

ТЕХНИЧЕСКИ УНИВЕРСИТЕТ ВАРНА



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С МЕЖДУНАРОДНО УЧАСТИЕ



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XXXII НАУЧНО -ТЕХНИЧЕСКА КОНФЕРЕНЦИЯ С МЕЖДУНАРОДНО УЧАСТИЕ

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Съдържание

1.	POTENTIAL FOR REDUCTION OF UNBURNED FUEL EMISSIONS FROM LARGE GAS ENGINES THROUGH CHANGES IN VALVE TIMINGS USING SIMULATION	5
	Dimitar Dimitrov, Martin Pichler, Blaz Terlevic, Constantin Kiesling, Nicole Wermuth	
2.	ASSESSMENT OF DEGRADATION AND CYCLICITY OF THE RESOURCE OF A CAR'S TRACTION BATTERY	10
	Alexey Bazhinov, Eduard Kykla	
3.	DETONATION AND METHODS AND EQUIPMENT FOR ITS TESTING	13
	Volodymyr Bezridnyi, Ihor Kondysiuk	
4.	DIAGNOSIS OF THE SHOCK ABSORBER AS PART OF THE CAR SUSPENSION	16
	Alexander Dityatiev	
5.	CHANGING THE VEHICLE'S FUEL EFFICIENCY UNDER SIMULATED CONDITIONS ON A ROLLER STAND	23
	Yurij Gorbik, Sergy Krivoschapov	
6.	RECORDING AIR TEMPERATURE IN THE FUEL RATIONING PROCEDURE FOR MOTOR VEHICLES	29
	Sergy Krivoschapov , Yurij Gorbik	
7.	STUDY OF THE COMPLEX EFFECT OF GRINDING PARAMETERS ON THE QUALITY OF SURFACES PROCESSED WITH A SILICATE ABRASIVE COMPOSITION	36
	Volodymyr Makarov, Serhii Mastepan, Mykola Vynogradov, Mykola Mastepan, Mykyta Pantyushenko	
8.	IMPROVING THE MATHEMATICAL MODEL OF FUEL CONSUMPTION FOR CITY BUSES	46
	Ihor Marmut	
9.	OPTIMIZATION OF RUNNING-IN PARAMETERS USING A SILICATE ABRASIVE COMPOSITION VIA A FULL FACTORIAL EXPERIMENT	55
	Serhii Mastepan, Mykola Vynogradov, Mykola Mastepan, Mykyta Pantyushenko	
10.	CHANGE IN THE ELECTRICAL CONDUCTIVITY OF VEHICLE COOLANTS DEPENDING ON THE AMOUNT OF FUEL CONSUMED	64
	Mikhail Nahliuk	
11.	COMPARISON OF THE SERVICE LIFE OF DISC AND DRUM BRAKE MECHANISMS IN PASSENGER CARS	68
	Oleksandr Nazarov, Mykola Serhiienko, Yevgen Kalashnikov, Illia Pavlenko	
12.	THE IMPACT OF ADAPTIVE SUSPENSION ON VEHICLE SERVICE LIFE AND FUNCTIONAL SAFETY	74
	Vyacheslav Pavlenko, Vitaliy Pavlenko	
13.	JUSTIFICATION AND SELECTION OF KPIS FOR TRAFFIC MANAGEMENT SYSTEMS IN URBAN NETWORKS	78
	Hennadii Ptytsia, Liudmyla Abramova	
14.	EFFICIENCY OF LOGISTICS SYSTEMS IN LAST-MILE DISTRIBUTION	84
	Natalia Ptytsia, Olha Abramova, Valeriia Ptytsia	
15.	DIAGNOSTIC FEATURES OF THE CYLINDER PISTON GROUP OF GASOLINE ENGINES OF A CAR	89
	Irina Saraieva, Kozlov Oleksandr, Andrii Sokhin, Leontiev Denis	
16.	FEATURES OF DIAGNOSING AUTOMOTIVE MECHATRONIC SYSTEMS	93
	Vladimir Volkov, Tetiana Volkova, Maksym Levchuk, Ruslan Shulha	
17.	VEHICLE DRIVE AND TRANSMISSION SYSTEMS	99
	Yuriy Zybtsev, Pavlo Voroshylov	
18.	EXPERIMENTAL STUDY OF BICYCLE BRAKING DECELERATION	105
	Daniel Ivanov, Stoyan Stoyanov	

POTENTIAL FOR REDUCTION OF UNBURNED FUEL EMISSIONS FROM LARGE GAS ENGINES THROUGH CHANGES IN VALVE TIMINGS USING SIMULATION

Dimitar Dimitrov, Martin Pichler, Blaz Terlevic, Constantin Kiesling,
Nicole Wermuth

Abstract. The main subject of this paper is the reduction of unburned fuel emissions from large gas engines, using simulation and experimental investigations on a single-cylinder research engine. The design of experiments (DoE) method is used in connection with 1D simulation to find the optimal valve timing. The minimum of the unburned fuel emissions is found while taking into account the targeted operating conditions (brake mean effective pressure and NO_x level) and limitations (knock limit and zero penalty in indicated efficiency). The calibration of the combustion and scavenging sub-models is based on measurements on a single-cylinder research engine with a displacement of approximately 3 liters. The simulation results are validated with experimental measurements on the same research engine.

1. Introduction

Nowadays, the development of large gas engines is widely driven by the trend towards higher performance, greater efficiency and stricter emission regulations. Pollutant emissions from large engines and nitrogen oxide emissions (NO_x) in particular are currently strictly regulated. Limiting greenhouse gas emissions from large gas engines, such as carbon dioxide (CO_2) and methane has been pursued only more recently, e.g., via Germany's "Bundes-Immissionsschutzgesetz" for medium- and large-size combustion, gas turbine, and internal combustion engine plants [1], [2]. It seems likely that other countries and international institutions will also pass laws in the future to significantly reduce greenhouse gas emissions [3], [4], [5]. Lean-burn gas engines can reach the NO_x limits with relatively high excess air ratios (EAR), but this often leads to comparatively high unburned fuel (HC) emissions [6]. Methane as the main component of natural gas makes up the largest amount of hydrocarbon emissions and its impact on the greenhouse effect is around 30 times higher than that of CO_2 (on a 100-year time-horizon) [11]. It is thus essential to reduce the level of HC emissions from the engines to a minimum [7].

There are three main sources of HC emissions in lean-burn gas engines: scavenging, incomplete combustion and crevices in the combustion chamber [6, 8, 9, 10]. Scavenging refers to the flow of unburned mixture from the intake port to the main combustion chamber and then to the exhaust port during the valve overlap period. This process is favored by the pressure difference between the intake and exhaust port. Another source of HC emission is the incomplete combustion, mostly caused by flame quenching near the cylinder walls. This process is promoted by low pressures and temperatures at the end of combustion and by an increase in the EAR. The third source are small crevice volumes within the combustion chamber, which the flame does not propagate and thus fuel-air mixture remains unburned [5]. Such crevices exist between piston top land and liner, between piston ring and piston and in the vicinity of the cylinder head gasket. During the exhaust stroke the pressure in the combustion chamber drops and the unburned fuel from the crevices flows into the exhaust ports, mixed with the burned mixture.

This work presents the development of a camshaft with new valve timings targeting at the reduction of the unburned fuel emissions using simulation. The HC emissions from the scavenging process are represented by a 1D simulation model with a calibrated scavenging sub-model. The emissions from incomplete combustion and crevices are modeled in the 1D-framework via a conversion rate in the combustion chamber. The required conversion rate is described by a polynomial function, which is fitted to a measurement database obtained from measurements on single-cylinder research engine (SCE). The comprehensive measurement data base enables precise calibration of the sub-models.

2. Experimental investigations

Measurement data is required for the development and calibration of simulation models as well as for the calculation of the rate of heat release curves for direct specification in 1D simulation. The database for the calibration of the conversion rate in the combustion chamber and the scavenging sub-model consists of 27 steady-state single-cylinder research engine operating points. Each operating point was measured at least three times. The available database includes measurements of different parameter variations: indicated mean effective pressure (IMEP), NO_x level, ignition timing and scavenging pressure (difference between intake and exhaust pressure levels). The required rate of heat release curves for the 1D simulation were derived from the measured cylinder pressure curves.

All experimental investigations presented in this paper were carried out on a high-speed four-stroke single-cylinder research engine with a displacement of approximately 3 liters. A prechamber spark ignition concept and a camshaft with early intake valve closing (IVC) timing, also known as Miller valve timing, were used.

Charge air is supplied to the engine by an electrically driven compressor. The air humidity is controlled by means of a steam generator that admixes water steam to the cooled, dehydrated and re-heated intake air. Cooling water and lubricating oil are conditioned before being supplied to the engine. A back pressure flap in the exhaust piping is used to set the back pressure that would be generated by the turbine of a turbocharger on a multi-cylinder engine. The exhaust gas analysis is carried out with an AMA i60 exhaust gas measurement system from AVL List GmbH. The sampling point for HC emissions is located in the exhaust pipe, 650 mm from the cylinder head. Time-based measurement data is recorded and processed by the software Tornado from Kristl, Seibt & Co GmbH; crank angle-resolved data is recorded and processed using the indicating system IndiSet Advanced Gigabit and the software package IndiCom from AVL List GmbH.

3. Development of the 1D simulation model

For determination of the HC emissions, a 1D single cylinder engine model has been used. This model corresponds to the engine integrated on the test bed. Figure 1 shows the topology of the model in the commercial software package GT Power. The objective pursued with the model is to determine the conditions during the entire cycle not only in the main combustion chamber, but also in the prechamber (PC) as accurately as possible. Therefore, besides the main chamber the prechamber and the related gas valve (PCGV) are considered in detail in the 1D model. Air and fuel are mixed in the intake system. The prechamber mixture flows through the gas rail and the prechamber gas valve. The model uses measured pressure traces from the measured operating points for the intake as well as the exhaust manifold and in front of the PCGV. Three different controllers are integrated into the model for IMEP, PC mass flow and for the scavenging pressure. The combustion process is modeled with a two-zone sub-model. This enables determination of the maximum temperature in the unburned combustion zone (T_{u_max}), which is considered as a reasonable indicator of conditions for knock events and thus as a criterion for the knock limit. It is anticipated that the knock behavior of the engine stays unchanged in case that this temperature is constant. All measured operating points from the database were subsequently simulated using the 1D SCE model.

The goal of the 1D model is to predict the unburned fuel fraction in the exhaust gas for operating points with different boundary conditions than the measured ones, e.g., NO_x level. In this case, the rate of heat release curve, which is decisive in particular for combustion phasing, is derived from measured operating points with similar boundary conditions. The unburned fuel sensor in the simulation model is located in the exhaust pipe, at the same location as on the test engine. The unburned fuel fraction from the combustion is considered in the combustion sub-model and is a function of EAR. The function is determined using the MODDE software based on the 27 measured operating points. The unburned fuel fraction from valve overlap is taken into account in the scavenging sub-model.

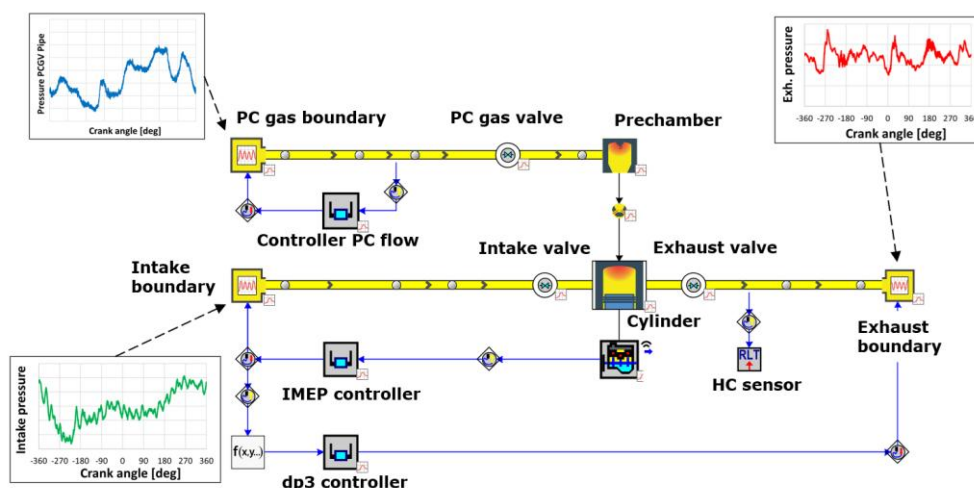


Fig. 1. Topology of the 1D model of the employed single-cylinder research engine with a prechamber

Fig. 2 shows an accurate overall agreement between measured and simulated HC emissions. Operating points 20 and 25 featured unstable combustion with a very high coefficient of variation of IMEP. Likely for this reason, the prediction of HC emissions for these operating points is inaccurate.

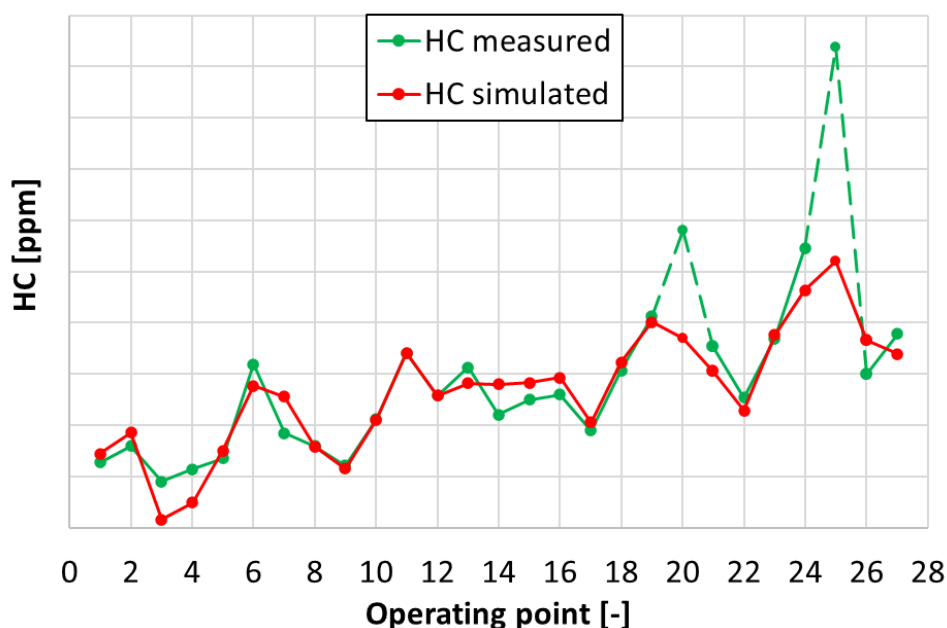


Fig. 2. Calibration of the 1D model: measured and simulated HC emission values

4. Optimization of the camshaft

The camshaft was optimized using 1D engine cycle simulation. A baseline operating point was selected from the measurements. The pre-chamber mass flow, IMEP and pressure gradient were kept constant for all simulation cases. Due to the high complexity of the system, the statistical method Design of Experiments (DoE) was used. Figure 3 provides an overview of the optimization process.

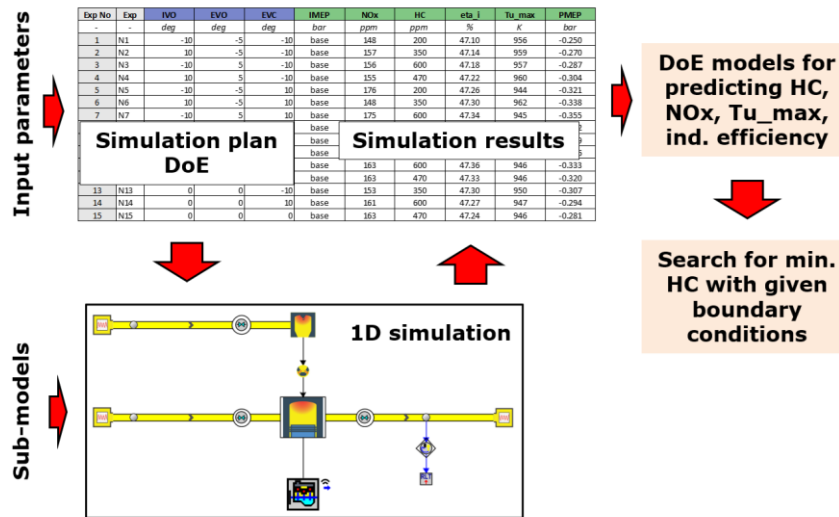


Fig. 3. Procedure for applying DoE method for 1D simulation.

The free parameters optimized on the basis of this approach were intake valve opening (IVO), exhaust valve opening (EVO) and exhaust valve closing (EVC). Intake valve closing (IVC) was not changed because this parameter has a very strong influence on NO_x formation. With the help of the commercial software MODDE, a simulation plan with 15 cases with different valve lift curves was generated. The cases were simulated and mathematical models for prediction of indicated engine efficiency, temperature of the unburned zone, NO_x emissions and HC emissions were created. Smaller valve overlap generally leads to lower HC slip while Tu_{max} and residual gas amount in the combustion chamber increase, thus resulting in a higher tendency to knock and limiting the HC slip reduction potential. The challenge in this task is to reduce the valve overlap phase for low HC emissions and to restrict the Tu_{max} and residual gas quantity for acceptable knocking tendency. Figure 4 illustrates the generally accurate agreement between results from the 1D simulation and the results predicted by the DoE mathematical model.

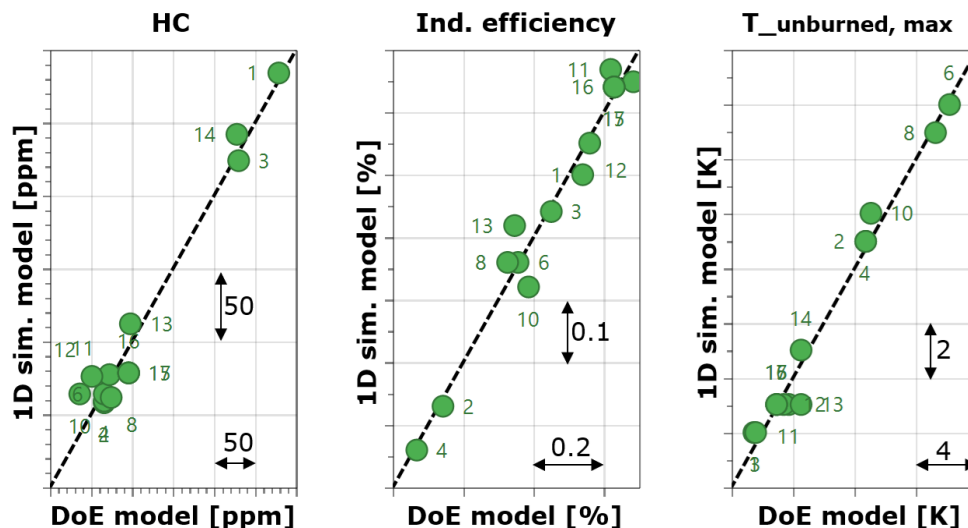


Fig. 4. Comparison of results from the 1D simulation and the DoE mathematical models.

With the help of the optimizer tool in the software MODDE, the optimal valve timing variant was determined. The goal of optimization was to minimize HC emissions while not affecting indicated engine efficiency, NO_x emissions and knock tendency. The optimized camshaft has a shorter valve overlap compared to the base version. In addition, the EVO timing and the valve overlap are retarded.

4. Verification of the simulation results

With the newly designed and machined camshaft installed on the SCE test bed, repetition testing has been conducted. The updated camshaft leads to the expected reduction in HC emissions compared to the base variant, while no loss in efficiency was observed. This confirms that the design yields a positive outcome in the primary investigated parameter (HC emissions) without negatively impacting other performance parameters.

5. Summary

This paper presented a development of a new camshaft for a large gas engine with the aim of reducing the unburned fuel emissions using simulation and experimental investigations. The simulation models were calibrated and validated with experimental measurements on a single-cylinder research engine. With the help of the Design of Experiments method, the optimal variant was determined and the valve timings for the new camshaft were specified. The operation with the newly developed camshaft leads to the expected reduction in HC emissions compared to the base variant.

Acknowledgments

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ОЦЕНКА НА ДЕГРАДАЦИЯТА И ЦИКЛИЧНОСТТА НА РЕСУРСА НА ТЯГОВАТА АКУМУЛАТОРНА БАТАРИЯ НА АВТОМОБИЛ

ASSESSMENT OF DEGRADATION AND CYCLICITY OF THE RESOURCE OF A CAR'S TRACTION BATTERY

Alexey Bazhinov¹, Eduard Kykla²

Abstract. The scientific and applied problem of increasing the accuracy and reliability of diagnosing traction batteries of electric vehicles is considered. The main result of the work is the development and experimental verification of a comprehensive technology that allows performing a joint high-precision assessment of the state of charge (SOC) and technical condition (SOH) of the battery in real time.

Keywords: Battery, Car, CAN, Kalman Filter.

1. Introduction:

To predict SOH, BMS uses models that describe the evolution of degradation as a function of the number of cycles and operating conditions.

The main degradation factors include the following. Number of cycles (Cycle Aging): Direct dependence of SOH on the number of charge/discharge cycles. Calendar Aging: Degradation during battery downtime occurs mainly from the storage temperature and the storage SOC level. Storage at high SOC (e.g. 90-100%) and high temperature (above 40 °C) significantly accelerates the growth of the SEI layer.

2. Exhibition:

The capacity loss formula is often modeled as a function of time t and the number of ampere-hours consumed $A_{h,passed}$:

$$\Delta C = k_{c,calendar} \cdot \sqrt{t} + k_{c,cycle} \cdot A_{h,passed} \quad (1)$$

where: $K_{c,calendar}$, $K_{c,cycle}$ - aging coefficients, which are functions of temperature and SOC.

For adaptive estimation of SOC and SOH, the Kalman filter algorithm is used. The Kalman Filter (KF) is an optimal recursive algorithm for estimating the state of a dynamic system that is subject to random noise. In the diagnosed TAB, the KF acts as an ideal tool for data fusion, integrating dynamic measurements (current) with nonlinear physical models (voltage), which provides the highest accuracy of SOC and SOH estimation in conditions of operational noise.

The principle of operation and advantages of the Kalman Filter (KF) are as follows. BMS uses KF to overcome the key shortcomings of coulometry (drift) and OCV methods (quiescence requirement and insensitivity in the mid-range SOC). The principle of recursion - the Kalman filter works in two sequential stages. The first stage involves prediction: based on the previous state and a mathematical model (e.g., equivalent circuit model, ECM), the filter predicts the current state of the system (SOC). The second stage performs correction. The filter uses actual measurements (voltage) to correct the predicted state. This correction is weighted, where the weight is determined by the covariance of the prediction and measurement errors. The key advantages of the Kalman Filter are as follows. Optimality: KF provides the minimum mean square error (MMSE) of the state estimation. Noise immunity: Effectively filters out random noise originating from current, voltage, and temperature sensors. Drift control: By periodically correcting the coulometric SOC with voltage data, KF prevents the accumulation of integral error.

The operation of the filter is described by two basic equations representing the dynamics of the system:

The first state equation describes how the internal state of the system (e.g., SOC) changes over time under the influence of a control input (current u_{k-1}):

$$x_k = f(x_{k-1}, u_{k-1}) + w_{k-1} \quad (2)$$

where: x_k - state vector at time k (includes SOC and polarization parameters); u_{k-1} - input influence (charge/discharge current); w_{k-1} - process noise (model inaccuracies).

The second measurement equation (Measurement Equation) relates the internal state of the system to what can be directly measured (voltage y_k):

$$y_k = h(x_k) + v_k \quad (3)$$

where: y_k - measured voltage at the terminals; $h(x_k)$ - nonlinear function using the ECM model and OCV-SOC dependence; v_k - measurement noise (sensor error).

Since the functional dependence of voltage on SOC is nonlinear, the standard (linear) Kalman filter cannot be applied directly. Therefore, its adaptive variants are used in BMS.

1. Extended Kalman Filter (EKF): linearizes nonlinear functions around the current evaluation point using Jacobi matrices.

2. Unscented Kalman Filter (UKF): uses the deterministic sampling method, providing higher estimation accuracy than EKF for complex nonlinearities. Co-estimation of the state of charge (SOC) and state of health (SOH) is the most complex, but the only correct diagnostic strategy in real-world conditions. This approach is necessary because the SOC and SOH parameters are strongly interdependent: an accurate calculation of SOC (by coulometry) requires knowledge of the current maximum capacity $C_{current}$, which is a direct indicator of SOH. To overcome the interdependence problem, modern BMSs use Dual State and Parameter Estimation algorithms, where an adaptive filter estimates not only the dynamic state (SOC), but also the slowly changing parameters $C_{current}$.

Instead of estimating only SOC, an adaptive filter (e.g., Extended Kalman Filter, EKF) operates on an extended state vector x that includes degradation parameters:

$$x = [SOC, U_{pol}, C_{current}, R_o] \quad (4)$$

where: U_{pol} – polarization voltage (dynamic element of the ECM model); $C_{current}$ – current maximum capacitance, which is a direct manifestation of SOH; R_o – ohmic resistance, which increases irreversibly with aging.

The mechanism of KO-estimation is as follows. SOC dynamics: the Kalman filter uses the coulometry equation to predict SOC. Adaptation of $C_{current}$ and R_o : the parameters $C_{current}$ and R_o are modeled as stationary processes with slow change. The filter uses the voltage measurement error $U_{measured} - U_{predicted}$ to recursively refine these parameters. For example, if the filter constantly overestimates the voltage prediction, this may indicate that the real resistance R_o (and, accordingly, SOH) is higher than predicted by the model. Using RLS: Often, a Recursive Least Squares (RLS) algorithm is used in parallel with EKF to adaptively estimate the $C_{current}$ and R_o parameters. RLS effectively identifies the model parameters in real time, and EKF uses these updated parameters to estimate the SOC state.

The BMS diagnostic circuit is a multi-layer architecture that provides continuous monitoring and control of the TAB. The data acquisition layer (Hardware Layer) includes the following.

Voltage and temperature: High-precision Analog-to-Digital Converters (ADCs) measure the voltage of each cell, and thermistors (NTC resistors) monitor the temperature at critical points of the modules.

