m}_j$ are stated for quasi-stationary treatment of the air flow through valves. The set for $\dot{m}_j$ has to be solved always with the current values of pressures and temperatures in selected elements of the engine. This set of equations is employed in simulations of the working fluid flow through the engine elements. The numerical models of the expander, the compressor, the heater and the cooler are presented in the subsections below.

2.3 Modelling Operation of the Engine Parts

2.3.1 Expander

The unknown variables for the expander are as follows: the pressure p_E, the temperature T_E and the density ρ_E. The time-dependent volume of the expander comes from the simple geometry

$$V_E(t) = A_{CE}\left(h_{E0} + \frac{s}{2}(1 - \cos\,\omega t)\right) \tag{2.6}$$

where d_C is cylinder diameter, h_{E0} – clearance over the position at its upper point, s – piston stroke and the cross-section area is defined by $A_{CE} = \frac{\pi d_C^2}{4}$. The angular velocity of the crankshaft is ω.

The current value of the crankshaft angle is denoted by $\alpha = \omega t$ and the cross-section area of governed valves is described by the formula

$$A_j(\alpha) = \frac{A_{j\,max}}{2}\left(1 - \cos\pi\frac{\alpha - \alpha_{j,k}}{\Delta\alpha_{j,k}}\right). \tag{2.7}$$

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

It is assumed for the expander that the heat produced inside the cylinder by friction forces is transferred out through the cooled walls of this cylinder, which can be described as

$$\dot{Q}_A + \dot{Q}_V = 0. \tag{2.8}$$

This is an approximation because $\dot{Q}_V$ contains not only friction effects but also a small amount of the heat transferred with hot fluid during the expansion phase.

The set of Eqs. (2.3)–(2.5) applied to the expander can be presented in the form

$$p_E = \rho_E T_E (c_p - c_v), \tag{2.9}$$

$$\frac{\mathrm{d}T_E}{\mathrm{d}\alpha} = \frac{1}{\omega \rho_E V_E} \left(\dot{m}_1 (\kappa T_H - T_E) - \dot{m}_2 (\kappa - 1) T_E - \frac{\omega p_E}{c_v} \frac{\mathrm{d}V_E}{\mathrm{d}\alpha} \right), \tag{2.10}$$

$$\frac{\mathrm{d}\rho_E}{\mathrm{d}\alpha} = \frac{1}{\omega V_E} \left(\dot{m}_1 - \dot{m}_2 - \rho_E \omega \frac{\mathrm{d}V_E}{\mathrm{d}\alpha} \right). \tag{2.11}$$

The algebraic formulae for modelling the flows $\dot{m}_1$ and $\dot{m}_2$ (and consequently $\dot{m}_3$ and $\dot{m}_4$ discussed in the next subsection) are built with Bernoulli and continuity equations for the working fluid. This attempt includes losses determined below by the coefficients ζ_j, $j = 1, \ldots, 4$. The equation for these rates is as follows

$$\dot{m}_j = \frac{\zeta_j A_j p_b}{\sqrt{R T_b}} \left(\frac{p_a}{p_b} \right)^{\frac{1}{\kappa}} \sqrt{\frac{2\kappa}{\kappa - 1} \left[1 - \left(\frac{p_a}{p_b} \right)^{\frac{\kappa - 1}{\kappa}} \right]} \quad \text{for } j = 1, 2, 3, 4. \tag{2.12}$$

The above formula takes into account the ratio of pressures on both sides of any valve, p_a and p_b. If

$$\frac{p_a}{p_b} \leq \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}}, \tag{2.13}$$

then Eq. (2.12) takes a different form

$$\dot{m}_j = \frac{\zeta_j A_j p_b \sqrt{\kappa}}{\sqrt{R T_b}} \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa + 1}{2(\kappa - 1)}}. \tag{2.14}$$

In such an attempt, the flows through the valves are treated as the quasi-stationary ones. The $\dot{m}_j$ values depend on the surrounding pressures, temperatures and valve cross-sections $A_j(\alpha)$. The equation of conservation described here does not take into account the so called inverse flow between the engine parts. Such a flow is impossible in the case of self-acting valves. After a careful analysis, it is also possible to set opening and closing times of the governed valves in the way which allows for avoiding any return flow through them. The values of the coefficients of losses ζ_j were assumed to be equal to 0.85, cf. [6], as typical in similar designs. It should be mentioned that flows through valves cause also different dynamic effects, which apart from the mentioned losses, are not included in modelling. If the high speed behaviour of wave generation is to be included, any changes have to include very high velocity ranges estimated as 400–800 m/s along the length of heat exchangers. Their length is assumed to be equal to 2 m at maximum, a travelling wave can cover it many times within a single cycle of work. Therefore, our quasi-static attempt to modelling shows an averaged behaviour and such a detailed description would change also some minor characteristics and not the general view of the cycle. In slow speed engines like the EHVE, dynamic effects are usually inconsiderable, the expected Mach number is low if compared to the value when dynamic effects prevail over the static ones.

We need to calculate the pressures p_a and p_b and the temperatures T_a, T_b on both sides of all valves as

$$\text{for valve 1}:$$

$$\text{if } p_H > p_E \ : \ p_b = p_H, \quad T_b = T_H, \quad p_a = p_C, \tag{2.15a}$$

$$\text{if } p_E > p_H \ : \ p_b = p_E, \quad T_b = T_E, \quad p_a = p_H, \tag{2.15b}$$

$$\text{for valve 2}:$$

$$\text{if } p_E > p_{Cl} \ : \ p_b = p_E, \quad T_b = T_E, \quad p_a = p_{Cl}, \tag{2.15c}$$

$$\text{if } p_{Cl} > p_E \ : \ p_b = p_{Cl}, \quad T_b = T_{Cl}, \quad p_a = p_E, \tag{2.15d}$$

In the above equations values of temperatures and pressures in heat exchangers are considered as those related to cooperating elements. This averaged attempt does not include any non-stationary behaviour in the close vicinity of the valves. When such effects are included, it is followed by qualitative only and not quantitative changes.

2.3.2 Compressor

The unknown variables for the compressor are as follows: the pressure p_C, the temperature T_C and the density ρ_C. The time-dependent compressor volume is, similarly to the expander, denoted as

$$V_C(t) = A_{CC}\left(h_{C0} + \frac{s}{2}(1 + \cos \omega t)\right), \tag{2.16}$$

where $A_{CC} = \frac{\pi}{4}\left(d_C^2 - d_{PR}^2\right)$, d_{PR} – piston rod diameter (Fig. 2.1), h_{C0} – clearance under the piston at its lowest position.

