USE OF EXHAUST GASES IN PISTON INTERNAL COMBUSTION ENGINE BY IMPLEMENTATION OF THE STIRLIG CYCLE

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ABSTRACT
The improvement of the fuel economy of the piston internal combustion engine is always a question of present interest. One way of solving it is the use the energy of the exhaust gases. The scope of the study is a test of an engine, which is a set of four-cycle piston internal combustion engine and an engine connected to its shaft, operating according to the Stirling cycle and using the heat of exhaust gases of a piston internal combustion engine.

Key words: internal combustion engine, Stirling engine, fuel economy.

ABSTRAKT
Zlepšenie spotreby paliva piestového spaľovacieho motora je stále aktuálna téma. Jeden zo spôsobov riešenia spočíva vo využívaní energie z výfukových plynov. Predmetom príspevku je test motora tvoreného štvordobým piestovým spaľovacím motorom a motorom pripojeným k jeho hriadeľu, pracujúcim podľa Stirlingovho cyklu, za použitia tepla z výfukových plynov piestového spaľovacieho motora.

Kľúčové slová: spaľovací motor, Stirlingov motor, spotreba paliva.

1 INTRODUCTION

There are many plants in literature that uses the energy of the exhaust gases of the piston internal combustion engine (ICE). All of them improve fuel economy to a certain degree. There are plants that operate according to the Rankine cycle with a steam turbine and there are plants with a gas turbine.

All these plants have the disadvantage that the vane and plate machines are external to the piston engine combustion, which is a precondition for increasing the thermal and mechanical losses. The speed of the gas and steam turbines is ten to

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twenty times higher than that of the piston engine, which is why optimal collaboration occurs only in the calculation mode or in a very narrow working range around it.

These shortcomings can be avoided to some extent, if the energy of the exhaust gases is used in a cycle of Stirling, as the cylinders of the internal combustion engine and the Stirling cylinders are combined into a single housing with a common crankshaft. The scheme of such a combined engine is shown on Fig. 1.

2 HEADING

The combustion engine is a double-cylinder, four-tact engine, where both cylinders move synchronously, but their work is out of phase at *360⁰*. The knee of the Stirling cylinder 3 is dephased to the knees of the engines at *+45⁰*, and that of the Stirling cylinder 4 - at *-45⁰*, where the angle between them is *90⁰*.

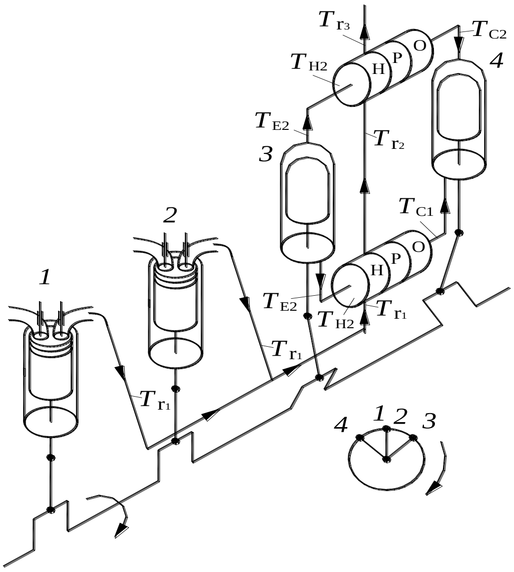

Figure 1

The exhaust gases from the internal combustion engine pass successively through the two heat-exchange groups, each of which consists of a heater H, regenerator R and cooler O, and then come out into the atmosphere. The spaces above the pistons of both Stirling cylinders are interconnected through the upper heat exchanger. Subpiston spaces are similarly connected through the lower heat exchanger. Thus both Stirling pistons form the hot volume E and cold volume C.

For defining the average torque and the effective power of the internal combustion engine the methodology given in [1] shall be applied. The inertial force of the movement of a crankshaft group is located by the statement:

$$P_j = \omega^2 R m_j \left[\cos\varphi + \lambda \frac{\cos 2\varphi\left(1-\lambda^2\sin^2\varphi\right) + \dfrac{\lambda^2}{4}\sin^2 2\varphi}{\left(1-\lambda^2\sin^2\varphi\right)\sqrt{1-\lambda^2\sin^2\varphi}} \right], \qquad (1)$$

where m_j is the reduced mass of the *j* piston, *R* and *L* – respectively the lengths of the knee and the connecting rod, $\lambda = \frac{R}{L}$, φ – the angle of the rotation of the knee, ω – the angle speed.

