ANALYSIS OF THERMODYNAMICAL PARAMETERS OF COMBUSTION ENGINE WORKING CYCLE

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Abstract

Thermodynamics systems may be classified as isolated, closed, or open based on the possible transfer of mass and energy across the system boundaries. An isolated system is one that is not influenced in any way by the surroundings. This means that no energy in the form of heat or work may cross the boundary of the system. In addition, no mass may cross the boundary of the system. A thermodynamic system is defined as a quantity of matter of fixed mass and identity upon which attention is focused for study. A closed system has no transfer of mass with its surroundings, but may have a transfer of energy (either heat or work) with its surroundings. An open system is one that may have a transfer of both mass and energy with its surroundings. Described system being an object of the paper is an open system for some phenomena.

An object the paper is the thermodynamical analysis of the filling process of the cylinder which occurs in calculated period from the percussion cap of the position the piston at TDC to it position at BDC, at constant average value of the pressure in the cylinder. During durations of the filling process of the cylinder, dirt of the working charge combustion products are not taken into account. One accepts that the quantity of the working charge in the established point placed on the compression line by piston BDC is equal quantity of the working charge in the end point of the filling process, answering to the start point of the real compression process. The state of the working charge in cylinder described is four parameters: a pressure, a temperature, quantity moles of the working charge and its volume. For appointment those parameters, earlier worked out dependences are put-upon.

Keywords: *combustion engine, engine thermodynamics, engine working cycle*

1. Introduction

Thermodynamics systems may be classified as isolated, closed, or open based on the possible transfer of mass and energy across the system boundaries. An isolated system is one that is not influenced in any way by the surroundings. This means that no energy in the form of heat or work may cross the boundary of the system. In addition, no mass may cross the boundary of the system. A thermodynamic system is defined as a quantity of matter of fixed mass and identity upon which attention is focused for study. A closed system has no transfer of mass with its surroundings, but may have a transfer of energy (either heat or work) with its surroundings. An open system is one that may have a transfer of both mass and energy with its surroundings. Described system being an object of the paper is an open system for some phenomena.

What about piston engine, when analysing an ideal cycle, it is important to note that a piston engine is not a closed system, and therefore cannot be considered a heat engine as defined in classical thermodynamics. Rather, a piston engine is an open system that exchanges heat and work with the environment. The two reactants in this system are the fuel and air mixture which flows into the system and the exhaust gas by products which flow out. However, the analyses used within the individual processes are based on thermodynamic principles. When developing the ideal cycles the designer must determine which model to use for the working fluid within the cylinder. By defining the fluid's thermodynamic properties, the cycle can be simplified using various assumptions.

All reciprocating engines are characterized by a piston that moves back and forth in a cylinder. This piston movement in turn drives a crankshaft, which transmits the power to a drive shaft or transmission of some type. The most important component of the engine is the piston / cylinder combination, which is the focus of the thermodynamic analysis. Although the piston is dependant on the crankshaft for movement during the non-power strokes, each piston operates independent of the others. For this reason, the thermodynamic analysis of reciprocating engines is not dependant on the number of cylinders or even engine geometry. While these parameters are extremely important from a structures and materials perspective, they are irrelevant in the performance analysis.

Reciprocating engines are categorized by the number of piston strokes required to complete the engine cycle, which is either two or four. The best way to measure engine work is through the use of a p-v diagram. However, in vehicle design this is not a viable option and the designer must predict the engine's performance based on a few critical engine parameters. This is done through the use of ideal engine cycles. By dividing the engine operating cycle into a sequence of separate processes (compression, combustion, expansion, and exhaust) and modelling each process, the designer can simulate the complete engine cycle. These simulated engine cycles in turn allow the user to estimate the engine performance. Ideal cycles are categorized based on the method used to model the combustion process. The three most common models are constant volume - Otto cycle, constant pressure - Diesel cycle and dual cycle. In each cycle, the processes other than the combustion process remain the same. The assumptions associated with the various cycles should take into account these kinds work.

