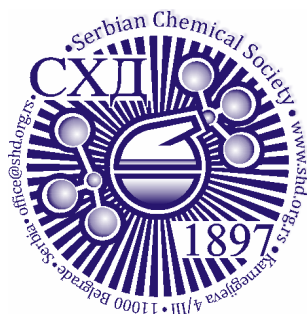




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A detailed thermodynamic analysis of a diesel engine at different compression ratios

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Abstract: This study investigates experimentally effects of compression ratio (CR) on the combustion, energy, and exergy characteristics of a diesel engine. A four-stroke, naturally aspirated, water-cooled, direct-injection, single-cylinder diesel engine with variable CR capacity was employed. Experiments were conducted at a constant speed of 1500 rpm under various loads, with CRs set to 15:1, 16:1, and 17:1. The software in the experimental setup calculates the arithmetic mean of ten cycles to minimize the effects of cycle-to-cycle variations and enhance the accuracy of the results. A detailed thermodynamic analysis was carried out to examine the combustion, energy, and exergy characteristics of the tested CRs using data obtained from the experiments. The results demonstrated that CR has a significant influence on combustion, energy, and exergy characteristics. Combustion parameters such as mass burned fraction, heat release rate, cylinder pressure, and gas temperature increased with rising CR. Similarly, performance indicators including efficiency, fuel consumption, and the overall energy and exergy performance of the engine also tended to improve as CR increased. Among the tested CRs, a CR of 16 provided the most favorable exergetic outcomes. Specifically, exergy efficiency, exergy performance coefficient, and sustainability index improved by 9.9–22.1 %, 45.3–110.9 %, and by 0.6–6.9 %, respectively, depending on engine load for CR of 16.

Keywords: combustion; exergy analysis; irreversibility; sustainability.

INTRODUCTION

Diesel engines are extensively employed in heavy-duty applications such as transportation, construction, and agriculture due to their low fuel consumption and high torque output and overall efficiency.¹ Single-cylinder diesel engines are

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frequently preferred as power sources in the agricultural sector, where they are commonly used to operate farming machinery such as water pumps, generators, and tillers, as well as in fishing activities.² However, the continuous rise in oil prices and the growing problem of environmental pollution have made it necessary to strengthen legal regulations and conduct technical studies aimed at improving the efficiency and environmental sustainability of internal combustion engines (ICEs).^{3,4} Hence, intensive efforts have been directed towards utilizing renewable alternative fuels, improving the properties of fossil fuels, and developing ICEs in line with current technological advancements.^{5,6} Improving fuel properties has made research on enhancing and optimizing engine design features increasingly important.^{7,8} The compression ratio (CR) is one of the fundamental design parameters of ICEs and has significant impact on combustion, performance, and emission characteristics. Therefore, CR must be determined appropriately accordance with the engine's structure and type, its charging system, and the fuel characteristics.⁹⁻¹¹ Conversely, research on engines is conducted through theoretical and/or experimental approaches. While experimental studies provide more realistic results, they are often time-consuming and costly. Advances in computer and software technologies have increased the reliability of theoretical studies, making it possible to achieve desired outcomes with fewer experimental studies.^{12,13} Theoretical and experimental studies on engines are primarily grounded in the principles of thermodynamics and fluid mechanics. Thermodynamics-based investigations often employ energy analysis derived from the first law of thermodynamics. However, it has been emphasized that exergy analysis, which supplements first-law approaches with those the second law of thermodynamics, can yield more accurate results. Moreover, exergy analysis enables the identification of losses caused by system irreversibilities and supports efficiency improvements through studies aimed at reducing these losses.¹⁴⁻¹⁶ For all these reasons, a detailed thermodynamic analysis was conducted in this study to investigate the effects of CR on the combustion, performance, energy, and exergy characteristics of a diesel engine.

EXPERIMENTAL

The experimental setup comprises of three primary components: the engine, the dynamometer, and the measurement and control panel. An image of the test system is provided in Fig. 1. As illustrated, an electric dynamometer was incorporated into the system to impose load to the engine, with the load being regulated through a dedicated switch.

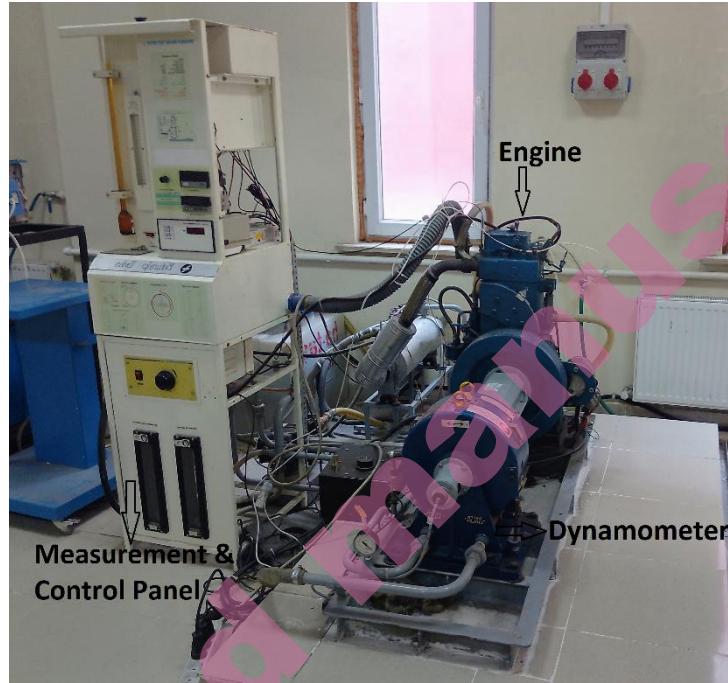


Fig 1. Image of the experimental setup

The system equipped with air, fuel, and coolant tanks, along with measurement devices for air, fuel, and coolant flow rates, torque, rotational speed, cylinder gas and fuel pressures, and temperatures at various points. A data acquisition unit is integrated to transfer the recorded measurements to a computer. The performance parameters are subsequently calculated from the collected data using ENGINESoft software, which operates on the LABVIEW platform. The engine employed in the system is a four-stroke, naturally aspirated, water-cooled, direct-injection diesel engine, with its technical specifications presented in Table 1. The CR of the engine can be adjusted through a mechanism located on the cylinder block. Experiments were carried out at a constant speed of 1500 rpm under varying load conditions ranging from 0 % to 100 %. Measurements were recorded once the engine reached a steady-state operating regime, and the collected data were utilized for a comprehensive thermodynamic analysis covering the combustion, performance, energy, and exergy characteristics.¹⁷