The mass flow rates $\dot{m}_3$ and $\dot{m}_4$ through self-acting valves 3 and 4 are calculated similarly as for the governed valves [1]. However, their cross-section areas are calculated simultaneously during the engine operation process, because A_3 and A_4 clearly depend on pressures in the compressor p_C, the heater P_H and the cooler p_{Cl}. In the case of these valves, the pressures p_a and p_b and the temperatures T_a, T_b on both sides of all valves are as follows

Table 2.1 Physical data of the self-acting valves

Valve No.	Mass m_i (kg)	Spring stiffness k_i (N/m)	Damping coefficient c_i (Ns/m)	Max. lift h_i (m)	Diameter d_i (m)
3	0.0755	5000	15.75	0.0074	0.030
4	0.0755	5000	15.75	0.0074	0.030

$$\text{for valve 3}:$$
$$\text{if } p_C > p_H : \; p_b = p_C, \quad T_b = T_C, \quad p_a = p_H, \tag{2.17a}$$
$$\text{if } p_H > p_C : \text{no flow}, \tag{2.17b}$$
$$\text{for valve 4}:$$
$$\text{if } p_C > p_{Cl} \; : \; \text{no flow}, \tag{2.17c}$$
$$\text{if } p_{Cl} > p_C \; : \; p_b = p_{Cl}. \quad T_b = T_{Cl}, \quad p_a = p_C, \tag{2.17d}$$

The self-acting valve movement is determined with an additional set of differential equations, taking into account the inertia of its movable parts. Each mass is supported by a spring and viscous damping is also introduced into its model. For the self-acting valves, we assume the data shown in Table 2.1.

The angle of valve 3 opening $\alpha_{3,0}$ is determined from the condition

$$p_C \geq p_H$$

and similarly, $\alpha_{4,0}$ from

$$p_C \geq p_{Cl}.$$

Also for the compressor, it is assumed that

$$\dot{Q}_A + \dot{Q}_V = 0.$$

As a result, we get the following set of equations describing the working fluid inside the compressor volume

$$\frac{\mathrm{d}T_C}{\mathrm{d}\alpha} = \frac{1}{\omega \rho_C V_C}\left(\dot{m}_4(\kappa T_{Cl} - T_C) - \dot{m}_3(\kappa - 1)T_C - \frac{\omega p_C}{c_v}\frac{\mathrm{d}V_C}{\mathrm{d}\alpha}\right), \tag{2.18}$$

$$\frac{\mathrm{d}\rho_C}{\mathrm{d}\alpha} = \frac{1}{\omega V_C}\left(\dot{m}_4 - \dot{m}_3 - \rho_C \omega \frac{\mathrm{d}V_C}{\mathrm{d}\alpha}\right), \tag{2.19}$$

$$p_C = \rho_C T_C (c_p - c_v). \tag{2.20}$$

The algebraic formulae for the flows $\dot{m}_3$ and $\dot{m}_4$ are given when the pressures p_C, p_H, p_{Cl}, temperatures and the actual cross-section values $A_j(\alpha)$ are calculated together.

2.3.3 Cooler

The cooler volume is much higher than the volumes of other engine parts. Taking into account the isobaric model of the working fluid, the cooling process under the constant pressure p_{Cl} is assumed. The fluid mass contained in the cooler is almost constant. The pressure p_{Cl} is the lowest, basic pressure of the engine cycle. There exists a link between p_{Cl} and the total fluid mass enclosed in all 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 fluid into the engine volumes before it starts. The cooler works as a stationary heat exchanger, which causes the working fluid temperature drop from $T_{E\,min}$ to T_{Cl}, which is the temperature of the fluid at the compressor inlet, the pressure p_{Cl} is assumed constant in this version of the EHVE.

2.3.4 Heater

The unknown variables for the heater are: the pressure p_H, the temperature T_H and the density ρ_H. They depend on time and space. The heater volume V_H is assumed constant. The surfaces surrounding V_H are not mobile. The heat generation connected with the friction forces acting inside the heater $\dot{Q}_{VH}$ is neglected as it is very inconsiderable when compared to the heat flux $\dot{Q}_{AH}$ delivered to the heating fluid. In the range of the crankshaft angle rotation $\alpha \geq \alpha_{3,o}$, after opening of valve 3, the working fluid exchange occurs. The hot fluid flowing into the expander is pushed out by a comparatively cool fluid coming from the compressor.

The next issue concerns the heaters H_1 and H_2, generally referred to as heaters. In this stage of the development of the EHVE, we considered two different models of the heater system. The first model treats thermodynamic parameters of the heater as time-dependent only. This assumption allows the treatment which will be explained later. The other one divides the heater volume into n sub-volumes which transfer the working fluid from one to another and different averaged conditions are met in each of them. Thus, the number of differential equations describing the state of any heater increases by $2n$ but the solution is expected to be more accurate. The second model takes into account a dependence of its parameters on time and space, which is the tube length.

The first heater operational model will be considered now. It is treated as a counter-current heat exchanger where changes of its parameters are almost constant along its length. This allows one to calculate its changes in a single plane, as being time-dependent only. This plane is chosen at the outlet of the heater.

For the first model in the range $\alpha_{1,o} \le \alpha \le \alpha_{3,c}$, when valves 1 and 3 are open, the heat flux $\dot{Q}_{AH}$ delivered through the heater surface area

$$A_H = n_H \pi d_H l_H,$$

(n_H is here a number of tubes of the diameter d_H and the length l_H) is described by the equation

$$\dot{Q}_{AH} = \alpha_A A_H (T_{w_H} - T_H), \tag{2.21}$$

where T_{wH} is the heater wall temperature. The heat transfer coefficient α_A is determined as a function of the Nusselt number Nu, i.e.,

$$\alpha_A = \frac{\mathrm{Nu}\lambda_H}{d_H},$$

The current value of the Nusselt number is varying as $\mathrm{Nu} = f(\mathrm{Re}, \mathrm{Pr})$. Values of λ, Re and Pr are the heat conductivity, the Reynolds and Prandtl numbers, respectively, all for the air flowing inside the heater tubes. The values of the specific heat c_p, the viscosity μ and the above quoted λ used in the calculations of Re and Pr numbers are taken as averaged values of these parameters in the range of temperatures and pressures occurring for $\alpha_{1,o} \le \alpha_{3,c}$. The velocity of the working fluid inside the tubes necessary for determination of the Reynolds number has also been taken as the mass averaged value for $\alpha_{1,o} \le \alpha \le \alpha_{3,c}$. Any unsteadiness of the flow inside the tubes which can cause an increase in α_A is not included in this procedure.