The gas force is defined applying the same procedure [2] using the indicator diagram from Fig. 2.

The imported quantity of heat Q_1 is determined by the expression:

$$Q_1 = g_ц H_u, \tag{2}$$

where $g_ц$ is the fuel portion of the cycle, and H_u - the lower specific heat of fuel combustion.

The pressures in specific points of the indicator chart types have the expression:

$$p_c = cp_a\varepsilon^k, \tag{3}$$

$$p_y = cp_a\varepsilon^k + \frac{nQ_1 y(k-1)\varepsilon}{V_a}, \tag{4}$$

$$p_b = \frac{b}{\varepsilon^{k-1}k^k V_a}\frac{\left[(1-n+kn)(k-1)Q_1 yz + c\varepsilon^{k-1}kp_aV_a\right]^k}{\left[n(k-1)Q_1 y + c\varepsilon^{k-1}p_aV_a\right]^{k-1}}, \tag{5}$$

where *c* is a factor, which determines the losses during compression, *k* is the factor of the adiabatic, $p_a = 0{,}9.10^5[Pa]$, $\varepsilon = \frac{V_a}{V_c}$ – compression ratio, *n* – distribution coefficient of imported heat, *y* – factor, which determines the losses during the combustion at constant volume, *b* characterizes the losses during release.

At the so determined pressure $p_г$, the gas force is determined by the following expression:

$$P_г = \frac{\pi D^2}{4}p_г. \tag{6}$$

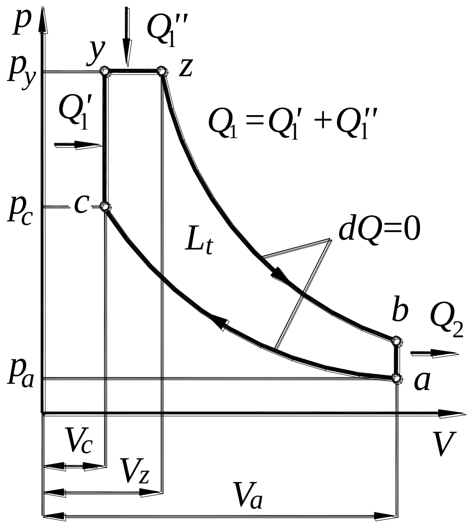

Figure 2

The total force acting on the crank mechanism is defined as the amount of the gas and the inertia forces:

$$P_\Sigma = P_г + P_j. \tag{7}$$

The torque of the one-cylinder engine is determined by the expression:

$$M_в^1 = P_\Sigma R\frac{\sin[\varphi + \arcsin(\lambda\sin\varphi)]}{1-\lambda^2\sin^2\varphi}. \tag{8}$$

The torque of the two-cylinder engine is determined by the expression:

$$M_в^{1+2} = M_в^1(\varphi_0) + M_в^1(\varphi_{360}). \tag{9}$$

The average torque of the internal combustion engine is determined by the expression:

$$M_{в,ср}^{1+2} = \frac{M_{в}^{1+2}}{720}. \tag{10}$$

And the effective power of ICE is:

$$N_{e}^{1+2} = \frac{M_{в,ср}^{1+2}\omega}{1000}0{,}82. \tag{11}$$

Here the coefficient *0,82* gives the thermal and mechanical losses.

To determine the pressure in the Stirling sector it is assumed that the working substance in the Stirling groups is an ideal gas with constant mass. Further the following indexes are placed: for parameters associated with the hot volume - index E, for those associated with the cold volume – index C and for those associated with the dead - index D. We come to the expression for the total mass of the working substance:

$$m = m_E + m_C + m_D. \tag{12}$$

Taking into account the equation of the condition the expression (12) shall be modified like that:

$$m = \frac{p}{R}\left(\frac{V_E}{T_E} + \frac{V_C}{T_C} + \frac{V_D}{T_D}\right), \tag{13}$$

where p is the pressure in the system, and with V and T the respective volumes and temperatures are marked.