The total current of the package is measured using a high-precision shunt or Hall sensor.

Sampling time (Δt), data collection occurs at a high frequency (from 100 ms to 1 s), since the accuracy of coulometry directly depends on the sampling frequency.

The calculation layer (Software/Modelling Layer) is the basis of the diagnostic logic. Model library: stores the identified parameters of the ECM model and the dependency tables $U_{OCV}(SOC, T)$. KO estimation algorithm (EKF/UKF): performs a dual SOC and SOH estimation based on the model and input data. RUL (Remaining Useful Life) estimation: predicts the remaining service life based on the current SOH decline rate

$$RUL = f(SOH_{\text{ПОТОЧНИЙ}}, C_{\text{rated}}, N_{\text{ЦИКЛІВ}}) \quad (5)$$

The Decision-Making Layer uses the following.

The BMS uses the estimated parameters for active control:

Charging control: determining the optimal charge rate (C-rate) depending on SOC and T to prevent lithium plating at low temperatures. Thermal control: activating the cooling system pumps and valves if T_{max} exceeds the optimal range (25-35 °C). Safety: opening or closing the high-voltage contactors when critical faults are detected (U_{max} or U_{min} exceeds the threshold).

The final result of the diagnostic loop is a set of standardized parameters that are transmitted to external systems via the CAN bus.

3. Conclusions:

An improved algorithm for joint estimation of SOC and SOH based on the Extended Kalman Filter (EKF) has been developed. The key difference and novelty of the algorithm is the implementation of a mechanism for adaptive correction of the covariance matrices of process noise (\$Q\$) and measurements (\$R\$) in real time depending on the dynamics of the load current and temperature. This allowed solving the problem of filter divergence during sharp changes in the operating modes of the electric vehicle and increasing the stability of the estimate.

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ДЕТОНАЦИЯ И МЕТОДИ И ОБОРУДВАНЕ ЗА НЕГОВИЯТА ИЗПИТВАНИЯ

DETONATION AND METHODS AND EQUIPMENT FOR ITS TESTING

Volodymyr Bezridnyi¹, Ihor Kondysiuk²

Abstract. Detonation is the supersonic propagation of a chemical reaction in an explosive substance or fuel mixture. It occurs as a blast wave (shock wave + reaction), causing a sharp pressure surge and destructive consequences (in engines) or an instantaneous release of energy (explosives). Key aspects of detonation: Physical nature: Unlike normal combustion (subsonic speed), detonation travels at supersonic speeds — from 1000 to 8000 m/s and higher. In internal combustion engines (ICE): It is an uncontrolled "explosion" of the fuel mixture, occurring before or instead of normal combustion from the spark plug. Signs of detonation in a car: A metallic knocking sound ("ringing") is heard, engine overheating is possible, power loss, vibration, and black soot from the exhaust pipe. Causes in ICE: Use of low-octane fuel, high load, engine overheating, improper ignition timing. Consequences: Destruction of pistons, piston rings and connecting rods due to impact loads.

Keywords: Damage, Detonation, Engine, Car, Combustion, Fuel.

1. Introduction:

Engine detonation is a serious problem, but many motorists do not know the reason for its occurrence. It occurs when the fuel is unevenly distributed. There are several ways to get rid of this problem. The essence of this combustion is that part of the fuel, due to a number of factors in the combustion chamber, spontaneously ignites. Moreover, this happens before a spark is formed between the spark plug electrodes, that is, the mixture ignites prematurely.

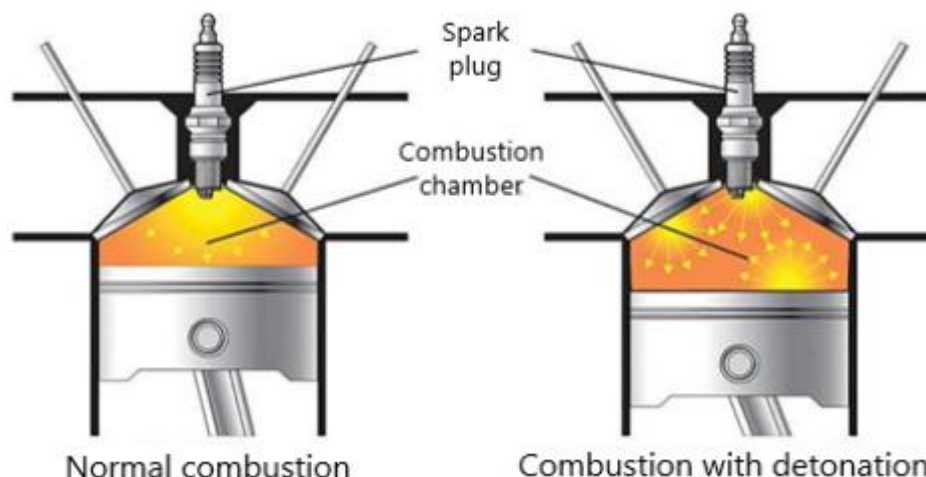


Fig. 1. Occurrence of detonation in the cylinder

2. Exhibition:

As installations for studying the detonation properties of fuels, most laboratories use single-cylinder engines of various designs. Special bombs are used much less often for this purpose.

Engines used for detonation experiments are most often made with a variable compression ratio, which allows testing fuels that are very different in terms of their detonation properties, because it is possible to take the compression ratio, which varies within very wide limits, as a variable factor when obtaining a scale of equivalents.

The results of experiments on the detonation of fuels obtained with the help of special bombs are of less practical value than the results obtained when testing engines of a special design. This is explained mainly by the fact that the combustion conditions in a bomb differ significantly from those in the combustion chamber of the engine; for example, there are absolutely no vortices that arise in the engine during the intake process, and the shape of the

combustion chamber does not have the same effect on the combustion process as it does in an engine.

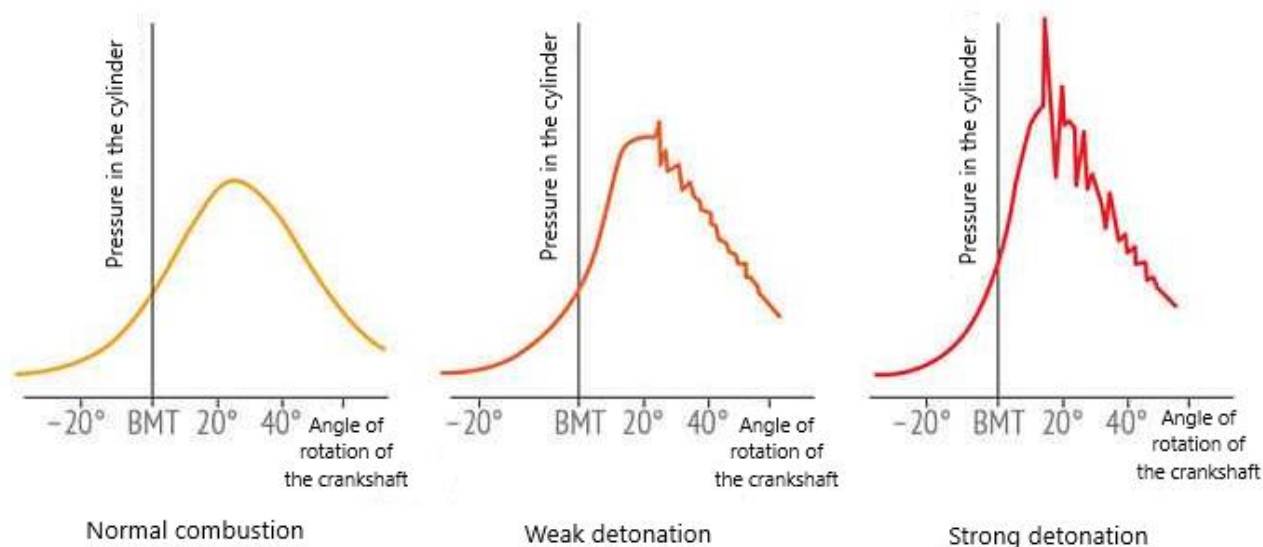


Fig. 2. Detonation gain dependence

Since the essence of the detonation phenomenon has not yet been fully elucidated, it is somewhat difficult to extend the results obtained in bomb experiments to the engine's operating process, and the assessment of detonation qualities from a practical point of view under these conditions is not entirely objective. On the other hand, the use of a special design for bomb detonation experiments is desirable, because in this case a more objective and constant environment for the combustion process can be obtained, the temperature conditions of the combustion process can be changed more widely, the fuel and air content can be dosed more precisely, and, finally, in this case, one does not have to take into account the mechanical condition of the engine (tightness of the piston rings, oil, carbon deposits in the combustion chamber, etc.), which when testing fuels on the engine can affect the combustion process and also distort the results of the experiment.



Fig. 2. Types of damage

In experiments to determine the detonation properties of fuel – with a bomb or an engine – it is necessary to make a comparative assessment of the intensity of detonation with the reference fuel, on the one hand, and with the tested fuel – on the other. The assessment of the intensity of detonation is made in different laboratories in different ways. The following can be mentioned among the most commonly used methods of assessing the intensity of detonation:

- a) by ear;
- b) by the type of exhaust gases;
- c) by the height of the ball bouncing on an elastic plate;
- d) by the rupture of the plate on which the flashes act;
- e) by the indicator diagram of the combustion process;
- f) by the electrical resistance of the mixture.

3. Conclusions:

At the first sign of detonation, reduce engine load and contact a service center for diagnostics as soon as possible. Specialists will perform a computer scan, identify the cause of the problem, and suggest solutions. In some cases, it may be necessary to restore the engine to factory settings or even partially disassemble it to assess the condition of the internal components.

Detonation is a serious enemy of any internal combustion engine. This phenomenon can cause irreparable damage to the engine in minutes, turning a routine drive into a costly repair. However, by understanding the causes of detonation and how to prevent it, you can significantly reduce the risk of encountering this problem. Proper maintenance of your car, using high-quality fuel, and regular maintenance are the keys to long and reliable operation of your engine.

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ДИАГНОЗА НА АМОРТИСЬОРА КАТО ЧАСТ ОТ ОКАЧВАНЕТО НА АВТОМОБИЛА

DIAGNOSIS OF THE SHOCK ABSORBER AS PART OF THE CAR SUSPENSION

Alexander Dityatiev¹

Abstract. Various methods are used to determine the technical condition of a vehicle's suspension and shock absorbers. A faulty suspension causes high body acceleration amplitudes, impaired vehicle handling, and reduced driving comfort. Braking distances increase due to wheel lift, and the vehicle's lateral and longitudinal roll during cornering and braking also increases. All of this increases the risk of accidents, as braking distances, roll on bends, and the operation of ABS, ESP, and other assistance systems directly depend on the damping properties of the suspension. The importance of these functions and the high costs of accident recovery necessitate regular suspension maintenance. A key challenge is the accurate diagnosis of the shock absorber. The fundamental difficulty lies in the fact that the shock absorber is diagnosed as part of the suspension, and the diagnostic results depend to varying degrees on the condition of other suspension components. The degree of dependence and the range of factors vary across different diagnostic methods, which is the driving force behind the development of new diagnostic methods. In this paper, along with traditional EuSAMA and MAXA methods, we consider new proposals – the phase method and the half-power method (Calvo method). The diagnostic parameters used here include the EuSAMA coefficient (ES parameter), unsprung mass amplitude at resonance, phase angle, and relative damping Θ . The large set of methods and corresponding diagnostic parameters indicates the lack of a method that meets all the requirements of suspension and shock absorber diagnostics.

Keywords: shock absorber diagnostics, methods, advantages, disadvantages.

1. Introduction:

Over time, shock absorber performance deteriorates, reducing safety and comfort. This necessitates periodic monitoring of shock absorber condition through diagnostics. In-place diagnostics are widely used, as assembly and disassembly are labor-intensive and require specialized equipment and qualified specialists. Certain diagnostic methods, to varying degrees, reduce the negative impact of external factors during shock absorber diagnostics. New methods are proposed with advantages in performance and diagnostic accuracy, based on the theory of mechanical system dynamics and spectraprocessing of diagnostic information.

Of the wide range of diagnostic methods [1], two main ones are currently widespread: methods using harmonic influence on the suspension: EuSAMA [2] and the resonance method [3]. The automotive service equipment market offers only stands implementing these methods in various modifications. It is estimated that 80% of the market segment in the European Union is occupied by stands offering the EuSAMA method.

2. Presentation

The EuSAMA method involves assessing the "adhesion coefficient" (hereinafter, the ES parameter) of a tire with a support surface equal to the ratio of the minimum vertical force during the test to the static load on the wheel. The wheel suspension is diagnosed by placing the wheel on a platform and exciting it from the platform with a sinusoidal vertical displacement signal of constant amplitude and descending frequency. The oscillating system includes the sprung and unsprung mass, the spring and tire stiffness, and the shock absorber damping. In Figure 1, the wheel pressure force on the platform in a static state (static load) is denoted by G_a ; the dynamic force due to the inertial forces of the unsprung mass is denoted by F_d ; and the "grip reserve," as the difference between the static load and the dynamic force, is denoted by F_x . Then, according to the definition of the parameter ES:

$$ES = \frac{G_a - F_d}{G_a} = 1 - \frac{F_d}{G_a} = 1 - \frac{A * C}{m * g}, \quad (1)$$

Where: A - is the amplitude of the displacement due to the force acting on the wheel; C - is the reduced suspension stiffness; M - is the mass of the unsprung parts of the wheel.

This is a very simplified model, but it helps us understand the nature of the ES parameter.

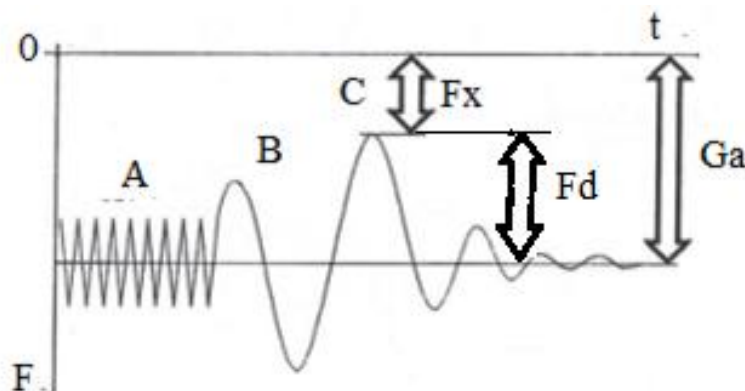


Fig. 1. Graphical interpretation of the EuSAMA parameter

As follows from formula (1), the ES parameter does depend on the wheel load and suspension stiffness (the state of the elastic device and tire pressure). The ES parameter also significantly depends on the magnitude of the input signal – the wheel displacement amplitude. Due to the dependence of the diagnostic result on the wheel load and tire pressure, the EuSAMA method is characterized by low accuracy [4]. As a result, serviceable shock absorbers in a suspension with a high load may receive a negative conclusion. The modernized method includes a coefficient correction depending on the axle overload [1]. Furthermore, it was noted that the EuSAMA method, both in its original source and with its modifications, allows for an assessment of the vehicle's driving safety potential through the "adhesion coefficient." The ES parameter, due to its dependence on a significant number of vehicle parameters and its mediocre sensitivity, has proven to be of little use for determining the performance of a shock absorber. A more thorough modernization of the method involved incorporating phase relationships of oscillating masses [5, 6]. However, this involves a combination of two methods.

The EuSAMA method's extensive application experience, technological advancement, and focus on driving safety indicators support its selection.

Disadvantages include the dependence of the diagnosis on external factors and low sensitivity to damping.

Resonance Method. The physical principle of suspension diagnostics using the resonance method is less clear. The fundamental difference from the EuSAMA method is that the supporting surface mass and the elastic structure of the surface are additionally introduced into the load-bearing structure. This means that we have a three-mass dynamic system in which the oscillation energy is distributed differently than in the two-mass EuSAMA, which contributes to obtaining qualitatively different results. The diagnostic parameters used in the method include the amplitude of unsprung mass oscillations at resonance, the percentage damping index, and the relative damping Θ . The method is implemented as follows.

Diagnostic parameter – unsprung mass oscillation amplitude at resonance. After acceleration, the electric motor generates oscillations with a maximum frequency of 10 Hz, after which the frequency begins to decrease. The initial amplitude of the support pad oscillations is 6.5 mm, and the frequency decreases to 2 Hz, thereby scanning the vehicle suspension with a frequency spectrum of 10...2 Hz. When the disturbing oscillation frequency becomes equal to the natural frequency of the "support pad mass - elastic structure of the support pad" system, a frequency value of approximately 10 Hz, resonance occurs and the amplitude of the support pad oscillations increases to values determined by the damping properties of the system, i.e.,

including the absorption properties of the shock absorber. The maximum amplitude of the support pad oscillations is recorded and subsequently interpreted as a diagnostic parameter for the condition of the shock absorber.

Diagnostic parameter – relative damping Θ . This parameter, which characterizes the properties of the suspension, has been known for a long time [7]. In DSTU, it is defined as "Relative damping - the ratio of the system's damping coefficient to the critical damping coefficient" [8]. In Ukraine, the auto service equipment market offers products from several companies, for example: Hofmann, the Contactest 3800 PC stand; EURO MAHA, the MSD 3000 stand for diagnosing vehicle suspensions using the Θ method.

The "relative damping Θ " parameter for suspension is determined by the formula:

$$\Theta = \frac{d}{C_c} = \frac{d}{2\sqrt{C_{\Pi} * m_1}}, \quad (2)$$

Where: d - is the damping coefficient, N*s/m; C_c - is the critical damping coefficient, N*s/m; C_{Π} - is the suspension stiffness, N/m; m_1 - is the sprung mass, kg.

The denominator of the formula is the critical damping coefficient, i.e. Its limiting value, at which the system ceases to oscillate, can be determined statically. The numerator of formula d represents the current (measured) damping coefficient of the vehicle's suspension. It is equal to:

$$d = \frac{b}{2 * m_1}, \quad (3)$$

where: b - is the viscous drag coefficient of a linear system with one degree of freedom, N*s/m.

The problem of determining the technical condition of a shock absorber without removing it from the vehicle ultimately comes down to correctly determining the relative damping coefficient d , the value of which is determined dynamically. There are known difficulties associated with the fact that testing is performed as part of the suspension system. Therefore, strictly speaking, when testing on a rig, the damping of the entire suspension (including the shock absorber) is determined, and this is specified in the rig operating instructions.

There are several methods for determining the relative damping coefficient d (formula 3) under these conditions.

The MAXA method is based on the principle corresponding to formula (2), where the C_{Π} component of the formula denotes the spring stiffness of the rig in N/m. The damping coefficient d is determined as follows:

$$d = \frac{C_g * r}{2\pi * f_{mes} * X_1} - d_{cm}, \quad \text{H*c/m}, \quad (4)$$

where: C_g - is the sum of the spring stiffnesses of the rig, in N/m; r - is the amplitude of the bearing pad travel from TDC to BDC at low frequency, mm; f_{mes} - is the frequency at which the amplitude of the oscillating bearing pads is greatest, 1/s; X_1 - is the double amplitude of the bearing pad oscillations at the resonant frequency, mm; d_{cm} - is the rig's damping coefficient (natural damping), N s/m.

It is easy to see that the values of C_p , C_g , and r are constants for the rig, while the values of m_1 , f_{mes} , and X_1 are measured during testing. The measurement diagram and the measured values are shown in Figure 2.

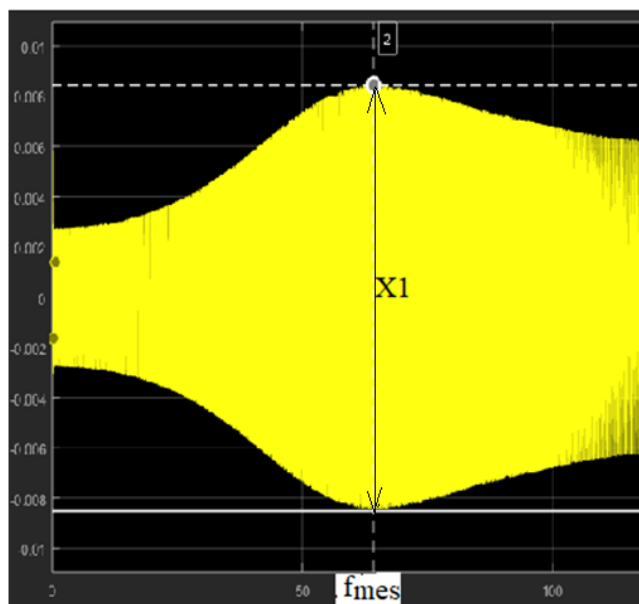


Fig. 2. Measurement diagram of the damping coefficient using the MAXA EUROSISTEM method - $X1$ - is the double amplitude of the platform vibrations at resonance, mm; f_{mes} - is the resonance frequency, Hz.

Figure 3 shows the dependence of the relative damping at the output on the relative damping at the input, which is essentially the gain of the methods. This data is necessary for correcting the relative damping coefficient measurement results.

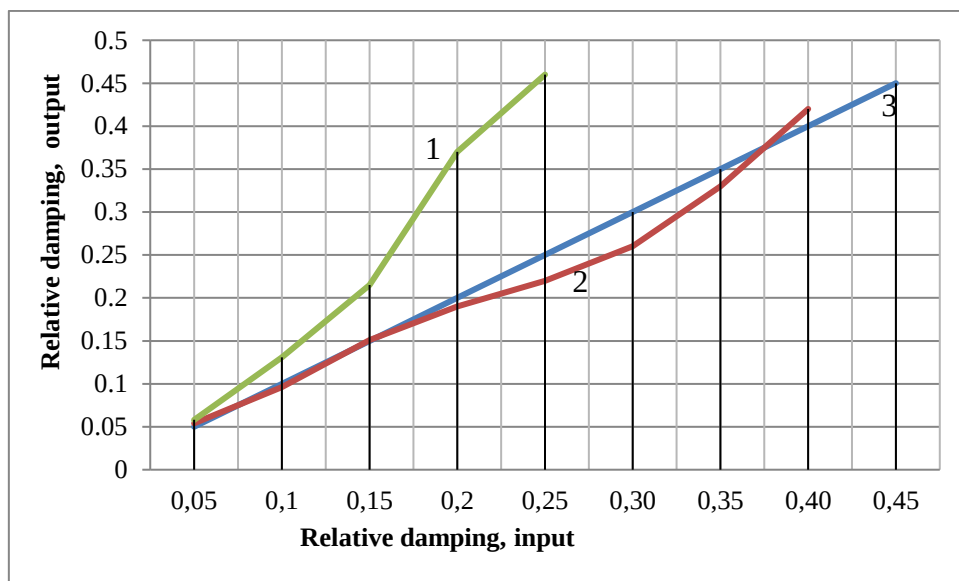


Fig. 3. Relative Output Damping vs. Relative Input Damping - 1 – Half-Power Method; 2 – MAXA EUROSISTEM Method, MSD 3000 Test Rig; 3 – Preferred Characteristic

The advantage of this diagnostic parameter in the resonance method is its physical meaning, understandable to systems engineers, and its basis in system dynamics theory.

The disadvantages include the need to constantly maintain the dynamic parameters of the test rig.

Phase Angle Method. Shock absorber diagnostics utilize phase relationships proposed by the American company Hunter Engineering Co. [9]. It proposes measuring the phase difference (phase angle) between the phase of the shock absorber oscillation and the phase of the

dynamic force oscillation at the tire-to-suspension contact. The phase angle complements the ES diagnostic parameter, allowing for more accurate determination of the shock absorber's technical condition [10].

The formula for calculating the phase in a second-order oscillatory system is:

$$\operatorname{tg} \varphi = \frac{\omega_0^2 - \omega^2}{2 * \xi * \omega}, \quad (5)$$

where: ξ - is the damping coefficient; ω - is the frequency of the driving force; ω_0 - is the natural frequency of the oscillatory system.

Recall that, according to the classical theory of mechanical systems, the phase frequency response of an oscillatory link depends significantly on the magnitude of the damping in the link. In particular, each of the two components of the suspension structural diagram is a second-order oscillatory link. For such links, the classical phase frequency response demonstrates the dependence of the phase lag of the displacement and velocity on the driving force with different damping. Moreover, for a frequency equal to the resonance frequency, the phase angle is fixed at $\pi/2$. The wheel suspension, as a fourth-order oscillatory system, has a similar characteristic, Fig. 4.

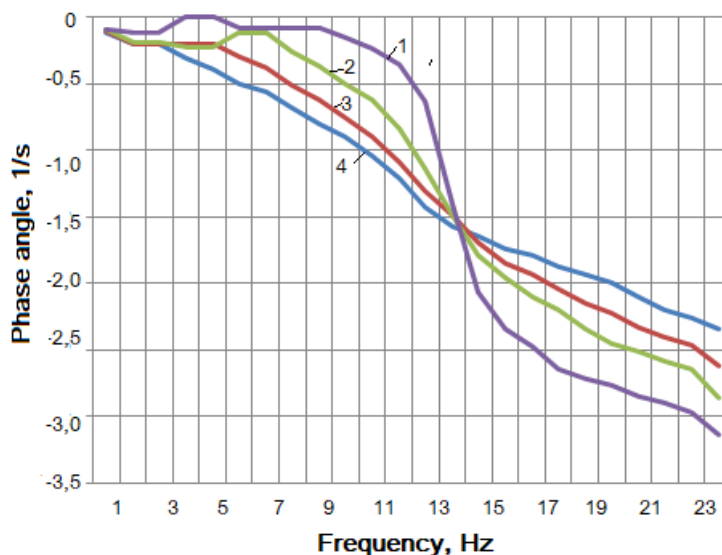


Fig. 4. Phase angle versus wheel excitation frequency for different damping settings (suspension phase frequency response) - Shock absorber damping ξ , N s/m: 1 – 275; 2 – 1100; 3 – 1925; 4 – 2750.

Note that the process phase (formula 5), and hence the phase difference, are independent of the driving force amplitude. The phase angle depends indirectly on the wheel load due to its dependence on ω_0 and ξ .

Advantage of this method: the amplitude of the input harmonic excitation of the wheel, the stiffness of the elastic element of the suspension, and the damping of the tire do not affect the phase angle.

Disadvantages: the influence of tire stiffness and the mass of the unsprung parts of the suspension.