TABLE 1. The specifications of test engine¹⁷

Brand and Model	Kirloskar TV1
Number of cylinder	1
Bore x Stroke	87.5 mm x 110 mm
Cylinder volume	661 cm ³
Compression ratio	Variable
Engine power	3.5 kW @ 1500 rpm

The combustion, performance, energy, and exergy characteristics were identified and thoroughly analyzed using the equations provided below within the framework of

thermodynamic evaluation. Various engine performance parameters, including power, efficiency, specific fuel consumption, and the air excess coefficient, were determined accordingly. The indicated (in-cylinder) power (IP), brake power (BP), and friction (lost) power (FP) were calculated as follows.^{18,19}

$$IP = \frac{W \times N \times Z}{60 \times CC \times 1000} \quad (1)$$

$$BP = \frac{2 \times \pi \times N \times \tau}{60 \times 1000} \quad (2)$$

$$FP = IP - BP \quad (3)$$

The indicated specific fuel consumption ($ISFC$) and brake specific fuel consumption ($BSFC$) were calculated as follows.^{18,19}

$$ISFC = \frac{\dot{m}_f}{IP} \quad (4)$$

$$BSFC = \frac{\dot{m}_f}{BP} \quad (5)$$

The indicated thermal efficiency (ITE or η_i), mechanical efficiency (ME or η_m), and brake thermal efficiency (BTE or η_e) were calculated as follows.^{18,19}

$$ITE = \eta_i = \frac{3600 \times IP}{\dot{m}_f \times LHV_f} \quad (6)$$

$$BTE = \eta_e = \frac{3600 \times BP}{\dot{m}_f \times LHV_f} \quad (7)$$

In the energy analysis, the fuel energy (E_f), energy converted into work (E_W), energy lost with heat transfer (E_{cw}), energy lost with exhaust (E_{exh}), and other energy losses (E_{other}) were determined as follows.^{19–21}

$$E_f = \dot{m}_f \times LHV_f \quad (8)$$

$$E_W = 3600 \times BP \quad (9)$$

$$E_{cw} = \dot{m}_{cw} \times c_{p,cw} \times (T_2 - T_1) \quad (10)$$

$$E_{exh} = (\dot{m}_f + \dot{m}_h) \times c_{p,exh} \times (T_3 - T_0) \quad (11)$$

$$E_{other} = E_f - (E_W + E_{cw} + E_{exh}) \quad (12)$$

In the exergy analysis, the fuel exergy (Ex_f), exergy converted into work (Ex_W), exergy lost with heat transfer (Ex_{cw}), exergy lost with exhaust (Ex_{exh}), and exergy destruction (Ex_{dest}) were determined as follows.^{19–21}

$$Ex_f = \dot{m}_f \left[1.0401 + 0.1728 \frac{h}{c} + 0.0432 \frac{o}{c} + 0.2169 \frac{s}{c} \left(1 - 2.0628 \frac{h}{c} \right) \right] \quad (13)$$

$$Ex_W = E_W \quad (14)$$

$$Ex_{cw} = E_{cw} + \left[\dot{m}_{cw} \times c_{p,w} \times T_2 \times \ln \left(\frac{T_1}{T_2} \right) \right] \quad (15)$$

$$Ex_{exh} = E_{exh} + \left[(\dot{m}_f + \dot{m}_h) \times c_{p,exh} \times T_0 \times \ln(T_0/T_3) \right] \quad (16)$$

$$Ex_{dest} = Ex_f - (Ex_W + Ex_{cw} + Ex_{exh}) \quad (17)$$

The exergy efficiency (η_{ex}), exergy performance coefficient (EPC), and sustainability index (SI) were also calculated as follows.^{20–22}

$$\eta_{ex} = \frac{Ex_W}{Ex_f} \quad (18)$$

$$EPC = \frac{Ex_W}{Ex_{dest}} \quad (19)$$

$$SI = \frac{1}{1-\eta_{ex}} \quad (20)$$

				Uncertainty (%)
Engine load	Strain gauge	0–50 kg	±0,1 kg	±0,5
Engine speed	Magnetic sensor	0–10000 rpm	±10 rpm	±1
Pressure	Piezo sensor	0–5000 psi	±0,1 psi	±1
Crank angle	Magnetic sensor	–	±1 °CA	±0,2
Air flow rate	Pressure transducer	200 mm	1 mm	±1
Fuel flow rate	Pressure transducer	500 mmWC	1 mm	±1
Exhaust gas temperature	K–type thermocouple	0–1200 °C	±1 °C	±0,2
Brake torque	–	–	–	±0,5
Brake power	–	–	–	±1
Air excess coefficient	–	–	–	±1
Specific fuel consumption	–	–	–	±1,1
Brake thermal efficiency	–	–	–	±1,2

Uncertainties inevitably arise during experimental measurements due to the sensitivity of instruments, possible measurement errors, and environmental conditions. Hence, uncertainty analysis is essential to demonstrate the accuracy of experimental results. In this study, the uncertainty analysis was carried out using the method described below. Any result parameter R is expressed as a function of the independent variables $x_1, x_2, \dots, x_n$ expressed as follows.²²

$$R = R(x_1, x_2, \dots, x_n) \quad (21)$$

Accordingly, the uncertainty U_R associated with any result parameter is determined as follows.

$$U_R = \sqrt{U_{R,1}^2 + U_{R,2}^2 + \dots + U_{R,n}^2} = \sqrt{\sum_{i=1}^n U_{R,i}^2} \quad (22)$$

This approach, also referred as the partial uncertainty method, determines the uncertainty $U_{R,i}$ in equation (23) arising from any independent variable x_i as follows.²²

$$U_{R,i} = \left| \frac{\partial R}{\partial x_i} \right| U_i \quad (1 \leq i \leq n) \quad (23)$$

The scale, sensitivity, and uncertainty values of the measurement devices, along with the uncertainty values of the main parameters calculated using this method, are shown in Table 2.