The basic equations of state, energy and mass for the first model of the heater operation derived from Eqs. (2.3)–(2.5) take the following form for the range of the rotation angles $\alpha_{1,o} \le \alpha \le \alpha_{3,c}$

$$\frac{\mathrm{d}T_H}{\mathrm{d}\alpha} = \frac{1}{\omega \rho_H V_H} \left(\dot{m}_3 (\kappa T_C - T_H) - \dot{m}_1 (\kappa - 1) T_H + \frac{\dot{Q}_{AH}}{c_v} \right), \tag{2.22}$$

$$\frac{\mathrm{d}\rho_H}{\mathrm{d}\alpha} = \frac{1}{\omega V_H} (\dot{m}_3 - \dot{m}_1), \tag{2.23}$$

$$p_H = \rho_H T_H (c_p - c_v). \tag{2.24}$$

For $\alpha = \alpha_{3,c}$, the H_1 heater volume is closed. The density of the fluid inside the heater H_1 is constant and the heat transfer is assumed as 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 until $\alpha = 4\pi$. At the value $\alpha = 2\pi$, the heating period of the second heater H_2 is finished and it is switched into the engine operation. The next period begins for $\alpha = 2\pi$, valves 1_2 and 3_2 work in the range $360° \le \alpha \le 540°$. Then, for the heater H_2, isochoric heating lasts until 1080°. It is assumed that during the isochoric process of heating, a kind of whirl movement of the air inside the heater tubes takes place. The experimental investigations of the prototype, which were performed a few years later, show, however, the fact that the amount of heat delivered to the engine in real conditions was insufficient. The value of the heat transfer coefficient $\alpha_{A,iso}$, which was expected to be 1200 W/m^{2}K, was much lower. For a solution of this problem, see Chap. 4.

When the isochoric process begins at α_{3c}, the equation of continuity yields $\rho_{H,iso} =$ constant. This value can be calculated from the equation of state for α_{3c}, e.g.,

$$\rho_{H,iso} = \frac{p_H(\alpha_{3c})}{T_H(\alpha_{3c})R},$$

then,

$$M_{H,iso} = \rho_{H,iso} V_H, \tag{2.25}$$

where the mass locked in the heater is $M_{H,iso}$. In [1, 2], Eq. (2.21) was used, hence the value of $\alpha_{A,iso}$ is applied in this equation.

We do not know what type of flow would appear during the isochoric process, thus we use a general formula

$$\dot{Q}_{iso} = M_{H,iso} c_v \left(T_H - T_H(\alpha_{3,c})\right) \frac{\omega}{\alpha}. \tag{2.26}$$

The EHVE has to work continuously and therefore

$$T_H(4\pi) = T_H(0).$$

We can calculate the value of α_A from the equality of Eqs. (2.21) and (2.26)

$$\alpha_A = \frac{M_{H,iso} c_v \left(T_H - T_H(\alpha_{3,c})\right) \frac{\omega}{\alpha}}{A_H(T_{wH} - T_H)}. \tag{2.27}$$

For $\alpha_{3,c} \leq \alpha \leq 4\pi$ at the engine rotational speed of 1500 rpm, the value of α_A should be about 1200 W/m^{2}K if the condition of the continuous work of the engine is to be assured. This was not confirmed in the experiment performed.

The pressure is calculated also from the isochoric formula

$$p_H(\alpha) = p_H(\alpha_{3,c}) \frac{T_H(\alpha)}{T_H(\alpha_{3,c})}. \tag{2.28}$$

The second model of the heater constitutes a certain modification 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 $p_H(\alpha)$ is taken from the first model and α_A assures continuous work of the engine. The amount of heat delivered to the working fluid is the same in both models. The characteristic feature of the second model is such that the whole space of the heater $V_H(\alpha)$ is divided into n elementary control volumes and the temperature differences not only in time but also along the heater tube length are determined. The elementary control volumes and the elementary heating surfaces are

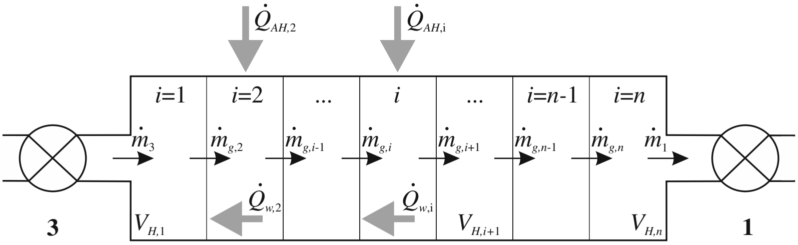

Fig. 2.2 Schematic diagram of the heater divided into n elementary control volumes

$$V_{H,i} = \frac{V_H}{n}, \qquad A_{H,i} = \frac{A_H}{n}. \tag{2.29}$$

A scheme of the heater division into the elementary volumes and the nomenclature used are explained in detail in Fig. 2.2.

Solutions to the equations of temperatures $T_{H,i}(\alpha)$, $i = 1, 2, \ldots, n$ are received on the basis of the following set of energy conservation equations

$$\begin{aligned}
\frac{\mathrm{d}T_{H,1}}{\mathrm{d}\alpha} &= \frac{1}{\omega \rho_{H,1} V_{H,1}} \left(\dot{m}_3 (\kappa T_C - T_{H,1}) - \dot{m}_{g,2}(\kappa - 1) T_{H,1} + \frac{\dot{Q}_{w,1}}{c_v} + \frac{\dot{Q}_{AH,1}}{c_v} \frac{\rho_{H,1}}{\rho_H} \right), \\
&\quad \text{for the first cell}: \ i = 1 \\
\frac{\mathrm{d}T_{H,i}}{\mathrm{d}\alpha} &= \frac{1}{\omega \rho_{H,i} V_{H,i}} \left(\dot{m}_{g,i} (\kappa T_{H,i-1} - T_{H,i}) - \dot{m}_{g,i+1}(\kappa - 1) T_{H,i} + \frac{\dot{Q}_{w,i}}{c_v} + \frac{\dot{Q}_{AH,i}}{c_v} \frac{\rho_{H,i}}{\rho_H} \right), \\
&\quad \text{for the } i\text{−th cell}: \ i = 2, 3, \ldots, n-1, \\
\frac{\mathrm{d}T_{H,n}}{\mathrm{d}\alpha} &= \frac{1}{\omega \rho_{H,n} V_{H,n}} \left(\dot{m}_{g,n} (\kappa T_{H,i-1} - T_{H,n}) - \dot{m}_1(\kappa - 1) T_{H,n} + \frac{\dot{Q}_{w,n}}{c_v} + \frac{\dot{Q}_{AH,n}}{c_v} \frac{\rho_{H,n}}{\rho_H} \right), \\
&\quad \text{for the } n\text{−th cell}: \ i = n.
\end{aligned} \tag{2.30}$$

