Original Research Article: full paper

(2025), "EUREKA: Physics and Engineering" Number 4

IDENTIFYING THE REGULARITIES OF POWER CHANGE AT THE VEHICLE'S WHEELS WHEN ONE OR SEVERAL ENGINE CYLINDERS ARE DISABLED

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Abstract

The object of this study is the power system of internal combustion engines in wheeled vehicles operating under conditions of partial cylinder deactivation. The study addresses the problem of decreased wheel power due to mechanical (pumping) losses in non-firing cylinders, which limits the efficiency of cylinder deactivation strategies aimed at improving fuel economy.

A new method has been developed to determine the work and power output of the engine during acceleration with one or more cylinders deactivated. This method enables direct and accurate assessment of vehicle performance under real-world operating conditions.

Experimental tests were conducted on two vehicle types: a Daewoo Lanos 1.5i passenger car and a KrAZ-255B heavy-duty truck. In both cases, cylinder deactivation was implemented by cutting off fuel supply, and on the KrAZ-255B, additional drainage of residual fuel from high-pressure lines was used to eliminate injector resistance. The results revealed a notable reduction in wheel power when cylinders were disabled; however, eliminating mechanical losses in the deactivated cylinders significantly reduced this power drop.

In the Daewoo Lanos 1.5i, the power loss decreased by factors of 1.44 and 2.98 when one or two cylinders were deactivated. For the KrAZ-255B, reductions of 1.1 and 3.09 were recorded. These results are explained by the mechanical drag caused by the pumping action of non-firing cylinders. When this resistance is removed, power transmission becomes more efficient.

The distinctive feature of this study is the quantification of mechanical losses and their effect on vehicle performance under real conditions. The method can be practically applied to optimize powertrain efficiency, especially in heavy vehicles and systems operating under varying load conditions.

Keywords: engine cylinder deactivation, energy efficiency, mechanical losses, dynamics, fuel economy.

DOI: 10.21303/2461-4262.2025.003882

1. Introduction

Improving the fuel efficiency and emissions of internal combustion engines remains a priority under increasingly stringent environmental regulations. Cylinder deactivation (CDA) – the temporary shutting off of one or more engine cylinders under light-load conditions – has emerged as an effective measure to reduce fuel consumption and CO_2 emissions in both gasoline and diesel engines. For instance, [1] reports that CDA can decrease fuel use by roughly 8–20% in spark-ignition engines at partial load, while also curbing exhaust emissions. Automakers worldwide have implemented CDA in production engines to meet efficiency targets [2]. [3] demonstrated that even a partial deactivation (disabling only one out of four cylinders) can yield a notable improvement in brake thermal efficiency for a diesel engine, illustrating the broad potential of this technology beyond the conventional strategy of cutting out half the cylinders. This evidence underlines the relevance of CDA as a cost-effective approach to enhance vehicle fuel economy and comply with emissions standards.

The development and application of CDA technology have been advanced by researchers and manufacturers across the globe, particularly in the USA, Germany, and Japan. In the United States, in research [4] quantified significant fuel economy benefits of CDA in V8 engines, which informed its adoption in American passenger vehicles. Recent U.S. studies have extended CDA to diesel engines [5] implemented CDA via flexible valvetrain control in a heavy-duty diesel, achieving a 40% fuel consumption reduction during idle while maintaining aftertreatment temperatures. German engineers have likewise contributed to CDA development. [2] analyzed the "potentials and constraints" of cylinder deactivation in automotive engines, confirming substantial fuel savings (on the order of 15%) at low loads but cautioning about increased vibration and noise. Meanwhile, in Germany proposed an innovative CDA concept for commercial diesels, deactivating a single cylinder ("3/4-cylinder" mode) instead of the typical half-engine cut-off, and achieved efficiency gains without the harsher torque fluctuations associated with deeper deactivation [3]. Japanese manufacturers have also successfully implemented CDA systems in production engines. For example, Honda's Variable Cylinder Management system on V6 gasoline engines and Mazda's recent SKYACTIV engines employ cylinder deactivation to improve cruising fuel economy, contributing to international experience with this technology [6]. This global progress demonstrates that CDA has been recognized and utilized as a viable solution for improving engine efficiency across different vehicle classes and regions.

A substantial body of research has investigated the performance effects and optimization of CDA under various operating conditions. Numerous studies confirm that CDA yields significant fuel consumption reductions in light-load operation. For instance, in [7] demonstrated 9–12% lower fuel consumption in a heavy-duty compressed-natural-gas engine with half of its cylinders deactivated, with minimal impact on output power. Similarly, found that switching off two out of four cylinders in a spark-ignition engine improved fuel economy by approximately 6–13% at low loads [8]. These efficiency gains are primarily attributed to reduced pumping losses and improved thermodynamic efficiency when the firing cylinders operate at higher load per cylinder [1, 8]. Crucially, recent research shows that the fuel-saving benefits of CDA can extend into higher load ranges than previously expected. L. and Gosala [9] observed that even at moderately high loads (up to ~8 bar BMEP) and high engine speeds, a heavy-duty off-road diesel engine still realized measurable fuel savings from CDA, broadening the known envelope of CDA effectiveness.

Moreover, CDA has been applied not only to light-duty engines but also in large engines. [7, 9] verified the feasibility of CDA in heavy-duty engines (natural gas and diesel, respectively), indicating that the technique is effective across engine sizes and fuels. Despite these clear advantages, the literature also highlights several challenges and trade-offs associated with cylinder deactivation. One well-documented concern is the increase in engine vibration and cabin noise when cylinders are turned off. [10] showed that deactivating half the cylinders in a V6 engine can introduce undesirable torsional vibrations, which may degrade vehicle refinement. This has driven research into mitigation strategies. Authors of the article [10] suggested active vibration control measures that allowed a broader range of CDA operation without compromising driver comfort.

Another challenge is the uneven torque delivery and transient response when cylinders are intermittently deactivated and reactivated. [11] investigated a downsized turbocharged engine and observed transient spikes in emissions and cyclic variations during the transition between deactivated and all-cylinder modes. Paper [12] decided this issue by using a 48 V mild-hybrid system to smooth out torque deviations during CDA operation; their results showed that coordinated motor assistance can effectively compensate for the sudden torque drop when cylinders are turned off, thereby maintaining drivability.

Furthermore, researchers have explored advanced valvetrain strategies to minimize the pumping losses in deactivated cylinders. Completely shutting the intake and exhaust valves of an inactive cylinder (sometimes termed "dynamic skip fire") is crucial to maximize fuel savings and avoid pumping air through the disabled cylinders [13]. This concept has been validated experimentally in [7], who found improved efficiency when deactivated cylinders were isolated from gas exchange.

In summary, the recent literature suggests that while CDA can substantially improve efficiency, careful engine control and hardware adaptations (e.g. active mounts, variable valve actuation, or hybrid assist) are required to mitigate its side effects.

Another active area of research is the impact of CDA on engine internal processes and emissions beyond fuel economy. [14] examined the effect of cylinder deactivation on the friction and blow-by of piston rings, noting that prolonged deactivation of cylinders alters ring lubrication conditions which could affect wear and oil consumption. Likewise, in [15], an engine study was conducted running with CDA and changes were reported in tribological parameters (such as cylinder liner friction) as well as a reduction in exhaust emissions of certain pollutants, owing to lower fuel throughput.

In diesel engines, an important benefit of CDA is the ability to elevate exhaust gas temperatures at light load for aftertreatment systems. [5] demonstrated that selectively deactivating cylinders at idle can keep the exhaust hotter (by increasing load on the remaining cylinders), thus improving the efficiency of catalytic converters and particulate filters without consuming extra fuel. This finding is critical for meeting stringent emissions standards in heavy-duty applications, where maintaining aftertreatment activity during low-load operation is challenging.

Overall, the literature review indicates that researchers have attained a comprehensive understanding of the thermodynamic benefits of CDA and have proposed various solutions to its drawbacks. However, certain knowledge gaps remain, particularly in translating engine-level findings to the whole-vehicle performance in real driving scenarios.

Despite extensive prior research, two key deficiencies can be noted in the existing studies on cylinder deactivation. First, most experimental investigations have been conducted on engine test beds or through simulation, whereas the effect of CDA on the actual wheel output power of a vehicle under dynamic driving conditions is not well documented. In other words, there is a lack of data on how shutting off cylinders influences the power delivered to the drive wheels during vehicle acceleration or hill climbing. This gap is critical because vehicle performance as perceived by the driver depends on wheel power, and it may differ from engine bench results due to drivetrain losses and vehicle inertia.