RESULTS AND DISCUSSION

Fig. 2(a) presents the variation in mass burned fraction (MBF) with crank angle at full load for different CR s. For all CR s, MBF increases rapidly with rising of crank angle, as shown in the figure. However, MBF begins to rise earlier at higher CR s. This behavior is attributed to the shortening of the ignition delay period, which leads to earlier and faster combustion since the in-cylinder gas pressure and temperature reach higher values as the CR increases. Fig. 2(b) illustrates the variation in heat release rate (HRR) with crank angle at full load for different CR s. Similar to the MBF , the HRR rises earlier with increasing CR and it

begins to decline sooner due to the shorter combustion duration, as shown in the figure. Moreover, the maximum *HRR* value increases with higher *CR*s as a result of the reduced ignition delay period. The maximum *HRR* values were determined to be 63.65, 67.34 and 67.49 J/°CA for *CR*s of 15, 16, and 17, respectively. Fig. 2(c) depicts the variation in pressure rise rate (*PRR*) at full load for different *CR*s. As the *CR* increases, the *PRR* begins to rise earlier and reaches its highest value at a *CR* of 17, as shown in the figure. The maximum *PRR* values were determined to be 6.04, 6.15 and 7.05 bar/°CA for *CR*s of 15, 16, and 17, respectively. These results indicate that increasing the *CR* beyond a certain level may lead to knocking, which adversely affects engine performance.

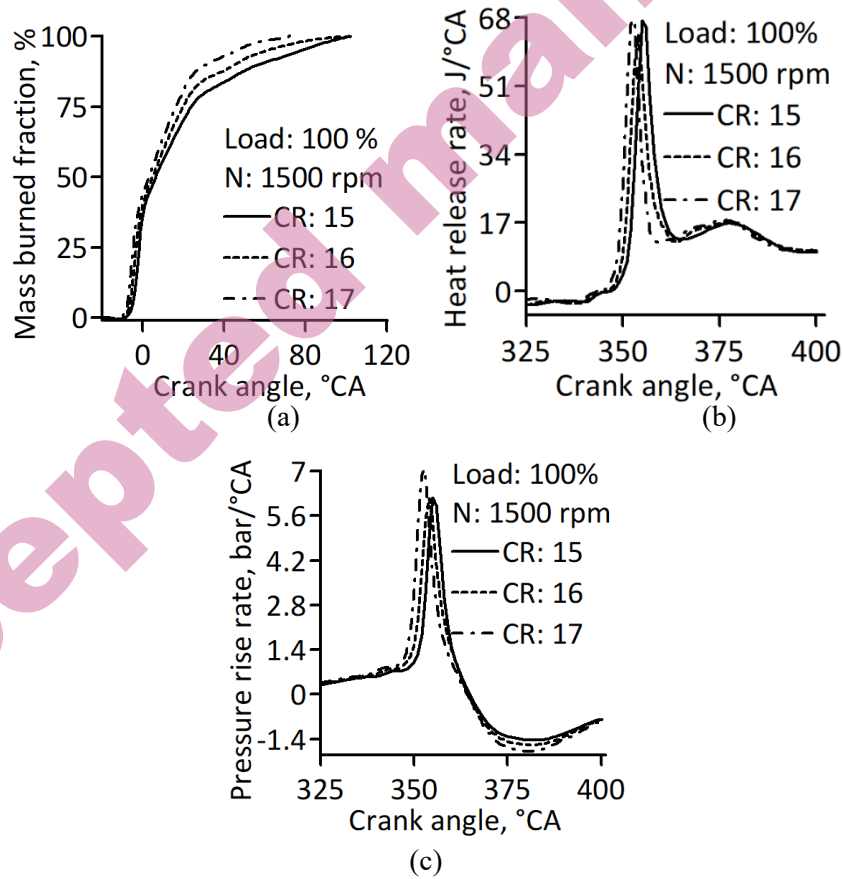


Fig 2. Variations in (a) mass burned fraction, (b) heat release rate and (c) pressure rise rate

Figs. 3(a) and 3(b) show the variations in cylinder pressure and temperature with crank angle respectively at full load for different *CR*s. Both cylinder pressure and temperature rise rapidly with the onset of combustion and this increase occurs

earlier as the CR rises, as illustrates in the figures. This behavior is attributed to the shortened ignition delay period at higher CR s, resulting in earlier combustion. Consequently, higher maximum values of cylinder pressure and temperature were achieved with increasing CR . Conversely, cylinder pressure and temperature began to decline earlier during the expansion process due to shorter ignition delay and combustion duration, leading to lower values at the end of the expansion. The maximum cylinder pressures were determined to be 61.81, 65.07 and 69.31 bar, while the maximum cylinder temperatures were found to be 1319.31, 1424.63 and 1456.38 °C for CR s of 15, 16, and 17, respectively.

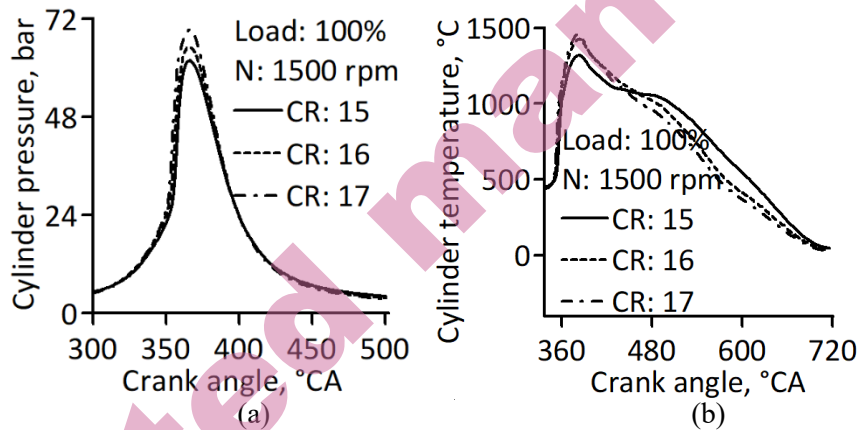


Fig 3. Variations in (a) cylinder pressure and (b) cylinder temperature

Figs. 4(a) and 4(b) present the variations in energy converted to work (E_w) and the ratio of E_w (E_w/E_t) with engine load at 1500 rpm for different CR s. Both E_w and its ratio increase significantly with rising engine load, which is attributed to the greater amount of fuel supplied to the cylinder at higher loads. Conversely, while the absolute E_w values did not vary considerably with CR , the ratio of E_w generally exhibited an increasing trend at higher CR s. This improvement is explained by enhanced combustion efficiency at elevated CR s, resulting in a larger fraction of fuel energy being converted into useful work.²³ It was determined that the E_w ratio increased by 4.7–9.9% and 4.6–17.4% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load, while a reduction of 2.3% was also observed at a CR of 16.