The density of the working fluid in each cell is calculated from

$$\rho_{H,i} = \frac{p_H(\alpha)}{T_{H,i}(c_p - c_v)}. \tag{2.31}$$

The density ρ_H used in Eqs. (2.30) is the same as the one defined in the first model. Additionally, it can be determined by the formula

$$\rho_H = \sum_{i=1}^{n} \frac{\rho_{H,i} V_{H,i}}{V_H}. \tag{2.32}$$

The internal mass flow rates between the cells are calculated from the equation of mass conservation for each of the volumes

$$\dot{m}_{g,i} = \dot{m}_{g,i-1} - \omega V_{H,i-1} \frac{d\rho_{H,i-1}}{d\alpha}, \quad \text{for } i = 1, 2, \ldots, n, \tag{2.33}$$

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

The heat fluxes $\dot{Q}_{AH,i}$ for a single cell are calculated from the formula analogous to Eq. (2.21), i.e.,

$$\dot{Q}_{AH,i} = \alpha_A A_{H,i} (T_{wH} - T_{H,i}), \tag{2.34}$$

where α_A is calculated according to the procedure applied in the first model.

For the isochoric process in the locked volume, i.e., in the range $\alpha_{3,c} \le \alpha \le 4\pi$, additional heat fluxes $\dot{Q}_{w,i}$ between cells caused by the above-mentioned whirl motion are introduced. These heat streams are proportional to the assumed local mass flow rates $\dot{m}_T(\alpha)$ of the fluid circulating between cells, hence we have the following set of equations

$$\begin{aligned} \dot{Q}_{w,1} &= \dot{m}_T(\alpha) c_p (T_{H,2} - T_{H,1}), && \text{for}: \ i = 1, \\ \dot{Q}_{w,i} &= \dot{m}_T(\alpha) c_p (T_{H,i-1} - 2T_{H,i} + T_{H,i+1}), && \text{for}: \ i = 2, 3, \ldots, i-1, \\ \dot{Q}_{w,n} &= \dot{m}_T(\alpha) c_p (T_{H,n-1} - T_{H,n}), && \text{for}: \ i = n. \end{aligned} \tag{2.35}$$

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

$$\dot{m}_T(\alpha) = \dot{m}_T(0) e^{-\frac{\Delta T}{\omega}(\alpha - \alpha_{3,c})}. \tag{2.36}$$

The exponent value determines the intensity of $\dot{m}_T(\alpha)$, diminishing in time.

The heater is a counter-current heat exchanger in the models. Unfortunately, the heat exchange in the range of isochoric heating described in Chap. 3 entails a rather painful conclusion that the heat exchange in this range was insufficient. For this reason, the model assumed in Chap. 3 should be changed to yield the flows in heater tubes of the Reynolds number higher by 10^4 at least. The first attempt to achieve this is made in Chap. 4.

2.4 Engine Power and Efficiency

The mechanical work produced or consumed in the range of $\alpha_A \le \alpha \le \alpha_B$ by the expander is expressed by the formula

$$L_{E\,A-B} = \int_{\alpha_A}^{\alpha_B} p_E(\alpha) \frac{dV_E}{dt} \frac{d\alpha}{\omega}, \tag{2.37}$$

the current value of varying $\frac{dV_E}{dt}$ comes from the derivation of Eq. (2.6).

The compressor work in the range of $\alpha_A \leq \alpha \leq \alpha_B$ is determined with a similar formula

$$L_{C\,A-B} = \int_{\alpha_A}^{\alpha_B} p_C(\alpha) \frac{\mathrm{d}V_C}{\mathrm{d}t} \frac{\mathrm{d}\alpha}{\omega}, \tag{2.38}$$

and the current value of $\frac{\mathrm{d}V_C}{\mathrm{d}t}$ is calculated from the derivative of Eq. (2.16).

The sum of these two parts represents the work for the whole cycle $0 \leq \alpha \leq 2\pi$

$$L_{E+C} = L_{E\,0-2\pi} + L_{C\,0-2\pi}. \tag{2.39}$$

Now, we can define the power created in the engine as

$$P = \frac{\omega}{2\pi} L_{E+C}. \tag{2.40}$$

The amount of heat introduced by the heater in the range of $\alpha_{1,o} \leq \alpha \leq \alpha_{3,c}$ is calculated by the following integral

$$Q_{H\,I} = A_H \int_{\alpha_{1,o}}^{\alpha_{3,c}} \alpha_A \left(T_{wH} - T_H\right) \frac{\mathrm{d}\alpha}{\omega}. \tag{2.41}$$

For the range of $\alpha_{3,c} \leq \alpha \leq 4\pi$, when isochoric heating takes place in the closed volume of H_1, the amount of the delivered heat is

$$Q_{H\,II} = c_v M_H \left(\alpha_{3,c\,iso}\right) \left(T_H(4\pi) - T_H\left(\alpha_{3,2}\right)\right), \tag{2.42}$$

where $M_H(\alpha_{3,c})$ is the constant amount of air mass closed in the heater during the isochoric heating period. The next cycle of the EHVE begins when valves 1_2 and 3_2 are open at $2\pi \leq \alpha \leq 3\pi$. Then, heating of the H_2 heater starts at 3π and continues until 6π. The temperature $T_H(4\pi)$ should be close to $T_H(0)$ if the heat delivered to the EHVE is sufficient, as has been assumed here. The total amount of heat delivered during the cycle to the working fluid is a sum of

$$Q_H = Q_{H\,I} + Q_{H\,II}. \tag{2.43}$$

Finally, the engine cycle efficiency is calculated as a ratio of amounts of the generated work to the delivered heat

$$\eta_C = \frac{L_{E+C}}{Q_H}. \tag{2.44}$$

The engine cycle efficiency calculated according to Eq. (2.44) does not take into account yet mechanical losses caused mainly by friction between piston rings and cylinders. These losses can consume up to 10 % of the generated engine power. Therefore, the work and power calculated according to Eqs. (2.39) and (2.40), respectively, can be determined as indicative only.

2.5 Thermodynamic Cycle of the Early EHVE

This cycle is elaborated numerically in [3]. The thermodynamic cycle of a new type of the heat engine has been presented in the traditional coordinate systems: $p - \frac{1}{\rho}$ (pressure – specific volumes), Fig. 2.3, and $T - s$ (temperature – specific entropy), Fig. 2.4. The cycle is determined on the basis of the solution to the energy and mass conservation equations and the equation of state.