Half-Power Method, Calvo Method [11, 12]. The method draws on modal analysis and antenna technology. The relative damping Θ is used as a diagnostic parameter. The advantage of this method is that it measures the absolute value of only one parameter—frequency. Measuring the absolute magnitude of the pad's displacement is not required; it is sufficient to ensure that the displacement is linear depending on the dynamic force. This significantly simplifies and reduces the cost of operating the shield. Furthermore, the test result is

independent of vehicle mass fluctuations and tire pressure. The result is determined by the formula:

$$\Theta = \frac{f_1 - f_2}{f_m}, \quad (6)$$

where: f_1 and f_2 - are the frequencies corresponding to the intersection points with the resonance curve at 0.707 or $(1/\sqrt{2})$ of the maximum pad displacement, Hz; f_m - is the resonance frequency, Hz.

The measurement diagram and measured values are shown in Figure 5.

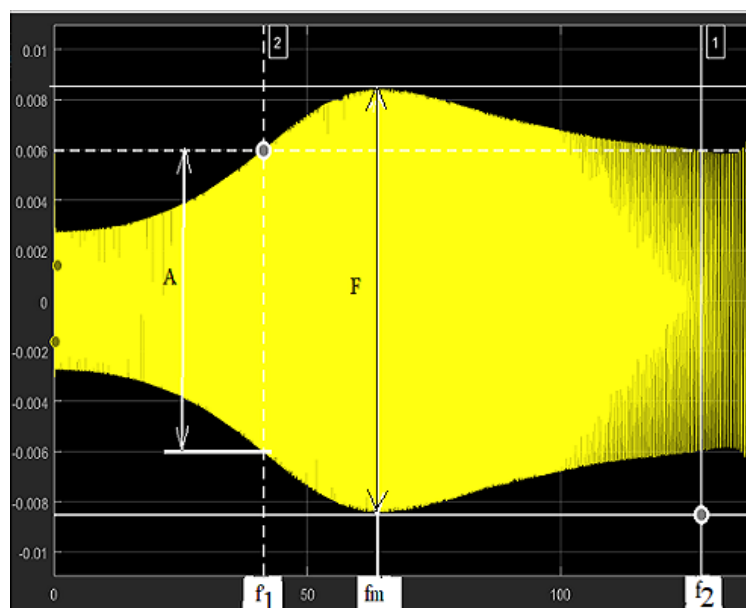


Fig. 5. Measurement scheme using the half-power method - F - is the amplitude of the platform oscillations at resonance, mm; A - is the double amplitude at the $F/\sqrt{2}$ level, mm; f_1 , f_2 , f_m - are the frequencies according to formula (6).

The half-power method is applicable for relative damping values Θ from 0 to 0.25, i.e., for cases with extremely unsatisfactory and satisfactory shock absorber conditions, Fig. 2. In this same range, the method demonstrates higher relative damping values Θ_{vihad} , which can lead to an erroneous diagnosis and acceptance of a faulty shock absorber for operation. The method is insensitive to changes in sprung mass within +40% of the nominal value and is slightly sensitive (within 15%) to changes in tire inflation pressure [13].

Comparative characteristics of the methods are presented in the table 1.

Table 1. Comparative characteristics of suspension/shock diagnostic methods

Indicators	Diagnostic methods				
	EuSAMA	Resonant		Phase	Half Power
Diagnostic parameter	ES parameter	Amplitude at resonance	Relative damping	Phase angle	Relative damping
Parameter control	Traffic safety	Technical condition	Technical condition	Technical condition	Technical condition
Availability of test rigs on the market	yes	yes	yes	No	No

Indicators	Diagnostic methods				
	EuSAMA	Resonant		Phase	Half Power
Results dependence on load	yes	yes	yes	yes	No
Results dependence on tire pressure	yes	yes	yes	yes	Weak [10]
Damping sensitivity	0,174	0,705	0,12	0,24	0,2
Oscillatory frequency during testing	25...0	10...2	10...2	Interresonance region	25...0

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ПРОМЯНА НА ГОРИВНАТА ЕФЕКТИВНОСТ НА ПРЕВОЗНОТО СРЕДСТВО ПРИ СИМУЛИРАНИ УСЛОВИЯ НА РОЛКОВ СТЕНД

CHANGING THE VEHICLE'S FUEL EFFICIENCY UNDER SIMULATED CONDITIONS ON A ROLLER STAND

Yurij Gorbik¹, Sergey Krivoshapov²

Abstract. This article proposes a method for determining the energy performance of a vehicle equipped with a fuel injection system using a roller dynamometer. It also examines the design of vehicle fuel efficiency dynamometer tests. A method for measuring fuel consumption is proposed based on the pulse duration of the electrical signal supplied to the fuel injector from the engine control unit. The injector orifice opening and closing delays are taken into account using a special coefficient. The results of a fuel consumption experiment for a Skoda Octavia Elegance with an AGU 1.8 20V Turbo on a roller dynamometer are presented. The roller dynamometer braking forces are determined to ensure consistency between stand and road fuel efficiency tests.

Keywords: car, fuel injector, fuel consumption, stand losses.

1. Introduction:

The functioning of an internal combustion engine is based on the transformation of the chemical energy of the fuel into mechanical energy. The efficiency of this conversion directly determines the key performance parameters of the power train, including its fuel efficiency and the environmental safety of its exhaust gases.

The widespread adoption of microprocessor-based control systems in automotive vehicles significantly improves the performance characteristics of internal combustion engines. This is achieved by optimizing fuel delivery algorithms that adapt to current load conditions and destabilizing external factors.

To assess fuel efficiency and regulate fuel consumption, instantaneous fuel consumption can be measured with high accuracy. This method is based on monitoring the injector opening time, taking into account their specifications. Measurements are taken while the vehicle is moving at a stable speed on the road or on a rolling test stand.

2. Exhibition:

The basic principles of fuel consumption regulation and fuel efficiency analysis are formulated in the study [1]. In this work, the design and operational efficiency indicators of automobiles are examined from the perspective of an energy approach and systems engineering. In turn, publication [2] presents an alternative method for determining fuel consumption, applied directly to vehicle diagnostics on roller stands.

The injector control algorithm is based on a simplified model for determining the cyclic feed based on the static performance of the injector and the duration of the control pulse. In this model, a constant effective flow area of the injector nozzle is assumed. The duration of the injector valve open state is assumed to be equal to the duration of the control pulse, which is determined using the principle of the base and correction matrices.

It is proposed to measure fuel consumption in a vehicle with an injection fuel system based on the duration of the electrical pulse that the engine control unit sends to the injector.

According to this model, the cyclic fuel supply [3].

$$G_c = g_c \cdot \tau_c, \quad (1)$$

where: g_c – static injector performance, τ_c – duration of the control pulse on the injector.

The injector static performance g_c is its ability to pass a certain amount of fuel G_n with the valve constantly open over a certain period of times (τ_v):

$$g_c = \frac{G_n}{\tau_v} . \quad (2)$$

The amount of fuel:

$$G_n = \mu \cdot f_{con} \cdot \sqrt{2 \cdot \rho_n \cdot \Delta P_n} \cdot \tau_v , \quad (3)$$

where: μf_{con} - constant effective flow area of the injector nozzle with the valve fully open, ΔP_n - constant fuel pressure drop across the injector, τ_v - measurement time when spilled.

To improve measurement accuracy, the Bosch ASNU-01 test bench was used to evaluate the static performance of Skoda Octavia Elegance injectors. In this mode, a constant voltage was applied to the windings, fully opening the valves. The injector flow time (10–30 ms) was automatically counted by the electronic unit via a frequency generator and a pulse counter.

Taking into account formulas (2) and (3), the static performance of the injector is the mass flow rate of fuel through the atomizer and is numerically equal to:

$$g_c = \mu \cdot f_{con} \cdot \sqrt{2 \cdot \rho_n \cdot \Delta P_n} . \quad (4)$$

In the equation (4) μf_{con} , ρ_n and ΔP_n – constant quantities.

To maintain stable fuel injector flow characteristics under all engine operating conditions, the control system maintains a constant difference between the fuel pressure and the pressure in the intake manifold behind the throttle valve. To this end, the fuel system implements pressure feedback using a differential pressure stabilizer. Its pressure-reducing valve is equipped with a diaphragm actuator and is spring-loaded, and the above-diaphragm cavity is directly connected to the area downstream of the throttle valve.

It is known that the anchor of the valve solenoid lags in time when rising by an amount τ_n and when lowering τ_o . This leads to the fact that the duration of the open state of the injector valve differs from the duration of the electrical control pulse supplied to the winding of the injector electromagnet by an amount

$$\tau_v = \tau_k - \tau_n + \tau_o \quad \text{or} \quad \tau_v = \tau_k - \Delta \tau_n , \quad (5)$$

where: $\Delta \tau_n = \tau_n - \tau_o$ – takes into account the mismatch of the control impulse τ_k to real injection time τ_v .

Taking this condition into account, formula (1) will have the form

$$G_c = g_c \cdot (\tau_k - \Delta \tau_n) . \quad (6)$$

The discrepancy $\Delta \tau_n$ can be determined by dynamically testing the injectors using the ASNU-01 injector testing and cleaning stand from Bosch. This injector operating mode was achieved by maintaining constant fuel pressure in the rig's fuel rail. The valves opened for short intervals ($\tau_k = 1 \dots 20$ ms) under the influence of control pulses generated by the electronic unit based on reference time intervals.

In this mode, standard pulse counter records the number of cycle feeds. This method eliminates the need for a stopwatch, allowing calibrated time intervals to be set directly using the switch, ensuring high reproducibility of test conditions [3].

Then the cyclic feed G_c can be determined based on the results of dynamic spillage:

$$G_c = \frac{C_{nd}}{j} . \quad (7)$$

where: C_{nd} – the amount of fuel accumulated in the measuring vessel of the stand during the dynamic spill, j – the number of injections (cycle feeds) made during the spill.

Based on the results of static and dynamic spillage, we determine the amount of discrepancy for each nozzle:

$$\Delta\tau_n = \tau_k - \frac{G_{nd} \cdot \tau_v}{G_n \cdot j} . \quad (8)$$

In practice, it is more convenient to use a correction factor that takes into account the delay in the operation of the injectors:

$$k_\tau = 1 - \frac{\Delta\tau_n}{\tau_k} . \quad (9)$$

The cyclic fuel supply through the injector spray hole with a constant fuel pressure drop will be determined by the formula:

$$G_c = g_c \cdot \tau_k \cdot k_\tau = \mu \cdot f_{con} \cdot \sqrt{2 \cdot \rho_n \cdot \Delta P_n} \cdot \tau_k \cdot k_\tau . \quad (10)$$

Experimental studies were conducted using a front-wheel drive Skoda Octavia Elegance with an AGU 1.8 20V Turbo power train. The data acquisition system was based on a personal computer, an ADC module, and the PowerGraf Professional software application.

Identification of engine operating modes was performed using signals from sensors for rotation speed, coolant temperature, throttle position, and exhaust gas composition and starter activation.

To determine the fuel consumption of a vehicle on a roller stand, the following signals were recorded: drum rotation frequency (voltage value on tachogenerators), vehicle speed, first injector injection duration, and camshaft position signal.

Using mathematical models, it is possible to determine fuel consumption at each speed and given load.

Specific fuel consumption in l/100 km can be determined using the formula:

$$Q_{l/100km} = \frac{100 \cdot Q_n}{V_a} , \quad (11)$$

where: Q_n – the volumetric hourly fuel consumption according to the formula, l/h; V_a – vehicle speed, km.

The volumetric hourly fuel consumption is determined by the formula

$$Q_n = g_c \cdot (\tau_k - \Delta\tau_n) \cdot 1.2 \cdot n \cdot 10^{-4} = 0.0005 \cdot n \cdot (\tau_k - \Delta\tau_n) \quad (12)$$

For the Skoda Octavia Elegance $\Delta\tau_n = 0.04$ ms.

The crankshaft rotation frequency is determined by the formula

$$n = \frac{1.2 \cdot 10^5}{T_c} . \quad (13)$$

The PowerGraf Professional software suite processes a data array for a fixed gear ratio and a given vehicle speed. Within the selected engine operating mode, the recorded signals are identified and their electrical parameters are measured. Specifically, the "Injector Voltage" graph is used to determine the injector control pulse duration (τ_k , ms); the "Drum Rotation Speed" graph is used to determine the voltage (U_B , V); and the "Phase Sensor" graph is used to determine the camshaft rotation period (T_C , ms). The final fuel consumption is calculated by substituting the obtained values into mathematical relationships (11), (12), and (13).

Fig. 1 shows the results of fuel consumption tests for a Skoda Octavia on a roller stand for A-95 gasoline using the proposed method for determining energy indicators.

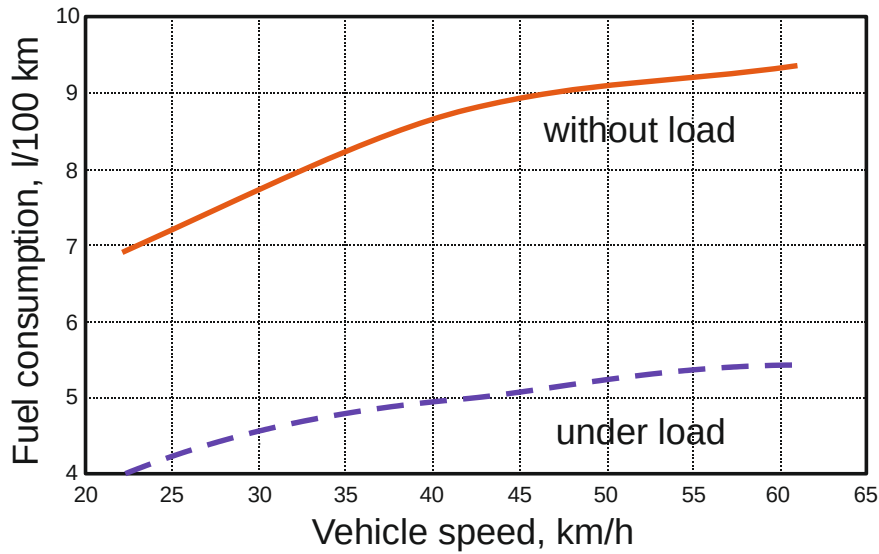


Fig. 1. Skoda Octavia fuel consumption graphs (under and without load) on a roller test stand

Fuel consumption depends on the load characteristics and parameters of the stand itself. Fuel consumption (in l/100 km) can be determined using the formula:

$$Q_s = \frac{[A \cdot i_s + B \cdot i_s^2 \cdot V_a + C \cdot (G_s \cdot f_s + P_B)]}{\eta_{is}} , \quad (14)$$

where: i_s – gear ratio set in the gearbox during vehicle testing on a test bench; G_s – curb weight of the vehicle per drive axle, N; f_s – coefficient of total resistance to movement of wheels on the roller stand; P_B – force absorbed by the stand brake, N.

On the road, at speeds over 45 km/h, fuel consumption increases sharply due to increasing air resistance. There is no air resistance on the stand. However, the rolling resistance coefficient f_s depends on the design of the roller stand. In order for the physical processes of fuel consumption on the road and on the test bench to be similar, the following condition must be met [4]:

$$G_a \cdot \psi + P_w = G_s \cdot f_s + P_s + P_B , \quad (15)$$

where: ψ - total road resistance coefficient; P_w - aerodynamic air resistance ($P_w = 0.077 \cdot k \cdot F \cdot V_a^2$)

Hence the braking force on the stand

$$P_B = G_a \cdot \Psi + 0.077 k F \cdot V_a^2 - G_s \cdot f_s - P_s \quad (16)$$

$$P_s = (m_s + 2m_z) \cdot j_z, \quad (17)$$

where: m_s - the stand inertial mass, kg; m_z - the wheel inertial mass, kg; j_z - the free deceleration of stand rollers, m/s².

In this case, the mathematical model of fuel consumption on a test stand is presented in the following form:

$$Q_s = \frac{1}{\eta_{is}} [A \cdot i_s + B \cdot i_s^2 \cdot V_a + C \cdot (G \cdot f_s + P_s + P_B)] . \quad (18)$$

Therefore, to determine load conditions, the following data must be entered:

- the average travel speeds;
- the average total drag coefficient values;
- the air drag coefficient;
- the total drag coefficient on the test bench.

3. Conclusion:

The developed methodology eliminates the need for labor-intensive road testing, which requires specialized testing sites, and offers a number of advantages: it ensures an individual approach to each vehicle when rationing fuel, the ability to create route standards, independence of testing from weather conditions, and the ability to promptly change standards when vehicle operating conditions change based on a comprehensive criterion — average speed.

The presented algorithm for assessing vehicle energy parameters is applicable to experimental measurements of fuel consumption in vehicles with multipoint injection. Furthermore, this methodology allows for the calculation or adjustment of baseline fuel consumption standards for transportation, taking into account actual vehicle operating conditions.

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ЗАПИСВАНЕ НА ТЕМПЕРАТУРАТА НА ВЪЗДУХА В ПРОЦЕДУРАТА ЗА РАЗПРЕДЕЛЕНИЕ НА ГОРИВОТО ЗА МОТОРНИ ПРЕВОЗНИ СРЕДСТВА

RECORDING AIR TEMPERATURE IN THE FUEL RATIONING PROCEDURE FOR MOTOR VEHICLES

Sergey Krivoschapov¹, Yuriy Gorbik²

Abstract. The influence of air temperature on vehicle fuel consumption is examined. Methods for obtaining air temperature information from the internet by interacting with servers via APIs or by parsing HTML pages are proposed. A database structure for storing air temperature information is developed for subsequent adjustment of fuel consumption in travel documentation. Digital and analog sensors for measuring vehicle temperature are analyzed. A microcontroller-based measuring system structure is proposed that allows for recording and storing current temperature information. The possibility of remotely monitoring a vehicle en route using GPS and GSM-GPRS network technologies is explored. Conclusions are drawn and avenues for further research are outlined.

Keywords: fuel efficiency, vehicle, fuel consumption, operating conditions, air temperature, sensors, weather API, parsing.

1. Introduction:

The overall efficiency of a vehicle's operation can be considered from financial, technical, energy and environmental perspectives. Financial efficiency is assessed through the cost per unit of vehicle mileage, which takes into account fixed and variable costs. One of the cost items is fuel costs, which account for 40-60% of all costs [1]. Technical efficiency is assessed by reliability indicators, which are also related to the distribution of energy in the car. In the work [2] a relationship was established between the durability indicator - the vehicle's service life and fuel consumption. Energy efficiency is assessed by specific energy consumption, while environmental efficiency is determined by the volume of emissions (CO₂ release per unit distance). Energy consumption and emissions of harmful substances directly depend on fuel consumption. For every liter of gasoline, 2.3...2.4 kg of CO₂ is released, and for every liter of diesel fuel, 2.6...2.7 kg of CO₂ is released.

The performance and efficiency of vehicles also depend on the operating conditions. There are four groups of operating conditions for road transport vehicles [3]: road conditions, transport conditions, atmospheric and climatic conditions, and work culture. Atmospheric and climatic conditions are difficult to influence, but they can and should be recorded. The main parameters of atmospheric and climatic conditions are temperature and pressure, humidity, and air dustiness. Temperature significantly affects vehicle fuel consumption, so this parameter is the focus of this study.

2. Exhibition:

Atmospheric air temperature affects the vehicle from different directions.

Firstly, internal combustion engines require air for fuel combustion. Air is mixed with fuel, and the resulting mixture is injected into the cylinders, where it burns, releasing the necessary energy. Changes in air temperature in the intake system affect the excess air ratio α and the indicated engine efficiency η_i . For a gasoline engine, the relationship between these coefficients is noted [2]: $\eta_i = 0.32 \cdot \alpha$. In turn:

$$\alpha = \alpha_0 \cdot \sqrt{\frac{T_0}{T}}, \quad (1)$$

where α_0 - excess air coefficient under normal conditions; T_0 - air temperature under normal conditions ($T_0 = 293$ °K); T - current air temperature, °K.

Meaning $\sqrt{T/T_0}$ is indicated by a separate coefficient:

$K_m = \sqrt{T/T_0} = \sqrt{(273+t)/273} \approx (0.96 + 0.0017 \cdot t)$, which shows the effect of temperature on the quality of the fuel and air mixture.

Secondary, air exerts resistance as a car moves along the road. Temperature changes air density, which changes the aerodynamic drag coefficient. From the Mendeleev-Clapeyron equation, air density: $\rho = \frac{P \cdot M}{R \cdot T}$, where P - absolute pressure, Pa; M - molecular weight ($M = 0.029$ kg/mol); R - universal gas constant ($R = 8.314$ J/(mol·K)); absolute temperature, °K.

$$\text{Then: } \rho = \frac{101325 \cdot 0.029}{8.314 \cdot (273+t)} = \frac{353.4}{273+t}.$$

Third, temperature changes the density of the fuel. The following formula is used: $\rho_t = \rho_{20} - \gamma \cdot (t - 20)$, where ρ_{20} - fuel density at standard temperature +20 °K; γ - coefficient of volumetric expansion. For gasoline: $\rho_{20} = 0.76$ g/cm³ and $\gamma = 0.0011 \dots 0.00125$ 1/°C.

Fourth, air interacts with all vehicle components and systems. Changes in air temperature lead to changes in the temperature of these components, which can alter some of their properties. Engine and transmission lubricants, the elastic and absorbent properties of tires, and suspension components are all affected by temperature.

The overall impact is taken into account by the following coefficient: $K_a = (1.05 - 0.007 \cdot t)$.

Changes in ambient temperature will affect the energy transfer from the engine to the vehicle's wheels. This will clearly alter the vehicle's fuel-energy and environmental performance. In [1], fuel consumption is adjusted using a general temperature coefficient, which can be determined using the formula: $K = K_m \cdot K_p \cdot K_a \approx (1 - 0.005 \cdot t)$.

In the recommendations [4], the linear fuel consumption is adjusted for air temperature. At sub-zero temperatures (below 0 °C), fuel consumption increases by 2 % for every 5 degrees. The maximum increase is 12 % at temperatures below -25 °C. The increase in fuel consumption for windshield heating in the temperature range of +5 °C and below is taken into account. Fuel consumption also increases at temperatures above +20 °C for cooling the vehicle's interior when using the air conditioner or climate control system..

The use of correction factors that increase fuel consumption due to air temperature necessitates temperature recording at motor transport enterprises. Temperature values are established based on data from the Hydrometeorological Center, the State Service, other reliable official sources, or based on the results of our own measurements.

The internet offers numerous resources providing temperature data [5]. On these servers, you need to select a region and city. The page will then display the current temperature, along with additional information such as time, wind speed, atmospheric pressure, humidity, precipitation, and more. Manually tracking temperature is not practical.

It makes sense to automate the process of obtaining weather data from the internet. There are several programmatic methods for obtaining temperature information in a given region. These include requesting an HTML page from the server, parsing it, and extracting the necessary data; interacting with the server via an API (Application Programming Interface); or receiving files in CSV or other format from the server.

The API access algorithm consists of four steps: selecting a data provider, registering to receive a token (API key), constructing a URL request, and processing the resulting response in JSON (JavaScript Object Notation) or XML (eXtensible Markup Language) format.

Table 1 provides a comparative overview of service providers providing weather information via API. To select the optimal weather API, it is necessary to compare them based on four critical criteria: cost (free limits), coordinate accuracy, forecast depth, and ease of integration.

Table 1. Weather API Comparison Chart

Provider	Free plan (Limits)	Peculiarity	Coordinate format / Geo	Data source	Link
Open-Meteo	Up to 10,000 requests/day	No API key or registration required	Strictly according to lat / lon	National weather services (DWD, NOAA, ECMWF)	https://open-meteo.com/
OpenWeatherMap	60 requests/minutes (Current)	Huge community, plugins	City, ID, lat / lon	Own model + radars	https://openweathermap.org/
WeatherAPI	Up to 1,000,000 requests/month	Sports and astro-weather	City, IP-address, Zip- code, lat / lon	Global and local models	https://www.Weatherapi.com/
Visual Crossing	Up to 1,000 requests/month	The best free archive	City, lat / lon	Historical stations, NOAA	https://www.Visualcrossing.com/
Yandex.Weather	Up to 50,000 requests/month (Test)	Hyperlocality in the CIS	Strictly according to lat / lon	Meteum model + AI + user data	https://yandex.ru/
Tomorrow	Up to 500 requests/month	Weather on Routes mode	Strictly according to lat / lon	Own data from satellites	https://www.tomorrow.io/

The Open-Meteo service is completely free for non-commercial use, requires no registration or API key, provides detailed forecasts (up to 15 days), updates hourly, and accepts data only in latitude/longitude coordinates.

The OpenWeatherMap service offers a free current weather limit and a basic 5-day forecast, updated minute by minute.

The WeatherAPI provider has a large observation history database, a query builder, and an API key, with the ability to upload CSV files through the dashboard.

Yandex.Weather provides point forecasts based on a coordinate grid, focusing on the CIS countries.

The Tomorrow.io service allows you to pass an array of coordinates (a logistics route track) to the API, and the server will return a weather forecast for each point along the route, taking into account the travel time.

To work with weather servers via APIs in Python, libraries of two types are used: HTTP clients (for sending direct requests to any API) and specialized SDKs (ready-made libraries from weather service developers or the community). Examples of general-purpose libraries include requests, httpx, and aiohttp. Specialized libraries are provided for Tomorrow.io (tomorrow-io), for working with Open-Meteo (openmeteo-requests), and for OpenWeatherMap (pyowm).

To work with JSON data in Python, built-in tools (json) are used for simple queries, while pydantic, pandas, orjson, or ujson are used for complex queries. Some servers may receive data in XML format. In this case, Python can use the standard library: xml.etree.ElementTree, as well as external libraries: lxml or xmldict.

To work with CSV files in Python, the csv library is used for simple queries, while pandas or polars are used for complex queries.

If a website doesn't have an official API, weather data can be retrieved from the web page's code. To parse HTML pages in Python, use the BeautifulSoup4 (bs4) library for simple requests, and lxml.html or Selectolax for complex ones. For dynamic HTML (using JavaScript/React/Vue), where data is loaded using JavaScript (SPA technology), use browser automation tools such as Playwright or Selenium.