2. Mean pressure in engine cylinder

Fig. 1 presents the diagram of a computational engine work cycle utilised for its thermal calculations. In Fig. 1, the characteristic points of the computational work cycle are designated: w - beginning of opening of the outlet valve, w' – closing of the outlet valve, d – opening of the inlet valve, d' – closing of the inlet valve, a – conventional end of the cylinder filling process, f – conventional end of the process of giving up the heat at p = const, ps – beginning of the combustion process, c – conventional end of the compression process, z' – conventional end of the heat delivery at V = const, z – end of the heat delivery at p =const, ks – end of the combustion

process, b – conventional end of the expansion process, p and k – beginning and end of the working medium mean pressure isobar in the cylinder during the filling process.
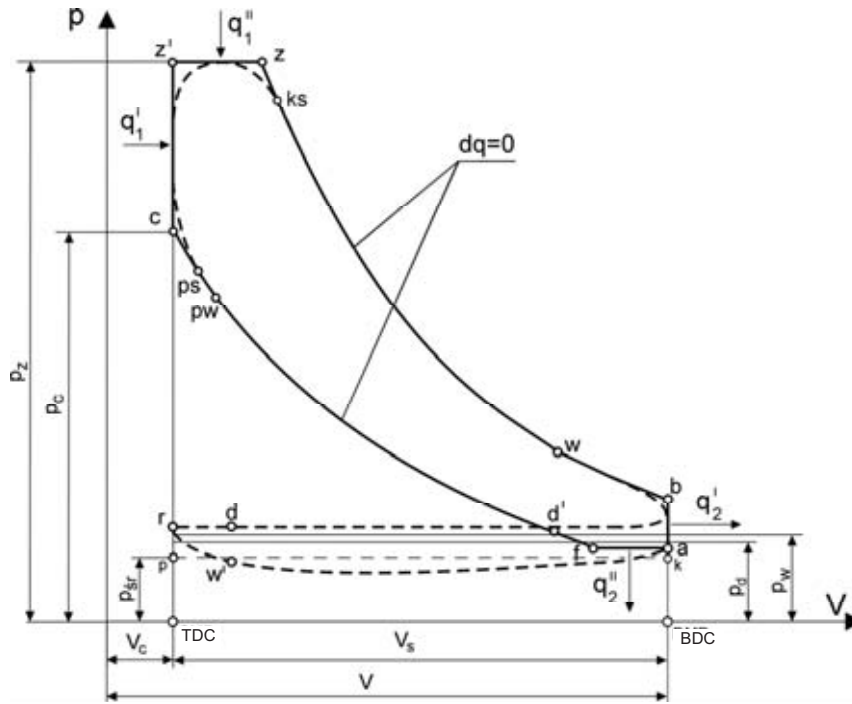

Fig. 1. Thermodynamical engine working cycle

3. Pressure in cylinder during filling

Assumptions:
1. The cylinder filling process proceeds during the time calculated from the piston TDC position to its BDC position, at constant mean value of the pressure in the cylinder $p_{śr}$ which should be determined computationally.
2. Fouling of the working medium with the combustion products is not taken into account during the cylinder filling process.
3. It is assumed that the quantity of the working medium in the conventional point "a" lying on the compression line at the piston BDC is equal to the quantity of the working medium at the point of the end of the filling process, corresponding to the beginning point of the effective compression process.
4. The state of the working medium in the cylinder is uniquely described by four parameters: pressure p, temperature T, quantity of moles of the working medium M and its volume V. The previously indicated equations and the state equation are used for determination of those parameters.

Those equations have the form:

$$
\left.
\begin{aligned}
V &= \frac{V_s}{\varepsilon - 1}\left[1 + (\varepsilon - 1)\frac{\sigma}{2}\right], \\
M &= \int_d^{d'}(\delta M_d - \delta M_w), \\
\pm \int_d^{d'}\delta Q_{sc} &+ \int_d^{d'}\delta H_d - \int_d^{d'}\delta H_w = U_{d'} - U_d + \int_{v_d}^{v_{d'}}pdV = H_{d'} - H_d - \int_d^{d'}Vdp, \\
pV &= \bar{R}MT,
\end{aligned}
\right\}
\tag{1}
$$

where:

$$\sigma = (1-\cos\alpha)+\frac{\lambda_k}{4}(1-\cos 2\alpha),$$

$$\lambda_k = \frac{R}{L} \quad (R – \text{crankthrow of the engine connecting rod, } L – \text{length of the connecting rod}).$$

According to the assumptions taken, the quantity of the working medium that has flown into the cylinder is:

$$M_d = \int_{\alpha_p}^{\alpha_k} \frac{f_{śr} w_{śr} \rho_{śr}}{\mu_{czr}} \frac{d\alpha}{6n}, \tag{2}$$

where:

μ_{czr} - molar mass of the working medium,

$f_{śr} = \alpha f_g$ - minimum section of the stream of gases,

α - narrowing coefficient of the stream of gases,

f_g - geometrical passage section of the inlet valves,

$\rho_{śr} = \dfrac{p_{śr}}{R_{czr} T_{śr}}$ - density of the working medium in the minimum section of the gas stream,

$$p_{śr} = p, \quad T_{śr} = T_d \left(\frac{p}{p_d}\right)^{\frac{\kappa_d-1}{\kappa_d}}.$$