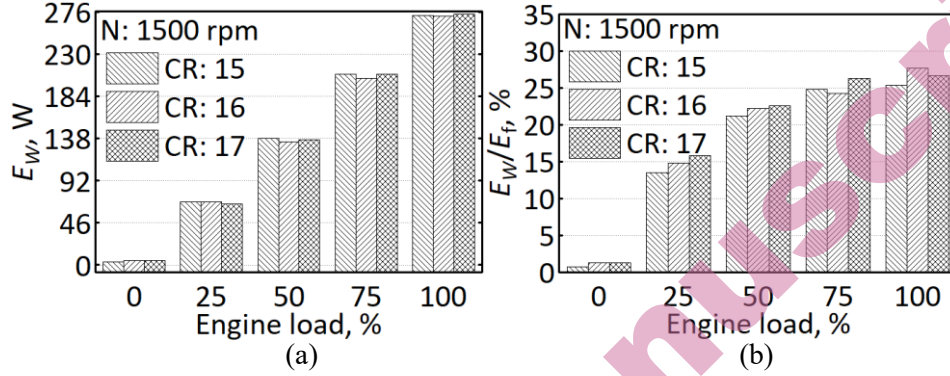


Fig 4. Variations in (a) energy converted to work and (b) ratio of energy converted to work

Figs. 5(a) and 5(b) show the variations in energy lost by heat transfer (E_{cw}) and the ratio of E_{cw} (E_{cw}/E_f) with engine load at 1500 rpm for different CRs. As shown in Fig. 5(a), the amount of E_{cw} increases notably with rising load, which is attributed to higher combustion temperatures resulting from greater fuel supply to the cylinder. The E_{cw} ratio was lower at minimum and maximum loads (0% and 100%), while it exhibited an increasing trend at medium loads (25–75%). This behavior is explained by lower combustion temperatures at low loads and shorter combustion durations at high loads. Conversely, although the absolute E_{cw} values increased with CR, the E_{cw} ratio generally showed a stronger tendency to rise with higher CRs. This is attributed to increased heat lost caused by elevated combustion temperatures and longer combustion durations at higher CRs.²⁴ It was determined that the E_{cw} ratio increased by 6.4–19.5% and 5.8–24.7% for CRs of 16 and 17, respectively, compared to a CR of 15 depending on engine load, while a reduction of 2.3% was also observed at a CR of 16.

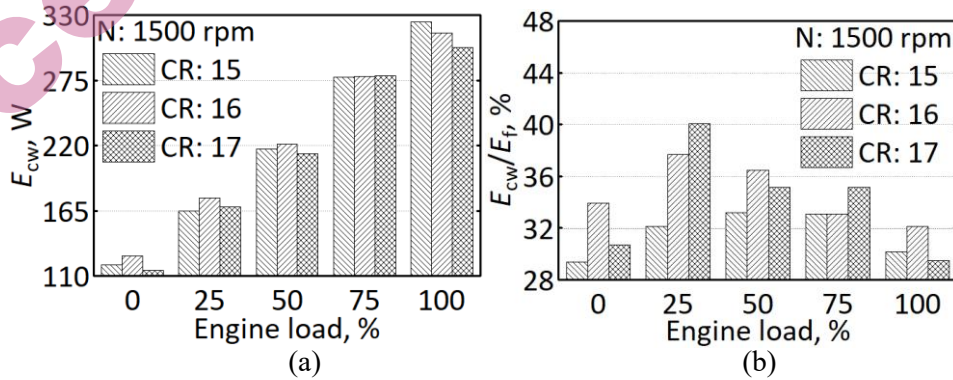


Fig 5. Variations in (a) energy lost with heat transfer and (b) ratio of energy lost by heat transfer

Figs. 6(a) and 6(b) show the variation in energy lost through exhaust (E_{exh}) and the ratio of E_{exh} (E_{exh}/E_f) with engine load at 1500 rpm for different CRs. As illustrated, both the amount and ratio of E_{exh} increase notably with rising load. This is attributed to the greater fuel supply at higher loads, which results in an increased in combustion energy release, elevated combustion temperatures, and extended combustion duration.²⁰ Conversely, the amount and the ratio of E_{exh} generally exhibited a slight upward trend with increasing CR, which is explained by higher in-cylinder pressures and temperatures at elevated CRs. These conditions lead to higher exhaust gas temperatures, and consequently greater energy carried away by the exhaust gases.²⁵ It was determined that the E_{exh} ratio increased by 4.4–17.1% and 13–21% for CRs of 16 and 17 respectively, compared to a CR of 15 depending on engine load, while reductions of 0.6–3.6% and 0.7% were also observed at the same CRs.

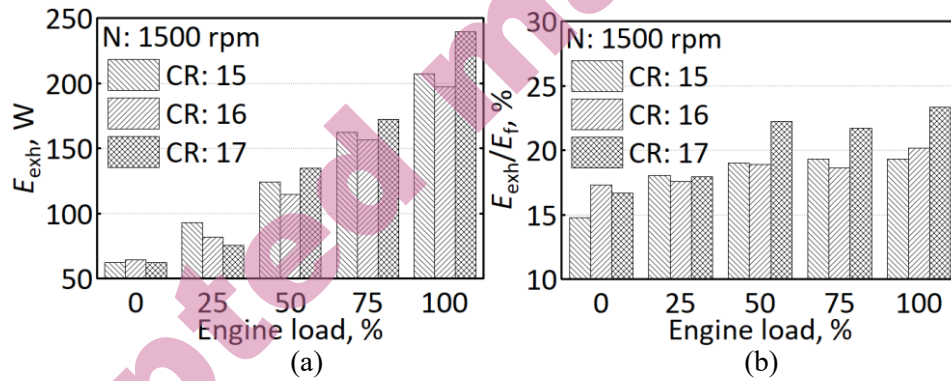


Fig 6. Variations in (a) energy lost with exhaust and (b) ratio of energy lost with exhaust