The received weather data is entered into a database, where it is accumulated for further processing. The database structure is shown in Figure 1.

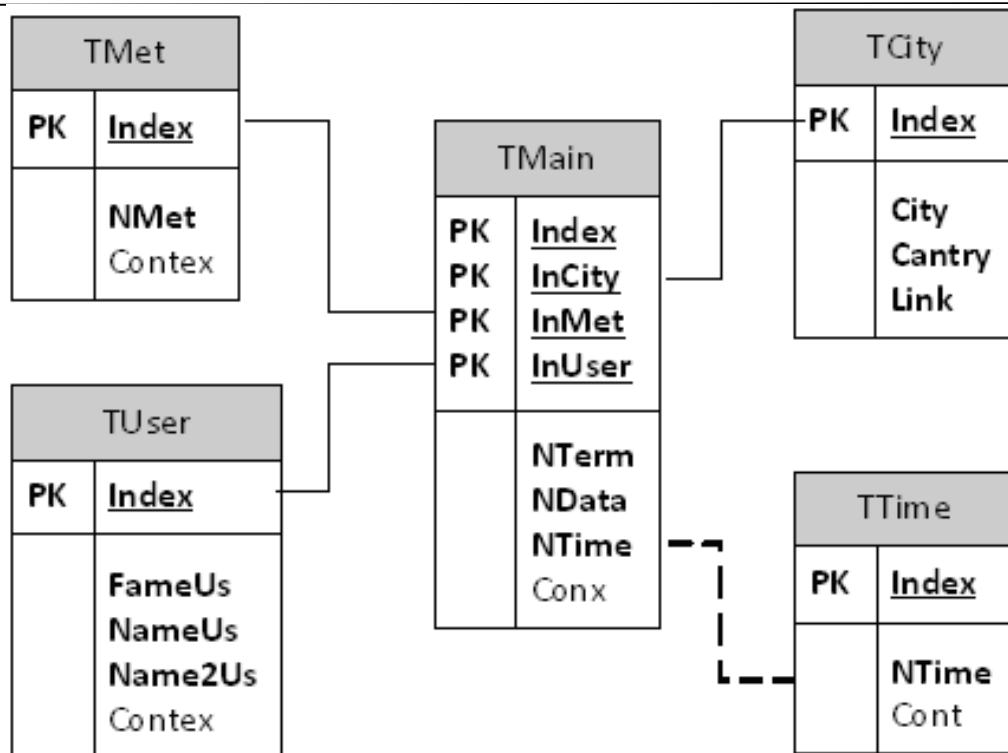


Fig. 1. Structure of the data base for temperature logging

In addition to the temperature value, the database also records the date, time, and location of the update. The temperature database is linked to the company's overall process automation system for the automobile transport enterprise. Each temperature record is linked to a specific vehicle, its travel documentation, and its route. Temperature information is used in processing primary documentation. Standard fuel consumption is adjusted based on temperature.

Temperature information can be obtained not only from the internet, but also by direct measurement. It is advisable to measure temperature in an area as close as possible to the vehicle's location. Ideally, on the vehicle itself during operation.

It is proposed to install temperature measurement sensors on the vehicle, as well as storage facilities for the obtained information. The measurement system is proposed to be controlled using an ATmega328P microcontroller [6]. Arduino UNO breadboards are built on this microcontroller, which facilitates the design and debugging process [7].

There are a wide variety of temperature sensors available, varying in accuracy, connection interface, and measurement range. Table 2 lists some common air temperature sensors.

Table 1. Comparison of air sensors

Sensor	Accuracy	Interface	Measurements
DHT11	±2.0 °C	1-Wire (Data)	Temperature + humidity (low accuracy)
DHT22	±0.5 °C	1-Wire (Data)	Temperature + humidity (low accuracy)
BMP280	±0.5 °C	I ² C / SPI	Temperature + pressure
BME280	±0.5 °C	I ² C / SPI	Temperature + humidity + pressure (low accuracy)
DS18B20	±0.5 °C	1-Wire (Dallas)	Temperature
LM35	±0.75 °C	Analog	Temperature
TMP36	±1.0 °C	Analog	Temperature
NTC-thermistor	±1.0 °C	Analog	Temperature

The DS18B20 digital temperature sensor [8] has a 1-Wire interface. The DS18B20 chip can be locally powered from 3.0 V to 5.5 V or powered via a 1-wire data line. The DS18B20 temperature sensor is connected to the board via three pins: the VDD pin of the sensor is connected to the 5V pin on the board, the GND pins are combined, and the DQ pin of the sensor is connected to a digital pin on the board. A 4.7 k Ω resistor must be connected between the DQ and VDD pins of the sensor.

Each DS18B20 sensor has unique built-in addressing. This allows multiple sensors to be connected to a single line. The DHT11 and DHT22 sensors [9] also transmit data via the 1-Wire interface, but they do not have unique addressing, so only one sensor can be connected to a single line. However, the DHT11 and DHT22 sensors can transmit humidity data in addition to temperature. The DHT22 sensor offers greater accuracy and cost than the DHT11.

The BMP280 and BME280 sensors use different interfaces: I2C or SPI. The ATmega328P microcontroller supports both of these interfaces at the hardware level. The BME280 sensor measures temperature, pressure, and humidity, while the BMP280 is a simplified version without humidity measurement.

The TMP36 [10] and LM35 [11] analog temperature sensors produce a continuous voltage signal at their output. This signal must be converted from analog to digital form. The ATmega328P microcontroller has 8 ADC input lines. The voltage on the sensor changes linearly: 10 mV equals 1 °C.

To convert voltage to temperature, the dependence is used:

$$T = (V - 500) / 10, \quad (2)$$

where: T - temperature, °C; V - voltage at the middle connector of the sensor, mV.

The analog voltage value will be in the range from 0 to 1.75 V.

In the ATmega328P microcontroller program, which is loaded into the board, the port is initialized. In the loop, it is necessary to enter data from the analog ADC pin and convert 10-bit analog values into voltage according to the formula:

$$V = N \cdot (5000 / 1024), \quad (2)$$

where: V - voltage at the middle sensor connector, mV; 5000 - reference voltage, mV (5V); 1024 - maximum ADC value.

The difference between the TMP36 [10] and LM35 [11] sensors is that the LM35 sensor starts counting from 0 °C and at 25 °C the sensor generates a voltage of 250 mV, while the TMP36 sensor counts from -50 °C and at 25 °C the sensor output will be 750 mV. A negative voltage cannot be applied to the ADC pins of the ATmega328P microcontroller, but the LM35 sensor output can produce such a voltage. If the LM35 sensor is to operate in a temperature range where the signal from the LM35 sensor is negative, then this signal will need to be shifted to the positive side using resistors. In this case, it is necessary to take this into account in the ATmega328P microcontroller program using correction factors.

An NTC-thermistor can be thought of as a resistor, since temperature changes the resistance. The voltage divider method is most often used to measure resistance with an Arduino. An unknown resistor is connected in series with a reference resistor. The Arduino applies voltage, measures the voltage drop across the unknown resistor via an analog input, and calculates its resistance using Ohm's law.

The ATmega328P microcontroller stores temperature data from the sensor on a USB flash drive via a USB Host module (e.g., CH376S or MAX3241E USB Host Shield). Data from the ATmega328P to the CH376S is transmitted via the TXD and RXD pins. The supply voltage is 5V. Time and temperature data, as well as the vehicle code, are written line by line to a file on the flash drive.

The ATmega328P does not have an integrated real-time clock. Therefore, an external real-time module is connected to the microcontroller: DS1307, DS3231 (with temperature compensation, very accurate), or DS1302. I2C or SPI protocols are used for data exchange.

The general schematic diagram for connecting all sensors and modules to the Nano V3.0 AtMega328P breadboard is shown in Figure 2. Here in the figure: the blue color indicates the direction of power supply voltage transfer; the red color indicates the transfer of information in the simplified diagram; the green color indicates the direction of information transfer in the expanded diagram.

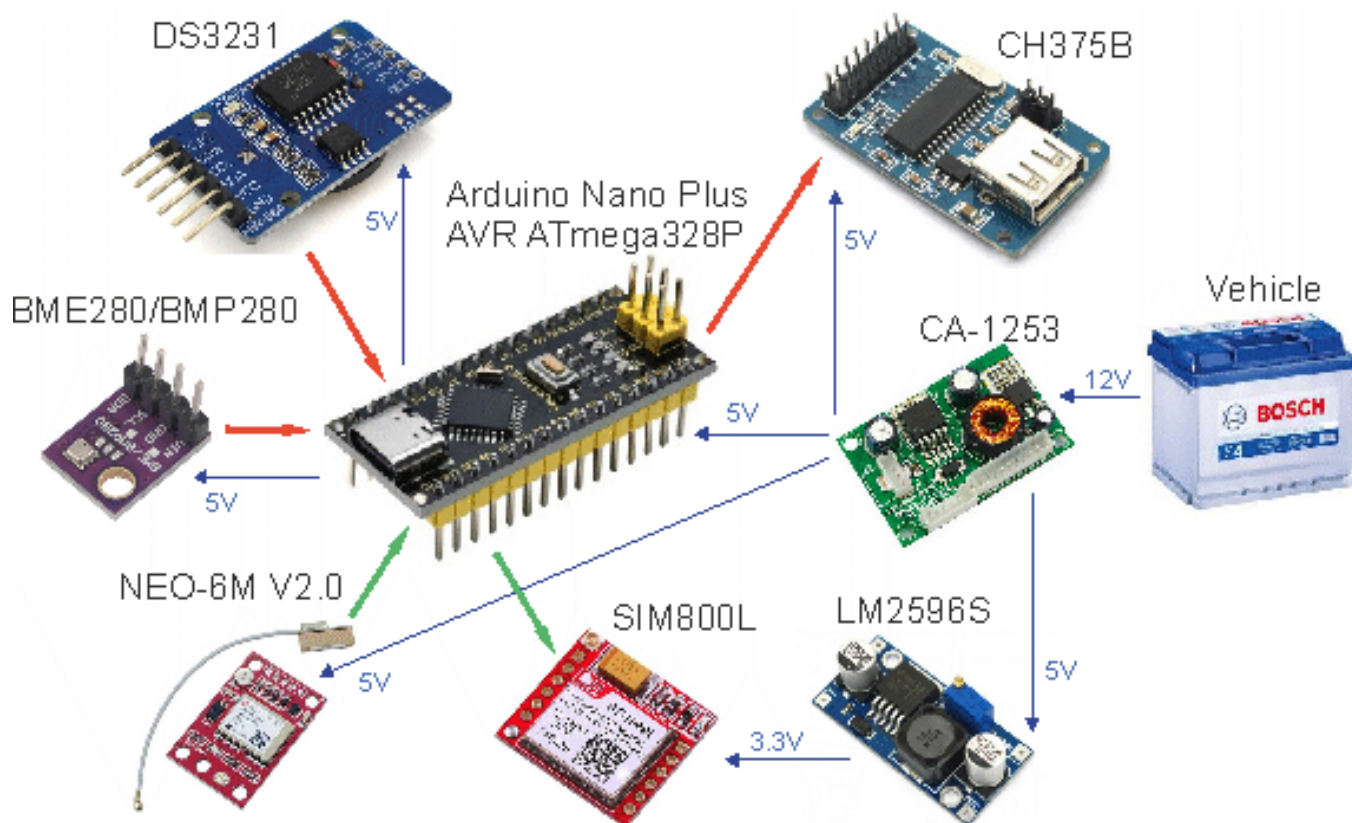


Fig. 2. Diagram of a measuring complex for recording temperature on a vehicle.

Periodically, for example, at the end of a shift, the operator retrieves the USB flash drive from the vehicle. This data is sent to the computer, where it is transferred to the temperature recording database for subsequent processing and fuel consumption adjustments.

To determine the location of the vehicle on the route, it is advisable to connect a GPS (Global Positioning System) module to the system. The NEO-6 u-blox 6 GPS Modules [12] was used with the ATmega328P microcontroller. Data transfer occurs via the UART (RX/TX) software interface. The module requires up to 10-15 minutes for the initial satellite acquisition. The minimum satellite acquisition speed is 1 second. Data from the NEO-6 module will also be written to a file on the USB flash drive, after the time and temperature readings.

Wireless data transmission via GSM (Groupe Spécial Mobile) channels can be added to the vehicle temperature measurement system. In this case, the SIM800L [13] GSM-GPRS module or any similar module should be connected to the system. This module communicates with the ATmega328P microcontroller via RX/TX channels. The SIM800L requires a power supply voltage of 3.7–4.2 V, while the ATmega328P is designed for a reference voltage of 5 V. In addition, the SIM800L consumes a peak current of up to 2 A. Therefore, the SIM800L cannot be powered from the ATmega328P; i.e., the SIM800L must have its own power source. A step-down converter (e.g., LM2596) can be used. The RX and TX signals of the SIM800L also operate with a voltage of 3.3 V. For reliability of the circuit, it is recommended to install a voltage

divider on 1 kOhm and 2 kOhm resistors between the TX pin of the Arduino and the RX pin of the SIM800L. Although the communication interface: UART (Serial) allows operation with a reference voltage of 3.3 V.

Using a GSM/GPRS module in the measuring system allows for real-time temperature data transmission. However, a server must be installed on-site or in the cloud to receive this data.

3. Conclusion:

Information received from the Internet or a temperature sensor installed on the vehicle, transmitted via a USB flash drive or via GSM-GPRS cellular communication channels, allows increasing the accuracy and reliability of recording the atmospheric and climatic conditions of operation of transport vehicles.

An automated data processing system should create conditions for more accurate determination of the actual and standard fuel consumption values on a route.

The vehicle temperature recording system can be integrated with the general system for remote monitoring of the transport process and the system for assessing the technical condition of the vehicle.

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7. Software: Download the Arduino IDE. URL: <https://www.arduino.cc/en/Main/Software>;
8. DS18B20 - Programmable Resolution 1-Wire Digital Thermometer. Maxim Integrated. URL: <https://datasheets.maximintegrated.com/en/ds/DS18B20.pdf>;
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11. BMP280 Digital Pressure Sensor. URL: <https://www.bosch-sensortec.com/media/boschsensortec/downloads/datasheets/bst-bmp280-ds001.pdf>
12. NEO-6 u-blox 6 GPS Modules. Data Sheet. URL: https://content.u-blox.com/sites/default/files/products/documents/NEO-6_DataSheet_%28GPS.G6-HW-09005%29.pdf
13. GSM-GPRS SIM800L URL: https://www.makerhero.com/img/files/download/Datasheet_SIM800L.pdf;

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ИССЛЕДОВАНИЕ КОМПЛЕКСНОГО ВЛИЯНИЯ ПАРАМЕТРОВ РЕЖИМА ПРИТИРКИ НА КАЧЕСТВО ОБРАБОТАННЫХ ПОВЕРХНОСТЕЙ СИЛИКАТНЫМ АБРАЗИВНЫМ СОСТАВОМ

STUDY OF THE COMPLEX EFFECT OF GRINDING PARAMETERS ON THE QUALITY OF SURFACES PROCESSED WITH A SILICATE ABRASIVE COMPOSITION

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Abstract. This study confirms that the high surface quality achieved after lapping with a silicate abrasive composition is due to the formation of silicon oxide. Under the action of abrasive particles, SiO_2 penetrates the depressions of the workpiece surface and bonds to it. With repeated exposure to the particles, SiO_2 tightly “packs” the deep depressions of the surface irregularities. It has been established that surface roughness during lapping with a silicate compound and “KT” paste decreases initially with an increase in specific pressure and the speed of the sample’s reciprocating-rotational motion, but begins to increase after a certain “optimum” due to deep scratches and a lack of a lubricating layer. It has been confirmed that the lapping productivity with a silicate abrasive composition is 1.5 times higher than when using “KT” paste. It has been confirmed that when lapping the surfaces of a friction pair with a silicate composition, metal removal is 1.4 times less than when using “KT” paste.

Keywords: lapping, silicate abrasive composition, research results, valve, valve seat, surface finish quality.

1. Введение:

Под технологическими факторами процесса притирки понимается совокупность параметров и условий, влияющих на ход процесса образования и удаления микронеровностей поверхности при взаимодействии абразивных зерен с материалом детали. От этих факторов зависит интенсивность снятия материала, качество обработанной поверхности и износостойкость сопряжения.

К основным технологическим факторам притирки относятся:

1. Режимные параметры:

- усилие прижима клапана к седлу. Небольшое (слабый пружинный прижим, менее 0,01 МПа) – низкая производительность процесса. Чрезмерное усилие (более 0,05 МПа) недопустимо – оно вызывает ускоренный износ седла и клапана, возникают задиры, детали перегреваются. Оптимальное давление обеспечивает равномерный контакт без задириров;

- скорость относительного движения (возвратно-вращательная). Высокие скорости (более 400 мин⁻¹) запрещены, поскольку они вызывают перегрев и повреждение поверхности в виде микротрещин. Низкие обороты (менее 150 мин⁻¹) – низкая производительность процесса;

- время притирки. Продолжительность притирки зависит от состояния поверхности клапана и седла. Короткий процесс притирки не обеспечивает герметичности соединения. Процесс заканчивается, когда на поверхности появляется равномерный матовый пояс шириной 1,5-2 мм.

2. Характеристики абразивного материала:

- зернистость абразива (размер частиц). Грубая (крупнозернистая) – применяется в начале для устранения неровностей. Тонкая (мелкозернистая) – используется для окончательной доводки и обеспечения плотного прилегания.

- твердость и форма зерен;

- концентрация абразивных частиц в пасте или суспензии.

3. Состав и свойства рабочей абразивной пасты:

- тип смазочно-абразивной смеси (паста, суспензия);

- вязкость, смазывающие свойства, способность удерживать абразив.

4. Температурные условия процесса:

- выделение тепла в зоне обработки;
- условия охлаждения и смазки.

5. Свойства материала детали:

- твердость и микроструктура;
- склонность к пластической деформации или растрескиванию.

Влияние характеристик абразивного материала было исследовано в предыдущих работах [1, 2].

В работе [1] исследовано влияние зернистости абразива силикатного состава на время формирования поверхностного слоя и количество снимаемого металла. Доказано, что высокое качество поверхности после обработки силикатным составом обусловлено появлением в процессе притирки аморфного кремнезема (SiO_2), который усиливает действие абразивных зерен и приводит к уменьшению царапающего действия абразива.

В работе [2] установлено, что при увеличении твердости абразива силикатного состава время формирования поверхностного слоя находится в пределах 4-6 минут, шероховатость притертой поверхности – в пределах 0,11-0,15 мкм. Доказано, что высокое качество поверхности после притирки силикатной пастой формирует поверхность, близкую к оптимальной, за счет изменения структуры поверхностного слоя притертой поверхности и насыщения ее оксидом кремния. Химический анализ подтвердил, что абразивное зерно, использованное в силикатном составе при обработке поверхности, имеет округлую форму и покрыто пленкой SiO_2 . Выявлено, что при притирке поверхностей силикатной пастой с абразивом электрокорунда снятие металла в 1,35-1,5 раза меньше, чем при обработке пастой «КТ».

Таким образом, в работе предлагается провести исследование влияния параметров режима притирки на качество поверхностей, обработанных силикатным абразивным составом.

Целью работы является определение оптимальных значений параметров режима притирки силикатным абразивным составом, которые обеспечивают максимальное качество обработанной поверхности клапанного соединения по критерию шероховатости поверхности (R_a).

Объектами исследования являются рабочие поверхности обрабатываемых деталей. В нашем случае это образец, который модулирует клапан, и контрольный образец – седло головки цилиндров автомобильного двигателя.

Предмет исследования: влияние параметров режима притирки силикатным абразивным составом на шероховатость обработанных поверхностей.

2. Изложение:

Исследования проводились на специальной установке, позволяющей с высокой точностью воспроизвести процесс притирки образца и контрольного образца абразивной пастой в лабораторных условиях. Установка смонтирована на станине универсального станка ОПР-1841А, предназначенного для притирки клапанов автотракторных двигателей [3].

Цель исследования – установить, как давление (P) на образец, относительная скорость (V) вращения образца и продолжительность (t) притирки влияют на шероховатость R_a обрабатываемой поверхности.

В качестве образца использовали конус из стали 40Х (HRC 48) диаметром 23 мм, шириной 15 мм. Контрольным образцом служило кольцо из чугуна СЧ 18-36 (HRC 32) диаметром 50 мм, шириной 10 мм с шириной рабочей поверхности 3 мм. Выбор материала данных пар трения был обусловлен его использованием для изготовления тарелки клапана и седла клапана. Исходная шероховатость поверхностей трения во всех образцах была одинаковой и составляла $R_a = 1,25$ мкм [3].

Для проведения исследования использовали состав силикатной пасты с абразивом карбид кремния зеленый зернистостью 50/40 мкм, что рекомендовано в работе [2].

Для сравнения работоспособности силикатного абразивного состава использовали существующий состав пасты «КТ» с абразивом зернистостью 50/40, который применяется в настоящее время для притирки клапанов автомобильных двигателей. Представленные два состава отличаются тем, что силикатный состав изготовлен на водной основе, а состав пасты «КТ» – на основе индустриального масла.

При определении влияния времени (t) притирки на качество обработанной поверхности (R_a) давление (P) на образец составляло 0,03 МПа, скорость вращения ($V_{в-п}$) образца составляла 0,4 м/с, время притирки (t) варьировалось от 120 с до 600 с.

Оценка проводилась по качеству обрабатываемой поверхности и съему металла. Снятие металла определяли весовым методом, то есть взвешиванием на аналитических весах типа AS 520.R2 «Radwag» с точностью $\pm 1 \cdot 10^{-4}$ г (класс точности согласно ДСТУ EN 45501-1) до и после испытаний. После обработки образцы промывались водой и ацетоном.

С помощью профилографа-профилометра контролировали профиль притираемых поверхностей. Результаты испытаний определяли как среднее арифметическое пяти опытов с погрешностью $\pm 3\%$.

Наилучшую работоспособность имел тот состав пасты, после притирки которой образец и контрообразец имели наименьшую шероховатость, и стабилизация коэффициента трения происходила быстрее. Результаты исследований приведены на рис. 1.

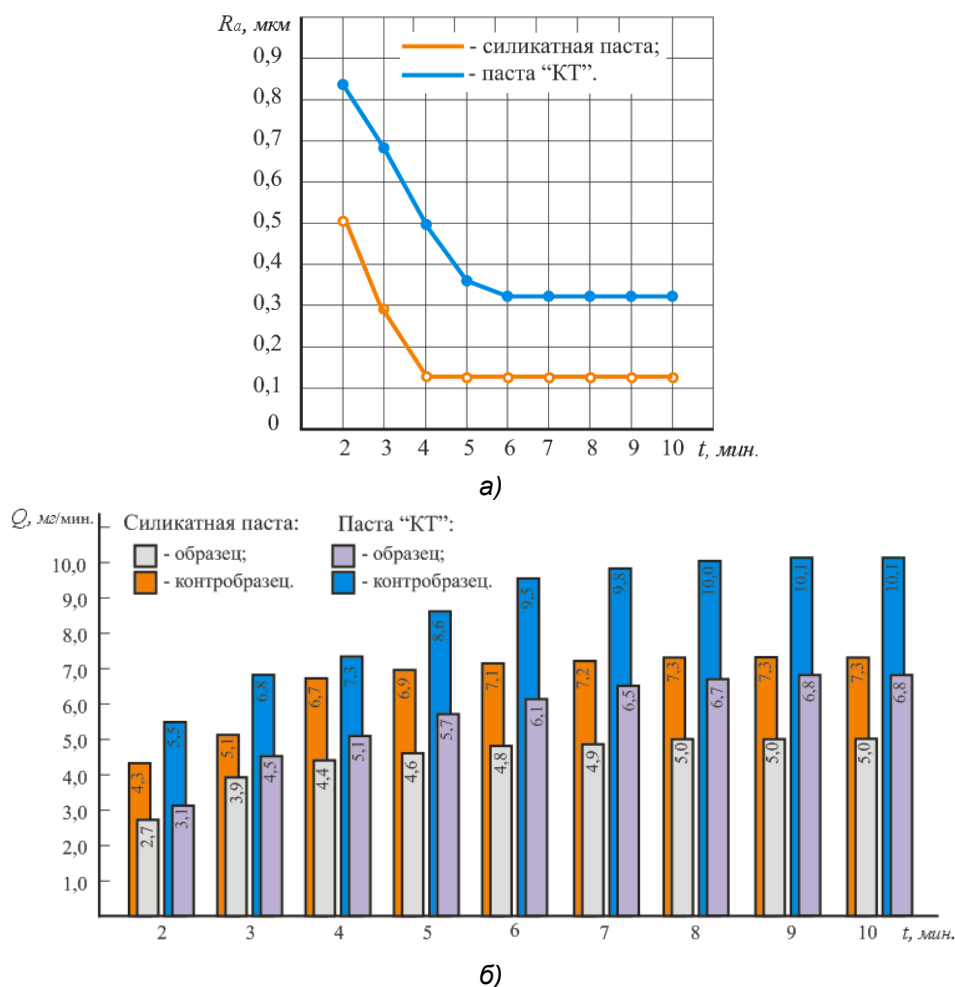


Рис. 1. Зависимость шероховатости поверхности (R_a , мкм) (а) и съема металла (Q , мг/мин.) (б) от времени притирки (t , мин.) силикатным абразивным составом и пастой «КТ» пары трения сталь 40Х (HRC 48) – чугун СЧ 18-36 (HRC 32)

Анализ рис. 1а показал, что при изменении времени притирки силикатным абразивным составом с 2 до 4 минут шероховатость поверхности (R_a) резко изменяется: она уменьшается до $R_a = 0,5$ мкм (время притирки 2 мин.), $R_a = 0,3$ мкм (время притирки 3 мин.) и $R_a = 0,12$ мкм (время притирки 4 мин.). В это время абразивные зерна быстро срезают выступы микронного уровня. После 4-х минут притирки силикатным абразивным составом до 10 минут, шероховатость поверхности не изменяется и остается на уровне $R_a = 0,12$ мкм, что свидетельствует о стабилизации параметра шероховатости R_a и окончании процесса формирования качества поверхностей.