The value of gas velocity in the minimum stream section is calculated from the formula:

$$w_{śr} = \varphi w_t = \varphi \sqrt{2R_d T_d \frac{\kappa_d}{\kappa_d-1}\left[1-\left(\frac{p_{śr}}{p_d}\right)^{\frac{\kappa_d-1}{\kappa_d}}\right]}, \tag{3}$$

where:

φ - coefficient of reduction of the flow velocity,
w_t – theoretical velocity of the gas,
R_d – individual constant of the gas flowing in the inlet system,
T_d – temperature in the inlet system,
κ_d – exponent of the adiabatic curve of the medium flowing through the inlet system,
$p_{śr}$ – mean pressure in the cylinder,
p_d – pressure in the inlet system.

Substituting to equation (2) the expressions determining $f_{śr}$, $w_{śr}$, and $\rho_{śr}$ for the cylinder filling process, and making transformations, we obtain:

$$\delta M_d = \frac{\sqrt{2R_d}}{R \cdot 6n}(\mu_d \cdot f_d)y_d \frac{p_d}{\sqrt{T_d}} d\alpha, \tag{4}$$

where:

$\mu_d = \alpha \cdot \varphi$ - coefficient of the flow throughput through the valve,

$$y_d = \left(\frac{p_{śr}}{p_d}\right)^{\frac{1}{\kappa_d}} \sqrt{\frac{\kappa_d}{\kappa_d - 1}\left[1 - \left(\frac{p_{śr}}{p_d}\right)^{\frac{\kappa_d - 1}{\kappa_d}}\right]}$$ - function of the flow throughput through the inlet valve.

Integrating expression (4) in the limits from $\alpha = 0$ to $\alpha = 180°$ of crankshaft rotation we obtain:

$$M_d = \int_{\alpha=0}^{\alpha=180°} \frac{\sqrt{2R_d}}{\overline{R} \cdot 6n}(\mu_d f_d)y_d \frac{p_d}{\sqrt{T_d}} d\alpha = \frac{\sqrt{2R_d}}{\overline{R} \cdot 6n}(\mu_d f_d)_{śr} y_{dśr} \frac{p_d}{\sqrt{T_d}} \cdot 180 = a_d y_{dśr}, \tag{5}$$

where:

$$a_d = \frac{30\sqrt{2R_d}(\mu_d f_d)_{śr} p_d}{\overline{R} \cdot n \cdot \sqrt{T_d}}, \tag{6}$$

$$y_{śr} = \left(\frac{p_{śr}}{p_d}\right)^{\frac{1}{\kappa_d}} \sqrt{\frac{\kappa_d}{\kappa_d - 1}\left[1 - \left(\frac{p_{śr}}{p_d}\right)^{\frac{\kappa_d - 1}{\kappa_d}}\right]}. \tag{7}$$

The value of the cylinder filling ratio can be calculated according to the relationship proposed:

$$\eta_v = \frac{T_o}{T_d + \Delta T} \frac{1}{\varepsilon - 1}\left[\frac{\varepsilon p_a}{p_o} - \frac{p_r}{p_o}\left(\frac{p_a}{p_r}\right)^{\frac{n_1 - 1}{n_1}}\right], \tag{8}$$

where:

n_1 - exponent of the compression polytrope,
$p_r = p_w (1 + a)$ - pressure of the exhaust gas residue in the cylinder, according to [2] $a = 0.3 - 0.5$,
$p_w = p_o (1 + \delta)$ - pressure in the outlet system of an unsupercharged engine,
p_o - ambient pressure,

$\delta = \frac{\Delta p_w}{p_o} = 0.01 - 0.03$.

The value of pressure in the conventional end of the filling process is:

$$p_a = \frac{1}{2}(p_d + p_{śr}). \tag{9}$$

The value of mean pressure in the cylinder during the cylinder filling process is calculated according to the following algorithm:

1. We assign the value $p_{śr}$ and calculate p_a according to equation (9).
2. We calculate η_v according to equation (8).
3. Knowing the value η_v, we calculate $M_d = \eta_v \cdot \frac{p_o V_s}{RT_o}$.
4. From formula (6) we calculate a_d, and then from equation (5) we determine $y_{dśr}$.
5. According to equation (6) we determine the pressure ratio $\left(\frac{p_{śr}}{p_d}\right)$, and then we calculate:

$$p_{śr} = p_d \left(\frac{p_{śr}}{p_d} \right). \tag{10}$$

If the calculated value $p_{śr}$ does not coincide with the value assumed for calculations, we repeat the calculations with a new value of $p_{śr}$. We continue the calculations until the value $p_{śr}$ stabilises.