Figs. 7(a) and 7(b) illustrate the variations in other energy losses (E_{other}) and the ratio of E_{other} (E_{other}/E_f) with engine load at 1500 rpm for different CRs. In ICEs, most of other energy losses arise from combustion inefficiencies, friction losses, auxiliary power usage, and fundamental thermodynamic limitations.²⁶ As shown in Fig. 7(a), the amount of E_{other} initially decreases with rising load, then increased again, resulting lower values at medium loads (50–75%). This behavior is attributed to the fact that energy dissipated to auxiliary equipment remains relatively constant, even though the engine produces more work as load increases. Conversely, both the amount and ratio of E_{other} generally tend to decrease with higher CRs. This reduction is explained by the increase in E_w and the improvement in combustion efficiency at elevated CRs, which lead to lower losses.²⁷ It was determined that the E_{other} ratio decreased by 15.5–20.2% and 8.6–28% for CRs of 16 and 17, respectively, compared to a CR of 15 depending on engine load, while an increment of 5.4% was also observed at a CR of 16.

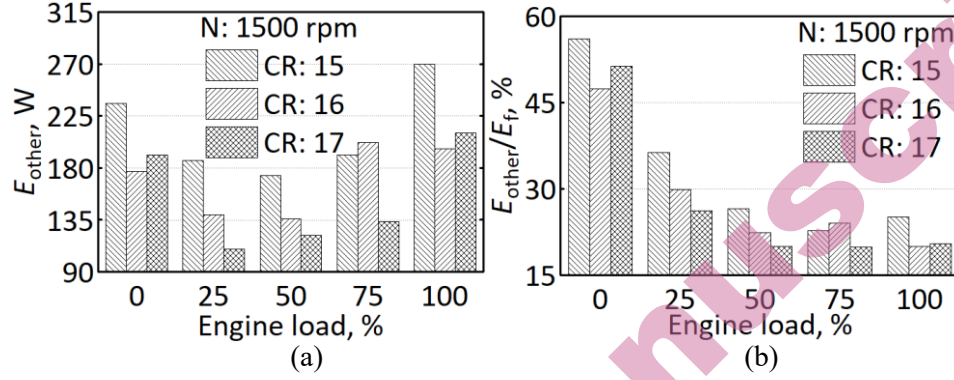


Fig 7. Variations in (a) other energy losses and (b) ratio of other energy losses

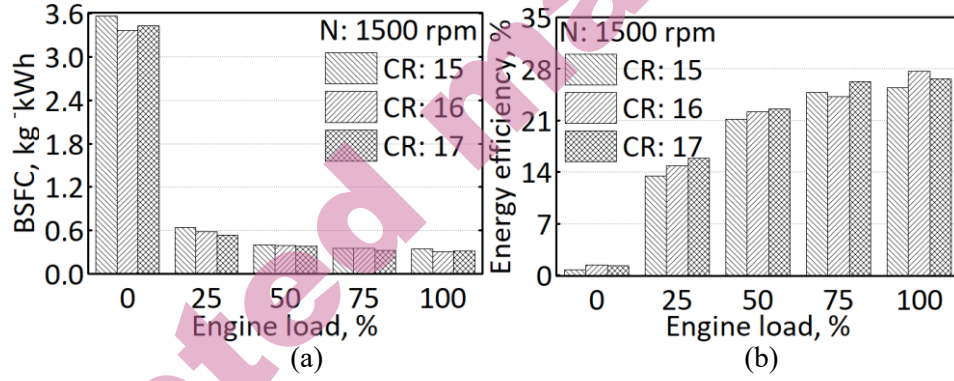


Fig 8. Variations in (a) BSFC and (b) energy efficiency

Figs. 8(a) and 8(b) show the variations in brake specific fuel consumption (*BSFC*) and energy (thermal) efficiency with engine load at 1500 rpm for different *CR*s. As shown in Fig. 8(a), *BSFC* decreases significantly with rising engine load, which is attributed to the increase in work (power) produced at higher loads. Conversely, *BSFC* generally tends to decrease with increasing *CR*, due to elevated cylinder pressure and temperature that promote a more homogeneous fuel–air mixture and enhance combustion efficiency.²³ *BSFC* was reduced by 2.5–9.4% and 5–15.6% for *CR*s of 16 and 17, respectively, compared to a *CR* of 15 depending on engine load. Furthermore, higher *CR*s were noted to improve volumetric efficiency and reduce *BSFC*, as shortened ignition delay enhances combustion efficiency.²¹ A higher *CR* compresses the air–fuel mixture further before ignition, leading to increased cylinder temperature and pressure, which results in greater energy conversion and higher power output. As shown in Fig. 8(b), energy efficiency increased both rising load and *CR*, attributed to improved combustion efficiency and consequently higher work output. It was observed that increasing *CR* reduces the fuel required to achieve the same power, while simultaneously

improving combustion efficiency and reducing losses, thereby enhancing energy efficiency.²⁸ It was determined that energy efficiency increased by 4.7–9.8% and 4.9–17.4% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load, while a reduction of 2.2% was also observed at a CR of 16.

Figs. 9(a) and 9(b) present the variations in exergy converted to work (Ex_w) and the ratio of Ex_w (Ex_w/Ex_f) with engine load at 1500 rpm for different CR s. As seen in Fig. 9(a), both the amount of Ex_w and its ratio increase significantly with rising engine load. This behavior is attributed to the greater fuel supply at higher loads, and the enhanced work output resulting from improved combustion efficiency. Conversely, although the absolute Ex_w values did not vary considerably with CR , the Ex_w ratio generally exhibited an increasing trend with higher CR s. This improvement is explained by the rise in combustion efficiency at elevated CR s, which enables a larger fraction of exergy input (Ex_f) to be converted into useful work. The Ex_w ratio was found to increase by 9.9–22.1% and 4.9–17.4% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load.