These diagrams are only illustrative and they give a general view to the engine heat cycle character. The mechanical work gained and the heat delivered during the cycle is calculated on the basis of the numerical solution to the presented set of model equations.

For this version of the EHVE, the timing of the valves is chosen in such a way that it is foreseen that valves 1 and 3 slightly overlap, [1, 3]. In Fig. 2.3, a simple $p - \frac{1}{\rho}$ diagram of the engine thermodynamic cycle for a single heater obtained from the values read from the results of numerical simulations is presented. The essential assumption for this diagram is

$$T_H(4\pi) = T_H(0).$$

The $T - s$ and $p - \frac{1}{\rho}$ diagrams for the engine heat cycle are depicted as a kind of an envelope formed by means of loops illustrating the operation of the engine parts, i.e., the compressor, the heater, the expander and the cooler. The loops present the thermodynamic state of the air remaining inside the volumes of the above-mentioned engine parts, although the mass of the air in these volumes changes within very wide limits (except for the cooler).

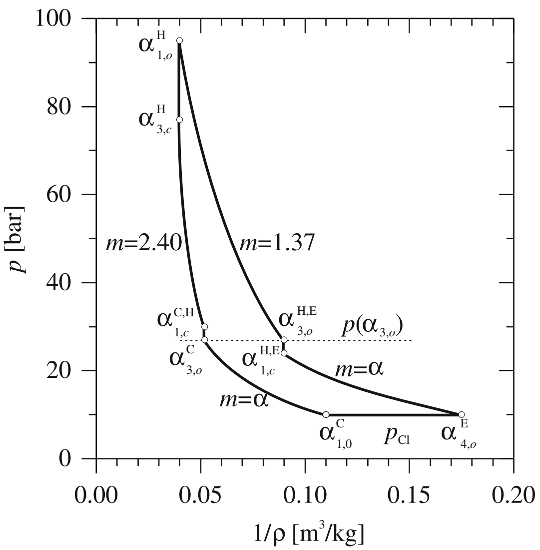

Fig. 2.3 $p - \frac{1}{\rho}$ diagrams of the engine thermodynamic cycle

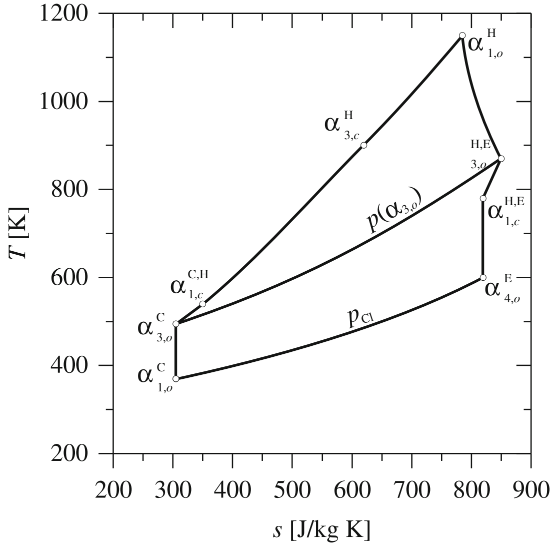

Fig. 2.4 $T - s$ diagram of the engine thermodynamic cycle

The $T - s$ diagram from Fig. 2.4 for the engine thermodynamic cycle is plotted in the way presented below. The diagram starts from the point $\alpha^{C}_{1,o}$ of the compressor – see the left bottom point on the diagram. Then, isentropic compression to the point $\alpha^{C}_{3,o}$ takes place. For $\alpha > \alpha^{C}_{3,o}$, the air flows through valve 3 from the compressor to the heater and is compressed and heated there. Therefore, the substitutive processes for the compressor-heater assembly should connect the point $\alpha^{C}_{3,o}$ of the compressor with the point $\alpha^{H}_{3,c}$ of the heater, cf. Fig. 2.3. The first part of this substitutive process lies between the points $\alpha^{C}_{3,o}$ and $\alpha^{C,H}_{1,c}$. It is assumed that this part of the process is nearly isochoric because volumes of the compressor, the heater and the expander are connected with one another. The second part of the substitutive process from $\alpha^{C,H}_{1,c}$ to $\alpha^{H}_{3,c}$ has a form of the polytrope with the compression index $m \approx 2.40$, again we recall Fig. 2.3. Then, isochoric heating is observed from the point $\alpha^{H}_{3,c}$ to the point $\alpha^{H}_{1,o}$ of the heater. The substitutive process for the heater-expander assembly runs from the point $\alpha^{H}_{1,o}$ of the heater to a certain point situated between the points $\alpha^{H}_{3,o}$ of the heater and $\alpha^{E}_{3,o}$ of the expander on the isobar $p(\alpha_{3,o})$, see Fig. 2.4. This point is denoted by $\alpha^{H,E}_{3,o}$. The process from $\alpha^{H}_{1,o}$ to $\alpha^{H,E}_{3,o}$ can be modelled by a polytrope with the expansion index $m \approx 1.37$. Then, a nearly isochoric process between the substitutive point $\alpha^{H,E}_{3,o}$ and the substitutive point $\alpha^{H,E}_{1,c}$ is assumed. The reason for this assumption is the same as for the compressor-heater assembly. The positions of $\alpha^{H,E}_{3,o}$ and $\alpha^{H,E}_{1,c}$ are determined taking into account the air mass in the heater and expander volumes for $\alpha = \alpha_{3,o}$ and $\alpha_{1,c}$, respectively. The point $\alpha^{H,E}_{1,c}$ lies on the isentrope, illustrating thus an expansion in the closed cylinder of the expander. The lowest point of this expansion is determined by the point $\alpha^{E}_{4,o}$, which is situated on the isobar p_{Cl}. The isobar p_{Cl} connects the point $\alpha^{E}_{4,o}$ and the point $\alpha^{C}_{1,o}$, closing the engine cycle. If isochoric heating were insufficient, the EHVE would have to be further developed.

For better understanding of the attempt undertaken in our considerations for modelling of the engine work, we have assumed the working fluid to have properties of ideal gas. This is of course a simplification but the results obtained without using it show only some quantitative and not qualitative differences.