Knowing the values p_a, V_a and M_d from the state equation we calculate the value of temperature T_a and the coefficient of the exhaust gas residue γ.

We calculate the parameters of point f after prior determination of the exponent of the polytrope of compression n_1, utilising for that purpose the experimentally made indicator diagram. From the effective indicator diagram for $V_1 \leq V_{d'}$ we read the pressure p_1 and for the point $V_2 \geq V_{ps}$ we read the pressure p_2 and then using the polytrope equation $pV^{n_1} = const$, we determine the value of the exponent of the polytrope of compression n_1. Knowing the exponent of the polytrope n_1, we determine the point f as the point of intersection of the polytrope of exponent n_1 with the isobar $p_a = const$ (the cylinder volumes, for which the engine work cycle is realised by a closed thermodynamic system, should be utilised for calculations).

4. Parameters during compression

We calculate the parameters with the assumption that the compression process is reversible and that it proceeds adiabatically. It means that the total quantity of the heat exchanged during that transformation $Q^{f-c} = 0$, and therefore $n_1 = \kappa_s$.

The equation of the first principle of thermodynamics for that process has the form:

$$0 = \bar{c}_{vsr}(M_c T_c - M_f T_f) - \bar{R} \frac{M_c T_c - M_f T_f}{\kappa_s - 1}. \tag{11}$$

The assumptions thus made permit the improvement of exactness of the value of the exponent n_1 calculated according to the methodology proposed in point 4.1 i.e.:

1. We assume the value $n_1 = \kappa_s$ calculated beforehand in the chapter 3. and calculate $\kappa_s - 1$.

2. From the relation $T_c = T_f \left(\dfrac{V_c}{V_f} \right)^{\kappa_s - 1}$ we calculate the value T_c.

3. We calculate $\bar{c}_{vsr} = a_{spr} + b_{spr} \dfrac{T_c + T_f}{2}$, and then $\kappa_s - 1 = \dfrac{\bar{R}}{\bar{c}_{vsr}}$.

If the calculated value $\kappa_s - 1$ differs from the assumed value, the calculations are repeated with a new value of $\kappa_s - 1$. The calculations are considered as completed if stabilisation of the values $\kappa_s - 1$ and T_c is obtained. Then we calculate the value:

$$p_c = \frac{\bar{R} M_c T_c}{V_c}. \tag{12}$$

5. Working parameters during combustion end expansion

The process of delivering heat to the computational work cycle (Fig. 1) occurs in isochoric manner – the transformation c-z' and in isobaric manner – the transformation z'-z. The expansion process from the point "z" to "w" is realised in a closed thermodynamic system. In the point "w" the process of the working medium exchange in the cylinder begins.

The parameters of the working medium at the conventional end of the effective combustion process, the point "z" is determined utilising the following system of equations:

$$\left.\begin{array}{l} M_z = \beta M_c = \beta\lambda M_o g_c (1+\gamma), \\[2mm] H_{pal} + Q_{c-z} = U_z - U_c + \int\limits_{V_c}^{V_z} pdV = H_z - H_c - \int\limits_{p_c}^{p_z} Vdp, \\[2mm] p_z = \lambda_p p_c, \\[2mm] p_z V_z = \overline{R} M_z T_z, \end{array}\right\} \qquad (13)$$

where:

$H_{pal} = c_{pal} g_c T_{pal}$ - enthalpy of the fuel delivered to the cylinder (c_{pal} - specific heat of the fuel,

g_c - fuel dose delivered per work cycle),

$Q_{c-z} = (x_z - y_z) g_c W_u = \xi g_c W_u$ - heat effectively emitted during the combustion process and delivered to the cycle in the section from the point,"c" to "z". In that relationship, x_z is the relative quantity of the heat emitted to the point "z", and y_z is the relative quantity of the heat carried away from the working medium to the cylinder walls during the period from the point "c" to the point "z", W_u – is the calorific value of the fuel.

