ISSN 1052-6188, Journal of Machinery Manufacture and Reliability, 2024, Vol. 53, No. 4, pp. 390–396. © Pleiades Publishing, Ltd., 2024.

EXPERIMENTAL MECHANICS, DIAGNOSTICS, AND TESTING

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

**J. Agalarov[a], T. Hasanova[b,*], and J. Imamalieva[c]**

[a] *National Academy of Sciences of Azerbaijan (NASA), Institute of Mechanics and Mathematics (IMM), Department of Wave Dynamics, Baku, Azerbaijan*

[b] *Department of Mechanics Azerbaijan University of Architecture and Civil Engineering (AzUAC), Baku, Azerbaijan*

[c] *Department of Exploitation and Reconstruction of Buildings and Structures, Azerbaijan University of Architecture and Civil Engineering (AzUAC), Baku, Azerbaijan*

**e-mail: atika2014@rambler.ru*

Received February 27, 2024; revised April 2, 2024; accepted April 19, 2024

**Abstract**—Currently, variable compression ratio engines are not used, presumably due to the complexity of the design. Currently, various engines are being developed that differ in design, kinematics, load control and compression ratio changes. The article presents the results of calculating the efficiency and fuel consumption for a crankshaft with a rotation speed of 2000 and 4000 rpm with a compression ratio of 8.6 and a rotation speed of 4000 rpm with a compression ratio of 13 and provides combustion characteristics $m = 3$ and 1. It is shown that these characteristics at higher speeds they approach the indicators at lower speeds with an increase in the compression ratio. The engine parameters were calculated for various modes based on Wiebe theory. Based on the calculations, the efficiency and fuel consumption are determined for the corresponding modes and the nature of the engine operation. It has been found that with an increase in the crankshaft rotation speed, the cycle efficiency will decrease by almost half and fuel consumption will increase accordingly; with an increase in the compression ratio, the efficiency is restored and fuel consumption decreases accordingly.

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

**DOI:** 10.1134/S1052618824700298

## 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.

Despite the fact that adjusting the compression ratio is a difficult task, known two-shaft designs can be used without significant complications [1–4]. Work [5] showed a decrease in engine efficiency with increasing crankshaft rotation speed, which can be corrected by compensating the compression ratio. The article [6] pointed out the prospects of using twin-shaft engines [1–5] with an adjustable compression ratio. Here, according to the data in article [6], the results of calculating the efficiency and fuel consumption for speeds of 2000 and 4000 rpm with a compression ratio of 8.6 and a speed of 4000 rpm with a compression ratio of 13 and combustion characteristics $m = 3$ and 1 are presented. The indicated characteristics at higher speeds are shown to approach characteristics at lower speeds with increasing compression ratio. Previously [5] engine parameters were calculated under various modes based on Wiebe theory. Using the data from the above calculations, the efficiency and fuel consumption are determined for the corresponding modes and nature of engine operation.

## MATHEMATICAL MODEL

The calculation is made under the following conditions:

Specific volume of working fluid

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

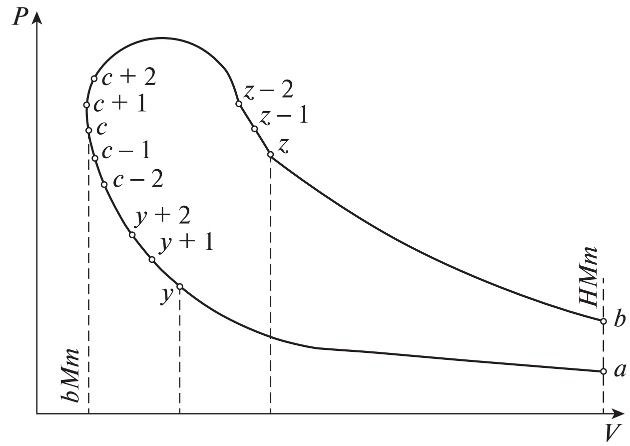


**Fig. 1.** Scheme for determining cycle operation: *bMm*—top dead center of the engine piston position; *HMm* is the bottom dead center of the engine piston position.

where $\lambda$ is the ratio of the crank radius.

Proportion of fuel burned at the site $i-(i-1)$

$$\Delta x_n = e^{-6.908(\Delta\varphi_n/\varphi_z)^{m+1}} - e^{-6.908(\Delta\varphi_{n-1}/\Delta\varphi_z)^{m+1}}.$$

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

$$P_n = \frac{0.0854 q_z \Delta x_n + P_{n-1}\left(kv_{n-1} - v_n\right)}{kv_n - v_{n-1}},$$

where $k$ 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 and 4000 rpm, combustion character indicators $m = 1$ and 3. Flame front inclination angle $\varphi_r = 46°$ and 92°.