Second, many studies implicitly assume ideal deactivation (often closing the valves of the inactive cylinders to eliminate pumping losses) [7, 13], but in practice a simpler approach of just cutting fuel to the cylinders is frequently used, which means the "deactivated" cylinders still pump air and impose drag on the engine. The magnitude of these pumping losses and their impact on overall performance have not been quantified sufficiently in published research. It remains unclear how much power loss is incurred due to pumping in dead cylinders and how much could be regained by alleviating those losses (for example, by venting or disabling the valve motion in those cylinders).

There is therefore a need for studies that measure vehicle performance with partial cylinder deactivation both with and without pumping losses, to directly assess the benefit of eliminating those losses. Moreover, previous works have not fully examined differences between vehicle types – for instance, how CDA effects might differ in a light passenger car versus a heavy truck engine. Addressing these deficiencies would provide more practical guidance for implementing CDA in various vehicles while maintaining acceptable performance.

The objective of this study is to ensure the reliability and functional stability of wheeled vehicles under conditions of partial engine cylinder deactivation. The proposed hypothesis assumes that it is possible to improve the energy efficiency of vehicles by:

1. Conducting theoretical research on determining engine power and work during vehicle acceleration.

2. Carrying out experimental research to determine the current effective engine power during operation.

3. Investigating methods to enhance vehicle energy efficiency by disabling engine cylinders while minimizing mechanical losses.

The object of research is the dynamic processes occurring in the motor-transmission units of wheeled vehicles during operation with partial engine cylinder deactivation.

This paper contributes to the body of knowledge by providing both theoretical development and practical experimental validation for methods aimed at increasing operational efficiency through cylinder deactivation strategies, considering both passenger cars and heavy trucks.

2. Materials and methods

2. 1. Research design

The experimental study aimed to determine the power developed by the vehicle engine during acceleration under conditions of partial cylinder deactivation, with and without mechanical (pumping) losses. Two types of wheeled vehicles were tested: the Daewoo Lanos 1.5i passenger car and the KrAZ-255B truck.

Experiments were conducted on road sections with longitudinal and transverse slopes not exceeding 1.5%, with smooth, dry asphalt surfaces free from potholes and irregularities. All tests complied with traffic regulations and were performed without interrupting traffic flow.

2. 2. Experimental procedure

The experimental procedure included:

1. Warm-up of the engine and transmission units before tests.

2. Acceleration cycles with full working cylinders and then with one or several cylinders disconnected.

3. Two-cylinder disconnection schemes:

– disconnection by fuel cut-off only (with mechanical losses);

– disconnection with mechanical loss elimination (using spark plug removal or injector removal).

Each test cycle involved four experimental runs: two in the forward and two in the reverse direction to eliminate road slope influence.

The Daewoo Lanos 1.5i was tested in second gear, initially accelerating from 7 km/h up to a crankshaft speed of 5600 min^{-1}. The KrAZ-255B was tested in second gear as well, starting from 3 km/h up to a crankshaft speed of 2100 min^{-1}.

2. 3. Parameters and equipment

The following parameters were recorded:

– number of active engine cylinders $\left(i''_{cyl}\right)$;

– gear engaged;

– vehicle mass m_a;

– vehicle speed V_{aut};
– vehicle longitudinal acceleration $\dot{V}_a$;
– time of motion t.

Equipment used:
– GPS/GLONASS receiver Transystem GM-3N for vehicle speed measurement;
– three-coordinate linear acceleration sensors;
– on-board data acquisition system for real-time parameter recording and storage.

The vehicle speed was determined by averaging three methods:
– speedometer readings;
– GPS/GLONASS receiver data;
– integration of the longitudinal acceleration.

To improve measurement accuracy, two longitudinal acceleration sensors were used simultaneously.

2. 4. Calculation methods

The power at the driving wheels during acceleration was calculated.

The coefficient of performance (COP) of the wheeled machine was determined based on the recorded data, depending on the power filling factor.

Data processing included graph plotting, regression analysis, and comparative evaluation of the effects of cylinder disconnection on vehicle performance.

3. Results and discussion

3. 1. Determination of engine power and work during acceleration

The movement of the vehicle on a measured area can be considered difficult. The movement that corresponds to the movement at the average speed on the measuring area should be considered as transferable. The parameter of relative motion should be considered the fluctuation of the car's linear speed relative to its average speed on the measuring section.

To implement the given task, let's use a complex movement model. The transfer speed is determined on a measuring section with a length equal to 1 km or 1000 m. Knowing the time T_M of passing the measuring section, it is possible to determine the portable linear speed of the car

$$V_{port} = \overline{V}_{aut} = S_M / T_M . \tag{1}$$

The average speed of the car on the measured section of the road will be the speed of the portable movement. In portable motion, the linear velocity is constant, which means the acceleration is zero. The constant of the traction force on the wheels, creates a portable movement and will be equal to

$$P_{wh\ port} = m_{\mathrm{a}} \cdot g \cdot f + \frac{C_x}{2} \cdot \rho \cdot F \cdot \overline{V}^2_{port} = m_a \cdot g \cdot f + \frac{C_x}{2} \cdot \rho \cdot F \cdot \frac{S_M^2}{T_M^2}, \tag{2}$$

where m_a – the mass of the car; g – free fall acceleration, $g = 9.81$ m/s^2; f – coefficient of rolling resistance; C_x – coefficient of frontal aerodynamic resistance; r – air density; F – frontal area of the car body.

Engine power spent on portable motion

$$N_{wh\ port} = P_{wh\ port} \cdot V_{port} = \frac{S_M}{T_M} \cdot \left(m_a \cdot g \cdot f + \frac{C_x}{2} \cdot \rho \cdot F \cdot \frac{S_M^2}{T_M^2} \right). \tag{3}$$

Work spent on portable movement on a measured section of the path

$$A_{wh\ port} = P_{wh\ port} \cdot S_M = S_M \cdot \left(m_{\mathrm{a}} \cdot g \cdot f + \frac{C_x}{2} \cdot \rho \cdot F \cdot \frac{S_M^2}{T_M^2} \right). \tag{4}$$

Engine power consumed by friction in the transmission

$$N_{fr} = P_{fr}^{giv} \cdot V_{port}, \tag{5}$$

where is P_{fr}^{giv} – the conventional frictional force in the transmission brought to the driving wheels.

Work caused by friction

$$A_{wh\,port} = P_{fr}^{giv} \cdot S_M. \tag{6}$$

The equation of vehicle dynamics [16–19]

$$m_a \cdot \dot{V}_a = P_{wh} - m_a \cdot g \cdot f - m_a \cdot g \cdot \sin \pm - \frac{C_x}{2} \cdot \rho \cdot F \cdot \left(V_{port} + \Delta V\right)^2, \tag{7}$$

where α – the angle of the final slope of the road; V_a – linear acceleration of a wheeled machine; ΔV – relative speed of movement

$$\Delta V = V_{0ut} - V_{port}. \tag{8}$$

From equation (7) it is possible to determine the total traction force on the drive wheels

$$P_{wh} = m_a \cdot \dot{V}_a + m_a \cdot g \cdot f + m_a \cdot g \cdot \sin \pm + \frac{C_x}{2} \cdot \rho \cdot F \cdot V_{port}^2 + $$

$$+ \frac{C_x}{2} \cdot \rho \cdot F \cdot \left(2\Delta V \cdot V_{port} + \Delta V^2\right). \tag{9}$$

The speed of relative movement

$$\Delta V = \Delta V_{\max} \cdot \sin\left(\Omega_v \cdot t\right), \tag{10}$$

where $\Delta V_{\max}$ – amplitude of relative velocity fluctuations; W_n – circular frequency of fluctuations of the relative speed of the car [17], $\Omega_v = 2\pi / T_v$; T_n – period of relative velocity fluctuations ΔV.