Для подтверждения окончания процесса формирования качества поверхностей при притирке силикатным абразивным составом было принято решение проследить динамику изменения коэффициента трения при притирке силикатным абразивным составом и определить шероховатость обработанной поверхности (R_a) и снятие металла (Q) при этом. Для сравнения результатов, как и в предыдущих экспериментах, использовали существующий состав пасты «КТ». Результаты исследования приведены на рис. 2.

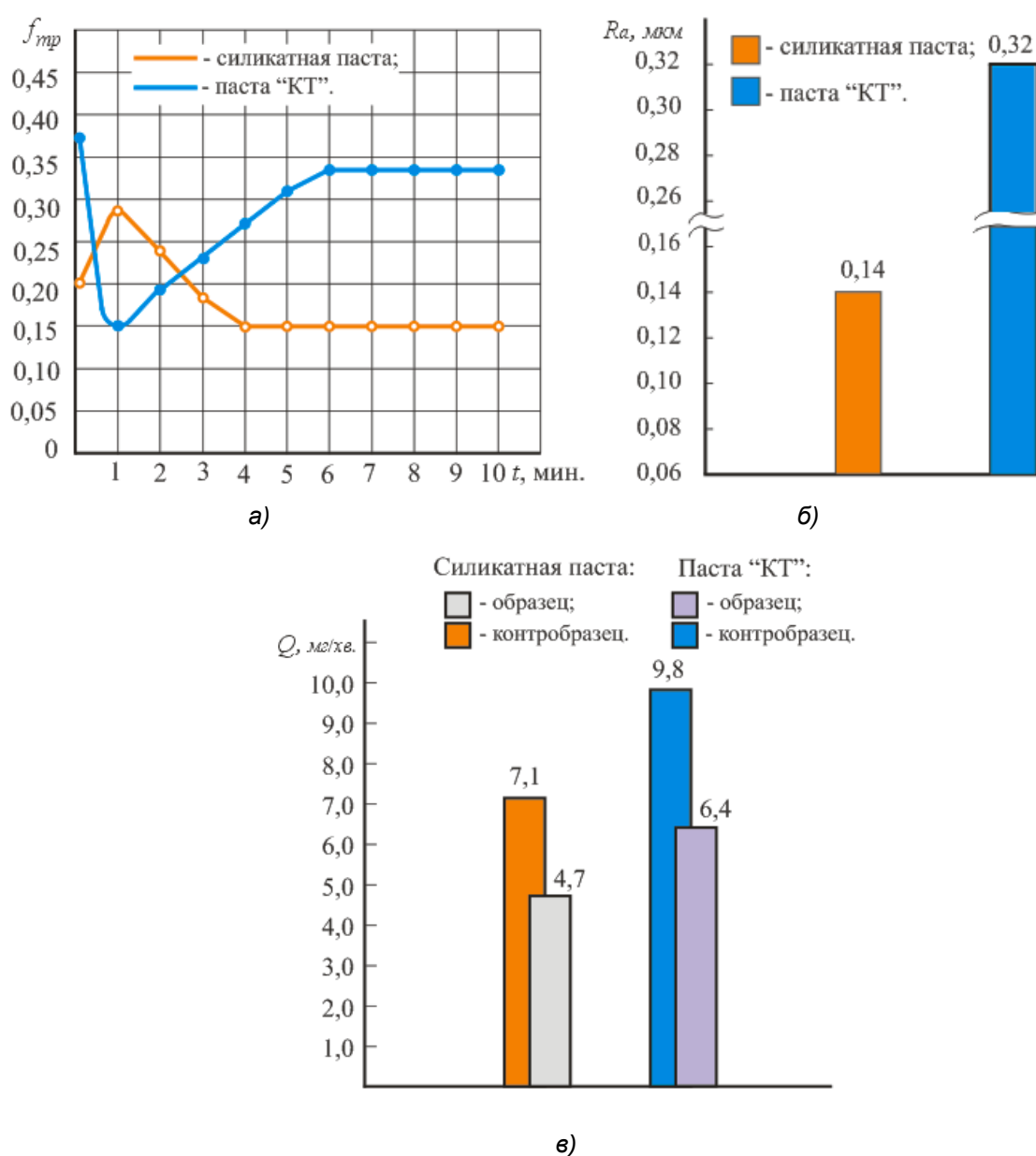


Рис. 2. Изменение коэффициента трения ($f_{тр}$) (а), шероховатости поверхностей (R_a , мкм) (б) и съема металла (Q , мг/мин.) (в) пары трения сталь 40Х (HRC 48) – чугун СЧ 18-36 (HRC 32) при притирке силикатным абразивным составом и пастой «КТ»

По результатам рис. 2а видно, что при притирке силикатным составом в начальный момент (первая минута) происходит увеличение коэффициента трения за счет действия острых кромок абразива. В дальнейшем, возможно, происходит затупление этих кромок и обволакивание абразива оксидом кремния, что приводит к снижению коэффициента трения [1].

Стабилизация коэффициента трения при притирке силикатным составом происходит на четвертой минуте. При этом значение коэффициента трения составляет $f_{тр} = 0,15$. Стабилизация коэффициента трения при притирке указывает на завершение процесса формирования поверхностного слоя и заканчивается на 4-й минуте.

При притирке пастой «КТ» сначала происходит снижение коэффициента трения за счет работы масляной пленки, поскольку состав изготовлен на основе индустриального масла. Начиная со второй минуты притирки, пленка изнашивается и в работу вступают абразивные зерна. Абразивные зерна внедряются в обрабатываемую поверхность и срезают вершины микронеровностей, что приводит к увеличению коэффициента трения. Этот процесс продолжается до шестой минуты (рис. 2а) и коэффициент трения составляет $f_{тр} = 0,34$. Как было указано выше, идет процесс формирования окончательной шероховатости поверхностей трения. Начиная с шестой минуты происходит стабилизация коэффициента трения, и в последующие три минуты притирки он остается неизменным ($f_{тр} = 0,34$) [1].

Анализ рис. 2б показал, что после притирки силикатным абразивным составом шероховатость поверхности составляет $Ra = 0,14$ мкм, а пастой «КТ» – $Ra = 0,32$ мкм.

На рис. 3 приведены фотографии поверхностей и профилограммы поверхностей образцов после притирки абразивными составами.

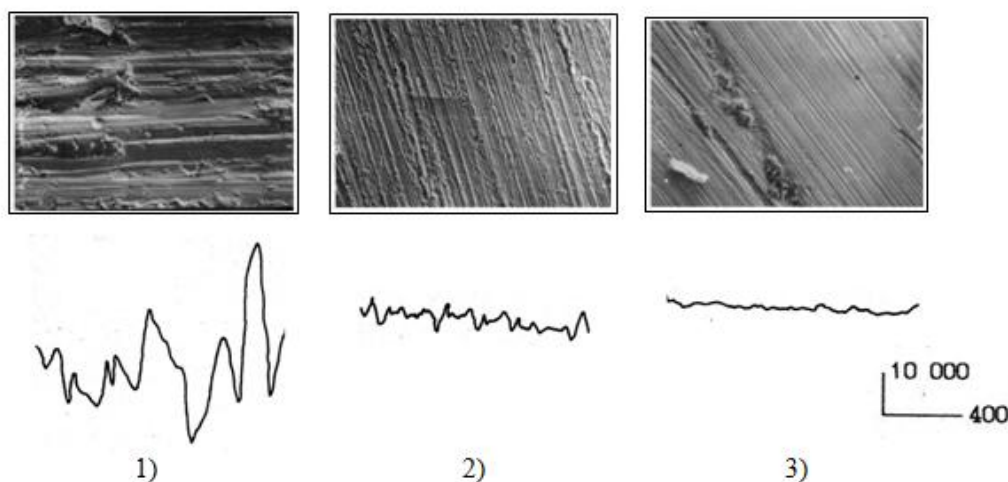


Рис. 3. Фотографии и профилограммы поверхностей образцов после притирки абразивными составами (x400): 1 – исходная; 2 – пастой «КТ»; 3 – силикатной пастой [4]

Высокое качество поверхности после обработки силикатным составом по сравнению с поверхностью, обработанной пастой «КТ», обусловлено тем, что при взаимном перемещении поверхностей происходит резание-царапание абразивными зернами самых высоких вершин неровностей. В точках соприкосновения абразивных частиц происходит повышение температуры, а поскольку вокруг абразива находится SiO_2 , то при попадании последнего в зону соприкосновения происходит образование аморфного кремнезема (SiO_2), который оказывает полирующее действие высокодисперсного, относительно мягкого абразива. И это приводит к высокому качеству поверхности после притирки силикатным составом [4].

Анализируя рис. 2в, видно, что после притирки образцов силикатным составом снятие металла составляет $Q = 4,7$ мг/мин для образца и $Q = 7,1$ мг/мин для контролбразца. При притирке пастой «КТ» съем металла составляет $Q = 9,8$ мг/мин для контролбразца и $Q = 6,4$ мг/мин для образца. На основании такого результата можно отметить, что при притирке поверхности силикатным составом съем металла Q в 1,4 раза меньше, чем при обработке пастой «КТ». Таким образом, при назначении припуска на обработку необходимо это учитывать.

Итак, можно отметить, что при притирке силикатным составом достаточно четырех минут, а пастой «КТ» – 6 минут. В результате этого можно утверждать, что производительность притирки силикатной пастой в 1,5 раза выше, чем при обработке пастой «КТ».

Анализируя рис. 1б, можно сказать, что при увеличении времени при притирке силикатным составом от $t = 2$ мин до $t = 4$ мин снятие металла значительно увеличивается и на $t = 4$ мин составляет $Q = 4,4$ мг/мин для образца и $Q = 6,7$ мг/мин для контролбразца. При притирке пастой «КТ» от $t = 2$ мин до $t = 6$ мин съем металла увеличивается и на $t = 6$ мин составляет $Q = 6,1$ мг/мин для образца и $Q = 9,5$ мг/мин для контролбразца. По нашему мнению, это происходит в результате того, что паста свежая, абразивные зерна в начале притирки острые, и в это время срезаются наиболее выступающие микронеровности.

При увеличении времени при притирке силикатным составом до $t = 5$ мин, а пастой «КТ» до $t = 7$ мин поверхность становится более ровной, высота микронеровностей снижается, количество «выступов», которые абразив может срезать, резко уменьшается, и снятие металла увеличивается незначительно. При дальнейшем увеличении времени притирки до $t = 7$ мин абразивные зерна притирочных составов закругляются, снятие металла становится минимальным или практически прекращается. Поэтому длительная притирка не приводит к повышению качества обработанной поверхности, а только снижает эффективность притирки.

По результатам анализа рис. 1б можно утверждать, что при притирке поверхности силикатным составом съем металла Q в 1,4 раза меньше, чем при обработке пастой «КТ».

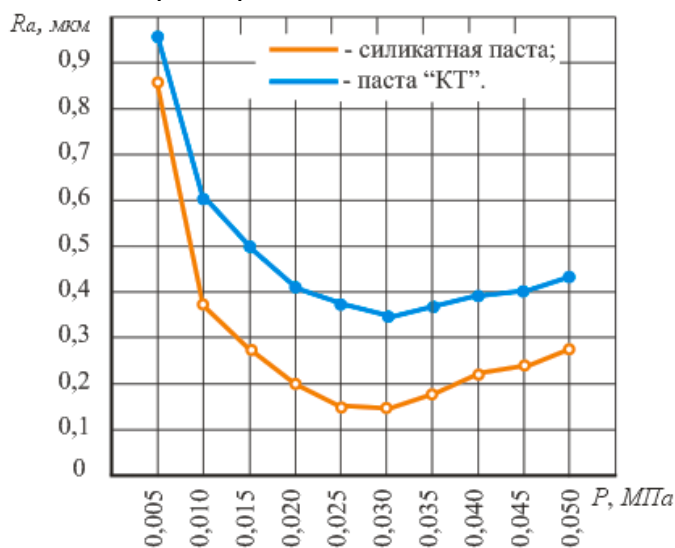
При определении зависимости качества обработанной поверхности (R_a) от удельного давления (P) на образец возвратно-вращательная скорость ($V_{в-в}$) образца составляла 0,4 м/с. Результаты предварительных исследований (рис. 2а) показали, что формирование шероховатости поверхностей (R_a) обработанных силикатным составом достигается на 4-й минуте, а при притирке пастой «КТ» – на 6-й минуте. Поэтому было принято решение о создании одинаковых условий обработки и проведении сравнения результатов притирки в течение 6 минут, то есть время притирки (t) равно – 360 с. Удельное давление (P) на образец в данном исследовании варьировалось от 0,005 МПа до 0,05 МПа. Результаты исследования приведены на рис. 4.

Анализируя рис. 4а, можно сказать, что при удельном давлении на образец $P = 0,005$ МПа зерна абразива при притирке плохо врезаются в поверхность металла и действуют преимущественно как полирующие. Поэтому при таком удельном давлении шероховатость обработанных поверхностей изменяется незначительно. При притирке силикатным составом она достигает $R_a = 0,86$ мкм, при обработке пастой «КТ» – $R_a = 0,95$ мкм.

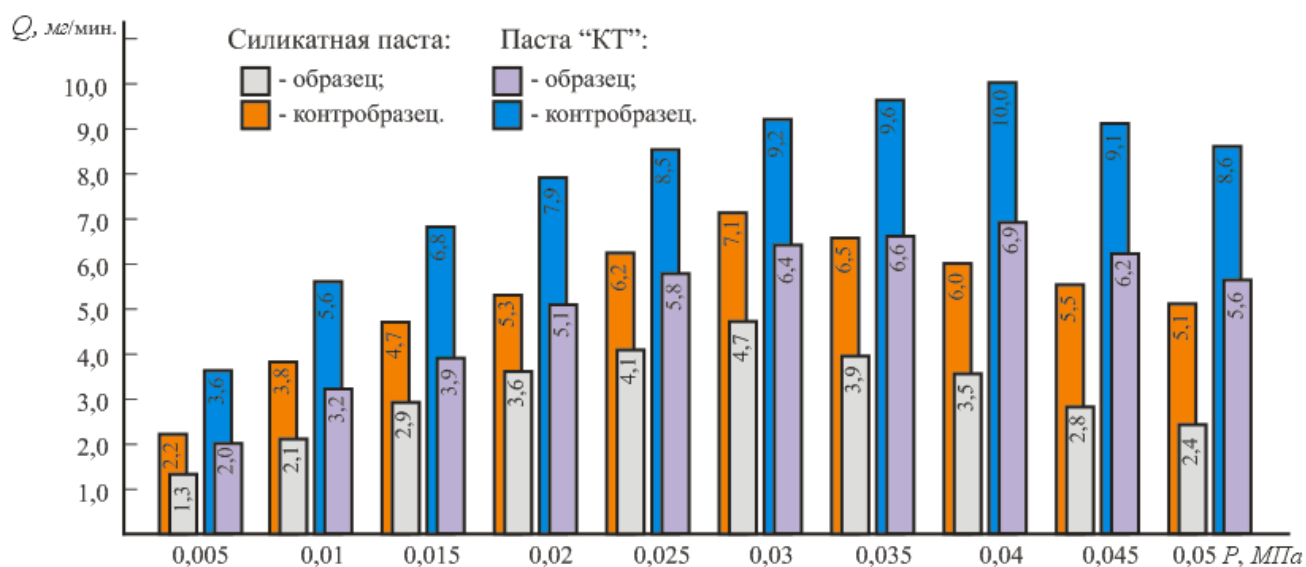
При увеличении удельного давления от 0,01 до 0,03 МПа шероховатость обработанной поверхности улучшается, как при притирке силикатным составом, так и пастой «КТ». При этом процессе зерна абразива вдавливаются настолько, чтобы происходило контролируемое микрорезание и пластическое сглаживание.

При притирке силикатным составом шероховатость поверхности уменьшается от $R_a = 0,38$ мкм до $R_a = 0,15$ мкм, и минимальное значение достигается при удельном давлении $P = 0,025$ -0,03 МПа. При притирке пастой «КТ» шероховатость поверхности уменьшается от $R_a = 0,6$ мкм до $R_a = 0,35$ мкм, и минимальное значение достигается при удельном давлении $P = 0,03$ МПа.

При последующем увеличении удельного давления от 0,03 до 0,05 МПа шероховатость обработанной поверхности увеличивается. При притирке силикатным составом – $R_a = 0,19-0,29$ мкм, при обработке пастой «КТ» – $R_a = 0,38-0,42$ мкм.



а)



б)

Рис. 4. Зависимость шероховатости поверхности (R_a , мкм) (а) и съема металла (Q , мг/мин) (б) от удельного давления (P , МПа) на образце при притирке силикатным абразивным составом и пастой «КТ» пары трения сталь 40Х (HRC 48) – чугун СЧ 18-36 (HRC 32)

Подводя итог анализа рис. 4а, можно отметить, что динамика изменения зависимости одинакова как при притирке силикатным составом, так и пастой «КТ».

Анализ рис. 4б показал, что в процессе притирки при наименьшем удельном давлении на образец ($P = 0,005$ МПа) снятие металла минимально, но абразив может «скользить», оставляя отдельные микроцарапины, что и влияет на шероховатость поверхности. При увеличении удельного давления увеличивается количество активных абразивных зерен в зоне контакта поверхностей и глубина их вдавливания, что приводит к повышению съема металла.

При притирке силикатным составом повышение съема металла происходит при удельном давлении до 0,03 МПа, при обработке пастой «КТ» – при удельном давлении до 0,04 МПа. При таком удельном давлении съем металла для силикатного состава составляет $Q = 4,7$ мг/мин для образца и $Q = 7,1$ мг/мин для контробразца. При притирке пастой «КТ» – $Q = 10,0$ мг/мин для контробразца и $Q = 6,9$ мг/мин для образца (рис. 5б). Видно, что при притирке поверхности силикатным составом снятие металла Q в 1,4 раза меньше, чем при обработке пастой «КТ».

При дальнейшем увеличении удельного давления на образец, по нашему мнению, может происходить разрушение абразивных зерен, абразивные зерна могут вдавливаются слишком глубоко, оставляя глубокие риски, паста при таких условиях может выдавливаться из зоны контакта, в этот момент снятие металла перестает расти и начинает уменьшаться, за счет этого качество поверхности ухудшается.

При определении влияния возвратно-поступательной скорости вращения ($V_{в-в}$) образца на качество обработанной поверхности (R_a) удельное давление (P) на образец составляло 0,03 МПа (при этом учитывали результаты предыдущего исследования), время притирки (t) – 360 с, скорость возвратно-вращательного движения ($V_{в-в}$) образца варьировалась от 0,1 м/с до 0,9 м/с. Результаты исследования приведены на рис. 5.

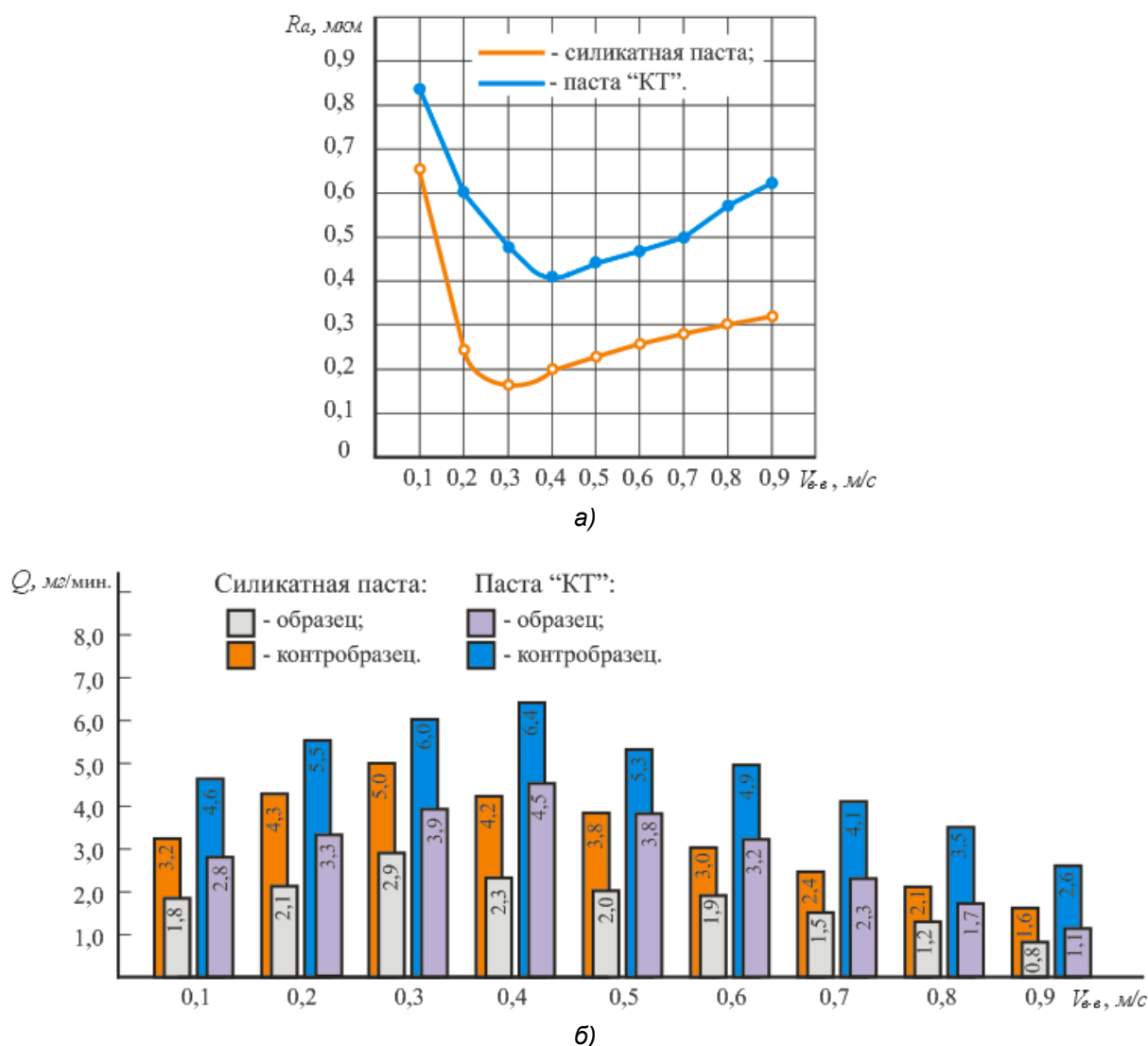


Рис. 5. Зависимость шероховатости поверхности (R_a , мкм) (а) и съема металла (Q , мг/мин) (б) от скорости возвратно-вращательного движения ($V_{в-в}$, м/с) образца при притирке силикатным абразивным составом и пастой «КТ» пары трения сталь 40Х (HRC 48) – чугун СЧ 18-36 (HRC 32)

Анализ результатов (рис. 5а) показал, что при скорости вращения образца $V_{в-в} = 0,1$ м/с шероховатость обработанной поверхности высокая. Для поверхностей, притертых силикатным составом, она составляет $R_a = 0,53$ мкм, а для поверхностей, обработанных пастой «КТ» – $R_a = 0,75$ мкм. При притирке поверхностей с такой скоростью возвратно-вращательного движения образца, по нашему мнению, абразивные зерна работают локально, абразивный состав плохо перемешивается. В результате этого возникают отдельные микроцарапины и «пятна» из-за неравномерного снятия металла. Поэтому шероховатость обработанной поверхности велика из-за появления единичных глубоких рисок.

При увеличении возвратно-вращательной скорости образца шероховатость обработанной поверхности уменьшается. При притирке силикатным составом со скоростью возвратно-вращательного движения образца $V_{в-в} = 0,3$ м/с шероховатость поверхности минимальна и составляет $R_a = 0,18$ мкм (рис. 5а). Минимальная шероховатость обработанной поверхности составляет $R_a = 0,4$ мкм при обработке пастой «КТ» со скоростью возвратно-вращательного движения образца $V_{в-в} = 0,4$ м/с. При таких скоростях обратного-вращательного движения образца, по нашему мнению, движение достаточно интенсивно, чтобы зерна абразива равномерно срезали микронные выступы, паста постоянно перемешивалась, продукты износа отводились.

Дальнейшее увеличение скорости возвратно-вращательного движения образца при притирке силикатным составом от $V_{в-в} = 0,3$ м/с до $V_{в-в} = 0,9$ м/с приводит к увеличению шероховатости обработанной поверхности. При обработке пастой «КТ» это происходит при повышении скорости возвратно-вращательного движения образца от $V_{в-в} = 0,3$ м/с до $V_{в-в} = 0,9$ м/с (рис. 5а). Ухудшение качества обработанной поверхности при высоких возвратно-вращательных скоростях притирки, по нашему мнению, возникает в результате того, что смазочный слой и паста частично выдавливаются из зоны контакта, абразивные зерна начинают действовать более агрессивно. Из-за этого могут образовываться бороздки, повышаться температура, появляется риск возникновения микрорадиов.

Анализируя рис. 5б, можно сказать, что при увеличении скорости вращения при притирке силикатным составом до $V_{в-в} = 0,3$ м/с, а при обработке пастой «КТ» до $V_{в-в} = 0,4$ м/с снятие металла увеличивается. В этом процессе абразивный состав равномерно распределяется и эффективность притирки увеличивается. Максимальный съем металла при притирке силикатным составом составляет $Q = 2,9$ мг/мин. для образца и $Q = 5,0$ мг/мин. для контрольного образца. При притирке пастой «КТ» максимальное снятие металла составляет $Q = 6,4$ мг/мин для контрольного образца и $Q = 4,5$ мг/мин для образца (рис. 5б).

При дальнейшем увеличении скорости вращения притирки снятие металла уменьшается, что приводит к снижению эффективности процесса (рис. 5б). Как и в предыдущих опытах, видно, что при притирке поверхности силикатным составом съем металла Q в 1,4 раза меньше, чем при обработке пастой «КТ».

Подводя итог анализа рис. 2-6, отмечаем, что при притирке силикатным абразивным составом качество обработанной поверхности выше, чем при притирке пастой «КТ». Это объясняется тем, что в процессе притирки силикатным абразивным составом происходит образование оксида кремния (SiO_2). Под действием абразивных частиц SiO_2 попадает в углубления обрабатываемой поверхности и сцепляется с поверхностью. При многократном воздействии частиц происходит плотная «упаковка» SiO_2 глубоких углублений неровностей, что позволяет получить высокое качество поверхности [1].