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

## NUMERICAL MODEL

Figure 1 shows a diagram for determining the operation of the cycle. Based on the points on the graph, 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

$$l_{yc} = \int_y^c p_y v dv_y \quad \text{or} \quad \sum \Delta p_y \Delta v_y. \tag{1}$$

The cycle work is composed of the sum of the combustion work during expansion

$$l_{cz} = \int_c^z p_c v_c dv_c \quad \text{or} \quad \sum \Delta p_c \Delta v_c. \tag{2}$$

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

$$l_{ay} = \frac{1}{n_1 - 1}\left(p_y v_y - p_a v_a\right), \quad n_1 = 1.35. \tag{3}$$

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

$$l_{zb} = \frac{1}{n_2 - 1}\left(p_z v_z - p_b v_b\right), \quad n_2 = 1.28, \tag{4}$$

**Table 1.** Pressure distribution depending on the specific volume for combustion types $m = 1$ and 3 at 2000 and 4000 rpm crankshaft for compression ratios $\varepsilon = 8.6$, $\varepsilon = 13$, $\varphi_z$ is the duration of combustion

| $\alpha_0$ | $\varphi_0$ | $\varepsilon = 8.6$; $P$, kg/cm$^2$ | | | | | $\varepsilon = 13$ | | |
|---|---|---|---|---|---|---|---|---|---|
| | | $v$, m$^3$/kg | $m = 3$ | | $m = 1$ | | $v$, m$^3$/kg | $p$, kg/cm$^2$ | |
| | | | 2000 rpm $\varphi_z = 46°$ | 4000 rpm $\varphi_z = 92°$ | 2000 rpm $\varphi_z = 46°$ | 4000 rpm $\varphi_z = 92°$ | | 2000 rpm | 4000 rpm |
| –24 | 0 | 0.2065 | 9.0 | 9.0 | 9.0 | 9.0 | 0.1607 | 12.70 | 12.70 |
| –22 | 2 | 0.1967 | 8.40 | 8.57 | 8.40 | 8.40 | 0.1500 | 10.07 | 12.56 |
| –20 | 4 | 0.1879 | 7.80 | 8.02 | 8.43 | 8.01 | 0.1411 | 9.14 | 10.53 |
| –18 | 6 | 0.1794 | 7.36 | 7.58 | 9.2 | 7.92 | 0.1324 | 8.38 | 12.10 |
| –16 | 8 | 0.1721 | 7.05 | 7.17 | 12.23 | 8.24 | 0.1248 | 7.70 | 12.06 |
| –14 | 10 | 0.1655 | 6.78 | 6.88 | 15.54 | 8.88 | 0.1179 | 7.10 | 13.89 |
| –12 | 12 | 0.1598 | 6.91 | 6.04 | 19.40 | 7.58 | 0.1125 | 6.77 | 16.57 |
| –10 | 14 | 0.1551 | 7.52 | 6.46 | 26.67 | 11.18 | 0.1070 | 6.39 | 19.80 |
| –8 | 16 | 0.1511 | 8.68 | 6.40 | 28.17 | 10.80 | 0.1028 | 6.50 | 23.23 |
| –6 | 18 | 0.1480 | 10.65 | 6.19 | 32.59 | 12.77 | 0.0996 | 6.89 | 27.03 |
| –4 | 20 | 0.1458 | 14.43 | 6.93 | 36.10 | 14.90 | 0.0973 | 7.79 | 31.47 |
| –2 | 22 | 0.1445 | 17.32 | 6.90 | 38.38 | 17.39 | 0.0958 | 9.38 | 35.90 |
| 0 | 24 | 0.1440 | 29.18 | 7.44 | 42.21 | 20.64 | 0.0954 | 11.60 | 40.28 |
| 2 | 26 | 0.1445 | 27.90 | 8.90 | 44.77 | 29.57 | 0.0958 | 14.75 | 45.57 |
| 4 | 28 | 0.1458 | 33.98 | 8.83 | 47.05 | 36.30 | 0.0973 | 8.12 | 53.84 |
| 6 | 30 | 0.1480 | 39.74 | 9.66 | 48.90 | 28.78 | 0.0996 | 22.00 | 52.73 |
| 8 | 32 | 0.1511 | 44.63 | 10.50 | 50.20 | 31.01 | 0.1028 | 25.74 | 55.07 |
| 10 | 34 | 0.1511 | 48.35 | 12.54 | 50.40 | 32.69 | 0.1070 | 29.28 | 56.60 |
| 12 | 36 | 0.1598 | 50.00 | 12.57 | 50.00 | 33.92 | 0.1125 | 33.00 | 56.62 |
| 14 | 38 | 0.1654 | 51.00 | 13.60 | 48.26 | 34.76 | 0.1179 | 36.43 | 54.87 |
| 16 | 40 | 0.1721 | 50.00 | 14.64 | 48.01 | 34.76 | 0.1248 | 39.34 | 53.20 |
| 18 | 42 | 0.1794 | 48.37 | 15.78 | 46.28 | 34.72 | 0.1324 | 41.85 | 51.12 |
| 20 | 44 | 0.1879 | 46.25 | 16.71 | 44.27 | 34.40 | 0.1411 | 42.97 | 48.08 |
| 22 | 46 | 0.1867 | 45.53 | 17.86 | 41.93 | 33.55 | 0.1500 | 42.70 | 44.77 |
| 24 | 48 | 0.2065 | 44.00 | 18.50 | 39.66 | 32.69 | 0.1608 | 40.88 | 42.96 |
| 26 | 50 | 0.2171 | – | 20.09 | – | 31.50 | 0.1718 | 39.00 | 38.49 |
| 28 | 52 | 0.2284 | – | 20.05 | – | 29.92 | 0.1836 | 37.33 | 35.36 |
| 30 | 54 | 0.2404 | – | 20.06 | – | 28.51 | 0.1962 | 35.23 | 32.41 |
| 32 | 56 | 0.2531 | – | 20.37 | – | 27.03 | 0.2095 | 33.59 | 29.38 |
| 34 | 58 | 0.2667 | – | 20.29 | – | 25.65 | 0.2237 | 31.95 | 26.88 |
| 36 | 60 | 0.2804 | – | 17.79 | – | 24.26 | 0.2378 | 29.11 | 24.59 |
| 38 | 62 | 0.2352 | – | 16.92 | – | 22.71 | 0.2535 | 26.85 | 22.61 |
| 40 | 64 | 0.3101 | – | 16.10 | – | 21.64 | 0.2690 | 24.66 | 20.53 |
| 42 | 66 | 0.3273 | – | 15.86 | – | 20.42 | 0.2856 | – | – |
| 44 | 68 | 0.3422 | – | 15.30 | – | 17.95 | 0.3027 | – | – |
| 46 | 70 | 0.3598 | – | 14.39 | – | 16.89 | – | – | – |
| 48 | 72 | 0.3760 | – | 13.84 | – | 16.15 | – | – | – |
| 50 | 74 | 0.3957 | – | 12.69 | – | 15.24 | – | – | – |
| 52 | 76 | 0.4121 | – | 12.60 | – | 13.92 | – | – | – |
| 54 | 78 | 0.4307 | – | 12.51 | – | 13.14 | – | – | – |