Coefficient of non-uniformity of the vehicle's linear speed [17]

$$\delta_v = \frac{V_{aut\max} - V_{aut\min}}{V_{port}} = \frac{2\Delta V_{\max}}{V_{port}}. \tag{11}$$

From equation (11) it is possible to determine

$$\Delta V_{\max} = \frac{\delta_v}{2} \cdot V_{port}. \tag{12}$$

Linear acceleration of the vehicle [16]

$$\dot{V}_a = d\left(\Delta V\right)dt = \Delta V_{\max} \cdot \Omega_v \cdot \cos\left(\Omega_v \cdot t\right) = \frac{\delta_v}{2} \cdot V_{port} \cdot \Omega_v \cdot \cos\left(\Omega_v \cdot t\right). \tag{13}$$

From equation (9), it is possible to determine the component of the traction force, which creates relative motion [18, 20]

$$P_{wh rel} = P_{wh} - P_{wh\,port} = m_a \cdot \dot{V}_a + m_a \cdot g \cdot \sin\alpha + \frac{C_x}{2} \cdot \rho \cdot F \cdot \left(2\Delta V \cdot V_{port} + \Delta V^2\right). \tag{14}$$

Equation (14) taking into account ratios (10), (12), (13) will take the following form

$$P_{whrel} = m_a \cdot \left[g \cdot \sin \pm + 0.5 \cdot \delta_v \cdot V_{port} \cdot \Omega_v \cdot \cos\left(\Omega_v \cdot t\right) \right] + \\ + \frac{C_x}{2} \cdot \rho \cdot F \cdot V_{port}^2 \left[\delta_v \cdot \sin\left(\Omega_v \cdot t\right) + 0.25 \cdot \delta_v^2 \cdot \sin^2\left(\Omega_v \cdot t\right) \right]. \quad (15)$$

Power on the wheels, which implements relative motion [21]

$$N_{whrel} = \int P_{whrel} \mathrm{d}\left(\Delta V\right). \quad (16)$$

Differentiating equation (10), it is possible to determine [21, 22]

$$d\left(\Delta V\right) = \Delta V_{\max} \cdot \Omega_v \cdot \cos\left(\Omega_v \cdot t\right) dt = \frac{\delta_v}{2} \cdot V_{port} \cdot \Omega_v \cdot \cos\left(\Omega_v \cdot t\right) dt. \quad (17)$$

After substituting expressions (15) and (17) under the sign of the integral in equation (16) and integrating, it is possible to obtain [16, 18]

$$N_{wh\,rel} = 0.5 \cdot \delta_v \cdot V_{port} \left\{ \begin{array}{c} m_a \left[\begin{array}{l} g \cdot \sin \pm \cdot \sin\left(\Omega_v \cdot t\right) + 0.25 \cdot \delta_v \cdot V_{port} \cdot \Omega_v \cdot \cos\left(\Omega_v \cdot t\right) + \\ +0.5 \cdot \sin\left(2 \cdot \Omega v \cdot t\right) \end{array} \right] + \\ +0.25 \cdot C_x \cdot F \cdot V_{port}^2 \cdot \delta_v \cdot \sin^2\left(\Omega_v \cdot t\right) \left[1 + \frac{\delta_v^2}{6} \cdot \sin\left(\Omega_v \cdot t\right) \right] \end{array} \right\}. \quad (18)$$

The work of the traction component during time T_M, performed in relative motion [16, 21]

$$A_{wh\,rel} = \int_0^{T_M} N_{wh\,rel} \cdot \mathrm{d}t. \quad (19)$$

After substituting (18), (19) and integrating, it is possible to obtain [16, 18]

$$A_{wh\,rel} = 0.5 \cdot \delta_v \cdot V_{port} \left\{ \begin{array}{c} 0.5 \cdot \delta_v \cdot V_{port} \left[\frac{T_M^2 \cdot \Omega_v^2}{2} + 0.25 + 0.25 \cdot \cos\left(2 \cdot \Omega_v \cdot T_M\right) \right] + \\ +0.25 \cdot C_x \cdot F \cdot V_{port}^2 \cdot \delta_v \left[\frac{T_M \cdot \Omega_v}{2} - 0.25 \cdot \sin\left(\Omega_v \cdot T_M\right) \right] + \\ + \frac{\delta_v^2}{6} \left(\frac{2}{3} + \frac{\cos^3\left(\Omega_v \cdot T_M\right)}{3} - \cos\left(\Omega_v \cdot T_M\right) \right) \end{array} \right\}. \quad (20)$$

Total work of traction forces [7, 16, 21]

$$A_{wh} = A_{wh\,port} + A_{wh\,rel} + A_{fr}. \quad (21)$$

Taking into account ratios (4), (20) and (6), expression (21) will take the form [4, 16, 21]

$$A_{wh} = S_M \left(m_a \cdot g \cdot f + \frac{C_x}{2} \cdot \rho \cdot F \cdot \frac{S_M^2}{T_M^2} \right) + P_{fr}^{giv} \cdot S_M + \\ + \frac{\delta_v}{2} \cdot V_{port} \left\{ \begin{array}{l} \frac{\delta_v}{4} \cdot V_{port} \left[\frac{T_M^2 \cdot \Omega_v^2}{2} + \frac{1}{4}\left(1 - \cos\left(2\Omega_v \cdot T_M\right)\right) \right] + \frac{\delta_v}{4} \cdot C_x \cdot F \cdot V_{port}^2 \times \\ \times \left[\frac{T_M \cdot \Omega_v}{2} - \frac{1}{4} \sin\left(2 \cdot \Omega_v \cdot T_M\right) + \frac{\delta_v^2}{6} \left(\frac{2}{3} + \frac{\cos^3\left(\Omega_v \cdot T_M\right)}{3} - \cos\left(\Omega_v \cdot T_M\right) \right) \right] \end{array} \right\}. \quad (22)$$

The performance indicators of the vehicle are not related to the work performed by the engine and ambiguously characterize the wear processes in their nodes, since they are indirect indicators. The work done by the engine is an objective indicator of the vehicle's performance.

The method of determining the work performed and the power developed by the engine, with a disconnected part of the cylinders during the acceleration of the wheeled machine, allows, in combination with the appropriate measuring complex, to evaluate the work done directly during the operation of the wheeled machine.

An analytical model was developed to estimate the effective engine work during vehicle acceleration, considering rolling resistance, aerodynamic drag, transmission losses, and the dynamic effects of relative speed fluctuations.

3. 2. Experimental studies on determining current effective engine power in operating conditions

When one or more cylinders are turned off by stopping the fuel supply, mechanical (pumping) losses associated with air compression during the compression stroke are observed. These losses can be eliminated by installing an additional bypass valve with a suitable control system on the shut-off cylinder. The purpose of experimental research is to determine the effect of the number of disconnected cylinders on the change in power on the driving wheels of the machine when moving in different gears and to estimate the amount of mechanical (pumping) losses when one or more cylinders of the wheeled machine engine are disconnected. Experimental studies are carried out separately for trucks and passenger wheeled vehicles.

Conditions for conducting road tests. Road experimental studies were carried out on a section of the road, the longitudinal and transverse slopes of which did not exceed 1.5%, which had a smooth dry asphalt concrete surface without potholes and irregularities. The length of the experimental section of the road in the first case was 700 m, and in the second – 500 m. The experimental studies were conducted in compliance with all traffic rules and did not require the blocking of traffic for other vehicles. Parameters recorded during the experimental study:

– the number of working cylinders of wheeled machine engine i''_{cyl};
– the gear on which wheeled machine moves;
– mass of the wheeled vehicle m_a;
– the speed of the wheeled vehicle V_{aut};
– longitudinal acceleration of the wheeled vehicle $\dot{V}_a$;
– time of movement t.

The power on the drive wheels of the machine, which is spent on acceleration during experimental studies, was calculated according to the following formula [8]

$$N_{wh} = N_{wh} \cdot \eta_{tr} = m_a \cdot V_{aut} \cdot \dot{V}_a. \qquad (23)$$

Measuring equipment [6] was used to register the above-mentioned parameters.

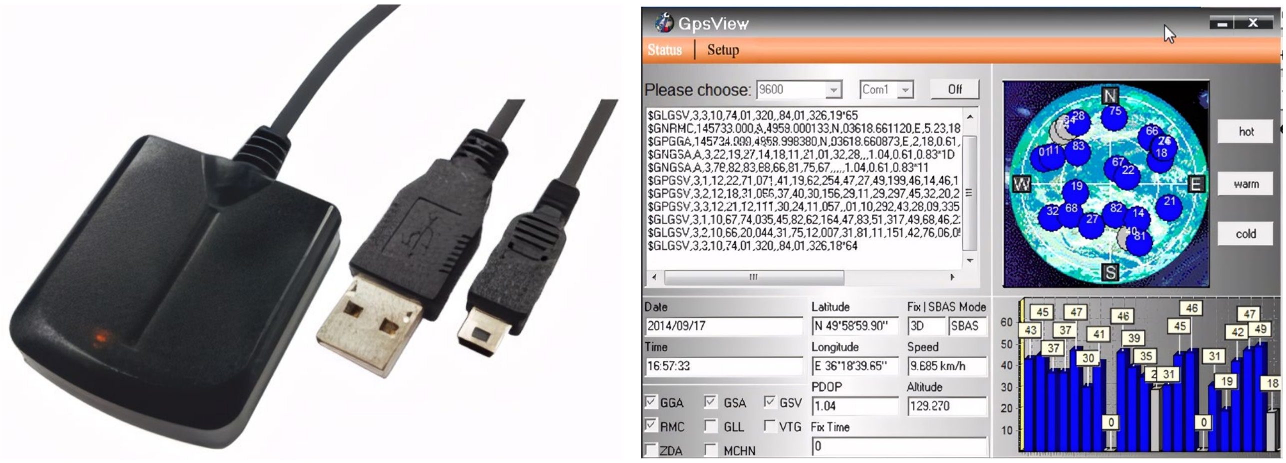
The speed of the wheeled vehicle was determined simultaneously in several ways:

– according to the speedometer readings;
– according to the indications of the GPS/GLONASS receiver Transystem GM-3N (**Fig. 1**).

