Influence of Compression Ratio on Pressure in the Cylinder of Internal Combustion Engine

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Research Article

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Influence of Compression Ratio on Pressure in the Cylinder of Internal Combustion Engine

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Abstract— Currently, engines with variable compression ratios are not used, presumably due to the complexity of the design. Regulating the compression ratio is a complex problem, for which various engines are being developed that differ in design, kinematics, method of regulating the load and changing the compression ratio. This paper presents the results of calculating efficiency and fuel consumption for crankshaft speeds of 2000 rpm and 4000 rpm with a compression ratio of 8.6 and a rotation speed of 4000 rpm with a compression ratio of 13 and combustion characteristics m= 3 and m=1 are presented. It is shown that these characteristics at higher speeds approach those at lower speeds with increasing compression ratio. The engine parameters were calculated for various modes based on Wiebe theory. Based on the above calculations, efficiency and fuel consumption are determined for the corresponding modes and nature of engine operation. As a result of the calculations, it was obtained: with an increase in the crankshaft rotation speed, the Cycle Efficiency Coefficient will decrease by almost half and fuel consumption will accordingly increase; with an increase in the compression ratio, the Efficiency Coefficient is restored and fuel consumption is accordingly reduced.

Keywords: twin-shaft engines, compression ratio, fuel consumption, combustion nature, crankshaft

Article Highlights: As the crankshaft rotation speed increases, the efficiency of the cycle decreases, and fuel consumption increases. As the compression ratio increases, efficiency is restored and fuel consumption decreases accordingly.

With flexible control of the compression ratio, it is possible to influence the parameters of physical processes in the engine that affect fuel consumption and emissions of toxic components.

The engine design must provide for the use of compression ratio control to restore pressure and, accordingly, efficiency.
1. INTRODUCTION

Thanks to flexible control of the compression ratio, it is possible to influence the parameters of physical processes in the engine that affect fuel consumption and the emission of toxic components: pressure and temperature at the end of the compression stroke; maximum combustion pressure and temperature; degree of expansion and indicator efficiency; combustion chamber volume; exhaust gas temperature.

2. MATHEMATICAL MODEL

The calculation is made under the following conditions:

Specific volume of working fluid

\[ \nu = \frac{V_z}{\varepsilon} \left( 1 + \frac{\varepsilon - 1}{2} \left[ 1 + \frac{1}{\lambda} \right] - \left( \cos \alpha + \frac{1}{\lambda} \sqrt{1 - \lambda^2 \sin^2 \alpha} \right) \right) \]

where \( \lambda \) - is the ratio of the crank radius.

Proportion of fuel burned at the site

\[ i - (i-1) \]

\[ \Delta x_n = e^{-6.908 \left( \frac{\Delta \phi}{\phi} \right)^{0.03}} - e^{-6.908 \left( \frac{\Delta \phi_{n-1}}{\phi} \right)^{0.03}} \]

Pressure of the working fluid in the cylinder during the combustion process

\[ P_n = \frac{0.0854 q_z \Delta x_n + P_{n-1} \left( k v_{n-1} - v_n \right)}{k v_n - v_{n-1}} \]

\( k \) - is the heat capacity factor, \( q_z \) - is the total heat of combustion used.

Ignition timing is 24°, compression ratios are 8.6 and 13, rotation speed is 2000 rpm and 4000 rpm, combustion character indicators \( m=1 \) and 3. Flame front inclination angle \( \varphi_r = 46^\circ \) and \( 92^\circ \).

Table 1 shows the data of this calculation. Based on these data, the corresponding efficiency factors and fuel consumption were determined.