The solution of the system of equations (13) makes it possible to determine the specific enthalpy of the working medium in the point,"z":

$$h_z = \frac{1}{\beta}\left[\frac{\xi W_u + c_{pal} T_{pal}}{\lambda M_o (1+\gamma)} + \overline{c}_{psr} T_c + \overline{R}(\lambda_p - 1) T_c\right]. \qquad (14)$$

The value of that enthalpy is also expressed by the formula:

$$h_z = \left(a_\gamma + \overline{R} + \frac{b_\gamma}{2} T_z\right) T_z. \qquad (15)$$

Therefore, knowing the value h_z, from equation (15) we can determine the value of temperature T_z from the formula:

$$T_z = \frac{-(a_\gamma + \overline{R}) + \sqrt{(a_\gamma + \overline{R})^2 + 2b_\gamma \cdot h_z}}{b_\gamma}. \qquad (16)$$

Knowing the values T_z and p_z and the quantity of the working medium in the point,"z", we calculate the volume of the working medium V_z from the state equation:

$$V_z = \frac{\overline{R} M_z T_z}{p_z} \qquad (17)$$

and also the ratio of initial and consecutive compression:

$$\rho = \frac{V_z}{V_c} \quad \text{and} \quad \delta = \frac{\varepsilon}{\rho}. \qquad (18)$$

The value of temperature of the working medium at the moment of opening the outlet valve in the point "w" is determined utilising the following system of equations:

$$Q_{z-w} = \left(1 - \xi - w_w\right)g_c W_u = U_w - U_z + \int_{V_z}^{V_w} pdV,$$

$$M_w = M_z = \beta\lambda M_o(1+\gamma),$$

$$V_w = f(\alpha), \tag{19}$$

$$p_w V_w = \bar{R}M_w T_w,$$

where: $w_w = \dfrac{\displaystyle\int_{\alpha_z}^{\alpha_w} \alpha_g F(T - T_{sc})d\alpha}{6 \cdot n \cdot \xi \cdot g_c \cdot W_u}$ - relative quantity of the heat carried away to the cylinder

walls during the process z-w,

The equation of the first principle of thermodynamics (the first equation of the system of equations (19) can be written in the form:

$$\frac{\left(1 - \xi - w_w\right)g_c W_u}{\beta\lambda M_o(1+\gamma)} = \bar{c}_{vw} T_w - \bar{c}_{vz} T_z + \int_{V_z}^{V_w} pdV. \tag{20}$$

The value of specific energy of the working medium at the moment of opening the outlet valve is:

$$u_w = \bar{c}_{vw} T_w = \frac{\left(1 - \xi - w_w\right)g_c W_u}{\beta\lambda M_o(1+\gamma)} + \bar{c}_{vz} T_z - \frac{\displaystyle\int_{V_z}^{V_w} pdV}{\beta\lambda M_o(1+\gamma)}. \tag{21}$$

Because:

$$u_w = \left(a_\gamma + b_\gamma T_w\right)T_w, \tag{22}$$

then knowing the value u_w determined from equation (20), we can calculate the value T_w:

$$T_w = \frac{-a_\gamma + \sqrt{a_\gamma^2 + 4b_\gamma u_w}}{2b_\gamma}. \tag{23}$$

The value of the exponent of the expansion polytrope is determined from the equation of the expansion polytrope:

$$p_z V_z^{n_2} = p_w V_w^{n_2}, \tag{24}$$

from where we can obtain:

$$n_2 = \frac{\log \dfrac{p_z}{p_w}}{\log \dfrac{V_w}{V_z}}. \tag{25}$$

The value poly-trails of the expansion determined in the paper can be put-upon also with reference to other uses.

6. Summary

An isolated system is one that is not influenced in any way by the surroundings. A closed system has no transfer of mass with its surroundings, but may have a transfer of energy with surroundings. An open system is one that may have a transfer of both mass and energy with its surroundings. Our system being an object of the paper is an open system for some phenomena.

The thermodynamical analysis of the filling process of the cylinder which occurs in calculated period from the percussion cap of the position the piston at TDC to it position at BDC, at constant average value of the pressure in the cylinder.

The quantity of the working charge in the established point placed on the compression line by piston BDC is equal quantity of the working charge in the end point of the filling process, answering to the start point of the real compression process.

The state of the working charge in cylinder described is four parameters: a pressure, a temperature, quantity moles of the working charge and its volume. For appointment those parameters, earlier worked out dependences are put-upon.

As result of the realization work presented in the paper one can mark amongst other things:
- the value of the average pressure in the cylinder during the filling process of the cylinder basing on presented in the paper algorithm,
- polytropic exponent for compression process at the implementation to this end of the experimentally elaborated indicator diagram,
- the value of the temperature of the working charge at point of the exhaust-valve opening by way implementation of worked out system equations,
- parameters of the working charge in the established end of the effective combustion process,
- the value specific energy of the working charge in the point of the exhaust-valve,
- the value polytropic exponent expansion appointed from polytropic equation of expansion.

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