At each moment of time during the experimental study, the linear speed of the wheeled vehicle was determined as the average of these three methods.

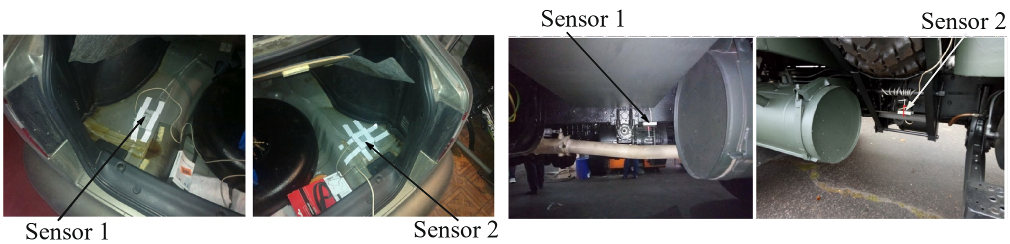
In order to increase the accuracy of the measurements due to the averaging of the obtained values, data on the longitudinal acceleration of the wheeled vehicle was collected simultaneously from two sensors (**Fig. 2**).

During the experimental research, individual cylinders of the wheeled machine engine were shut down. On the Daewoo Lanos 1.5i passenger car, such disconnection was carried out by stopping the fuel supply to the cylinder. To do this, the power supply (**Fig. 3**) of the fuel injector was turned off, which led to the termination of its operation and the blocking of the fuel supply to the cylinder. With this disconnection, significant mechanical losses from air compression in the combustion chamber of the disconnected cylinder during the compression stroke. To eliminate these mechanical losses, the spark plug was turned out of the cylinder that was being disconnected. To estimate the magnitude of these mechanical losses in the disconnected cylinder, experimental studies were conducted when the cylinders were disconnected according to both the first and second options. Disconnecting the cylinders was carried out in accordance with the sequence of their operation through one.



a *b*

Fig. 1. Registration of the longitudinal speed of the wheeled vehicle according to the readings of the GPS/GLONASS receiver: *a* – GPS/GLONASS receiver Transystem GM-3N; *b* – the window of the program that registers the readings of the GPS/GLONASS receiver – by integrating the linear longitudinal acceleration of the wheeled machine



a *b*

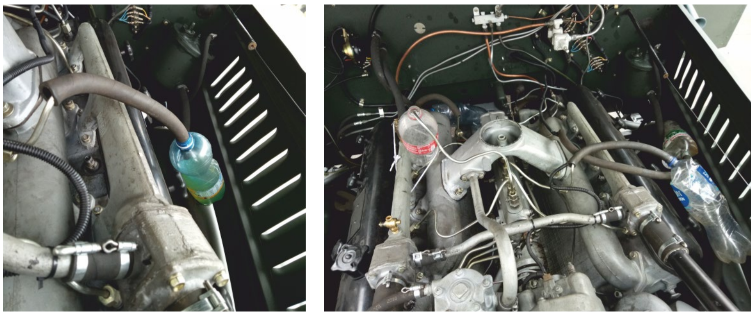
Fig. 2. Installation of three-coordinate sensors on wheeled vehicles during road experimental research: *a* – sensor installation locations on the Daewoo Lanos 1.5i; *b* – sensor installation locations on the KrAZ-255B

Fig. 3. Disconnecting the fuel supply to the engine cylinder of Daewoo Lanos 1.5i

Factors that vary during the experiment are the number of working cylinders of the wheeled machine engine and the type of their shutdown scheme. Thus, experimental studies are two-factorial.

The output parameter of the experimental study is the power on the driving wheels of the machine. At the same time, the factors are not correlated with each other and are compatible.

On the KrAZ-255B truck, the cylinders were turned off by stopping the fuel supply to the fuel injector and draining the fuel into the container from the connected high-pressure fuel pipe (**Fig. 4, *a*, *b***). To eliminate mechanical losses, the fuel injector was removed from the disconnected cylinder. Experimental studies were also carried out when the cylinders were turned off, both according to the first and second options. Disabling the cylinders was carried out in accordance with the sequence of their operation in such a way as to ensure more even operation of the engine.



a *b*

Fig. 4. Disconnection of fuel supply to the engine cylinder of the KrAZ-255B: *a* – disconnection of two cylinders; *b* – disconnection of four cylinders

The sequence of actions in the process of experimental research. Before starting the experimental research, the engine and transmission units of the wheeled machines were warmed up.

Each of the stages of the experimental study consisted of cycles. The first cycle represented acceleration of the wheeled machine with the maximum possible intensity (with full pressure on the gas pedal) to revolutions corresponding to the maximum power of the engine with all working cylinders. After that, the wheeled vehicle stopped, and in order to increase the reliability of the obtained data (to compensate for the error due to the angles of the road slope), the drive was repeated in the opposite direction of movement. Then the next cycles started with one or more cylinders being cut off. The disconnection occurred according to two schemes, as shown above (the first scheme assumes the presence of mechanical (pumping) losses in the disconnected cylinders, the second scheme assumes their absence). Each such cycle consisted of four runs (experiments). During the first two races (in the forward and reverse direction), the wheeled vehicle accelerated with one or more cylinders disconnected according to the first scheme of their disconnection. During the third and fourth runs (in the forward and reverse direction), the wheeled car accelerated with one or more cylinders disconnected according to the second scheme of their disconnection.

During experimental studies of the Daewoo Lanos 1.5i passenger car, acceleration in all cycles was carried out in second gear and with four, three and two working cylinders. Thus, the number of cycles consists three units. At the same time, in the second cycle, in accordance with the sequence of operation of the cylinders, the 1st cylinder was turned off, and in the third cycle, the 1st and 4th cylinders. The initial acceleration speed of the Daewoo Lanos 1.5i in second gear was 7 km/h. The final speed during acceleration corresponded to the engine crankshaft rotation frequency $n = 5600$ min^{-1} in the second gear. The time of one run (experiment) during acceleration and subsequent braking was up to 25 s.

At the second stage of experimental research (testing the KrAZ-255B), acceleration was also carried out in second gear and with eight, six and four working cylinders. Thus, the number of cycles also consists three units. At the same time, in the second cycle, in accordance with the sequence of operation of the cylinders, the 1st and 6th cylinders were turned off, and in the third cycle, the 1st, 4th, 6th and 7th cylinders. The initial speed during acceleration of the KrAZ-255B in second gear was 3 km/h. The final speed during acceleration corresponded to the engine crankshaft rotation frequency $n = 2100$ min^{-1} in the second gear. The time of one run (experiment) during acceleration and subsequent braking was up to 60 s.

The procedure and methods of recording the test results is the continuous process of recording the readings of the linear acceleration sensors and the GPS/GLONASS sensor, which is carried out with the help of a special program that allows to simultaneously register data and save them on the computer hard disk, as well as visualize the test process (**Fig. 1, *b***) [6].

Processing of experimental data. According to the results of the conducted experimental research, data were obtained, which were used to construct the graphs of changes in longitudinal acceleration and speed of the Daewoo Lanos 1.5i car over time during intensive acceleration in the second gear, which are shown in **Fig. 5**.