3. NUMERICAL MODEL

Based on the points on the graph (Fig. 1.), the state of the cycle is determined, from which the efficiency is calculated. The cycle work is composed of the sum of the combustion work during compression

\[ l_{xc} = \int_{y}^{c} p_y v_y dv_y \quad \text{or} \quad \sum \Delta p_y \Delta v_y \ldots \] (1)
The cycle work is composed of the sum of the combustion work during expansion

\[ l_{cc} = \int_c^z p_c v_c dv_c \quad \text{or} \quad \sum \Delta p_c \Delta v_c \ldots \]  

(2)

The work of the cycle is made up of the sum of the work of pure compression

\[ l_{ay} = \frac{1}{n_1-1} \left( p_y v_y - p_a v_a \right), \ldots (n_1 = 1.35) \ldots \]  

(3)

The work of the cycle is composed of the sum of the work of pure expansion

\[ l_{zb} = \frac{1}{n_2-1} \left( p_z v_z - p_b v_b \right), \ldots (n_2 = 1.28) \ldots \]  

(4)

where \( p_b = \left( \frac{v_z}{v_b} \right)^{\frac{n_2}{n_1}} \) \( p_z = \left( \frac{v_z}{v_a} \right)^{\frac{n_2}{n_1}} \)

Complete cycle work

\[ l = l_{yc} + l_{cc} + l_{ay} + l_{zb} \ldots \]  

(5)

Efficiency determined by the formula

\[ \eta_i = \frac{l(1+\gamma)(1+\alpha L'_0)}{427 H u} \]  

(6)

where \( H \) - is the lower calorific value of the fuel, \( H = 10500 \text{kcal/kg} \);  
\( 1+\gamma \) – coefficient of residual gases, \( 1+\gamma = 1.088 \);  
\( \alpha \) – excess air coefficient \( \alpha = 0.85 \);  
\( L'_0 \) - the theoretically required amount of air for complete combustion of 1 kg of fuel \( L'_0 = 14.8 \text{ kg g} \).

Indicated specific fuel consumption \( g = \frac{632}{\eta_i H u} \).

The figure shows a diagram for determining the operation of the cycle. \( \varphi_z \) – duration of combustion.

The table shows the values of the crankshaft rotation angles \( \varphi \), counted from the moment of ignition and \( \alpha \) - counted from the top dead center \( u \), respectively, working fluid pressure depending on the specific volume.

4. RESULTS
I. Consider the case \( \varepsilon = 8.6; m = 3; \varphi r = 46^\circ \)
According to formula (1) \( l_{ye} = -0,515 \times 10^4 \text{ kq/m \cdot kq} \)

according to formula (2) \( l_{ez} = 2,859 \times 10^4 \text{ kq/m \cdot kq} \)

according to formula (3) \( l_{ay} = -2,476 \times 10^4 \text{ kq/m \cdot kq} \)

according to formula (4) \( l_{zb} = 10,75 \times 10^4 \text{ kq/m \cdot kq} \)

as a result \( l_i = 10,63 \times 10^4 \text{ kq/m \cdot kq} \)

According to formula (6)

\[
\eta = \frac{10,63 \times 10^4 \cdot 1,088 (1 + 0,85 \cdot 14,8)}{427 \cdot 10500} = 0,35
\]

Indicated specific fuel consumption

\[
g = \frac{632}{\eta \cdot H_0} = \frac{632}{0,35 \cdot 10500} = 172 \text{ g/k.c.u}
\]

I. In case of \( \varepsilon = 8,6; m = 1; \varphi_r = 46^\circ \)

According to the formula (1) \( l_{ye} = -0,958 \times 10^4 \text{ kq/m \cdot kq} \)

according to the formula (2) \( l_{ez} = 2,7979 \times 10^4 \text{ kq/m \cdot kq} \)

according to the formula (3) \( l_{ay} = -2,476 \times 10^4 \text{ kq/m \cdot kq} \)

according to the formula (4) \( l_{zb} = 10,134 \text{ kq/m \cdot kq} \)

as a result \( l_i = 9,498 \times 10^4 \text{ kq/m \cdot kq} \)

\( l_i = 9,4979 \)

According to the formula (6)

\[
\eta = \frac{9,4979 \times 10^4 \cdot 1,088 (1 + 0,85 \cdot 14,8)}{427 \cdot 10500} = 0,31
\]

Indicated specific fuel consumption

\[
g = \frac{632}{\eta \cdot H_0} = \frac{632}{0,31 \cdot 10500} = 194 \text{ g/k.c.u}
\]