If the temperature in the dead volume is the average of the two temperatures $T_D = \frac{T_E + T_C}{2}$, then from (13) for the pressure in the system looks like that:

$$p = \frac{mR}{\frac{V_E}{T_E} + \frac{V_C}{T_C} + \frac{2V_D}{T_E + T_C}}. \tag{14}$$

The mass of the working substance does not change and is equal to its mass at the beginning, when occupying the maximum volume $V_{\max}$ by the initial values of the temperature T_0 and the pressure p_0, i.e. $m = \frac{p_0 V_{\max}}{RT_0}$. Thus, the final expression for the pressure in the cylinder Stirling takes the form:

$$p = p_0 \frac{V_{\max}}{T_0\left(\frac{V_E}{T_E} + \frac{V_C}{T_C} + \frac{2V_D}{T_E + T_C}\right)}. \tag{15}$$

The values of the hot and cold volumes are determined by the expressions:

$$V_E = \frac{\pi D_E^2}{4} x_E \text{ and } V_C = \frac{\pi D_C^2}{4} x_C, \tag{16}$$

where D_E and D_C are the diameters of the respective pistons, and x_E and x_C are their displacements.

For determining the displacements of the pistons of the 3th and 4th cylinder the following expressions are valid:

$$x_E = R\left[1 - \cos\varphi + \frac{1}{\lambda}\left(1 - \sqrt{1 - \lambda^2 \sin\varphi}\right)\right] \text{ and}$$

$$x_C = R\left[1-\cos\left(\varphi-\frac{\pi}{2}\right)+\frac{1}{\lambda}\left(1-\sqrt{1-\lambda^2\sin\left(\varphi-\frac{\pi}{2}\right)}\right)\right]. \tag{17}$$

The total volume is determined by the expression:

$$V = V_E + V_C + V_D. \tag{18}$$

Taking into account the already determined parameters the pressures in the upper part of the Stirling group (dephased by *+45⁰* from the first piston) and in the lower part of the group (dephased by *-45⁰* from the first piston) can be determined by (15). Thus for each of the pistons the gas forces from above and below shall be determined according to the expressions:

$$P_{\Gamma}^{\mathrm{up}} = \frac{\pi D^2}{4}p^{\mathrm{up}} \text{ and } P_{\Gamma}^{\mathrm{down}} = \frac{\pi D^2}{4}p^{\mathrm{down}}. \tag{19}$$

The inertion force for each stirlig piston is determined by (1), taking into account the respective dephasing (for the 3$^{\mathrm{th}}$ cylinder with $+45^0$, and for 4$^{\mathrm{th}}$ – with -45^0).

The total force acting on the crank mechanism for the Stirling cylinder is defined as the amount of the gas and inertia forces:

$$P_{\Sigma,3} = P_{\Gamma,3}^{\mathrm{up}} + P_{\Gamma,3}^{\mathrm{down}} + P_{j,3} \text{ and } P_{\Sigma,4} = P_{\Gamma,4}^{\mathrm{up}} + P_{\Gamma,4}^{\mathrm{down}} + P_{j,4} \tag{20}$$

The torque for each cylinder Stirling is determined by expressions:

$$M_{в}^{3} = P_{\Sigma,3}R\frac{\sin\left[\varphi+\frac{\pi}{4}+\arcsin\left(\lambda\sin\left(\varphi+\frac{\pi}{4}\right)\right)\right]}{1-\lambda^2\sin^2\left(\varphi+\frac{\pi}{4}\right)} \tag{21}$$

and

$$M_{в}^{4} = P_{\Sigma,4}R\frac{\sin\left[\varphi-\frac{\pi}{4}+\arcsin\left(\lambda\sin\left(\varphi-\frac{\pi}{4}\right)\right)\right]}{1-\lambda^2\sin^2\left(\varphi-\frac{\pi}{4}\right)}. \tag{22}$$

The cumulative torque of the Stirling group is determined by the expression:

$$M_{в}^{3+4} = M_{в}^{3} + M_{в}^{4}. \tag{23}$$

Thus the average torque of the engine of Stirling is determined:

$$M_{в,ср}^{3+4} = \frac{M_{в}^{3+4}}{720} \tag{24}$$

and the effective power of the engine of Stirling:

$$N_e^{3+4} = \frac{M_{в,ср}^{3+4}\omega}{1000}0{,}75. \tag{25}$$

Here the coefficient *0,75* marks the thermal and mechanical losses in this sector.