3. Заключение:

Таким образом, было подтверждено, что высокое качество поверхности после притирки силикатным абразивным составом достигается за счет образования оксида

кремния (SiO_2). Под действием абразивных частиц SiO_2 попадает в углубления обрабатываемой поверхности и сцепляется с ней. При многократном воздействии частиц происходит плотная «упаковка SiO_2 глубоких углублений неровностей. Установлено, что шероховатость поверхности при притирке как силикатным составом, так и пастой «КТ» с увеличением удельного давления в начальный момент уменьшается, но после определенного «оптимума» ($P = 0,03$ МПа) начинает расти из-за глубоких царапин и недостатка смазочного слоя. Установлено, что шероховатость поверхности при притирке абразивным составом с ростом скорости возвратно-вращательного движения образца в начальный момент уменьшается до оптимального значения, а при чрезмерном повышении скорости снова возрастает. При притирке силикатным составом оптимальное значение скорости возвратно-вращательного движения составляет 0,3 м/с. Подтверждено, что производительность притирки силикатным абразивным составом в 1,5 раза выше, чем при обработке пастой «КТ». Подтверждено, что при притирке поверхностей пары трения силикатным составом снятие металла в 1,4 раза меньше, чем при обработке пастой «КТ».

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ПОДОБРЯВАНЕ НА МАТЕМАТИЧЕСКИЯ МОДЕЛ НА РАЗХОДА НА ГОРИВО ЗА ГРАДСКИ АВТОБУСИ

IMPROVING THE MATHEMATICAL MODEL OF FUEL CONSUMPTION FOR CITY BUSES

Ihor Marmut¹

Abstract. To address the issues of rational use of fuel in road transport, it is of great importance to improve the methods for calculating fuel efficiency indicators of cars in order to more fully and accurately take into account the peculiarities of their movement in different operating conditions. When developing fuel consumption standards, an analytical method was used, which is based on a strict mathematical model of fuel consumption of diesel cars. The methodology is built taking into account a single classification of operating conditions, in which road and transport conditions are quantified by average technical speed, and atmospheric and climatic conditions are quantified by ambient air pressure and temperature. An important advantage of the developed methodology is the possibility of individual fuel consumption regulation at road transport enterprises using simple computing equipment based on actual vehicle speeds and daily air temperature.

Keywords: city bus, fuel consumption, mathematical model, average technical speed.

1. Introduction:

The issues of studying fuel efficiency and traction-speed properties of cars, improving these qualities and methods for determining them in operation are given great attention by various researchers [1-4].

Works devoted to studying fuel efficiency of cars, taking into account driving modes and final results, can be divided into the following areas: research in steady, unsteady and cyclic driving modes; development of methods for assessing fuel efficiency of cars; research into the influence of various factors on the operational qualities of cars; increasing fuel efficiency of cars.

The most fully studied issues of assessing the analyzed qualities of cars in steady modes. However, the qualitative influence of factors for these modes on roads with a horizontal profile can differ significantly in real operating conditions. In this regard, the problem of studying the fuel efficiency of cars in unsteady and cyclic driving modes attracts more and more researchers every year. The most complex are cyclic driving modes with stops, which include almost all phases of movement that occur in operation, as well as all transmission gears used in this case.

Taking into account this specificity, the factors affecting fuel consumption can be divided into three groups:

- a group of constant factors - the type of work performed, the organization of work, weather conditions, driver qualifications, technical condition of the vehicle, quality of fuel and lubricants, etc.;

- a group of factors that are taken into account - the same factors if they are subject to quantitative or qualitative accounting and have significant deviations in the relevant periods;

- a group of unaccounted factors - factors that may have random values (random fluctuations in load modes, vehicle movement resistance, climatic conditions, etc.).

Specific fuel consumption is determined, first of all, by three technical and operational indicators: mileage utilization coefficients, load capacity (passenger capacity) and load capacity (passenger capacity) of the vehicle.

The most common indicators of fuel consumption in road transport are: linear norms per 100 km of mileage, energy intensity of transportation, specific costs per unit of transport work [5-9, 11].

Linear norms should take into account the amount of transported cargo (passengers). The energy intensity of bus transportation characterizes the amount of energy or fuel that must be spent to perform a unit of work, for example, per 100 passenger-km (its unit of measurement is kcal/100 passenger-km). This indicator is rarely used, because it cannot be compared with other units of measurement of fuel (l, kg, etc.).

To address the issues of rational use of fuel in road transport, it is of great importance to

improve the methods of calculating fuel efficiency indicators of cars in order to more fully and accurately take into account the peculiarities of the process of their movement in different operating conditions.

When developing fuel consumption standards, three methods are mainly used: analytical, statistical, and experimental [1].

The most advanced and widespread method is analytical. Fuel consumption rates are determined based on an analysis of its components. This method provides the necessary efficiency of development and autonomy of fuel consumption planning at different levels. The accuracy of calculations depends on the perfection of the fuel consumption model. The statistical method is based on statistical data and is used mainly in the development of group fuel consumption rates. The difficulty of its application lies in the need to use a large number of factors that vary over a wide range. The experimental method allows you to develop standards that correspond to specific car models and their operating conditions. However, it has limited application, because it lacks a quantitative assessment of operating conditions, which forces you to determine fuel consumption experimentally for almost every route and every car model.

Prof. Govorushchenko M.Ya. proposed an analytical method for fuel consumption standardization, which fundamentally differs from existing ones in that it is based on a strict mathematical model of fuel consumption of gasoline and diesel cars. The method is built taking into account the developed unified classification of operating conditions, in which road and transport conditions are quantified by average technical speed, and atmospheric and climatic conditions are quantified by ambient air pressure and temperature [1].

An important advantage of the developed methodology is the possibility of individual fuel consumption regulation at road transport enterprises (RTE) using simple computing equipment based on actual vehicle speeds and daily air temperature.

Research objective. To develop fuel consumption standards for city buses and recommendations for the efficiency of their use, taking into account real operating conditions.

2. Presentation:

1. Mathematical model of fuel consumption for city buses.

The car-driver system, its inputs and outputs can be considered as interconnected objects, depending on which task is being solved (analysis, synthesis, measurement), certain variables are considered. When analyzing (for example, fuel consumption), the outputs are investigated in relation to changes in input variables and the system itself (car-driver).

The input variables (external influences) include: road, transport, atmospheric and climatic conditions and the culture of operation [1].

Internal state variables include: torque, engine speed, gear ratios of the main gear and gearbox, transmission efficiency, engine indicator efficiency, wheel radii, suspension parameters, etc.

The output parameters (response to external influences), in addition to fuel consumption, are the speed (productivity, cost) of the vehicle, the composition of the engine exhaust gases, the noise level, the frequency of failures, the consumption of spare parts, etc.

When developing analytical methods for fuel consumption normalization, it is necessary to establish a functional relationship between input variables and vehicle states.

For further analysis, selection and justification of factors (parameters) for inclusion in mathematical models of bus fuel consumption, taking into account operating conditions, we will write the fuel consumption equation proposed by Prof. Govorushchenko M.Ya. for steady-state traffic [1]:

$$Q = \frac{1}{\eta_i} [A \cdot i_k + B \cdot i_k^2 \cdot V_a + C(G_a \cdot \psi + 0,077kF \cdot V_a^2)]. \quad (1)$$

To build a mathematical model for the Mercedes-Benz O530/Citaro C2 bus, we take the following initial data (table 1).

Table 1. Initial data for the Mercedes-Benz O530/Citaro C2 bus

Indicator	Value
Indicative efficiency:	
– for a fully loaded bus	0,48
– for a fully loaded bus	0,51
Cylinder displacement V_h , l	14,618
Final drive ratio i_o	3,920
Lower calorific value of fuel H_H , kJ/kg	43000
Fuel density ρ_H , g/cm ³	0,84
Wheel rolling radius, m	0,508
Piston stroke S_H , m	0,142
Transmission efficiency η_{TP} :	
– for equipped bus	0,79
– for a fully loaded bus	0,88
Bus weight, H:	
– equipped	119584
– with full load	171675
Streamline factor kF , H·s ² ·m ⁻²	2,95

The indicator efficiency of the engine η_i is quite difficult to determine in engine theory. The formula η_i is written in general form as follows [1]:

$$\eta_i = \frac{R \cdot P_i \cdot L_0 \cdot T}{H_H \cdot \eta_v \cdot P} \cdot \alpha,$$

where: R – is the universal gas constant, which is equal to 8.31 J/(mol·K); P_i – is the average indicator pressure, determined from the experimental indicator diagram, MPa; L_0 – is the stoichiometric amount of air in the mixture per 1 kg of fuel, kmol; T – is the ambient temperature, K; H_H – is the lower specific heat of combustion of the fuel, kJ/kg; η_v – is the filling coefficient; P – is the ambient pressure, MPa; α – is the excess air coefficient, which is determined by the ratio of the actual amount of air L to the stoichiometric L_0 .

The indicator pressure P_i is calculated by the formula [10]

$$P_i = \frac{P_c}{\varepsilon - 1} \left[\frac{\lambda}{n_2 - 1} \left(1 - \frac{1}{\varepsilon^{n_2 - 1}} \right) - \frac{1}{K_1 - 1} \left(1 - \frac{1}{\varepsilon^{K_1 - 1}} \right) \right],$$

where: $P_c = P_a \cdot \varepsilon^{K_1} = 0,9P_0 \cdot \varepsilon^{K_1}$ – gas pressure in the cylinder at the end of the compression process (P_a – pressure at the beginning of compression; $P_0 = 0,1013$ MPa – ambient pressure under normal conditions); ε – compression ratio; λ – pressure increase rate in the cylinder; n_2 – polytrope index; K_1 – average adiabatic index in the compression process.

The degree of pressure increase λ is determined by the ratio of the pressure of gases in the cylinder at the end of combustion P_{zp} to the pressure of gases at the end of the compression process P_c . We use a number of dependencies of various quantities from engine theory to obtain a general formula for calculating λ [10]:

$$P_{zp} = \mu \cdot P_c \cdot T_z / T_c; \quad \mu = (\mu_0 + \gamma) / (1 + \gamma); \quad \gamma = \xi_{oq} \cdot \frac{T_0 + \Delta T}{\xi_{c.3} \cdot T_r} \cdot \frac{P_r}{\varepsilon \cdot P_a - \xi_{oq} \cdot P_r},$$

where: μ – is the coefficient of molecular change of the working gas mixture; T_z – is the maximum temperature of the cycle, K; T_c – is the temperature at the end of compression, K; μ_0 – is the coefficient of molecular change of the fresh mixture, determined by the ratio of the total amount of incomplete combustion products to the amount of fresh mixture per 1 kg of fuel; γ – is the coefficient of residual gases; ξ_{oq} – engine cylinder cleaning coefficient; T_0 – ambient temperature under normal conditions (293 K); ΔT – heating of the fresh charge from the cylinder walls, K; $\xi_{c.3}$ – coefficient that takes into account the change in the number of kmol of fresh charge during the

recharge period; T_r – residual gas temperature, K; P_r – residual gas pressure ($P_r = 1, 1P_0$), MPa; ξ – coefficient that takes into account the ratio of the heat capacities of the combustion products and the working mixture at constant pressure.

After substitution, we obtain an expression for calculating the degree of pressure increase in the cylinder:

$$\lambda = \frac{\left[\mu_0 + \frac{\xi_{0q}(T_0 + \Delta T)}{\xi_{c,3} \cdot T_r(0,82\xi - \xi \cdot \xi_{0q})} \right] \cdot T_z}{\left\{ T_0 + \Delta T + \xi_{c,3} \cdot \xi \left[\frac{\xi_{0q}(T_0 + \Delta T)}{\xi_{c,3}(0,82\xi - \xi \cdot \xi_{0q})} \right] \right\} \cdot \varepsilon^{K_1 - 1}}. \quad (2)$$

The stoichiometric amount of air, kmol, is determined by the formula [10]:

$$L_0 = \frac{1}{0,21} \left(\frac{C}{12} + \frac{H}{4} - \frac{O}{32} \right).$$

For 1 kg of fuel of average composition, $C = 0,885$ kg, $H = 0,145$ kg, therefore $L_0 = 0,5119$ kmol. Filling factor [10]: $\eta_v = \xi \cdot \frac{\varepsilon}{\varepsilon - 1} \cdot \frac{P_a}{P_0} \cdot \frac{T_0}{T_0 + \Delta T} \left(1 - \frac{\xi \cdot P_r}{\varepsilon \cdot P_a} \right)$.

After substituting the above formulas, we get:

$$\eta_v = \frac{0,9\xi_{c,3} \cdot \varepsilon \cdot T_0}{(\varepsilon - 1) \cdot (T_0 + \Delta T)} \cdot (1 - 1,222 \xi / \varepsilon). \quad (3)$$

To simplify the determination of the values of the indicator efficiency, we use experimental data for individual values [10]: $n_2 = 1,24$; $T_r = 900$ K; $\xi_{0q} = 1$; $\xi_{c,3} = 1$; $\xi = 1$; $T_z = 2597,8$ K; $\Delta T = 10$ K; $K_1 = 1,36$.

As a result of calculations for the Mercedes-Benz O530/Citaro C2 bus (at $\varepsilon = 9$ and $\mu_0 = 1,0985$) we obtain:

$$\eta_i = 0,36\alpha. \quad (4)$$

Similar calculations for other cars, in particular passenger cars, showed the possibility of using the obtained equation for practical calculations. The values of coefficients A , B , C for the Mercedes-Benz O530/Citaro C2 bus (diesel engine) are determined by the formulas [1]:

$$A_d = \frac{381V_h \cdot i_0}{H_H \cdot \rho_{\Pi} \cdot r_K}, \quad B_d = \frac{11V_h \cdot S_{\Pi} \cdot i_0^2}{H_H \cdot \rho_{\Pi} \cdot r_K^2}, \quad C_d = \frac{100}{H_H \cdot \rho_{\Pi} \cdot \eta_{TP}},$$

where: V_h – engine cylinder displacement, l; i_0 – final drive gear ratio; H_H – lower calorific value of fuel, kJ/kg; ρ_{Π} – fuel density, kg/m³; r_K – wheel radius, m; S_{Π} – piston stroke, m; η_{TP} – transmission efficiency.

Mathematical models of fuel consumption for a bus with variable values of gearbox ratios i_K , speed V_a , total road resistance ψ at curb weight Q_c and gross weight Q_{Π} take the form:

$$\begin{aligned} Q_c &= [1,19i_K + 0,038i_K^2 \cdot V_a + 3,5 \cdot 10^{-3}(119584\psi + 0,227V_a^2)] / \eta_{ic}; \\ Q_{\Pi} &= [1,19i_K + 0,038i_K^2 \cdot V_a + 3,15 \cdot 10^{-3}(171675\psi + 0,227V_a^2)] / \eta_{i_{\Pi}}. \end{aligned} \quad (5)$$

If it is necessary to determine fuel consumption for specific values of i_K , they can be expressed in terms of the average speed of the bus [1]:

$$i_K = K_{\Pi} \cdot V_{\max} \cdot i_{K\Pi} / V_a, \quad (6)$$

where: K_{Π} – speed coefficient; K_{Π} – maximum speed, km/h; $i_{K\Pi}$ – gear ratio of direct (increased)

transmission.

$$i_k = \frac{0,522 \cdot 117 \cdot 1}{V_a} = \frac{61,1}{V_a}. \quad (7)$$

The value of ψ can also be expressed in terms of V_a :

$$\psi = 0,8/V_a. \quad (8)$$

The fuel consumption equation (1) includes 13 parameters, three of which are constant for this vehicle (V_h , i_0 , kF) and 10 are variable. The parameters can also be divided into controlled (η_i , i_k , r_k , V_a , G_a) and uncontrolled. The average speed, indicator and transmission efficiency change with changes in the technical condition of the vehicle.

Some parameters have a negligible effect on fuel consumption and can be neglected in future operational calculations. Research by the Kharkiv National Automobile and Highway University has shown that all parameters can be arranged in the following order of importance (influence on fuel consumption): ψ , η_i , i_k , V_a , G_a , r_k , ambient air temperature t .

If we take $1/\eta_i$ as a constant value, then there will be a linear relationship between fuel consumption, total road resistance and vehicle weight:

$$Q = A_1 + B_1 \cdot G_a \cdot \psi, \quad (9)$$

$$\text{where } A_1 = \frac{1}{\eta_i} (A \cdot i_k + B \cdot i_k^2 \cdot V_a + 0,077C \cdot kF \cdot V_a^2) \text{ and } B_1 = C/\eta_i$$

Equation (9) shows that fuel consumption, on the one hand, increases with increasing vehicle load, and on the other hand, decreases due to an increase in the indicator efficiency due to an increase in load. The increase in fuel consumption per ton of cargo (passengers) transported over a distance of 100 km is also not a constant value and depends on the coefficients C , ψ , the total vehicle weight G_a and the speed V_a .

There is also no linear relationship between fuel consumption and ψ . According to equation (1), with increasing ψ , fuel consumption increases according to the law of a straight line. However, the load on the engine increases, which should lead to an increase in η_i and, consequently, to a decrease in fuel consumption Q .

2. Calculation of fuel consumption standards for city buses and recommendations for the efficiency of its use.

Scientifically and technically justified fuel consumption standards play a major role in implementing fuel economy. However, the foundations of the current fuel consumption standardization methodology are based on experimental data that do not fully take into account road and transport conditions, changes in passenger traffic on the route, and the specifics of the operating conditions of a particular route.

The basis for determining fuel consumption standards should be its strict mathematical model, taking into account a single classification of bus operating conditions, in which road and transport conditions are quantified by the average speed of their movement. This is due to the fact that there is a direct relationship between the speed of the vehicle, road and transport conditions and fuel consumption.

Fuel consumption standards are linear, route and group. The main task of fuel consumption management according to linear norms is to comply with the standard fuel consumption values for the bus mileage, determined by its technical characteristics and regulatory documents [11], through systematic monitoring of the technical condition of the vehicle and maintaining it at a high technical level. This work is carried out by the technical service of the automobile enterprise.

The main goal of route rationing is to bring the standard fuel consumption value as close as possible to the actual one on a specific route. The use of differentiated route ratios makes it

possible to reveal fuel economy reserves and to carry out a more rational internal redistribution of fuel along routes, depending on specific conditions that are not taken into account in linear ratios.

The criterion for economical and rational use of automobile fuel by rolling stock, taking into account the transport work performed, is the group rate of its consumption. Fuel consumption management according to group rates is carried out in two stages:

- improvement of the system of rationing and planning of group rates in RTE;
- performance of operational and periodic analysis of fuel efficiency.

Scientifically substantiated fuel consumption standards should be established on the basis of its mathematical model (formula (1)). In the current documents, only the average fuel consumption standard per 100 km of mileage is approved for buses. However, as noted in paragraph 1, in order to improve the standardization methodology, it is necessary to calculate the basic and additional fuel consumption standards for buses as well.

The basic standards for the Mercedes-Benz O530/Citaro C2 bus of empty and full weight should be calculated according to formula (5) taking into account formulas (7) and (8):

$$Q_c = \left[1,19 \cdot \frac{61,1}{V_a} + 0,038 \cdot \frac{61,1^2}{V_a^2} \cdot V_a + 3,5 \cdot 10^{-3} \left(119584 \cdot \frac{0,8}{V_a} + 0,227 V_a^2 \right) \right] / 0,48;$$

$$Q_{\Pi} = \left[1,19 \cdot \frac{61,1}{V_a} + 0,038 \cdot \frac{61,1^2}{V_a^2} \cdot V_a + 3,15 \cdot 10^{-3} \left(171675 \cdot \frac{0,8}{V_a} + 0,227 V_a^2 \right) \right] / 0,51.$$

The fuel consumption rate equations are presented in table 2.

Table 2. Fuel consumption rate equation for Mercedes-Benz O530 / Citaro C2 bus

Mass of the bus	
equipped	full
$H_{o.c} = \frac{1144}{V_a} + 0,0017 V_a^2$	$H_{o.\Pi} = \frac{1269}{V_a} + 0,0014 V_a^2$

Specific fuel consumption rates for this bus depending on operating parameters are given in table 3 and fig. 1.

Table 3. Results of fuel consumption calculation for the Mercedes-Benz O530/Citaro C2 bus

V_a , km/h	Fuel consumption rate	
	$H_{o.c}$	$H_{o.\Pi}$
20	57,88	64,01
30	39,66	43,56
40	31,32	33,96
50	27,13	28,88
60	25,19	26,19
70	24,67	24,99
80	25,18	24,82
90	26,48	25,44
100	28,44	26,69

The additional fuel consumption rate for buses in liters per 100 passenger-km can be established using the formula for calculating the additional fuel consumption per 100 t-km for trucks, l/100 t-km [1]: $H'_d = 10^6 \Psi / (\eta_i \cdot H_H \cdot \rho_{\Pi} \cdot \eta_{TP})$.

Number of passengers per 1 ton of their mass, pas. /t: $P = 1000/G_{\Pi}$, where $G_{\Pi} = 75$ kg – average mass of one passenger, kg.

Additional fuel consumption rate of the bus, l/100 passenger-km:

$$H_d = H'_d / P. \quad (10)$$

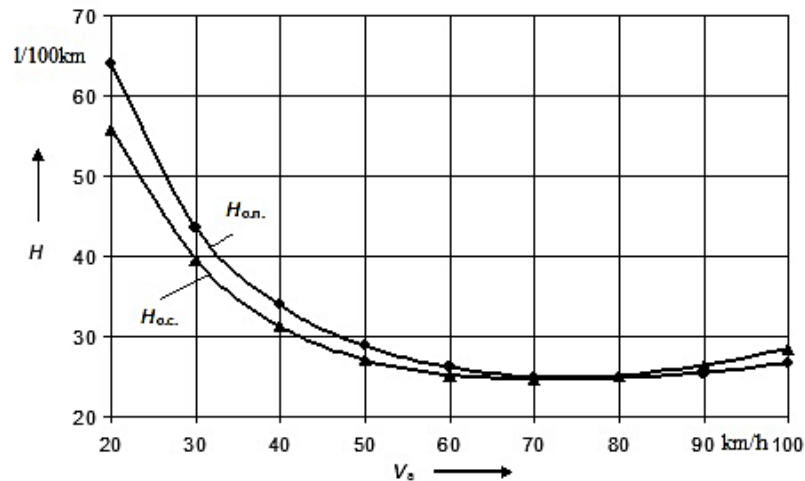


Fig. 1. Graph of fuel consumption changes for the Mercedes-Benz O530/Citaro C2 bus at curb and gross weight depending on speed

Total fuel consumption rate taking into account passengers transported, l/100 km:

$$H = H_{o.c} + H_{\Delta} \cdot q_a \cdot \gamma_a \cdot \beta. \quad (11)$$

Table 4 shows the calculated data for the Mercedes-Benz O530/Citaro C2 bus. The values of average speeds and total road resistance are taken from [1] (taking into account the classification of vehicle operating conditions proposed in the KhNAHU).

Table 4. Estimated fuel consumption rates for the Mercedes-Benz O530 / Citaro C2 bus

Category of operating conditions	Estimated values		$H_{o.c}$, l/100 km	H_{Δ} , l/100 pas. -km	General norm H , l/100 km
	ψ	V_a , km/h			
1	0,013	60	25,19	0,060	27,73
2	0,017	47	28,10	0,079	31,44
3	0,021	38	32,56	0,097	36,66
4	0,025	32	37,49	0,116	42,40
5	0,031	26	45,15	0,143	51,20

The analytical equation for fuel consumption rationing in liters for a certain period (for example, for a working day), which takes into account the distance traveled, the transport work performed, the road, transport and atmospheric and climatic conditions of bus operation, can be presented in the following form:

$$H_i = 0,01K_t \cdot K_h(H_{o_i} \cdot l_c + H_{\Delta_i} \cdot W), \quad (12)$$

where: K_t – is a coefficient that takes into account the ambient air temperature, °C; K_h – is a coefficient that takes into account the height above sea level, m; H_{o_i} and H_{Δ_i} – are, respectively, the main and additional fuel consumption rates for the i -th category of operating conditions (the category is determined by the average speed of the bus on the route for the analyzed period); l_c – is the total mileage of the bus for the analyzed period, km; W – is the transport work performed for the same period, passenger-km.

To manage fuel consumption, determine the amount of fuel losses due to non-fulfillment of technical and operational indicators (TOI) and take into account their impact on changes in the group norm, it is advisable to conduct both periodic and operational analysis. Periodic analysis is performed for a quarter, a year, and operational analysis – for a month or a decade. The proposed methodology is based on an analysis of the impact of TOI on the fulfillment of the group norm H_w , g/passenger-km, which for buses is calculated according to the generally accepted formula

$$H_w = 10\rho_{\pi}(1 + \Delta)\frac{H_0}{q_a \cdot \gamma_a \cdot \beta}, \quad (13)$$

where: ρ_{π} – fuel density, g/cm³; Δ – total surcharge, expressed in fractions of a unit; H_0 – weighted average fuel consumption rate by fleet structure, l/100 km.

If we examine equation (13) by partial derivatives, we can determine the magnitude of the influence of each of the TOI on the fulfillment of the group norm K . This method makes it possible to determine the sensitivity of the group norm due to non-fulfillment of the TOI, the change in the group norm in the event of non-fulfillment of the TOI in g/passenger-km (ΔH_w), and fuel losses due to non-fulfillment of the TOI in t (ΔQ).

The analysis of fuel efficiency is performed in three stages:

- calculation of the planned values of the weighted average linear norm H_0 and passenger capacity q_a (at the beginning of the current year);
- determination of the actual values of H_0 and q_a , the planned and actual values of the total allowance Δ (for a decade, month, quarter, year);
- solving the analysis tasks (for the same period).