I. In case of \( \varepsilon = 8,6; m = 3; \varphi_r = 92^\circ \)

According to the formula (1) \( l_{ye} = -0,4636 \times 10^4 \text{ kq/m \cdot kq} \)

according to the formula (2) \( l_{ez} = 3,4686 \times 10^4 \text{ kq/m \cdot kq} \)

according to the formula (3) \( l_{ay} = -2,4757 \times 10^4 \text{ kq/m \cdot kq} \)

according to the formula (4) \( l_{zb} = 4,1108 \times 10^4 \text{ kq/m \cdot kq} \)

as a result \( l_i = 4,632 \times 10^4 \text{ kq/m \cdot kq} \)

According to the formula (6)

\[
\eta = 0,1526
\]

\[
g = \frac{632}{0,15 \cdot 10500} = 0,401 \text{ g/k.c.u}
\]

II. In case of \( \varepsilon = 8,6; m = 1; \varphi_r = 92^\circ \)

According to the formula (1) \( l_{ye} = -1,354 \times 10^4 \text{ kq/m \cdot kq} \)
according to the formula (2) \( l_{cz} = 6,317 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (3) \( l_{ay} = -2,476 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (4) \( l_{zq} = 3,926 \cdot 10^4 \text{km} / \text{kg} \)
as a result \( l_i = 6,413 \cdot 10^4 \text{km} / \text{kg} \)

According to the formula (6)

\[
\eta = 0,2116
\]
\[
g = \frac{632}{0,21\cdot 10500} = 286 \text{ г} / \text{l.c.ч}
\]

III. In case of \( \varepsilon = 13; m = 3; \varphi_r = 64^\circ \)

According to the formula (1) \( l_{ye} = 0,44548 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (2) \( l_{cz} = 5,926 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (3) \( l_{ay} = -2,998 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (4) \( l_{zq} = 11,28 \cdot 10^4 \text{km} / \text{kg} \)
as a result \( l_i = 13,8 \cdot 10^4 \text{km} / \text{kg} \)

According to the formula (6)

\[
\eta = 0,455
\]
\[
g = \frac{632}{0,455\cdot 10500} = 132 \text{ г} / \text{l.c.ч}
\]

IV. In case of \( \varepsilon = 13; m = 1; \varphi_r = 64^\circ \)

According to the formula (1) \( l_{ye} = 0,9949 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (2) \( l_{cz} = 6,5014 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (3) \( l_{ay} = -2,998 \cdot 10^4 \text{km} / \text{kg} \)
according to the formula (4) \( l_{zq} = 9,39 \cdot 10^4 \text{km} / \text{kg} \)
as a result \( l_i = 13,78 \cdot 10^4 \text{km} / \text{kg} \)

According to the formula (6)

\[
\eta = 0,3923
\]
\[
g = \frac{632}{0,39\cdot 10500} = 154 \text{ г} / \text{l.c.ч}
\]

5. CONCLUSIONS

With an increase in crankshaft rotation speed from 2000 rpm to 4000 rpm, the efficiency of the cycle will decrease by almost half and fuel consumption will increase accordingly. By increasing the compression ratio from 8.6 to 13 at a speed of 4000 rpm, the efficiency is restored and fuel consumption decreases accordingly.
As the crankshaft rotation speed increases, the pressure in the cylinder drops and the efficiency decreases. In order to restore pressure and, accordingly, efficiency, compression ratio control can be applied, which must be provided for in the engine design.

**FUNDING**


**CONFLICT OF INTEREST**

The authors declare that they have no conflicts of interest.

**REFERENCES**

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Table 1. Pressure distribution depending on the specific volume for combustion types \( m=1 \) and \( m=3 \) at 2000 rpm. and 4000 rpm. crankshaft for compression ratios \( \varepsilon = 8,6 \) and \( \varepsilon = 13 \)

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<th>( \phi_0 )</th>
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<th>( P / kq / sm^2 ) ( m=3 )</th>
<th>( m=1 )</th>
<th>( Vm^3KQ )</th>
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FIGURE CAPTIONS

Fig. 1. Scheme for determining cycle operation
bMm – top dead center of the engine piston position,
HMm – bottom dead center of the engine piston position
Fig. 1.