The total torque of the combined engine is defined by summing up the expressions (9) and (23):

$$M_{в} = M_{в}^{1+2} + M_{в}^{3+4}. \tag{26}$$

The average torque is defined as:

$$M_{в,ср} = \frac{M_{в}}{720}, \tag{27}$$

while the total effective power is equal to:

$$N_e = \frac{M_{в,ср}\omega}{1000}0{,}82. \tag{28}$$

The total efficiency coefficient of this combined engine is obtained as a ratio between the total effective power and the power that would result from imported heat Q_1. This power is determined by the expression:

$$N_1 = \frac{Q_1 \omega i}{4\pi 1000}, \tag{29}$$

where $i=2$ is the number of cylinders of the internal combustion engine.

The share of the processed energy from Stirling group, which is obtained from the exhaust gases, is:

$$\eta_{cmup.} = \frac{N_e^{3+4}}{N_1/3}. \tag{30}$$

From (28) and (29) the total efficiency coefficient of the combined engine is obtained:

$$\eta = \frac{N_e}{N_1}. \tag{31}$$

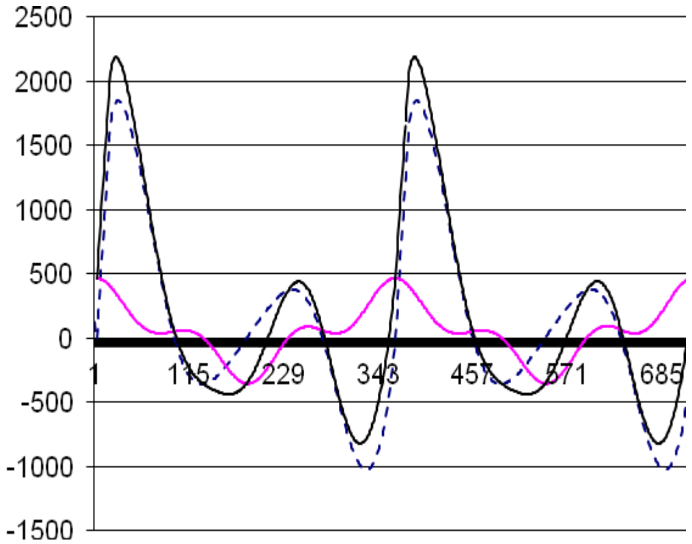

Figure 3

The above described methodology is applied by determining the torque of a specific two-cylinder four-cycle internal combustion engine and a two-cylinder Stirling group by the following numerical values of the parameters that are formed on the grounds of the diesel engine *D3900*:

$k=1{,}33$	$y=0{,}95$	$\varepsilon=16$	$D=0{,}09842[m^2]$	$n=0{,}52$	$V_a=0{,}001030597[m^3]$
$c=0{,}8$	$z=0{,}95$	$m_j=2[kg]$	$R=0{,}0635[m]$	$b=0{,}95$	

Through the conducted numerical study the characteristics of alternation of the torques of the ICE according to (9) (with the dotted line) are found (shown on fig. 3) of the Stirling group according to (23) (with the light line) of the total torque of the combined engine (26) (with the solid line). The total effective power of the combined engine is N_e=39,55 kW, and only of the Stirling group - N_e^{3+4}=11,29 kW. The share of the processed energy by the Stirling group is $\eta_{cmup.}$ = 0,434, and overall efficiency of the combined engine is η = 0,508.

3 CONCLUSION

We could draw the conclusion that the built in this manner combined engine uses effectively the fuel energy and realizes a good overall performance rate.

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Článok recenzovali dvaja nezávislí recenzenti.