**Table 1.** (Contd.)

| $\alpha_0$ | $\varphi_0$ | $\varepsilon = 8.6$; $P$, kg/cm$^2$ | | | | | $\varepsilon = 13$ | | |
|---|---|---|---|---|---|---|---|---|---|
| | | $v$, m$^3$/kg | $m = 3$ | | $m = 1$ | | $v$, m$^3$/kg | $p$, kg/cm$^2$ | |
| | | | 2000 rpm $\varphi_z = 46°$ | 4000 rpm $\varphi_z = 92°$ | 2000 rpm $\varphi_z = 46°$ | 4000 rpm $\varphi_z = 92°$ | | 2000 rpm | 4000 rpm |
| 56 | 80 | 0.4449 | – | 12.42 | – | 12.75 | – | – | – |
| 58 | 82 | 0.4685 | – | 12.04 | – | 12.19 | – | – | – |
| 60 | 84 | 0.4876 | – | 11.61 | – | 11.50 | – | – | – |
| 62 | 86 | 0.5087 | – | 11.32 | – | 10.90 | – | – | – |
| 64 | 88 | 0.5267 | – | 10.88 | – | 10.33 | – | – | – |
| 66 | 90 | 0.5482 | – | 10.60 | – | 9.82 | – | – | – |
| 68 | 92 | 0.5641 | – | 9.91 | – | 9.77 | – | – | – |

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

Complete cycle work

$$l = l_{yc} + l_{cz} + l_{ay} + l_{zb}. \tag{5}$$

Efficiency determined by the formula

$$\eta_i = \frac{l(1+\gamma)(1+\alpha L_0')}{427Hu}, \tag{6}$$

where $H$ is the lower calorific value of the fuel ($H = 10500$ kcal/kg); $(1 + \gamma)$ is the coefficient of residual gases, $(1 + \gamma) = 1.088$; $\alpha$ is the excess air coefficient ($\alpha = 0.85$); $L_0'$ is the theoretically required amount of air for complete combustion of 1 kg of fuel ($L_0' = 14.8$ kg g).