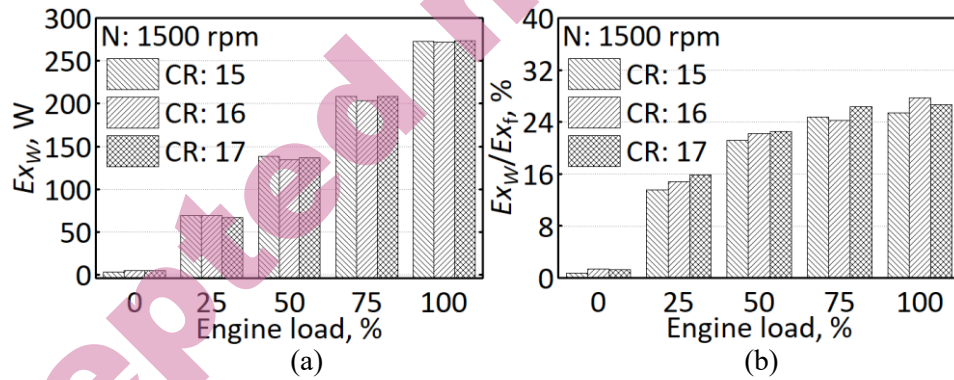


Fig 9. Variations in (a) exergy converted to work and (b) ratio of exergy converted to work

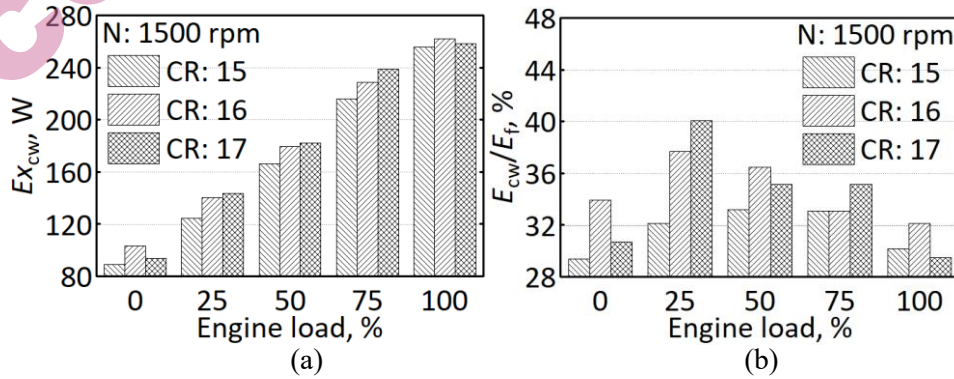


Fig 10. Variations in (a) exergy lost by heat transfer and (b) ratio of exergy loss by heat transfer

Figs. 10(a) and 10(b) illustrate the variations in exergy lost through heat transfer (Ex_{cw}) and the ratio of Ex_{cw} (Ex_{cw}/Ex_f) with engine load at 1500 rpm for different CRs. As shown in Fig. 10(a), both the amount and ratio of Ex_{cw} increased significantly with rising engine load. This behavior is attributed to higher combustion temperatures and changes in combustion duration as more fuel is supplied to the cylinder at increased loads. Conversely, although the amount of Ex_{cw} did not vary considerably with CR, the Ex_{cw} ratio generally exhibited a notable upward trend with increasing CR. This is explained by the rise in combustion temperature at higher CRs, which leads to greater heat losses.²⁹ The Ex_{cw} ratio was found to increase by 19.1–37.6% and 5.6–40.9 % for CRs of 16 and 17, respectively, compared to a CR of 15 depending on engine load.

Figs. 11(a) and 11(b) illustrate the variations in exergy lost through exhaust (Ex_{exh}) and the ratio of Ex_{exh} (Ex_{exh}/Ex_f) with engine load at 1500 rpm for different CRs. As shown in Fig. 11(a), both the amount and ratio of Ex_{exh} increased significantly with rising engine load. This is attributed to the greater fuel supply to the cylinder at higher loads, which results in elevated combustion temperatures and the expulsion of exhaust gases at higher pressures and temperatures. Conversely, although the amount of Ex_{exh} did not vary considerably with CR, the Ex_{exh} ratio generally exhibited a notable upward trend with increasing CR. This behavior is explained by the rise in combustion temperature at higher CRs, leading to exhaust gases being discharged at higher temperatures and pressures. It has been reported that improved combustion efficiency and elevated CRs contributes to increased combustion temperature and pressure, thereby causing higher Ex_{exh} . Moreover, it has been noted that the energy carried by exhaust gases can be recovered and converted into mechanical energy through a turbocharger system, thus enhancing engine performance.³⁰ It was determined that the Ex_{exh} ratio increased by 7.8–17.5% and 1.8–20.9% for CRs of 16 and 17, respectively, compared to a CR of 15 depending on engine load.

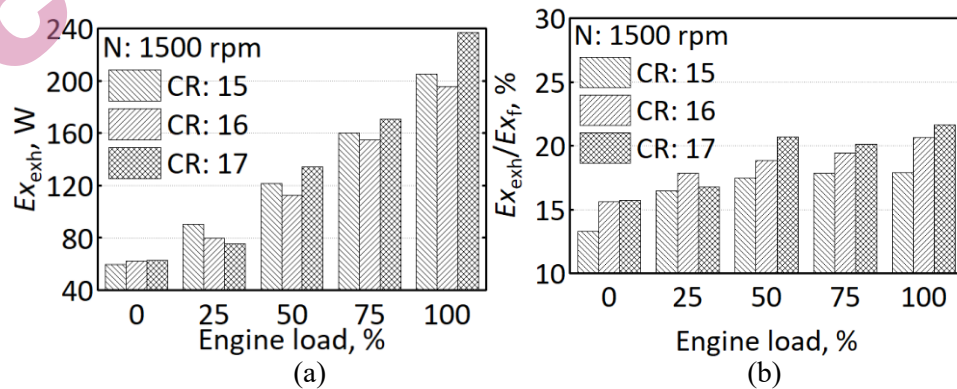


Fig 11. Variations in (a) exergy lost with exhaust and (b) ratio of exergy loss with exhaust