The indicated indicators are calculated using the formulas

$$H_0 = \frac{\sum_{i=1}^n A_{c_i} \cdot H_{0i}}{\sum_{i=1}^n A_{c_i}}, \text{ l/100 km}; \quad q_a = \frac{\sum_{i=1}^n A_{c_i} \cdot q_{ai}}{\sum_{i=1}^n A_{c_i}}, \text{ pas.}; \quad \Delta = \frac{H_w \cdot q_a \cdot \gamma_a \cdot \beta}{10\rho_{\pi} \cdot H_0} - 1 \text{ (from the formula (13))}.$$

The sensitivity of the group norm to TOI (denoted by the coefficient K with the index of the corresponding TOI) is determined using the following calculation formulas:

$$\begin{aligned} K_{H_0} &= \frac{10\rho_{\pi}(1+\Delta)}{q_a \cdot \gamma_a \cdot \beta}; & K_{\Delta} &= K_{H_0} \cdot \frac{H_0}{1+\Delta}; \\ K_{\beta} &= -K_{H_0} \cdot \frac{H_0}{\beta}; & K_{\gamma_a} &= -K_{H_0} \cdot \frac{H_0}{\gamma_a}; & K_{q_a} &= -K_{H_0} \cdot \frac{H_0}{q_a}. \end{aligned} \quad (14)$$

To determine the change in the group norm in case of non-fulfillment of the TOI in g/passenger-km, it is necessary to multiply the value of the studied TOI by the difference between the actual and planned values of this indicator: $\Delta H_{w_i} = K_i(i_{\phi} - i)$, where i – is the corresponding TOI.

Determination of fuel consumption in tons due to non-fulfillment of TOI is carried out according to the formula

$$\Delta Q_i = \Delta H_{w_i} \cdot W_{\phi} \cdot 10^{-6}, \quad (15)$$

where: W_{ϕ} – actual passenger turnover, passenger-km.

The proposed method provides sufficient accuracy for analysis, differs from existing ones in simplicity, greater clarity and compactness of calculations.

Based on the results of periodic and operational analyses, the causes of fuel losses are determined for any period of the year, which allows them to be eliminated in a timely manner.

3. Conclusions:

1. It is recommended to determine fuel consumption standards using a mathematical model in which all elements of the external environment of bus operation are quantitatively assessed by the average speed of their movement. This is due to the existence of a direct relationship between the speed of the vehicle, road and transport conditions and fuel consumption. When calculating the total fuel consumption rate for buses, it is proposed to take into account an additional fuel consumption rate that takes into account their transport work.

2. When calculating the total fuel consumption rate for buses, it is proposed to take into account an additional fuel consumption rate that takes into account their transport work.

3. To manage fuel consumption, it is proposed to conduct both periodic (quarterly, annual) and operational (monthly or decade) analyses.

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ОПТИМИЗАЦИЯ ПАРАМЕТРОВ РЕЖИМА ПРИТИРКИ СИЛИКАТНЫМ АБРАЗИВНЫМ СОСТАВОМ С ИСПОЛЬЗОВАНИЕМ ПОЛНОГО ФАКТОРНОГО ЭКСПЕРИМЕНТА

OPTIMIZATION OF RUNNING-IN PARAMETERS USING A SILICATE ABRASIVE COMPOSITION VIA A FULL FACTORIAL EXPERIMENT

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Abstract. The paper notes that the goal of optimizing the running-in parameters is to ensure processing conditions that result in a surface layer with optimal microgeometry and physical-mechanical properties, which help reduce contact stresses, increase the effective contact area, and decrease the wear rate of the valve connection under operating conditions. Optimization studies were conducted using the Box-Wilson method, which employs “active experimentation,” and the planning of points in the factor space under study is carried out in such a way as to obtain the maximum amount of information from the experiments and process the resulting data with the least computational effort. To determine the optimal parameters of the running-in process using a silicate abrasive composition that ensures the lowest surface roughness after processing, a full factorial experimental design of the $N = 2^3 = 8$ type was developed and implemented. During the implementation of this plan, experimental studies were conducted and a mathematical model equation for the silicate abrasive lapping process was derived. It was established that lapping time has the greatest influence on the roughness of the processed surface. Optimal parameters for the silicate abrasive lapping process are proposed to achieve the minimum surface roughness of the treated surfaces.

Keywords: lapping, silicate abrasive composition, surface roughness, full factorial experiment, mathematical model, lapping conditions.

1. Введение:

В современных научных исследованиях значительное внимание уделяется процессам абразивной обработки с использованием свободного абразива, в частности притирке, как одному из эффективных методов формирования высококачественных сопрягаемых поверхностей. Как показано в работах [1,2], параметры режима притирки существенно влияют на микрогеометрию поверхности, фактическую площадь контакта и эксплуатационные характеристики деталей машин.

В работе [1] исследован механизм притирки с использованием силикатного абразивного состава. Авторами установлено, что свойства связующего среды существенно влияют на стабильность абразивной пасты, характер взаимодействия абразивных зерен с поверхностью и интенсивность снятия материала. Показано, что применение силикатной композиции способствует формированию более однородного микрорельефа поверхности и уменьшению локальных контактных напряжений.

Обзор современных абразивных технологий, приведенный в работе [2], подтверждает, что процессы притирки остаются актуальными для прецизионной обработки ответственных поверхностей. Автор отмечает, что оптимизация параметров абразивной обработки должна учитывать не только интенсивность снятия материала, но и характеристики поверхностного слоя, которые определяют износостойкость деталей при дальнейшей эксплуатации.

В работе [3] рассмотрено применение методов экспериментального планирования и оптимизации параметров процесса полирования. Показано, что использование комплексных критериев оптимизации позволяет обеспечить рациональное сочетание качества поверхности и производительности процесса, что актуально для инженерных применений.

Таким образом, анализ современных научных источников свидетельствует об отсутствии универсальных рекомендаций относительно режимов притирки клапанных соединений с использованием силикатных абразивных составов. Это обуславливает целесообразность проведения дальнейших исследований, направленных на оптимизацию параметров режима притирки с учетом требований к качеству поверхности и износостойкости клапанных узлов.

Оптимизация параметров режима притирки силикатным абразивным составом – это научно обоснованный процесс определения рациональных значений контактной нагрузки, скорости и характера относительного движения притирочных поверхностей, продолжительности обработки, при которых обеспечивается формирование заданных показателей качества поверхности клапанного соединения и достигается повышение его износостойкости при минимальных материальных и энергетических затратах.

Целью оптимизации параметров режима притирки является обеспечение таких условий обработки, при которых формируется поверхностный слой с оптимальной микрогеометрией и физико-механическими свойствами, способствующими уменьшению контактных напряжений, повышению фактической площади контакта и снижению интенсивности износа клапанного соединения в эксплуатационных условиях.

Оптимизационные исследования проводились по методике Бокса-Уилсона [4], которая использует «активный эксперимент», а планирование точек в исследуемом факторном пространстве осуществляется так, чтобы получить максимальную информацию из поставленных опытов и обработать полученные данные с наименьшими вычислительными затратами.

Метод Бокса-Уилсона – метод оптимизации активного эксперимента путем крутого спуска по поверхности отклика (параметров оптимизации) к оптимуму, суть которого заключается в следующем: движение в направлении градиента при наличии линейного уравнения модели осуществляется из центра эксперимента последовательными шагами, которые пропорциональны произведению коэффициента регрессии каждого фактора на значение его интервала изменения.

Полным факторным экспериментом называется такой эксперимент, при реализации которого определяется значение параметра оптимизации при всех возможных сочетаниях уровней варьирования факторов. Если имеется k факторов, каждый из которых может устанавливаться на q уровнях, то для того, чтобы осуществить полный факторный эксперимент, необходимо провести $n=q^k$ опытов.

Целью работы является определение оптимальных значений параметров режима притирки силикатным абразивным составом, которые обеспечивают максимальное качество обработанной поверхности клапанного соединения по критерию шероховатости поверхности (R_a) с использованием полного факторного эксперимента.

2. Изложение:

С целью определения оптимальных параметров режима притирки силикатным абразивным составом, обеспечивающего наименьшую шероховатость поверхности после обработки, был разработан и реализован план полного факторного эксперимента типа $N = 2^3 = 8$.

Предельные значения независимых факторов (давление на образец (P), скорость вращения образца ($V_{в-в}$), время притирки (t)) выбирали из результатов предыдущих экспериментов.

Исследования проводились на специальной установке, смонтированной на станине универсального станка ОПР-1841А, предназначенного для притирки клапанов автотракторных двигателей.

Для проведения исследования использовался состав силикатной пасты с абразивом карбид кремния зеленый зернистостью 50/40, рекомендованный в работе.

В качестве образца использовали конус из стали 40Х (HRC 48) диаметром 23 мм, шириной 15 мм. Контрольный образец имел форму кольца из чугуна СЧ 18-36 (HRC 32) диаметром 50 мм, шириной 10 мм с шириной рабочей поверхности 3 мм. Исходная шероховатость поверхностей трения во всех образцах была одинаковой и составляла $R_a = 1,25$ мкм [5].

Время притирки во всех опытах составляло 10 мин. По окончании притирки определяли шероховатость поверхности после обработки и массовое снятие металла.

Среднее значение испытаний определялось как среднее арифметическое пяти опытов с погрешностью $\pm 3\%$.

Полученные значения параметра оптимизации – шероховатости поверхности образцов (R_a , мкм) и значения независимых факторов приведены в табл. 1.

Таблица 1. Значения независимых факторов и параметра оптимизации

Номер эксперимента	Значения независимых факторов			Шероховатость поверхности образцов, R_a , мкм.
	Давление например, МПа, x_1	Возвратно-вращательная скорость образца, м/с, x_2	Время притирки, с, x_3	
1	0,01	0,3	240	0,118
2	0,01	0,3	600	0,158
3	0,01	0,6	240	0,138
4	0,01	0,6	600	0,178
5	0,05	0,3	240	0,118
6	0,05	0,3	600	0,168
7	0,05	0,6	240	0,158
8	0,05	0,6	600	0,218

Был проведен эксперимент типа 2^3 , где число факторов $k=3$, количество уровней $p=2$, число опытов $N=8$, повторных опытов $n=5$.

Матрица планирования приведена в табл. 2.

Таблица 2. Матрица планирования

Номер эксперимента	Матрица планирования								Рабочая матрица			Результаты параллельных экспериментов y_i мкм	Среднее значение, $\bar{y}_i$, мкм
	x_0	x_1	x_2	x_3	$x_{1,2}$	$x_{1,3}$	$x_{2,3}$	$x_{1,2,3}$	Давление на образец, x_1	Возвратно-вращательная скорость образца, x_2	Время притирки, x_3		
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	1	-1	-1	-1	1	1	-1	-1	0,01	0,3	240	0,12 0,12 0,12 0,11 0,12	0,118
2	1	-1	-1	1	1	-1	1	1	0,01	0,3	600	0,16 0,15 0,16 0,16 0,16	0,158
3	1	-1	1	-1	-1	-1	1	1	0,01	0,6	240	0,14 0,14 0,14 0,13 0,14	0,138

Продолжение таблицы 2

1	2	3	4	5	6	7	8	9	10	11	12	13	14
4	1	-1	1	1	-1	1	-1	-1	0,01	0,6	600	0,18 0,18 0,18 0,18 0,17	0,178
5	1	1	-1	-1	-1	1	1	1	0,05	0,3	240	0,12 0,12 0,12 0,11 0,12	0,118
6	1	1	-1	1	-1	-1	-1	-1	0,05	0,3	600	0,17 0,16 0,17 0,17 0,17	0,168
7	1	1	1	-1	1	-1	-1	-1	0,05	0,6	240	0,16 0,16 0,16 0,16 0,15	0,158
8	1	1	1	1	1	1	1	1	0,05	0,6	600	0,21 0,22 0,22 0,22 0,22	0,218

После проведения экспериментов была выполнена статистическая обработка результатов. Сначала определялись погрешности повторных (параллельных) экспериментов. Среднеквадратичное отклонение определяется по формуле [6]:

$$S_i^2 = \frac{\sum_1^n (y_i - \bar{y})^2}{n-1}, \quad (1)$$

где $\bar{y}$ – среднее арифметическое значение параметра оптимизации по пяти повторным опытам (значения приведены в табл. 3).

Результаты расчетов сведены в табл. 3.

Для определения брака используем критерий Стьюдента:

$$\frac{y - \bar{y}}{s} \geq t, \text{ или } t_{\text{расч.}} \geq t_{\text{табл.}}, \quad (2)$$

где t – критерий Стьюдента, его значение для 5 повторных опытов и доверительной вероятности 0,95 равно 3,182 [6].

Для всех опытов $t_{\text{расч.}} = 1,79$ условие $t_{\text{расч.}} \geq t_{\text{табл.}}$ [6] не выполняется, следовательно, результаты повторных опытов нельзя считать ошибочными.

Для проведения регрессионного анализа рассчитали однородность дисперсии параллельных опытов по критерию Кохрена [6]:

$$G_{\text{роз.}} = \frac{S_{\text{max}}^2}{\sum_1^N S_i^2} = \frac{0,00002}{0,0016} = 0,125. \quad (3)$$

Расчетное значение критерия сравнивали с табличным для степеней свободы:

$f_1 = r - 1$; знаменателя $f_2 = N$ при выбранном уровне значимости $\alpha = 0,05$; $G_{\text{табл.}}$ – табличное значение критерия Кохрена – 0,516 [6].

$$G_{\text{табл.}}=0,516 > G_{\text{расч.}}=0,125. \quad (4)$$

Таблица 3. Результаты расчетов

Номер эксперимента	1	2	3	4	5	6	7	8	Сумма
S_i^2	0,00002	0,00002	0,00002	0,00002	0,00002	0,00002	0,00002	0,00002	0,00016
S_i	0,00447	0,00447	0,00447	0,00447	0,00447	0,00447	0,00447	0,00447	
$\frac{y_{\max} - \bar{y}}{S}$	0,45	0,45	0,45	0,45	0,45	0,45	0,45	0,45	
$\frac{\hat{y} - y_{\min}}{S}$	1,79	1,79	1,79	1,79	1,79	1,79	1,79	1,79	

Таким образом, гипотеза об однородности дисперсии параллельных опытов принимается. Отсюда дисперсия воспроизводимости составила $S_i^2=0,00002$. Погрешность эксперимента составила $S_i=0,00447$. Уравнение математической модели с учетом парных взаимодействий имеет вид [6]:

$$\hat{y} = b_0 + b_1x_1 + b_2x_2 + b_3x_3 + b_{12}x_1x_2 + b_{13}x_1x_3 + b_{23}x_2x_3 + b_{123}x_1x_2x_3. \quad (5)$$

Коэффициенты регрессии при полном факторном эксперименте определяются по формулам [6]:

$$b_0 = \frac{\sum_{i=1}^N \hat{y}_i}{N}; \quad (6)$$

$$b_i = \frac{\sum_{i=1}^N x_{iu} \hat{y}_i}{N}; \quad (7)$$

$$b_{ij} = \frac{\sum_{i=1}^N x_{iu} x_{ju} \hat{y}_i}{N}; \quad i \neq j; \quad (8)$$

$$b_{ijk} = \frac{\sum_{i=1}^N x_{iu} x_{ju} x_{ku} \hat{y}_i}{N}; \quad i \neq j \neq k. \quad (9)$$

Коэффициенты регрессии, рассчитанные по приведённым выше выражениям, равны: $b_0=0,15675$; $b_1=0,00875$; $b_2=0,01625$; $b_3=0,02375$; $b_{1,2}=0,00625$.

С учетом значения дисперсии воспроизводимости $S_i^2=0,00002$ при доверительной вероятности $\alpha=0,95$ находим границы доверительных интервалов для коэффициентов регрессии [6]:

$$\Delta b_i = \pm \frac{t \cdot S_i}{\sqrt{N}} = \pm \frac{2,78 \cdot 0,00447}{\sqrt{8}} = \pm 0,004. \quad (10)$$

Сравнивая значения коэффициентов регрессии с границами доверительных интервалов, видим, что коэффициенты $b_{1,2}$, $b_{1,3}$ и $b_{1,2,3}$ незначимы. Теперь уравнение математической модели имеет вид:

$$\hat{y} = 0,15675 + 0,00875x_1 + 0,01625x_2 + 0,02375x_3 + 0,00625x_{1,2}. \quad (11)$$

Проверяем адекватность полученного уравнения.

Вычисляем теоретические значения параметра оптимизации $\hat{y}$, величину погрешности $\Delta y = y - \hat{y}$; результаты приведены в табл. 4.

Таблица 4. Результаты расчетов

Номер эксперимента	1	2	3	4	5	6	7	8
$\hat{y}$	0,114	0,162	0,134	0,18	0,119	0,167	0,164	0,212
Δy	0,004	-0,004	0,004	-0,002	-0,001	0,001	-0,006	0,006
Δy^2	0,00001	0,00001	0,00001	0,000006	0,000002	0,000002	0,00004	0,00004

$$S_{ад}^2 = \frac{\sum_1^N \Delta y_i^2}{f}, \quad (12)$$

где $f=N-(k+1) = 8-(3+1) = 4$ – количество степеней свободы.

$$S_{ад}^2 = \frac{0,000129}{4} = 0,000032.$$

Адекватность математической модели определяем по критерию Фишера [6]:

$$F_{расч.} = \frac{S_{ад}^2}{S_y^2} = \frac{0,000032}{0,00002} = 1,6. \quad (13)$$

$F_{табл.} = 6,39$ [6]. $F_{расч.} \leq F_{табл.}$, следовательно, модель адекватна.

Анализируя уравнение 11 математической модели, можно отметить, что на параметр оптимизации влияют независимые факторы. Каждый коэффициент характеризует роль соответствующей переменной в процессе или силу влияния факторов. Чем больше численная величина коэффициента, тем больше влияет фактор. Если коэффициент имеет знак «+», то с увеличением значения фактора параметр отклика (оптимизации) увеличивается, а если «-» – уменьшается. Величина коэффициента соответствует вкладу данного фактора в величину параметра оптимизации при переходе фактора с нулевого уровня на верхний или нижний.

Качественный анализ полученного уравнения 11 позволяет сделать следующие выводы о влиянии независимых факторов при притирке сопряжения силикатным абразивным составом на шероховатость обработанной поверхности (R_a). Наибольшее влияние на шероховатость обработанной поверхности оказывает время притирки. Об этом свидетельствует значение коэффициента регрессии b_3 в уравнении 11, которое составляет 0,02375. Меньшее влияние на шероховатость обработанной поверхности оказывают скорость вращения образца ($b_2=0,01625$) и давление на образец ($b_1=0,00875$).

Из уравнения 11 также видно, что на шероховатость обработанной поверхности влияет взаимодействие возвратно-вращательной скорости образца и давления на образец, и это влияние является наименьшим ($b_{1,2} = 0,00625$).

С целью определения оптимальных значений параметров режима притирки силикатным абразивным составом проводили оптимизационное исследование методом «крутого спуска». При движении в факторном пространстве независимые переменные изменяли пропорционально величинам коэффициентов регрессии с учетом их знаков. Для этого определяли шаг движения по каждому фактору и проводили «воображаемые» эксперименты, которые заключались в вычислении значений функции отклика в точках факторного пространства, лежащих на пути к оптимуму. Осуществляли воображаемое движение к оптимуму. Для удобства расчетов кодовые значения факторов были переведены в натуральные путем декодирования по формулам [7]:

$$C_i = \frac{x_i - x_{i0}}{\Delta x_i}, \quad (14)$$

где C_i – кодовое значение i -го фактора; x_i – текущее значение i -го фактора; x_{i_0} – начальный (нулевой) уровень фактора; Δx_i – интервал варьирования i -го фактора:

$$\Delta x_i = \frac{x_{i_{max}} - x_{i_{min}}}{2}. \quad (15)$$

Полученные значения приведены в табл. 5.

Таблица 5. Стандартная матрица эксперимента

Факторы	Верхний уровень, C_i^+	Нижний уровень, C_i^-	Основной уровень, C_i^0	Диапазон варьирования, Δx_i	Зависимость кодированной переменной от натуральной
C_1	0,05	0,01	0,03	0,02	$x_1 = \frac{C_1 - 0,03}{0,02}$
C_2	0,6	0,3	0,45	0,15	$x_2 = \frac{C_2 - 0,45}{0,15}$
C_3	600	240	420	180	$x_3 = \frac{C_3 - 420}{180}$

В табл. 6 представлены результаты выполненных и воображаемых экспериментов по движению по линии «крутого подъема».

Значение параметра оптимизации в «условных» экспериментах на основе значений из табл. 5 определяли по следующей формуле:

$$R_a = 0,15675 + 0,00875 \cdot \left(\frac{C_1 - 0,3}{0,2} \right) + 0,01625 \cdot \left(\frac{C_2 - 0,7}{0,4} \right) + 0,02375 \cdot \left(\frac{C_3 - 420}{180} \right) + 0,00625 \cdot \left(\frac{C_1 - 0,3}{0,2} \right) \cdot \left(\frac{C_2 - 0,7}{0,4} \right), \text{ мкм};$$

$$R_a = 0,0762 - 0,0109C_1 + 0,0172C_2 + 0,0001C_3 + 0,0781C_1C_2, \text{ мкм}, \quad (16)$$

где C_1 – давление на образец (Р), МПа; C_2 – скорость вращения образца (Vв-в), м/с; C_3 – время притирки (t), с.

Таблица 6. Результаты выполненных и воображаемых экспериментов

Эксперименты	Незалежни фактори			Параметр оптимизации R_a , мкм
	C_1 Давление например, МПа (Р)	C_2 Возвратно-вращательная скорость, м/с (V ₃₋₀)	C_3 Время притирки , с (t)	
1) Воображаемый	0,05	1,1	600	0,193
2) Выполненный	0,05	1,1	600	0,218
3) Воображаемый	0,01	1,1	600	0,163
4) Выполненный	0,01	1,1	600	0,178
5) Воображаемый	0,05	0,3	600	0,148
6) Выполненный	0,1	0,3	240	0,168
7) Воображаемый	0,1	0,3	600	0,143
8) Выполненный	0,1	0,3	600	0,158
9) Воображаемый	0,5	1,1	240	0,157

Эксперименты	Незалежні фактори			Параметр оптимізації R_a , мкм
	C_1 Давление например, МПа (P)	C_2 Возвратно-вращательная скорость, м/с (V_{3-o})	C_3 Время притирки , с (t)	
10) Выполненный	0,5	1,1	240	0,158
11) Воображаемый	0,1	1,1	240	0,131
12) Выполненный	0,1	1,1	240	0,138
13) Воображаемый	0,1	0,3	240	0,107
14) Выполненный	0,1	0,3	240	0,118
15) Воображаемый	0,3	0,3	240	0,109
16) Воображаемый	0,3	0,3	600	0,145
17) Воображаемый	0,3	1,1	420	0,16

Виртуальные эксперименты были прекращены в эксперименте 17. Дальнейшее исследование было прекращено, поскольку, как видно из табл. 6, параметр оптимизации (R_a) начинает расти. Это дает основание полагать, что область оптимума достигнута.

Анализ результатов табл. 6 показал, что наименьшая шероховатость (R_a) обработанной поверхности получена в опытах 13 (мысленный, $R_a=0,107$) и 14 (осуществленный, $R_a=0,118$) при следующих значениях параметров режима: $P = 0,1$ МПа; $V_{3-o} = 0,3$ м/с; $t = 240$ с. При увеличении давления на образец ($P = 0,3$ МПа), при тех же значениях $V_{в-в} = 0,3$ м/с; $t = 240$ с, шероховатость обработанной поверхности незначительно увеличивается, как показано в таблице 3.6 (эксперимент 15, условный, $R_a=0,109$ мкм). При увеличении времени притирки (эксперимент 16, условный, $t = 600$ с) и скорости вращения (эксперимент 17, условный, $V_{в-в} = 1,1$ м/с) шероховатость обработанной поверхности значительно ухудшается, $R_a=0,145$ мкм и $R_a=0,16$, соответственно.

Таким образом, из анализа результатов выполненных и «воображаемых» опытов (табл. 6) видно, что в качестве оптимальных значений параметров режима притирки силикатным абразивным составом можно принять результаты опытов 13 (воображаемый) и 15 (воображаемый) на основании лучшей шероховатости обработанной поверхности.

Учитывая то, что при большем давлении на образец можно устранить более глубокие неровности, предлагаем для достижения минимального значения параметра оптимизации (R_a) следующие значения параметров режима притирки силикатным абразивным составом:

- удельное давление на образец – $P = 0,03$ МПа;
- скорость вращения образца – $V_{в-о} = 0,3$ м/с;
- время притирки – $t = 240$ с.

3. Заключение:

Таким образом, были проведены экспериментальные исследования и получены уравнения математической модели режима притирки силикатным абразивным составом. Установлено, что наибольшее влияние на шероховатость обрабатываемой поверхности оказывает время притирки. Предложены оптимальные параметры режима притирки силикатным абразивным составом для достижения минимального значения шероховатости обработанных поверхностей.

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ПРОМЯНА В ЕЛЕКТРИЧЕСКАТА ПРОВОДИМОСТ НА ОХЛАЖДАЩИТЕ ТЕЧНОСТИ НА ПРЕВОЗНИТЕ СРЕДСТВА В ЗАВИСИМОСТ ОТ КОЛИЧЕСТВОТО РАЗХОД НА ГОРИВО

CHANGE IN THE ELECTRICAL CONDUCTIVITY OF VEHICLE COOLANTS DEPENDING ON THE AMOUNT OF FUEL CONSUMED

Mikhail Nahliuk¹

Abstract. The article presents the results of the change in the electrical conductivity of antifreeze and coolants from the amount of fuel consumed, as well as their graphical dependencies. It was established that with the same amount of fuel consumed by cars, the change in the electrical conductivity of their antifreeze occurs differently.

Key words: antifreeze, cooling liquid, electrical conductivity, amount of fuel consumed.

1. Introduction:

The development of road transport is associated with the production of modern cars, and improving their quality, reliability and durability requires the use of modern high-quality performance materials. In addition, automobiles are subject to increasingly stringent and growing requirements, which are caused by the improvement of various systems and components of transport vehicles to improve safety, efficiency and environmental friendliness. One of the ways to improve the efficiency and environmental friendliness of a car is to improve the engine cooling system by using antifreeze for all-season operation in liquid cooling systems, as well as optimizing the timing of coolant changes.