Indicated specific fuel consumption $g = \frac{632}{\eta_i Hu}$.

## RESULTS

I. Consider the case $\varepsilon = 8.6$; $m = 3$; $\varphi_r = 46°$

according to formula (1) $l_{yc} = -0.515\times10^4$ kg m/kg;

according to formula (2) $l_{cz} = 2.859\times10^4$ kg m/kg;

according to formula (3) $l_{ay} = -2.476\times10^4$ kg m/kg;

according to formula (4) $l_{zb} = 10.75\times10^4$ kg m/kg,

as a result, $l_i = 10.63\times10^4$ kg m/kg.

According to formula (6)

$$\eta = \frac{10.63\times10^4\times1.088\,(1+0.85\times14.8)}{427\times10\,500} = 0.35.$$

Indicated specific fuel consumption

$$g = \frac{632}{\eta_1 H_0} = \frac{632}{0.35\times10\,500} = 172\ \text{g/(h.p h)},$$

where g/(h.p h)—grams per horsepower hour.

II. In case of $\varepsilon = 8.6$; $m = 1$; $\varphi_r = 46°$

$$\text{according to formula (1) } l_{ye} = -0.958\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (2) } l_{cz} = 2.7979\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (3) } l_{ay} = -2.476\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (4) } l_{zb} = 10.134 \text{ kg m/kg,}$$

$$\text{as a result, } l_1 = 9.498\times10^4 \text{ kg m/kg;}$$

$$l_i = 9.4979.$$

According to formula (6)

$$\eta = \frac{9.4979\times10^4\times1.088(1+0.85\times14.8)}{427\times10500} = 0.31.$$

Indicated specific fuel consumption

$$g = \frac{632}{\eta_1 H_0} = \frac{632}{0.31\times10500} = 194 \text{ g/(h.p h).}$$

III. In case of case $\varepsilon = 8.6$; $m = 3$; $\varphi_r = 92°$

$$\text{according to formula (1) } l_{ye} = -0.4636\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (2) } l_{cz} = 3.4686\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (3) } l_{ay} = -2.4757\times10^4 \text{ kg m/kg ;}$$

$$\text{according to formula (4) } l_{zb} = 4.1108\times10^4 \text{ kg m/kg,}$$

$$\text{as a result, } l_i = 4.632\times10^4 \text{ kg m/kg.}$$

According to formula (6)

$$\eta = 0.1526;$$

$$g = \frac{632}{0.15\times10500} = 0.401 \text{ g/(h.p h).}$$

IV. In case of $\varepsilon = 8.6$; $m = 1$; $\varphi_r = 92°$

$$\text{according to formula (1) } l_{ye} = -1.354\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (2) } l_{cz} = 6.317\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (3) } l_{ay} = -2.476\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (4) } l_{zb} = 3.926\times10^4 \text{ kg m/kg,}$$

$$\text{as a result, } l_i = 6.413\times10^4 \text{ kg m/kg.}$$

According to formula (6)

$$\eta = 0.2116;$$

$$g = \frac{632}{0.21\times10500} = 286 \text{ g/(h.p h).}$$

V. In case of $\varepsilon = 13$; $m = 3$; $\varphi_r = 64°$

$$\text{according to formula (1) } l_{ye} = 0.44548\times10^4 \text{ kg m/kg;}$$

$$\text{according to formula (2) } l_{cz} = 5.926 \times 10^4 \text{ kg m/kg;}$$

$$\text{according to formula (3) } l_{ay} = -2.998 \times 10^4 \text{ kg m/kg;}$$

$$\text{according to formula (4) } l_{zb} = 11.28 \times 10^4 \text{ kg m/kg,}$$

$$\text{as a result, } l_i = 13.8 \times 10^4 \text{ kg m/kg.}$$

According to formula (6)