Let us consider the process taking place inside the compressor volume when it is closed and all changes are forced by the movement of the piston, $\dot{m}_j = 0$. The equation of energy conservation (2.18) in this case can be written as

$$\mathrm{d}T_C = -\frac{1}{\rho_C V_C} \frac{p_C}{c_V} \mathrm{d}V_C, \tag{2.45}$$

whereas the equation of mass conservation (2.19) can take the form

$$\frac{\mathrm{d}\rho_C}{\rho_C} = -\frac{\mathrm{d}V_C}{V_C}. \tag{2.46}$$

From Eqs. (2.45) and (2.46) we get the relation

$$c_V \frac{\mathrm{d}T_C}{T_C} = R \frac{\mathrm{d}\rho_C}{\rho_C}, \tag{2.47}$$

which presents no increase in entropy. Then, after some transformations, with $R = c_p - c_V$, we get

$$\frac{\mathrm{d}T_C}{T_C} = (\kappa - 1) \frac{\mathrm{d}\rho_C}{\rho_C}. \tag{2.48}$$

The above equation can be finally transformed to

$$\frac{\mathrm{d}p_C}{p_C} = \kappa \frac{\mathrm{d}\rho_C}{\rho_C}, \tag{2.49}$$

which describes a differential form of an isentropic process inside the closed compressor volume. A similar discussion can be conducted for the expander. During the further development of the model, a corresponding attempt is undertaken in Sect. 5.3. for the improved version of the EHVE.

The $T - s$ and $p - \frac{1}{\rho}$ diagrams for a new type of the externally heated engine show that the cycle is composed of 8 processes – 2 isentropes, 3 isochores, 1 isobare and 2 polytropes. The presented diagrams have only an illustrative character and are discussed on the basis of the numerical solution to the set of the governing equations presented. Both diagrams are given for comparisons with the corresponding thermodynamic drawings illustrating the cycles of other known heat engines.

2.6 Results of the Simulations of the Early EHVE

The second part of the heat $Q_{H,II}$ is calculated with Eq. (2.42). An example of the engine subject to the presented computer simulations has the cylinder volume of 1 L with the data shown in Table 2.2, whereas the piston rod volume is ignored. For the heat exchangers, the data presented in Table 2.3 have been assumed, both the heaters H_1 and H_2 are here taken into account.

For all of the governed valves, the maximal flow area values are $A_{j\,max} = 7 \times 10^{-4}\,\mathrm{m}^2$. The data for the self-acting valves have already been given in Sect. 2.3.2. Additionally, there was an assumption of the required overlapping value of valves 1 and 3, $\delta_{ME} = \alpha_{1,2} - \alpha_{3,0} = 12°$. The engine rotational speed was set to $n = 1500$ rpm.

A wide discussion on the engine power and the efficiency η_C determined according to formulae (2.40) and (2.44) as functions of the angular velocity ω, the heater volume V_H and its wall temperature T_{wH} for the cooler pressure $p_{Cl} = 10$ bar has been presented in [1]. The preliminary optimization of the engine cycle for that value of cooler pressure shows that the power of 30 kW per 1 L of cylinder volume at 1500 rpm and the efficiency level η of about 38 % can be obtained, if the amount of heat delivered is satisfactory. For continuous operation, the following relationship should be held

$$T_H(4\pi) = T_H(0),$$

for the first heater, and

$$T_H(6\pi) = T_H(2\pi)$$

for the second one. All the results presented below are based on this assumption. Figures 2.5 and 2.6 present periodic solutions to the problem within one engine cycle for $p_{Cl} = 10$ and 30 bar, respectively. The first model of the heater operation has

Table 2.2 Dimensions of the engine cylinders

Cylinder	Diameter d (m)	Stroke s (m)	Dead height h_0 (m)
Expander	0.13	0.0753	0.0050
Compressor	0.13	0.0753	0.0120

Table 2.3 Dimensions of the heat exchangers

Exchanger	Diameter d (m)	Length l (m)	Number of pipes n (−)	Exchange area A_H (m^2)	Volume V (m^3)
Heater H	0.008	0.4	20	0.2	0.33×10^{-3}
Cooler Cl					5.00×10^{-3}

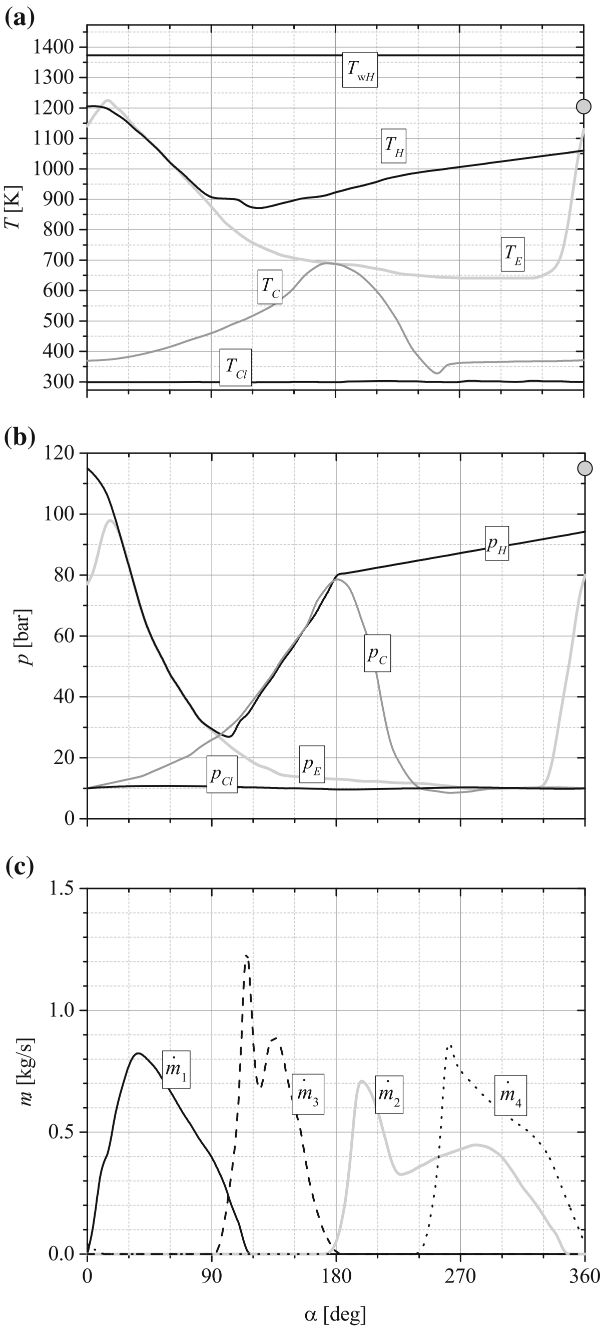

Fig. 2.5 Diagrams of **a** temperatures, **b** pressures and **c** valve mass flow rates as functions of the crankshaft angle α for the cooler pressure $p_{Cl} = 10\,\text{bar}$ and heating at $T_{wH} = 1373\,\text{K}$. Other important parameters were $T_{Cl} = 363\,\text{K}$, $V_H = 0.33 \times 10^{-3}\,\text{m}^3$, $n = 1500\,\text{rpm}$, $\delta_{ME} = 12°$ according to the first heater model. The circles in parts **a** and **b** show the final parameters of the heater H_2 at 360°