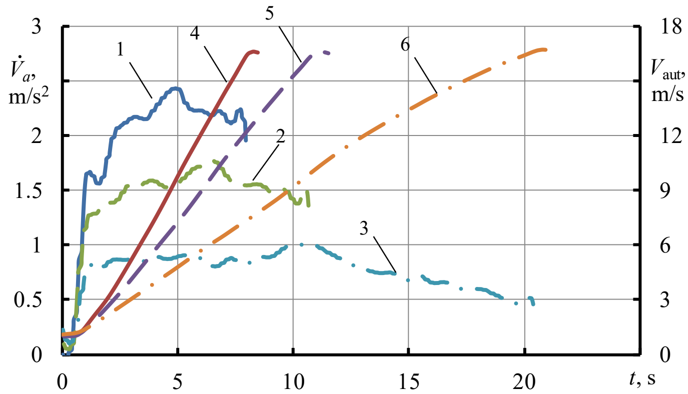


Fig. 5. Graphs of the dependence of linear speeds and acceleration on time during intensive acceleration in the second gear of Daewoo Lanos 1.5i car with different number of working cylinders (absence of mechanical (pumping) losses in disconnected cylinders): 1–3 – longitudinal linear acceleration; 4–6 – longitudinal linear speed; ——— – with all cylinders working; – – – – – with one cylinder disconnected; –·–·–– – with two cylinders disconnected

To further determine the effect of the number of disconnected engine cylinders and the effect of the presence or absence of mechanical (pumping) losses in these cylinders on the change in acceleration of the Daewoo Lanos 1.5i during intensive acceleration in second gear, graphs of the change in longitudinal acceleration over time were constructed, shown in **Fig. 6**.

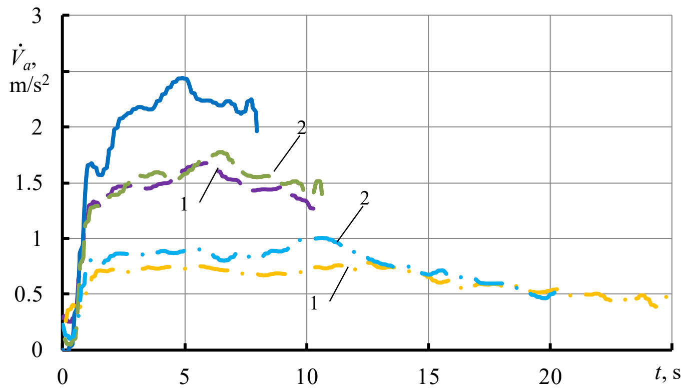


Fig. 6. Graphs of the dependence of linear longitudinal acceleration on time during intensive acceleration in the second gear of the Daewoo Lanos 1.5i with different number of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – – with one cylinder disconnected; –·–·–– – with two cylinders disconnected

Graphs of the change in the longitudinal acceleration of the specified car from its speed (during intense acceleration in second gear) with different number of working engine cylinders and the presence or absence of mechanical (pumping) losses in these cylinders are also plotted, which are shown in **Fig. 7**.

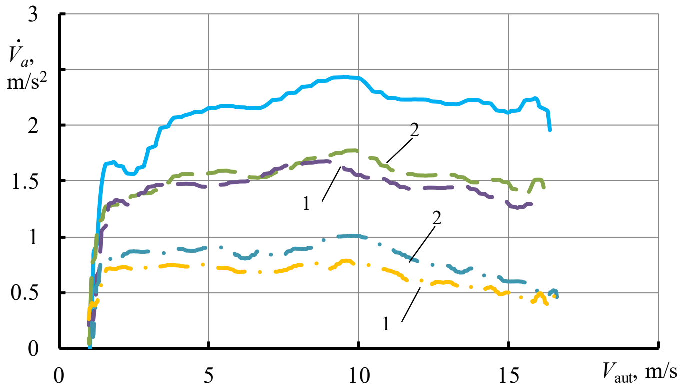


Fig. 7. Graphs of the dependence of linear longitudinal acceleration on the speed of movement during intensive acceleration in the second gear of the Daewoo Lanos 1.5i with different number of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – with one cylinder disconnected; –·–·–– with two cylinders disconnected

Using formula (23) and data obtained from the graphs in **Fig. 7**, the calculated amount of power on the driving wheels of the Daewoo Lanos 1.5i when moving at different speeds.

The power on the driving wheels of the car is calculated for all considered variants of the engine of the specified car (with all working cylinders, with one and two cylinders disconnected) and with the presence and absence of mechanical (pumping) losses in the disconnected cylinders. Based on the results of the calculations, the graphs shown in **Fig. 8**.

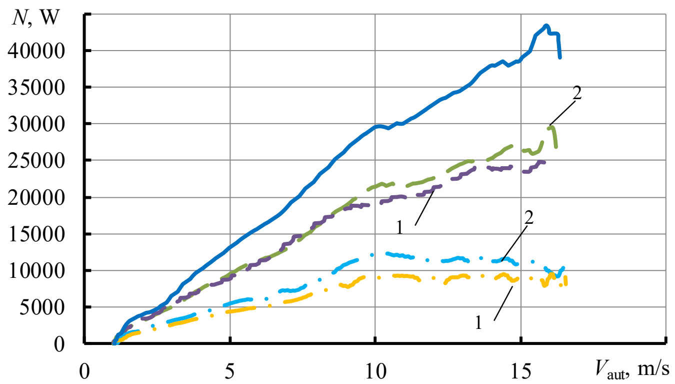


Fig. 8. Graphs of the dependence of the power on the driving wheels on the driving speed during intensive acceleration in the second gear of the Daewoo Lanos 1.5i with different number of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – with one cylinder disconnected; –·–·–– with two cylinders disconnected

Fig. 9 shows graphs and regression equations of the dependence of the power on the driving wheels on the driving speed during intensive acceleration in the second gear of the Daewoo Lanos 1.5i with different number of working cylinders.

Curves of the form $y = ax^3 + bx^2 + cx + d$, approximating the experimental data are obtained with high accuracy.

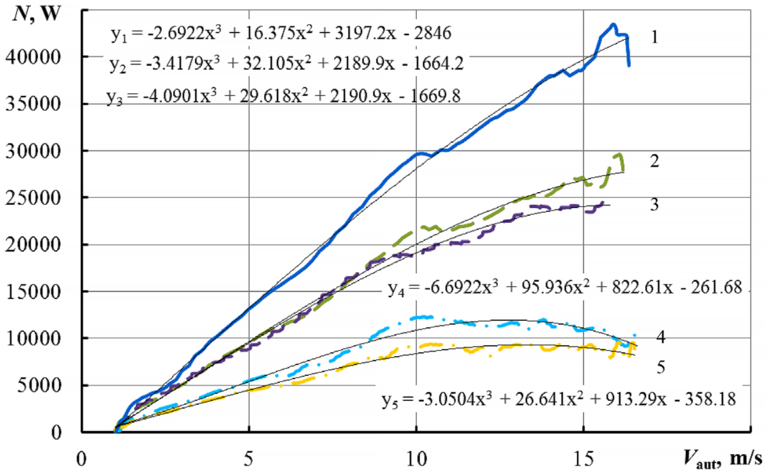


Fig. 9. Graphs and regression equations of the dependence of the power on the driving wheels on the driving speed during intensive acceleration in the second gear of the Daewoo Lanos 1.5i with different number of working cylinders: ——— – with all cylinders working; – – – – with one cylinder disconnected; – · – · – – with two cylinders disconnected

This is confirmed by the value of the regression coefficient R^2 (**Table 1**), the maximum value of which was 0.9973, and the minimum value was 0.9793. The results of the appraisal are summarized in **Table 1**.

Based on the results of the approximation, the maximum and average value of the engine power difference with the cylinders disconnected in the presence and absence of mechanical (pumping) losses was determined. For example, for the Daewoo Lanos 1.5i, when one cylinder was turned off, the average difference in the power value in the speed range V_{aut} = 2–13 m/s was ΔN_{aver} = 1405 W. The maximum difference was $\Delta N_{\max}$ = 1890 W. When the two cylinders were turned off, the average difference in the speed range V_{aut} = 2–13 m/s was ΔN_{aver} = 1720 W. The maximum difference was $\Delta N_{\max}$ = 2694 W, which is 1.43 times more than when only one cylinder was turned off. According to the results of experimental studies, the maximum power on the drive wheels used to accelerate the Daewoo Lanos 1.5i with all working cylinders was 44300 W. Thus, the difference in the power value in the presence and absence of mechanical (pumping) losses when one and two cylinders are turned off was 4.3% and 6%, respectively, of the maximum value indicated above.

The power on the drive wheels of a Daewoo Lanos 1.5i when one cylinder is disconnected (mechanical (pumping) losses are present) and when moving at a speed of 13 m/s drops by 1.56 times (from 35570 W to 22830 W). If mechanical (pumping) losses in the disconnected cylinder are removed, the power drop will decrease to 1.44 times (from 35570 W to 24720 W).

When two cylinders of the specified car's engine are turned off (mechanical (pumping) losses are present), the power on its driving wheels drops by 3.82 times from the initial values (from 35570 W to 93150 W). If mechanical (pumping) losses in disconnected cylinders are removed, the power drop will decrease to 2.98 times (from 35570 W to 11940 W).