Figs. 12(a) and 12(b) illustrate the variations in exergy destruction (Ex_{dest}) and ratio of Ex_{dest} (Ex_{dest}/Ex_f) with engine load at 1500 rpm for different CRs. Ex_{dest} represents the losses arising from irreversible processes such as mixture formation, combustion, friction, and power dissipated by auxiliary equipment and it is recognized as an important parameter for assessing the thermodynamic performance of an engine. Combustion, in particular, is a highly irreversible process; although the total energy remains constant during the conversion of fuel chemical energy to heat, exergy decreases significantly during combustion.¹⁸ As shown in Fig. 12(a), Ex_{dest} initially decreases with increasing engine load, reaches lower values at medium loads (25–75%) and then rises again. This trend is attributed to the greater fuel consumption at higher loads, which leads to increased entropy production and consequently higher irreversibilities.¹⁹ Moreover, longer combustion durations resulting from extended fuel injection at elevated loads cause the formation of more residual gas, thereby increasing exergy losses.²⁰ The Ex_{dest} ratio, however, decreases notably with rising engine load, which is explained by higher combustion temperatures and improved combustion efficiency at increased engine loads. Ex_{dest} was found to decrease by 23.1–39.7% and 20.1–38.3% for CRs of 16 and 17, respectively, compared to a CR of 15 depending on engine load. Conversely, both the amount and ratio of Ex_{dest} generally declined with increasing CR. This reduction is attributed to higher combustion temperatures and efficiency at elevated CRs. Enhanced thermal efficiency at higher CRs enables a larger portion of fuel chemical exergy to be converted into work and thereby reducing exergy losses.³¹ It was found that the Ex_{dest} ratio decreased by 13.5–36.4% and 10.1–24.5% for CRs of 16 and 17, respectively, compared to a CR of 15 depending on engine load.

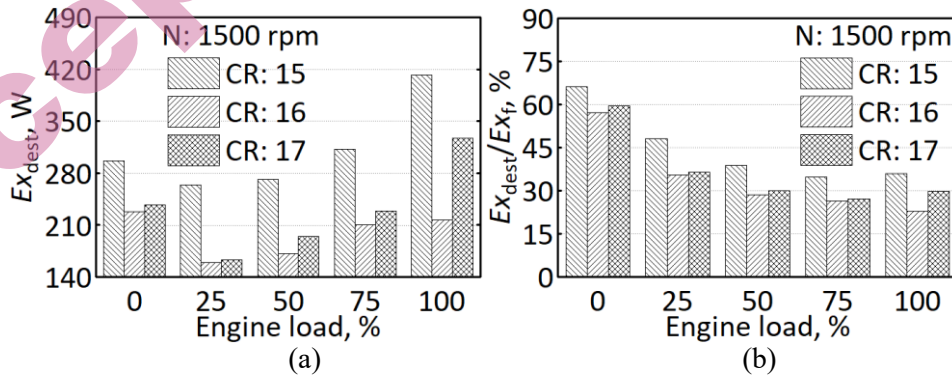


Fig 12. Variations in (a) exergy destruction and (b) ratio of exergy destruction

Figs. 13(a) and 13(b) present the variations in Ex_f and exergy efficiency with engine load at 1500 rpm for different CRs. Ex_f increases with rising engine load, while it tends to decrease slightly with higher CRs, as shown in Fig. 13(a). This

behavior is attributed to the greater amount of fuel supplied to the cylinder, and thus higher fuel energy at increased loads.³² Ex_f was found to decrease by 11.1–18.2% and 4.3–18.2% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load. This reduction is thought to result from the lower fuel energy associated with higher CR s. Conversely, exergy efficiency increased with both engine load and CR , which is explained by improvements in combustion efficiency under these conditions. It has been reported that high combustion instability and low combustion temperature reduce exergy efficiency at low loads, whereas rising combustion temperature enhances combustion efficiency and stability, thereby improving exergy efficiency as load increases. Additionally, the higher quantity and temperature of residual exhaust gases generated at elevated loads preheat the fresh charge, increasing combustion temperature and efficiency, and contributing to higher exergy efficiency.²⁰ Although higher CR s increase combustion temperature, leading to greater heat losses and thermal irreversibilities, exergy efficiency may still improve because the gains in combustion and thermal efficiency outweighed these losses.³³ Exergy efficiency was observed to increase by 9.9–22.1% and 4.9–17.4% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load. These variations highlight the dual effect of CR : although slight reductions in efficiency occur at highest CR , the overall trend demonstrates that increasing CR contribute to improved combustion efficiency and reduced irreversibilities, thereby enhancing exergy performance.

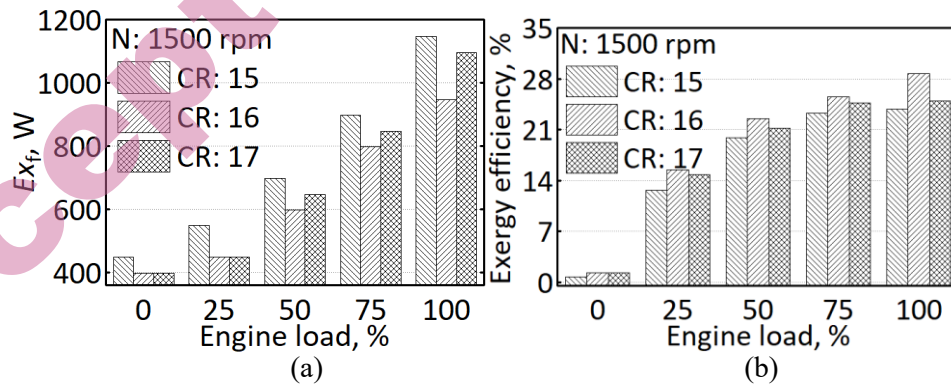


Fig 13. Variations in (a) fuel exergy and (b) exergy efficiency

Figs. 14(a) and 14(b) illustrate the variations in the exergy performance coefficient (EPC) and sustainability index (SI) with engine load at 1500 rpm for different CR s. Both EPC and SI increased with rising engine load and CR , which is attributed to improved combustion stability, resulting in higher work output and exergy efficiency. It has been reported that the greater amount of fuel and air supplied to the cylinder at higher loads enhances the energy potential in the

combustion chamber, thereby increasing EPC .¹⁸ Conversely, an SI value greater than one indicates more efficient energy utilization with minimal losses in ICEs. SI increases with higher power output while decreasing with rising exergy destruction. Exergy efficiency also positively influences SI , and a higher SI value makes the energy source or process under investigation more attractive.²² EPC was found to decrease by 45.3–110.9% and 26.7–99.9% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load. Similarly, SI increased by 0.6–6.9% and 0.5–2.6% for CR s of 16 and 17, respectively, compared to a CR of 15 depending on engine load. Based on these results, a CR of 16 appears to provide more favorable exergetic outcomes among the tested CR s. These findings indicate that although minor reductions occur at highest CR , the overall trend demonstrates that higher CR s enhance EPC and SI , reflecting improved combustion stability, reduced irreversibilities, and more suitable energy utilization.