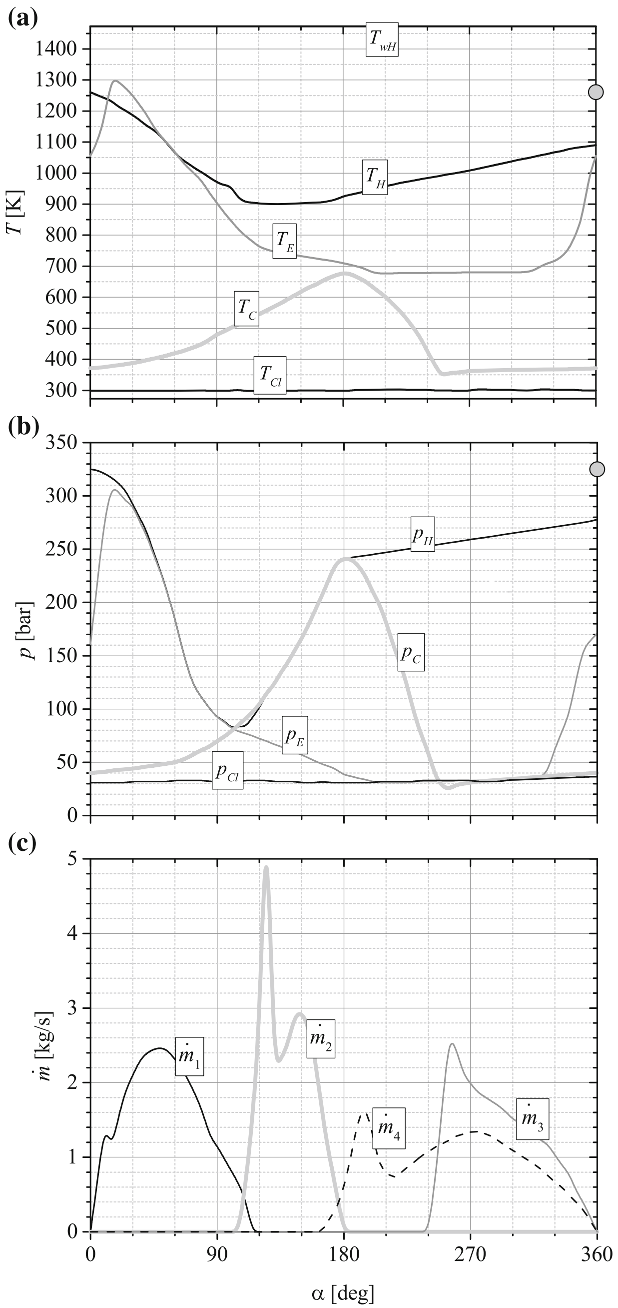

Fig. 2.6 Diagrams of **a** temperatures, **b** pressures and **c** valve mass flow rates as functions of the crankshaft angle α for the cooler pressure $p_{Cl} = 30\,\text{bar}$ and heater wall temperature $T_{wH} = 1473\,\text{K}$. Other important parameters were $T_{Cl} = 363\,\text{K}$, $V_H = 0.33 \times 10^{-3}\ \text{m}^3$, $n = 1500\,\text{rpm}$, $\delta_{ME} = 12°$ according to the first model. The circles in parts **a** and **b** show the final parameters of the heater H_2 at 360°

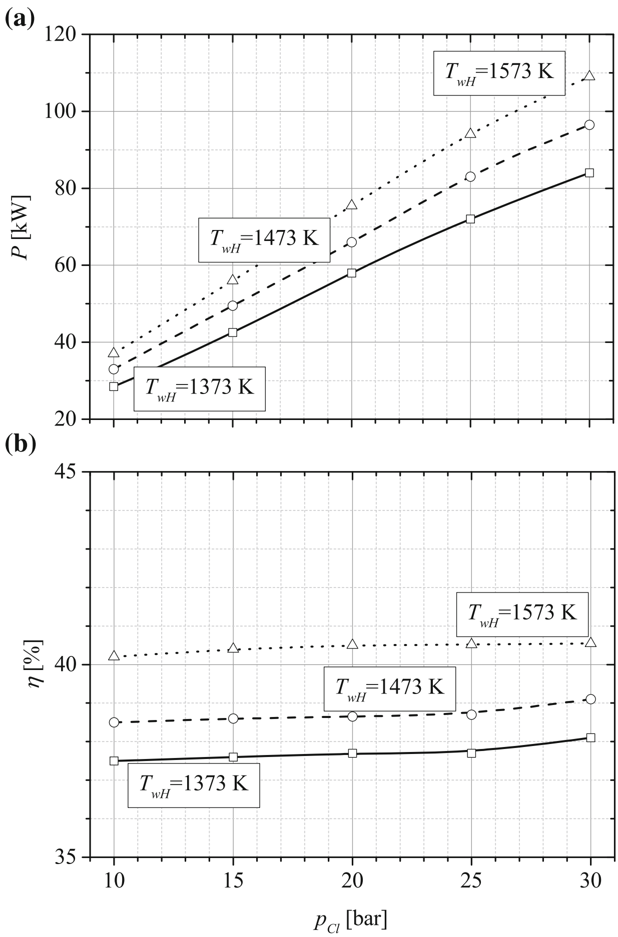

Fig. 2.7 Diagrams of the engine power P, (**a**), and and the efficiency, η_C (**b**) as functions of the cooler pressure p_{Cl} for the different heating wall temperatures $T_{wH} = 1373$, 1473 and 1573 K

been employed in this calculation. In both figures, the action of the second heater is marked with circles at the end of the cycle as the heaters switch off in this point.