Table 1
Results of the approximation of experimental data (**Fig. 9**) for determining the power on the drive wheels of the Daewoo Lanos 1.5i from its speed

No. of the curve in Fig. 9	Coefficients of the regression equation of the form $y = ax^3 + bx^2 + cx + d$				Coefficient
	a	b	c	d	R^2
1	–2.6922	16.375	3197.2	–2846	0.9972
2	–3.4179	32.105	2189.9	–1664.2	0.9954
3	–4.0901	29.618	2190.9	–1669.8	0.9961
4	–6.6922	95.936	822.61	–261.68	0.9798
5	–3.0504	26.641	913.29	–358.18	0.9793

The experimental data obtained from the results of the experimental road test of the KrAZ-255B truck were processed in a similar way. Using the obtained experimental data, graphs of changes in the longitudinal acceleration and speed of the KrAZ-255B over time with intensive acceleration in the second gear were constructed, which are shown in **Fig. 10, 11**.

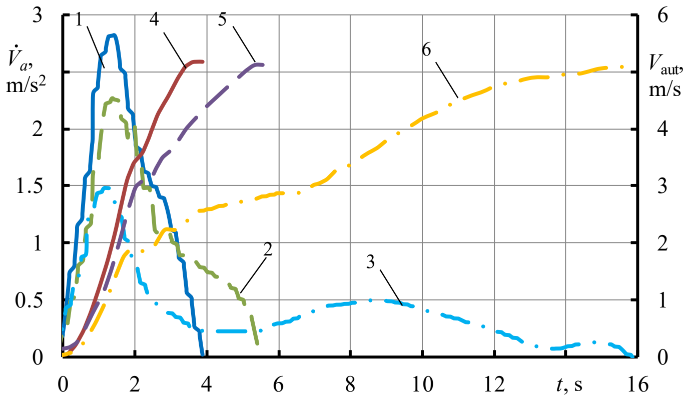


Fig. 10. Graphs of the dependence of linear longitudinal speeds and acceleration on time during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders (in the presence of mechanical (pumping) losses in disconnected cylinders): 1–3 – longitudinal linear acceleration; 4–6 – longitudinal linear speed; ——— – with all cylinders working; – – – – with two cylinder disconnected; –·–·–– with four cylinders disconnected

The obtained data confirm the expediency of eliminating mechanical (pumping) losses in disconnected cylinders, which can be disconnected for the purpose of saving fuel when the car engine is operating with a light load and idling.

The graphs are built for the cases of movement with all working cylinders of the engine (8 cylinders) of the KrAZ-255B, when the engine is working with two cylinders disconnected, and when the engine is working with four cylinders disconnected. Moreover, in **Fig. 10** shows the data for the case of mechanical (pumping) losses in the disconnected cylinders of the engine, and **Fig. 11** shows the data for the case of the absence of mechanical (pumping) losses in the disconnected cylinders of the KrAZ-255B engine.

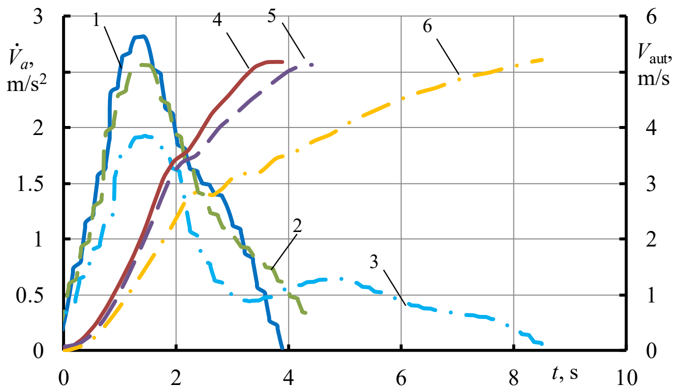


Fig. 11. Graphs of the dependence of linear longitudinal speeds and acceleration on time during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders (in the absence of mechanical (pumping) losses in disconnected cylinders): 1–3 – longitudinal linear acceleration; 4–6 – longitudinal linear speed; ——— – with all cylinders working; – – – – with two cylinder disconnected; –·–·–– with four cylinders disconnected

To further determine the effect of the number of disconnected engine cylinders and the effect of the presence or absence of mechanical (pumping) losses in these cylinders on the change in acceleration of the KrAZ-255B during intensive acceleration in second gear, by analogy with the Daewoo Lanos 1.5i, graphs of the change in longitudinal acceleration were constructed in times shown in **Fig. 12** and graphs of changes in longitudinal velocity over time shown in **Fig. 13**.

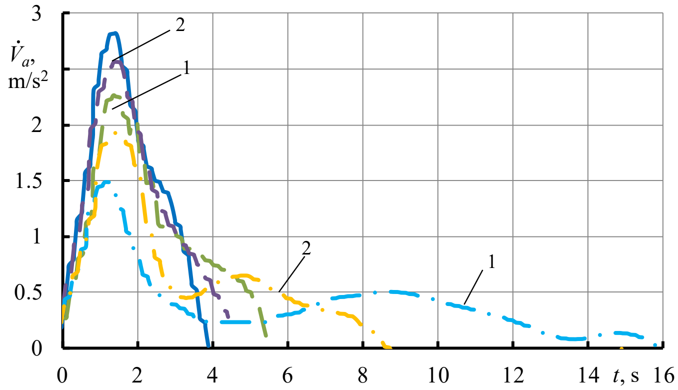


Fig. 12. Graphs of the dependence of linear longitudinal acceleration on time during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – with two cylinder disconnected; – · – · – – with four cylinders disconnected

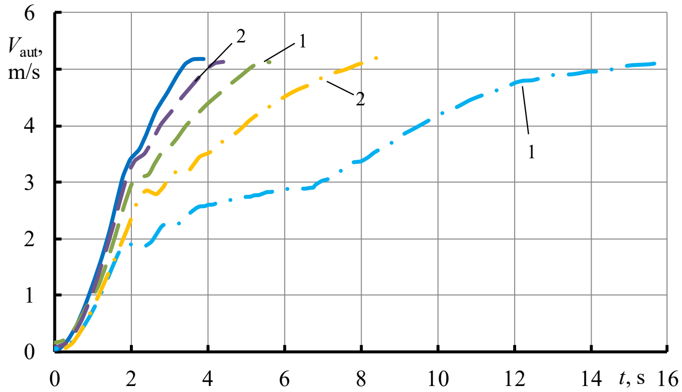


Fig. 13. Graphs of the dependence of linear longitudinal speed on time during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – with two cylinder disconnected; – · – · – – with four cylinders disconnected

Graphs of changes in longitudinal acceleration of the truck from its speed (during intense acceleration in second gear) with different number of working engine cylinders and the presence or absence of mechanical (pumping) losses in these cylinders are also plotted, which are shown in **Fig. 14**. Using formula (23) and data obtained from **Fig. 14**, the amount of power on the drive wheels of the KrAZ-255B when moving at different speeds was calculated.

The power on the driving wheels of the car is calculated for all considered variants of the engine of the specified vehicle (with all working cylinders, with two and four cylinders disconnected) and with the presence and absence of mechanical (pumping) losses in the disconnected cylinders. Based on the results of the calculations, the graphs shown in **Fig. 15** were constructed.

Fig. 16 shows graphs and regression equations of the dependence of the power on the driving wheels on the speed of movement during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders. The curves approximating the experimental data are obtained with high accuracy.