$$\eta = 0.455;$$

$$g = \frac{632}{0.455 \times 10\,500} = 132 \text{ g/(h.p h)},$$

VI. In case of $\varepsilon = 13$; $m = 1$; $\varphi_r = 64°$

$$\text{according to formula (1) } l_{ye} = 0.9949 \times 10^4 \text{ kg m/kg;}$$

$$\text{according to formula (2) } l_{cz} = 6.5014 \times 10^4 \text{ kg m/kg;}$$

$$\text{according to formula (3) } l_{ay} = -2.998 \times 10^4 \text{ kg m/kg;}$$

$$\text{according to formula (4) } l_{zb} = 9.39 \times 10^4 \text{ kg m/kg,}$$

$$\text{as a result, } l_i = 13.78 \times 10^4 \text{ kg m/kg.}$$

According to formula (6)

$$\eta = 0.3923;$$

$$g = \frac{632}{0.39 \times 10\,500} = 154 \text{ g/(h.p h)}.$$

## CONCLUSIONS

With an increase in crankshaft rotation speed from 2000 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

This research was supported by National Academy of Sciences of Azerbaijan (NASA), Institute of Mechanics and Mathematics (IMM), Baku, Azerbaijan and Patent: US 6039011. Date: Appl. March 4, 1998, Publ. March 21, 2000.

## CONFLICT OF INTEREST

The authors of this work declare that they have no conflicts of interest.

## REFERENCES

1. Moteki, K., Fujimoto, H., and Aoyata, S., Variable compression ratio mechanism of reprocating internal combustion engine, US Patent 6505581, 2002.
2. Cleevs, J.M., Variable compression ratio system for opposed piston and other methods of manufacture and use, US Patent 9206749, 2013.
3. Turner, J.W.G., Opposed piston internal combustion engine with variable timing, GB Patent 2428450, 2007.

4. Vibe, I.I., *Novoe o rabochem tsykle dvyhatelya* (New Information about the Engine Operating Cycle), Moscow: Mashgiz, 1962.
5. Klomp, E.D. and Rask, R.B., Variable compression ratio control system for an internal combustion engine, US Patent 6450136, 2002.
6. Agalarov, J., Nasibova, A., and Nuriyev, B., Internal combustion engine with opposed pistons, US Patent 6039011, 2000.
7. Dyachenko, V.G., *Teoriya dvigatelei vnutrennego sgoraniya* (Theory of Internal Combustion Engines), Kharkov: Kharkivsk. Nats. Avtomobil'no-Dorozhnyi Univ., 2009.
8. Pshikhorov, V.Kh., Dorukh, I.G., and Beresnev, A.L., Engine ignition system, RF Patent 2446309, 2010.
9. Beresnev, M.A., L-variation method for controlling internal combustion engines when operating on binary fuel, *Izv. Yuzhnogo Fed. Univ., Tekh. Nauki*, 2012, no. 3, pp. 251–256. https://elibrary.ru/ovyzcn.
10. Vagner, V.A., Application of alternative fuels in internal combustion engines, *Vest. Altaisk. Gos. Tekh. Univ. I.I. Polzunova*, 2000, no. 2, pp. 77–86. https://elibrary.ru/xyntxn.
11. Köse, H. and Ciniviz, M., An experimental investigation of effect on diesel engine performance and exhaust emissions of addition at dual fuel mode of hydrogen, *Fuel Process. Technol.*, 2013, vol. 114, pp. 26–34. https://doi.org/10.1016/j.fuproc.2013.03.023
12. Christodoulou, F. and Megaritis, A., Experimental investigation of the effects of simultaneous hydrogen and nitrogen addition on the emissions and combustion of a diesel engine, *Int. J. Hydrogen Energy*, 2014, vol. 39, no. 6, pp. 2692–2702. https://doi.org/10.1016/j.ijhydene.2013.11.124
13. Sharoglazov, B.A., Farafontov., M.F., and Klementyev, V.V., *Internal Combustion Engines: Theory, Modeling and Calculation of Processes*, Chelyabinsk: Yuzhno-Ural. Gos. Univ., 2004.
14. Baumgarten, C., *Mixture Formation in Internal Combustion Engines*, Heat and Mass Transfer, Berlin: Springer, 2006. https://doi.org/10.1007/3-540-30836-9
15. Saha, K. and Li, X., Assessment of cavitation models for flows in diesel injectors with single- and two-fluid approaches, *J. Eng. Gas Turbines Power*, 2016, vol. 138, p. 11504. https://doi.org/10.1115/1.4031224