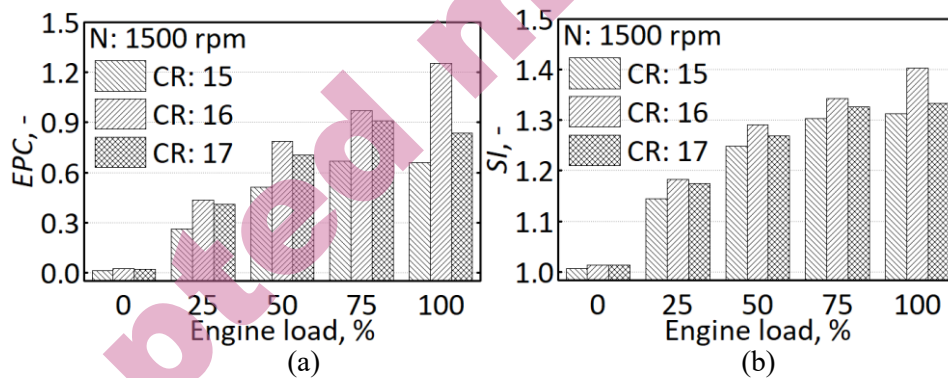


Fig 14. Variations in (a) exergy performance coefficient and (b) sustainability index

CONCLUSION

This study provides a comprehensive thermodynamic analysis of the effects of the compression ratio (CR) on the combustion, performance, energy, and exergy characteristics of a diesel engine. Based on the findings, the following conclusions can be drawn.

- It was found that increasing the CR shortened the ignition delay, initiated combustion earlier, and completed the combustion process in a shorter duration. Consequently, higher CR values resulted in increased cylinder pressure and temperature, leading to improvements in efficiency and reduced fuel consumption. However, elevated pressure and temperature levels may raise production costs, as the engine requires stronger and higher-quality components during manufacturing.
- It was observed that increasing engine load and CR enhanced the conversion of fuel energy into work, thereby improving thermal efficiency. On the other

hand, while energy losses due to heat and exhaust rose with higher engine load and CR , other forms of energy losses generally declined as CR increased.

- It was determined that increasing the engine load and CR enhanced the exergy converted into work and thereby improving exergy efficiency. On the other hand, while heat transfer and exergy losses through exhaust increased with higher CR s, exergy destruction decreased, and both the exergy performance and sustainability index showed notable improvements increased with rising CR .
- CR was identified as a key parameter influencing losses, efficiency, exergy, and sustainability. Among the tested CR s, a CR of 16 was found to deliver the most favorable particularly in terms of exergy performance. Specially, exergy efficiency, exergy performance and sustainability index increased by 9.9–22.1%, 45.3–110.9% and 0.6–6.9%, respectively, for a CR of 16 depending on engine load.

NOMENCLATURE

η_{ex}	exergy efficiency, %
BP	brake power, W
$BSFC$	brake specific fuel consumption, g ⁻¹ kWh
$BTE(\eta_e)$	brake thermal (energy) efficiency, %
$c_{p, cw}$	specific heat of cooling water, J ⁻¹ kgK
$c_{p, exh}$	specific heat of exhaust gases, J ⁻¹ kgK
CR	compression ratio, –
E_{cw}	energy loss with heat transfer, W
E_{exh}	energy loss with exhaust, W
E_f	fuel energy, W
E_{other}	other energy loss, W
E_w	energy converted to work, W
EPC	exergy performance coefficient, –
E_{xcw}	exergy loss with heat transfer, W
E_{xdest}	exergy destruction, W
E_{xexh}	exergy loss with exhaust, W
E_{xf}	fuel exergy, W
E_{xw}	exergy converted to work, W
FP	friction power, W
IP	indicated power, W
$ISFC$	indicated specific fuel consumption, g ⁻¹ kWh
$ITE(\eta_i)$	indicated thermal efficiency, %
CC	cycle coefficient, $CC=1$ and $CC=2$ for two and four stroke engines
LHV_f	lower heating value of fuel, J
m_{cw}	flow rate of cooling water, kg ⁻¹ s
m_f	flow rate of fuel, kg ⁻¹ s

N	engine speed, rpm
SI	sustainability index, –
T	temperature, °C or K
U	uncertainty, –
V_h	stroke volume, L or m ³
W	work, W
Z	number of cylinder, –

ИЗВОД

ДЕТАЉНА ТЕРМОДИНАМИЧКА АНАЛИЗА РАДА ДИЗЕЛ-МОТОРА ПРИ РАЗЛИЧИТИМ СТЕПЕНИМА КОМПРЕСИЈЕ

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У овом раду експериментално је испитан утицај степена компресије (CR) на карактеристике сагоревања, као и енергетске и ексергетске карактеристике дизел-мотора. У испитивањима је коришћен четворотактни, атмосферски, водом хлађени, једноцилиндрични дизел-мотор са директним убризгавањем и могућношћу промене степена компресије. Експерименти су спроведени при константној брзини обртања од 1500 min⁻¹, при различитим оптерећењима и степенима компресије од 15:1, 16:1 и 17:1. Софтвер у оквиру експерименталне поставке израчунава аритметичку средину резултата за десет радних циклуса како би се смањио утицај варијација између појединачних циклуса и повећала тачност резултата. На основу експериментално добијених података спроведена је детаљна термодинамичка анализа у циљу испитивања карактеристика сагоревања, као и енергетских и ексергетских карактеристика мотора при различитим степенима компресије. Резултати су показали да степен компресије има значајан утицај на наведене карактеристике. Са повећањем степена компресије повећавали су се параметри сагоревања, као што су удео сагореле масе, брзина ослобађања топлоте, притисак у цилиндру и температура гасова. Слично томе, са повећањем степена компресије уочена је и тенденција побољшања показатеља перформанси мотора, укључујући ефикасност, потрошњу горива, као и укупне енергетске и ексергетске перформансе. Од испитиваних степена компресије, степен компресије 16:1 показао је најповољније резултате са аспекта ексергетске анализе. Конкретно, при степену компресије 16:1, у зависности од оптерећења мотора, ексергетска ефикасност побољшана је за 9,9–22,1%, коефицијент ексергетских перформанси за 45,3–110,9%, а индекс одрживости за 0,6–6,9%.

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