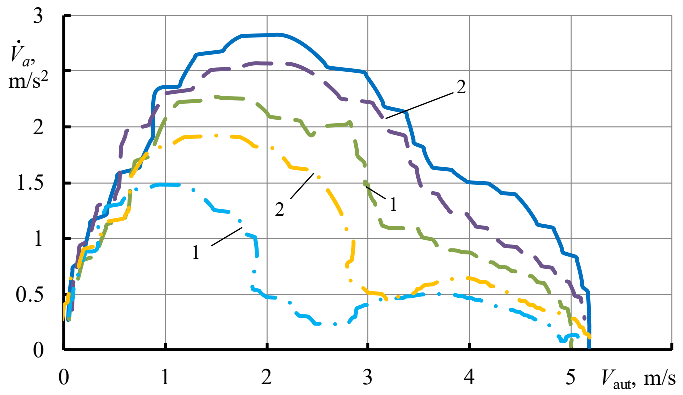


Fig. 14. Graphs of the dependence of linear longitudinal acceleration on the speed of movement during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – with two cylinder disconnected; –·–·– – with four cylinders disconnected

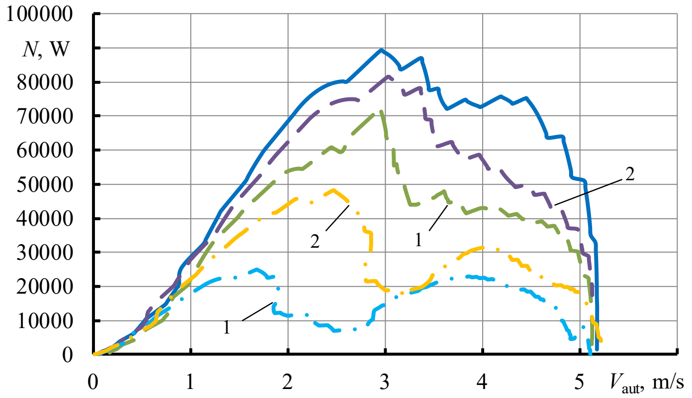


Fig. 15. Graphs of the dependence of the power on the driving wheels on the speed of movement during intensive acceleration in the second gear of the KrAZ-255B with different numbers of working cylinders: 1 – in the presence of mechanical (pumping) losses in the disconnected cylinder; 2 – in the absence of mechanical (pumping) losses in the disconnected cylinder; ——— – with all cylinders working; – – – – with two cylinder disconnected; –·–·– – with four cylinders disconnected

This is confirmed by the value of the regression coefficient R^2 (**Table 2**), the maximum value of which was 0.9915, and the minimum value was 0.8303. The coefficients of the obtained regression equations of the form $y = ax^6 + bx^5 + cx^4 + dx^3 + ex^2 + fx + g$ are summarized in **Table 2**.

The equations obtained as a result of the approximation make it possible to compare the values of acceleration power on the wheels of vehicles with different options for turning off the engine cylinders of wheeled vehicles when they are moving at a certain speed.

For the KrAZ-255B, when two cylinders were turned off, the average difference in the power value in the speed range V_{aut} = 1–5 m/s was ΔN_{aver} = 11151 W. The maximum difference was $\Delta N_{\max}$ = 19917 W. When four cylinders were turned off, the average difference in the speed range V_{aut} = 1–5 m/s was ΔN_{aver} = 13653 W. The maximum difference was $\Delta N_{\max}$ = 27437 W, which is 1.38 times more than when only two cylinders were turned off.

According to the results of experimental studies, the maximum power on the drive wheels used to accelerate the KrAZ-255B with all working cylinders was 89100 W. The difference in the power value in the presence and absence of mechanical (pumping) losses when two and four cylinders were turned off amounted to 22.3% and 30.8% of the maximum value indicated above, respectively.

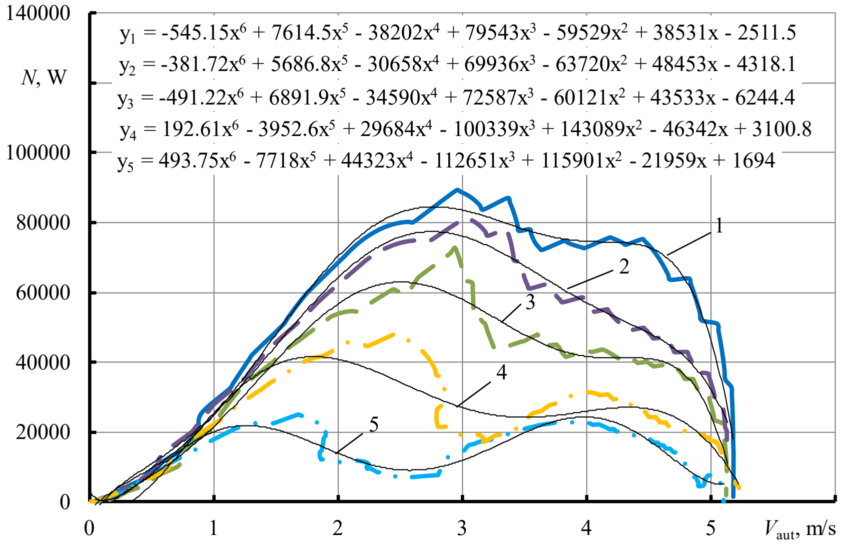


Fig. 16. Graphs and equations of the regression of the dependence of the power on the driving wheels on the speed of movement during intensive acceleration in the second gear of the KrAZ-255B with different number of working cylinders: ——— – with all cylinders working; – – – – – with two cylinder disconnected; – · – · – – with four cylinders disconnected

Table 2
Results of the approximation of experimental data (**Fig. 16**) for determining the power on the driving wheels of the KrAZ-255B from its speed

No. of the curve in Fig. 16	Coefficients of the regression equation of the form $y = ax^6 + bx^5 + cx^4 + dx^3 + ex^2 + fx + g$							Coefficient R^2
	a	*b*	*c*	*d*	*e*	*f*	*g*	
1	–545.15	7614.5	–38202	79543	–59529	38531	–2511.5	0.9817
2	–381.72	5686.8	–30658	69936	–63720	48453	–4318.1	0.9915
3	–491.22	6891.9	–34590	72587	–60121	43533	–6244.4	0.9655
4	192.61	–3952.6	29684	–100339	143089	–46342	3100.8	0.8303
5	493.75	–7718	44323	–112651	115901	–21959	1694	0.9036

The power on the drive wheels of the KrAZ-255B when two cylinders are disconnected (mechanical (pumping) losses are present) and when moving at a speed of 3 m/s drops by 1.44 times (from 83528 W to 57956 W). If mechanical (pumping) losses in the disconnected cylinder are removed, the power drop will decrease to 1.1 times (from 83528 W to 76153 W).

When two cylinders of the specified car's engine are turned off (mechanical (pumping) losses are present), the power on its driving wheels drops by 6.97 times from the initial values (from 83528 W to 11980 W). If mechanical (pumping) losses in disconnected cylinders are removed, the power drop will decrease to 3.09 times (from 83528 W to 27057 W).

The obtained data confirm the expediency of eliminating mechanical (pumping) losses in disconnected cylinders, which can be disconnected in order to save fuel when the truck engine is operating with a light load and idling. Moreover, the expediency of eliminating mechanical (pumping) losses in disconnected cylinders of a truck is significantly greater than for a passenger car.

Experimental results confirmed that mechanical (pumping) losses significantly affect the vehicle's driving power when engine cylinders are disabled. Eliminating these losses can greatly improve the effectiveness of cylinder deactivation strategies, especially for heavy trucks.

3. 3. Increasing the energy efficiency of wheeled vehicles by cylinder deactivation

The problem of rational use of the available power of the vehicle engine with a light profile of the path and a low speed of movement for underweight and empty wheeled vehicles was repeatedly raised in connection with freight transportation [6].

Only 4–5% of the duration of the traction mode [16, 18] is realized with power close to the nominal, 70–75% – with loads 0.5–0.8 of the nominal, the rest of the time using traction force and power of wheeled vehicles does not exceed 0,5 nominal values. Thus, the growing power of vehicles engines increases the problem of its full implementation in operation, and therefore the problem of increasing the operational COP of the traction mode of movement.

One of the ways to increase energy efficiency is the partial shutdown of the cylinders (or cycles) of vehicle engine during periods of operation with a clear excess of its traction force and power.

The COP of a wheeled vehicle can be represented as [16, 21]

$$\eta_{veh} = \frac{N_{us}}{N_{sp}} = \frac{N_{tr}}{N_{tr} + N_{loss} + N_{sp\,n}}, \tag{24}$$

where N_{us} – useful power; N_{sp} – spent power; N_{tr} – traction power of a wheeled machine; N_{loss} – power losses; $N_{sp\,n}$ – power spent on own needs.

The traction power of the wheeled vehicle N_{tr}, kW, is equal to

$$N_{tr} = F_{wh} \cdot V_{aut}, \tag{25}$$

where F_{wh} – traction force on the wheel of a wheeled vehicle, kN.

Power loss N_{loss} consists of mechanical energy, and losses on pumping strokes and the drive of auxiliary equipment. These losses are estimated using technical documentation.

By processing the records of the on-board recorder of motion parameters, the dependence of the COP of the wheeled vehicle on the instantaneous realized power was obtained. **Fig. 17** shows the dependence of the COP of vehicle engine on the power filling factor (PFF), equal to the ratio of the instantaneous power of the wheeled vehicle to its nominal power. The operation of the entire heat energy conversion system, mechanical and losses, as well as costs for own needs are reflected here. It can be said that this is the result of a field experiment repeated many times, the conditions of which are determined by the operating modes.