The next task of this section is to determine the engine power and efficiency according to Eqs. (2.40) and (2.44) for a higher value of the basic pressure of the cycle, i.e., the one in the cooler. We assume this increase from 10 up to 30 bar, and for the results see Fig. 2.7a, b. There is a well pronounced, almost linear increase in power values for both the cooler pressure and the heater wall temperatures, respectively. The obtained efficiency values remain almost constant for the considered range of the cooler pressure p_{Cl} and yield rather flat horizontal lines. However, a significant increase from about 37 % to almost 41 % occurs when the heater wall temperature increases from 1373 to 1573 K. The discussed output parameters of the engine model

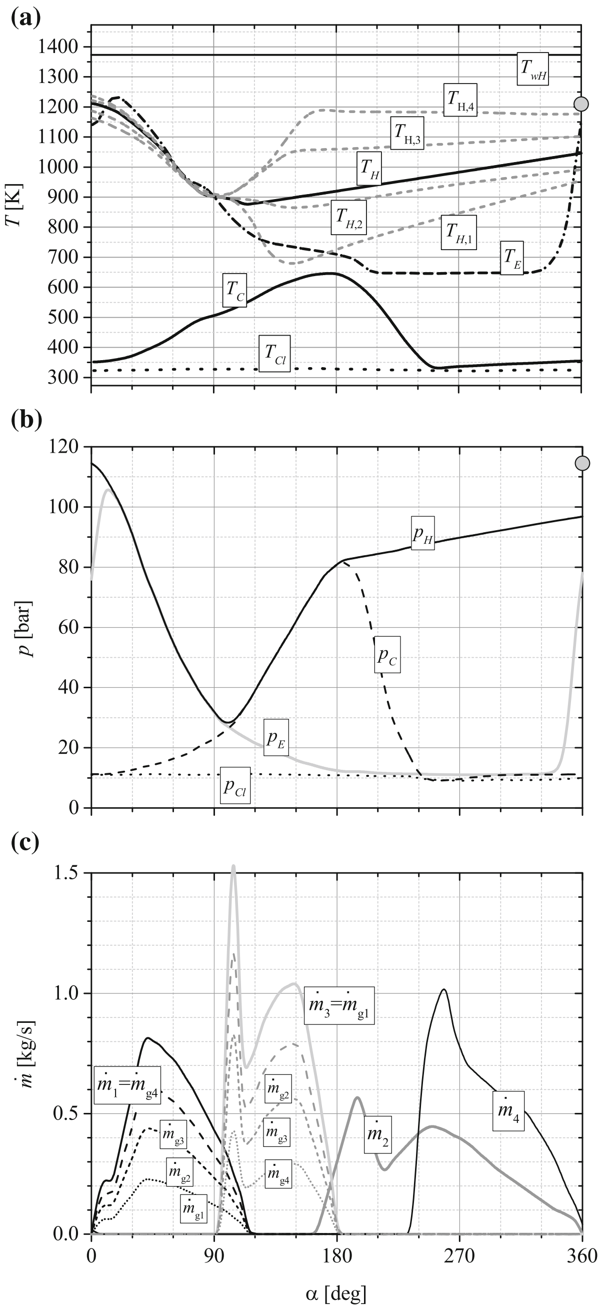

Fig. 2.8 Diagrams of **a** temperatures, **b** pressures and **c** valve mass flow rates as functions of the crankshaft angle α: $p_{Cl} = 10\,\text{bar}$, $T_{Cl} = 363\,\text{K}$, $T_{wH} = 1373\,\text{K}$, $V_H = 0.33 \times 10^{-3}\ \text{m}^3$, $n = 1500\,\text{rpm}$, $\delta_{ME} = 12°$. The temperatures T_H and the mass flow rates $\dot{m}_{g,i}$ are shown for $n = 4$ according to the second heater model. The circles in parts **a** and **b** show the final parameters of the heater H_2 at 360°

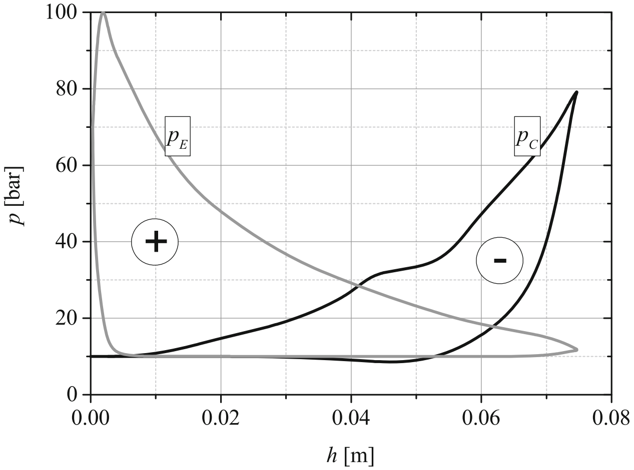

Fig. 2.9 Diagrams of the expander and compressor pressures as functions of the piston positions h according to the first model of the heaters, where $0 \leq h \leq s$, (p_C and p_E are taken from Fig. 2.5)

are calculated without taking into account any mechanical losses which exist in the real device, so these parameters are rather indicative, and not effective values.

Nevertheless, the power of about 100 kW per 1 L of the engine cylinder volume, which is theoretically achieved for $p_{Cl} = 30$ bar and $T_{wH} = 1473$ and 1573 K, (Fig. 2.7), shows that the presented engine performance can be potentially close to those achieved by best recently made internal combustion piston engines. The results definitely depend on an amount of the heat delivered to the EHVE. Once more, it is underlined that these are only theoretical results, $\alpha_A \geq 1200$, taking into account a very inconsiderable isochoric heating process, proven experimentally later and mentioned earlier in this chapter and formerly in [5].

The second model of the heater, which is an improved version of the first one, allows for estimation of spatial differences in the heater temperature during the engine cycle. A very simple example of such calculations has been performed according to the description presented in the previous section. The heater volume has been divided into only four parts, $n = 4$, for better clarity of the presentation of the idea. The results of the temperature distribution are shown in Fig. 2.8a. The pressures in respective elements are depicted in the part (b) and the mass flow rates are given in the part (c). When we compare this figure to Fig. 2.5, the character of all pressures is preserved, while both the temperatures and the mass flow rates present much more detailed characters, respectively. On average, the cell heater model simplifies the behaviour of that which is not divided into parts.

An important illustration of pressures inside the compressor and the expander as functions of the piston position h is depicted in Fig. 2.9. These relations can be used for calculations of the EHVE power as indicated by the + and − signs within both the expander and compressor volumes, respectively.

Two small heaters H_1 and H_2 are employed in this version of the EHVE. For a long time, as discussed above, their volumes are closed and only an isochoric heating process takes place inside their volumes.

Unfortunately, the experimental investigations described next in Chap. 3 show that isochoric heating is too poor for the EHVE to operate efficiently. Then, all of the above presented results exhibit mostly a theoretical character only as they were obtained with the numerical simulations based on the assumption that the heat delivered to the engine model was sufficient. With respect to this fact, the idea of the EHVE presented until now needs further improvements by replacing the isochoric heating process with a more effective one.

The first attempt is described in Chap. 4. Other, entirely different approaches are to be found in Chaps. 5–7.

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