An approximation of the experimentally obtained COP of vehicle engine η_{veh} depending on the coefficient of power filling by the method of least squares is constructed in the form

$$\eta_{veh} = \frac{\xi}{a\xi + b}, \tag{26}$$

where x – power filling factor.

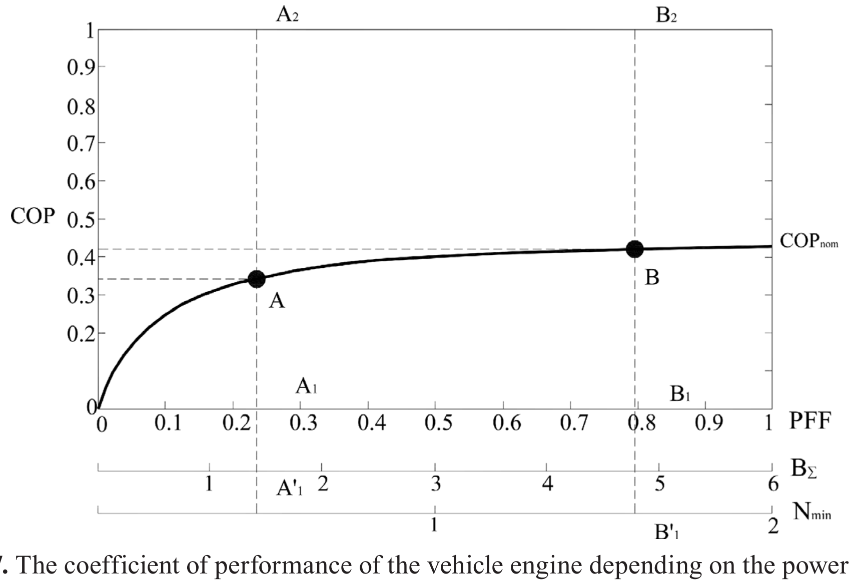


Fig. 17. The coefficient of performance of the vehicle engine depending on the power filling factor in traction mode

As a result, the COP of vehicle engine in the nominal mode is 25–40%.

Since the vehicle engine has the ability to adjust the traction force, the issue of operational disconnection/connection of one or more cylinders in automatic mode can be solved at the upper level of the control system.

The engine with a disconnected cylinder can be used to find the linear speed of the wheeled machine, which will increase the efficiency of the anti-skid system and allow to realize its traction properties as much as possible.

The number of cylinders of a vehicle engine in the general case is denoted as $i'_{cyl\Sigma}$ (**Fig. 17**) [6, 16–18], in this case $i'_{cyl\Sigma} = 8$. It is possible to denote the nominal power of one traction cylinder as $N_{tr\,cyl}$, let's take $N_{tr\,cyl}$ = 19.3 kW. Then the total nominal power of the engine will be $N_\Sigma = i'_{cyl\Sigma} \cdot N_{tr\,cyl} = 8 \cdot 19.3 = 154.5$ kW.

Suppose that at some point in time the power actually consumed by the vehicle engine for traction when all cylinders are working [6] is N_F, for example, N_F = 43 kW. In this case, the PFF of the wheeled machine will fold

$$\xi_\Sigma = OA_1 = \frac{N_F}{N_\Sigma} = \frac{43}{154.5} = 0.278, \tag{27}$$

this mode corresponds to the operational COP of the tractor engine (**Fig. 17**).

When using the proposed method of controlling the energy efficiency of the vehicle engine when working with partial load, the on-board computer program from the actual power N_F determines the minimum number of working cylinders $i_{cyl\,\min}$ necessary for the operation of the engine, in this case this number is equal to two (minimum possible), $i_{cyl\,\min} = 3$, and turns off other traction cylinders. In this case, the engine power will be $N_{\min} = i_{cyl\,\min} \cdot N_{tr\,cyl} = 3 \cdot 19.3 = 57.9$ kW.

The value of the PFF in this mode – when three traction cylinders are operating – will be [21, 22]

$$\xi_{\min} = OB_1 = \frac{N_F}{N_{\min}} = \frac{43}{57.9} = 0.742. \tag{28}$$

Thus, when determining the coefficient of power utilization, it is possible to judge the COP value of vehicle engine.

Cylinder deactivation under partial load conditions can substantially improve the operational energy efficiency of wheeled vehicles. Practical implementation requires dynamic assessment of actual power demand and intelligent control of the number of active cylinders.

The present study has several limitations. First, the results are based on experimental data obtained under specific conditions – flat road surface, moderate ambient temperature, and controlled gear-shifting – which may not fully represent all real-world driving scenarios. Second, the study was conducted on two types of vehicles: a passenger car and a heavy truck; therefore, extrapolation to other vehicle classes should be done with caution. Third, while the simulation of reduced pumping losses via venting methods is validated in principle, further testing on production engines is required to confirm its technical feasibility and long-term durability.

The obtained results are reproducible under the specified experimental protocols and vehicle configurations. To ensure reproducibility, details of the deactivation modes, data acquisition methods, and vehicle setup are clearly provided.

Future research should focus on expanding the range of tested operating conditions (e.g., hill climbing, variable loads, different climates), integrating CDA strategies with hybrid drivetrains, and assessing long-term effects on engine wear and maintenance. Moreover, real-time control algorithms for dynamic switching between CDA modes should be developed to optimize fuel savings and preserve performance across broader driving cycles.

4. Conclusions

The proposed method of determining the work performed and the power developed by the engine in the process of acceleration of the wheeled machine allows, in combination with the appropriate measuring complex, to evaluate the work done directly during the operation of the vehicle. The reliability of the obtained experimental data is ensured by the correct use of measuring equipment, obtaining and further processing of measurement information.

The values of the power drop on the driving wheels of the Daewoo Lanos 1.5i car and the KrAZ-255B truck, when part of their engine cylinders were disconnected, were determined, taking into account the presence or absence of mechanical (pumping) losses in the disconnected cylinders. The drop in power on the drive wheels of the Daewoo Lanos 1.5i when one or two engine cylinders are disconnected, taking into account the presence of mechanical (pumping) losses in the disconnected cylinders, was 1.56 times and 3.82 times, respectively. The elimination of mechanical (pumping) losses reduces the power drop to 1.44 times and to 2.98 times, respectively. The drop in power on the driving wheels of the KrAZ-255B when two or four engine cylinders were disconnected, taking into account the presence of mechanical (pumping) losses in the disconnected cylinders, was 1.44 times and 6.97 times, respectively. The elimination of mechanical (pumping) losses reduces the power drop to 1.1 times and to 3.09 times, respectively.

It was found that for the Daewoo Lanos 1.5i car, when one or two cylinders are disconnected in the presence and absence of mechanical (pumping) losses, the maximum difference in the power value in the speed range V_{aut} = 2–13 m/s (when moving in second gear) was, respectively $\Delta N_{\max}$ = 1890 W and $\Delta N_{\max}$ = 2694 W, which is 4.3% and 6.1% of the maximum power value at the wheels. For the KrAZ-255B, when two or four cylinders are disconnected in the presence and absence of mechanical (pump) losses, the maximum difference in the power value in the speed range V_{aut} = 1–5 m/s (when moving in the second gear) was, respectively, $\Delta N_{\max}$ = 19917 W and $\Delta N_{\max}$ = 27437 W, which is 22.3% and 30.8% of the maximum value.

The obtained data confirm the expediency of eliminating mechanical (pumping) losses in disconnected cylinders, which can be disconnected in order to save fuel when the engine of the car or truck is operating with a light load and at idle speed. Moreover, the expediency of eliminating mechanical (pumping) losses in disconnected cylinders of a truck is significantly greater than for a passenger car.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this study, whether financial, personal, authorship or otherwise, that could affect the study and its results presented in this paper.

Financing

The study was performed without financial support.

Acknowledgments

The authors are thankful to Kharkiv National Automobile and Highway University (Kharkiv, Ukraine) scientists for their cooperation in completing this research.

Data availability

Manuscript has no associated data.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

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Received 28.04.2025
Received in revised form 11.06.2025
Accepted 02.07.2025
Published 31.07.2025

How to cite: *Molodan, A., Podrigalo, M., Horielyshev, S., Abramov, D., Dubinin, Y., Polianskyi, O., Boikov, I., Medvid, M., Volkov, P., Poltavskyi, M. (2025). Identifying the regularities of power change at the vehicle's wheels when one or several engine cylinders are disabled. EUREKA: Physics and Engineering, 4, 26–46. http://doi.org/10.21303/2461-4262.2025.003882*