Antifreeze is a fluid used to cool the vehicle engine and other mechanical systems to prevent them from overheating and freezing at low temperatures. It is an important element of the cooling system that serves to keep the engine running at an optimal temperature. Modern antifreeze is a mixture of water with organic or inorganic additives such as ethylene glycol or propylene glycol. Ethylene glycol and propylene glycol play a key role in lowering the freezing point of the fluid and raising the boiling point, as well as affecting the corrosion resistance of the cooling system. Antifreeze may contain different chemicals depending on the manufacturer and type of antifreeze (e.g. organic acids, silicate compounds, etc.). These components can influence the conductivity of the fluid, which is important for evaluating the safety and functionality of the antifreeze under different conditions.

The conductivity of antifreeze depends on its chemical composition. Water, unlike pure organic compounds, has a certain conductivity due to dissolved minerals and salts. Additives in antifreeze, such as various chemicals (e.g. corrosion inhibitors), can alter its conductivity. Antifreeze generally does not conduct electric current as well as water, but it still has some conductivity. This is due to the fact that it may contain ions that allow the movement of electrical charge. This is especially true when mineral salts or other additives are used in antifreeze that increase its electrical conductivity.

For vehicles equipped with advanced electronic systems, the high electrical conductivity of antifreeze can be a source of potential problems. If antifreeze escapes from the cooling system and gets on electrical components, it can cause short circuits or corrosion, which can damage the vehicle's electrical system. Therefore, it is important to monitor the condition of the cooling system and avoid antifreeze leaks in the area where electrical components are located.

Antifreeze manufacturers conduct conductivity tests on their products to make sure they are safe for use in vehicles and will not damage engine management systems or other electronic components. Typically, antifreeze has a fairly low conductivity, but if it contains salts or other conductive additives, the conductivity can be higher. Interestingly, the conductivity of the antifreeze itself does not significantly affect its cooling capacity. The cooling capacity of an antifreeze is determined by its thermal characteristics, such as boiling and freezing point, as well as the efficiency of its heat dissipation properties. However, it is important that the

antifreeze is properly balanced, as additives that increase conductivity can also change the viscosity or other physical properties of the fluid, which in turn will affect the efficiency of the cooling system.

2. Description:

Nowadays, for each car there are service books that regulate the frequency of antifreeze replacement, taking into account the mileage of the car or the service life of the coolant in time. The replacement is carried out on a first-come, first-served basis. This is not entirely rational, as this approach does not take into account such factors as vehicle operating conditions, technical condition, load and speed conditions, and the actual condition of the antifreeze at the time of replacement. This approach leads to the fact that the fluid being drained may not have reached the end of its service life yet, or vice versa, it may have already begun to cause damage to the cooling system due to the production of additives.

All-season coolant is one of the main functional elements of the engine, which largely determines the reliability and efficiency of its systems. And, accordingly, like any functional element, it needs periodic diagnostics and quality control.

One of the indicators of coolant quality is the electrical conductivity of antifreeze. Issues related to the electrical conductivity of liquids have been considered by the scientific community for a long time. The results of measurements of the electrical conductivity of standard solutions at different temperatures, given in reference [1], show that with increasing liquid temperature, the conductivity also increases. Paper [2] describes studies and presents the results of changes in the electrical conductivity of clean and used samples of various oils. The authors of [3,4] studied the electrical conductivity of various hydrocarbon liquids.

To determine the actual condition of the coolant and its replacement, it is necessary to perform periodic monitoring. As a rule, the period is a value measured in kilometers or time (month, year). However, this does not take into account the fact that when the car is idle with the engine running (warming up, driving uphill, idling at a traffic light, etc.), the coolant continues to work and produce its service life. In work [5], the influence of the coolant on the amount of fuel consumed.

Amount of fuel consumed – is the total amount of fuel a vehicle has used over a period of time or over its entire life (e.g., per trip, day, month, or since purchase). It is usually measured in liters. The “amount of fuel consumed” metric has certain advantages over mileage, namely:

- It shows how much fuel has actually been consumed, rather than simply the distance traveled.
- It allows you to assess how fuel-efficient the car is under various conditions (city driving, highway driving, traffic jams).
- A sharp increase in total fuel consumption may signal engine malfunctions, fuel system issues, aggressive driving, etc.
- Mileage (odometer reading) shows only the distance traveled, while total fuel consumption shows the amount of fuel actually used.
- In addition, mileage alone is less informative because it does not account for driving style, traffic conditions, vehicle load, etc.

Thus, amount of fuel consumed is a comprehensive indicator that more objectively accounts for the actual operating conditions of a vehicle on a daily basis, including the load on its components, the driver's skill level, and road, traffic, and atmospheric and climatic conditions more so than mileage or operating hours.

The purpose of this article is to investigate changes in the electrical conductivity of domestic and foreign-made antifreeze when used in vehicles, depending on the amount of fuel consumed.

To study changes in the electrical conductivity of antifreeze based on the amount of fuel consumed, samples were collected for analysis from passenger cars using various brands and types of fluids, both domestic and foreign. All vehicles were operated under identical road and

climatic conditions. Sampling was conducted after a specific period of operation. At the same time, other quality indicators of the antifreeze were analyzed: alkalinity, corrosion potential of base metals, density, freezing point, and pH.

Based on the results obtained, graphs were plotted showing changes in the electrical conductivity of antifreeze as a function of fuel consumption (Figure 1).

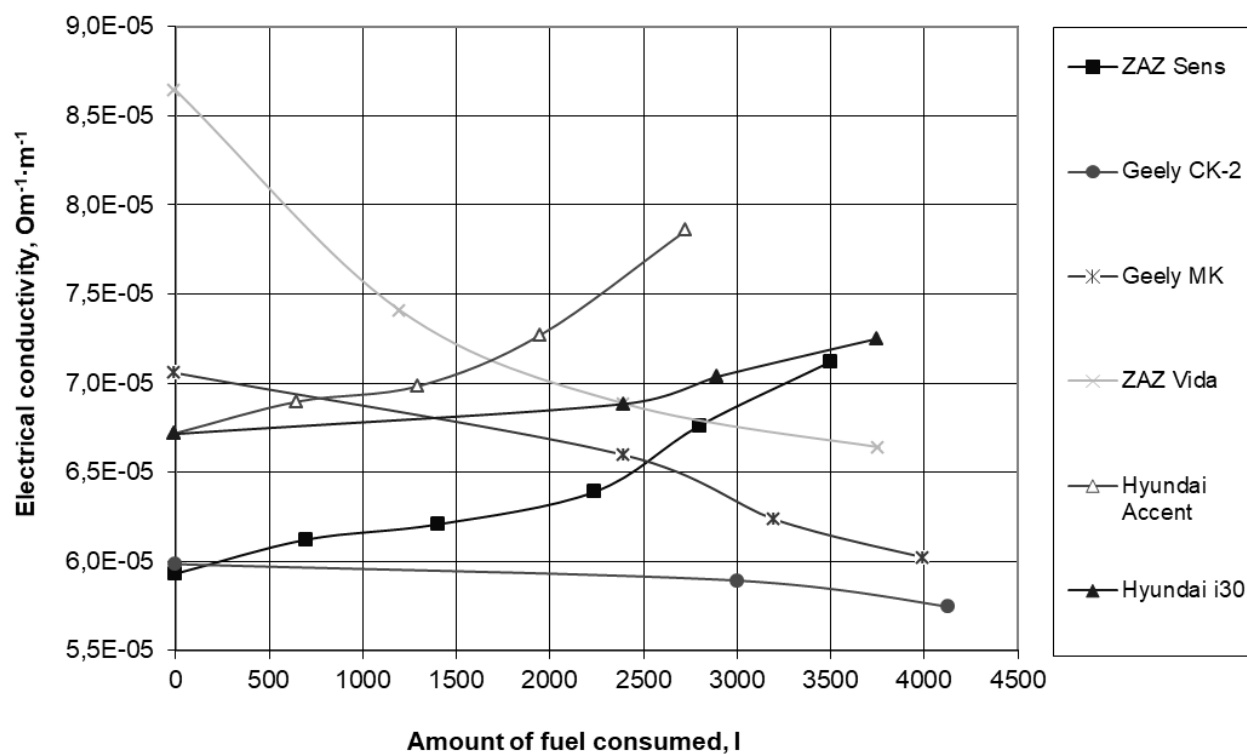


Fig. 1. Graph of the dependence of the change in electrical conductivity of antifreezes on the amount of fuel consumed

The graph shows that the electrical conductivity of pure antifreeze varies from $5,5 \cdot 10^{-5}$ to almost $9 \cdot 10^{-5} \text{ Om}^{-1}\text{m}^{-1}$, and behaves differently as the amount of fuel consumed increases. In ZAZ Sens, Hyundai Accent and Hyundai i30 cars, an increase in the electrical conductivity of coolants is observed. In Geely CK-2, Geely MK and ZAZ Vida cars, a decrease in the electrical conductivity of antifreezes is observed. This indicates that the change must be made individually for a specific car, taking into account the design features, the quality of the antifreeze used and the amount of fuel consumed.

3. Conclusions:

Diagnosing the quality of antifreeze by electrical conductivity, taking into account the amount of fuel consumed, you can more reliably judge the actual condition of the working antifreeze.

When specifying the frequency of diagnosing the coolant, it is necessary to set the limit value of electrical conductivity individually for each car, taking into account the technical condition of the engine and its systems, the quality of the antifreeze used, the amount of fuel consumed and external operating conditions.

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СРАВНЕНИЕ НА ЕКСПЛОАТАЦИОННИЯ СРОК НА ДИСКОВИТЕ И БАРАБАНИТЕ СПИРАЧНИ МЕХАНИЗМИ В ЛЕКИТЕ АВТОМОБИЛИ

COMPARISON OF THE SERVICE LIFE OF DISC AND DRUM BRAKE MECHANISMS IN PASSENGER CARS

Oleksandr Nazarov¹, Mykola Serhiienko², Yevgen Kalashnikov³, Illia Pavlenko⁴

Abstract. A mathematical model of wear on the friction surfaces of disc and drum brake mechanisms in the studied passenger cars is proposed. For the passenger cars under study, a methodology for determining theoretical wear under specific braking conditions is considered; results of theoretical and experimental studies of wear on the rotors of disc and drum brake mechanisms are presented, based on which conclusions regarding brake service life are drawn.

Keywords: passenger car, disc brake mechanism, drum brake mechanism, friction surface, wear, operation.

1. Introduction:

The practical operation of passenger cars, as well as an analysis of the causes of failure of various mechanisms and assemblies, clearly indicates that most of them lose their operational capability not due to breakdowns, but due to the wear of the working surfaces of individual parts and assemblies. Numerous studies [1–5] have shown that approximately 70% of vehicle failures during operation are caused by wear in friction assemblies, particularly in disc-type braking mechanisms, which, due to their high responsiveness, absorb the majority of the vehicle's kinetic energy.

Constantly increasing demands on the quality of vehicle repairs necessitate extending the service life of their mating parts.

An important factor in assessing the service life of brakes on passenger cars in operation is determining the wear ratio of individual brake mechanism components. Since the most critical part of a vehicle's braking system – the brake mechanism – operates under varying wear conditions, it is essential to identify the wear patterns of mating parts that generate friction, depending on changes in their geometric parameters and operating modes during service.

2. Description:

Analysis of recent achievements and publications. Due to a number of reasons (varying values of pressure and relative sliding speeds on the friction surface, uneven abrasive ingress during operation, etc.), the wear of passenger car brake mechanism components is not uniform. Therefore, to fully characterize the extent of component wear, it is necessary to know its distribution [6].

As a result of wear of the “rotor-pad” interface, a change in the relative position of the friction surfaces occurs, which we will hereafter refer to as interface wear; this is the geometric characteristic directly associated with the loss of the vehicle's or its mechanism's initial performance properties.

Purpose of the work and problem statement. The purpose of this work is to compare the service life of disc brake mechanisms installed on the front axle and drum brakes installed on the rear axle of passenger cars.

Mathematical model and algorithm for solving the problem. To determine the parameters that characterize the wear of the coupling, we will consider what changes in their relative positions may result from the wear of their surfaces.

In this context, the design and kinematic features of this pair are of great importance, as they determine the nature and direction of possible movement of the parts during wear [6].

Fig. 1 and Fig. 2 show diagrams for determining the wear of disc and drum brake mechanisms.

The wear laws, knowledge of which is necessary to solve the given problem, must, under the given conditions, determine the relationship between the wear rate of each of the mating

parts, the contact pressure on the friction surface p , and the relative sliding speed ϑ .

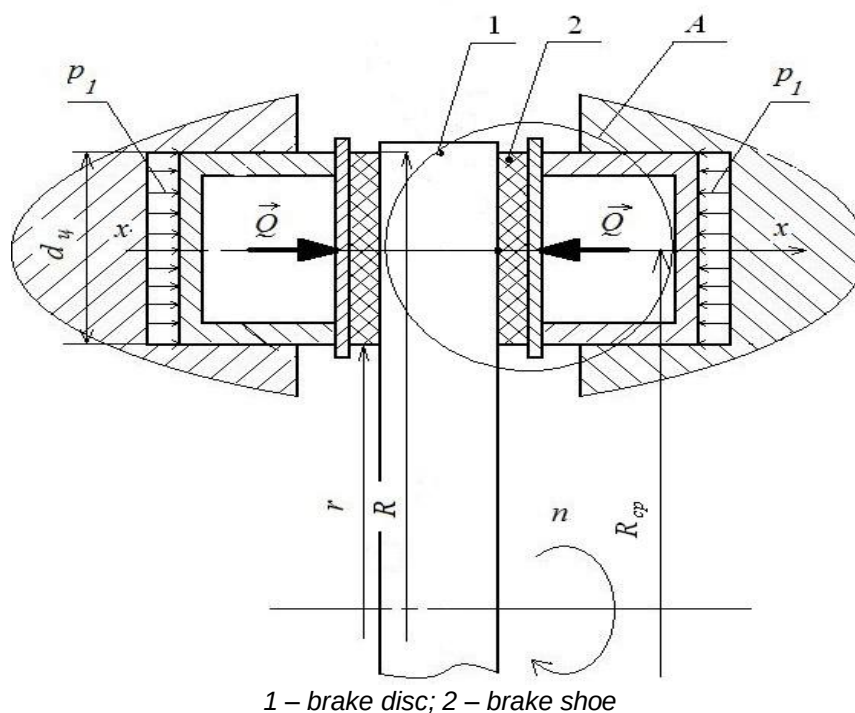


Fig. 1. Diagram for determining wear in a disc brake mechanism

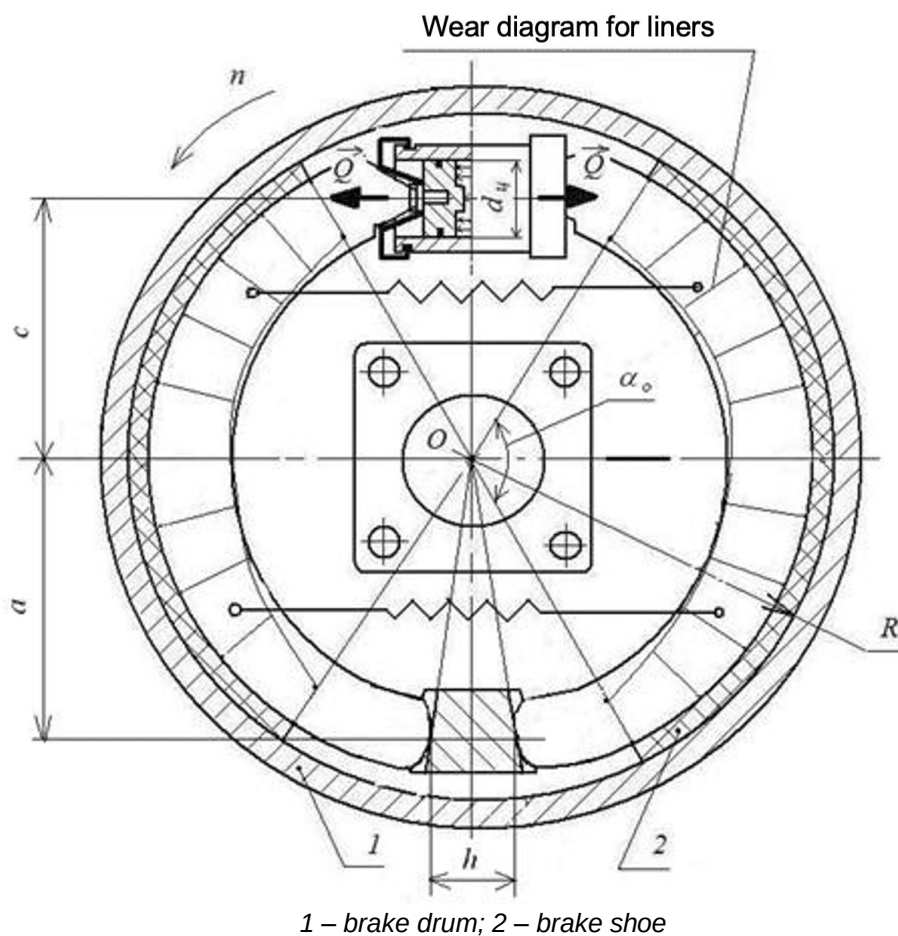


Fig. 2. Diagram for determining the wear of a drum brake

As studies [3–7] show, in many cases, power-law wear laws in general form can be accepted

$$\gamma_1 = k_1 \cdot p^{m_{p1}} \cdot g^{m_{v1}}; \quad (1)$$

$$\gamma_2 = k_2 \cdot p^{m_{p2}} \cdot g^{m_{v2}}; \quad (2)$$

where: k_1, k_2 are coefficients characterizing the wear rates of the friction surfaces of bodies 1 and 2, respectively.

In the case of abrasive wear, the amount of wear is proportional to the pressure and the friction distance (braking distance), and the exponents are $m_{pi} = m_{vi} = 1$ [7].

In this case, the dependencies take the form

$$\gamma_1 = k_1 \cdot p \cdot g; \quad (3)$$

$$\gamma_2 = k_2 \cdot p \cdot g. \quad (4)$$

For a given relative velocity of the mating parts, every point on the friction surface has a circular sliding velocity

$$g = \frac{d\varphi}{dt} \cdot \rho = \omega \cdot \rho, \quad (5)$$

where: $\omega = \frac{d\varphi}{dt}$ – the angular velocity of one of the mating components (brake disc) relative to the other component (brake pad) at the moment the vehicle brakes; ρ the radius of rotation of the point in question, for example, R and R_{cp} (see Fig. 1 and Fig. 2).

Taking into account the geometric parameters of the disc brake mechanisms used on the front axle and the drum brake mechanisms used on the rear axle of the passenger cars under study, the design coefficients of the brake mechanisms were established and summarized in Table 1.

Table 1. Values of design parameters for braking mechanisms

Vehicle	Braking mechanism								
	Disc			Drum					
	d_u , mm	R_c , mm	K_1 , mm ²	d_u , mm	D , mm	a , mm	c , mm	h , mm	K_2 , mm ²
Daewoo Lanos	48	104.6	423.5	19.05	200	77	74	17.7	291
Lada Priora	50	108.5	476.5	19.05	200	77	74	16	293
Chevrolet Aveo	52	104.6	497	20.64	200	77	74	17.7	341
Forza	52	104.6	497	20.64	200	77	74	18.6	340
Vida	52	104.6	497	20.64	200	77	74	17.7	341

We will determine the braking distance of a passenger car under the condition that the kinetic energy of the car's translational and rotational motion is dissipated by braking forces, taking into account aerodynamic drag

Analysis of experimental results. To study changes in the wear of the friction surfaces of the disc and drum brakes on the front and rear axles of the passenger cars under investigation

(Daewoo Lanos, Lada Priora, Chevrolet Aveo, Forza, Vida), equipped with 185/70 R14 tires, brake pads from two manufacturers were used (Fig. 3):

- Manufacturer 1;
- Manufacturer 2.

To avoid advertising, the manufacturers are not named.

Based on the experimental data, histograms of the wear distribution of the brake discs and drums for the left and right wheels of each axle of the vehicle were constructed, as shown in Fig. 3 in corresponding identical colors.

The wear of the brake discs and drums of the passenger cars under study occurred over a braking distance corresponding to a mileage of 90,000 km.

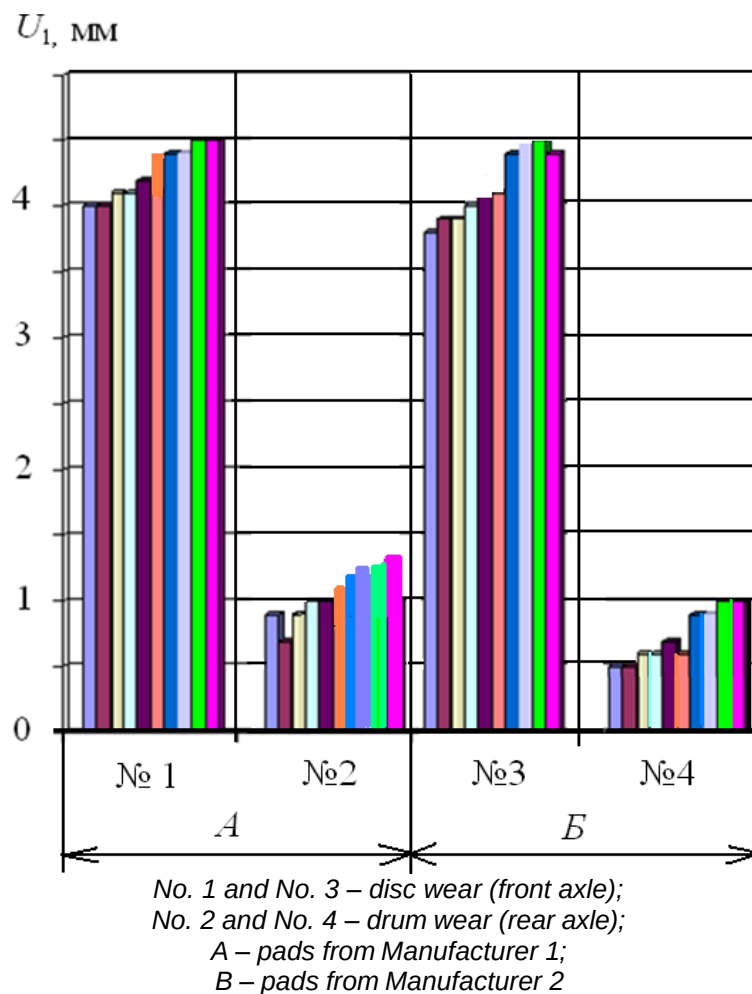


Fig. 3. Histogram of the distribution of brake mechanism wear

Based on experimental data from the passenger cars under study, histograms (Fig. 4, Fig. 5) were constructed showing the distribution of the ratio of drum brake rotor wear to disc brake rotor wear depending on the car model, designated as follows: 1 and 2 – Daewoo Lanos; 3 and 4 – Lada Priora; 5 and 6 – Chevrolet Aveo; 7 and 8 – Forza; 9 and 10 – Vida.

Analysis of the histograms (see Fig. 4 and Fig. 5) shows that under equal operating conditions for the passenger cars under study, in whose brakes friction material from Manufacturer 2 was used, the ratio of wear between drum and disc brake rotors ranges from 12.9% to 25.0%, whereas for vehicles equipped with friction material from Manufacturer 1, it ranges from 17.5% to 33.8%. This means that the wear rate of drum brake rotors is 4–7.75 times lower than that of disc brake rotors from Manufacturer 2, and 2.96–5.71 times lower than that of Manufacturer 1.

However, based on a comparison of the ratio of the corresponding minimum wear rates for

friction materials, for example, from Manufacturer 2 and Manufacturer 1, which is 0.737, and the corresponding maximum wear rates, which is 0.739, it can be concluded that the wear rate of the friction surfaces of brake rotors from Manufacturer 2, is almost identical to that of rotors from Manufacturer 1, and thus their service life is also nearly the same.

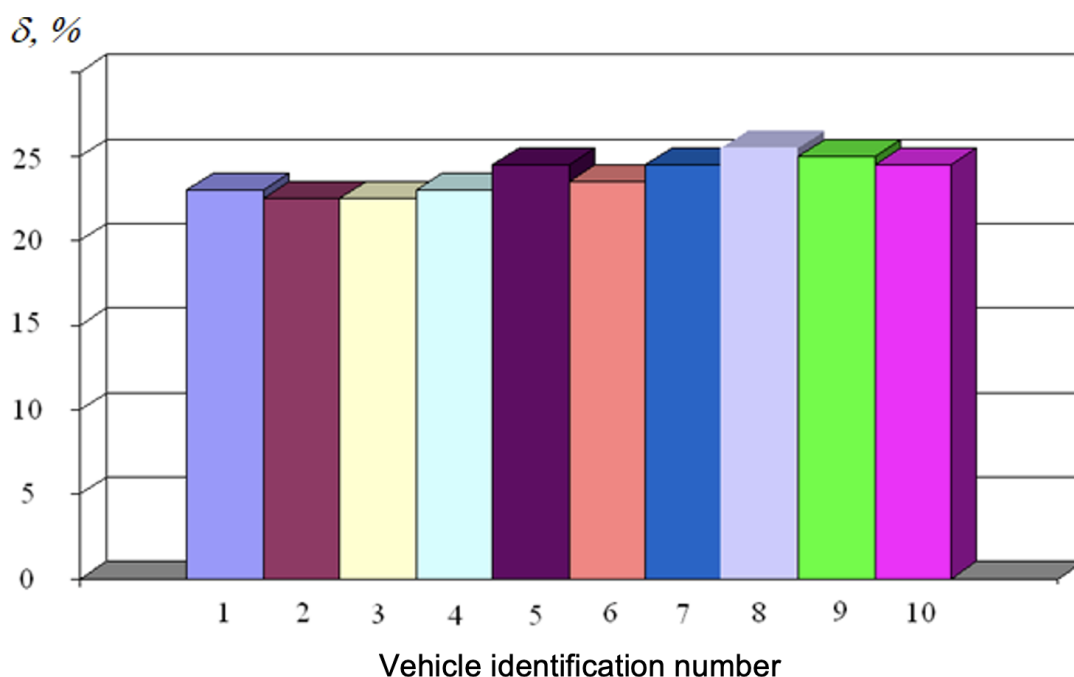


Fig. 4. Histogram of the wear ratio of drum and disc brake rotors (manufacturer 1)

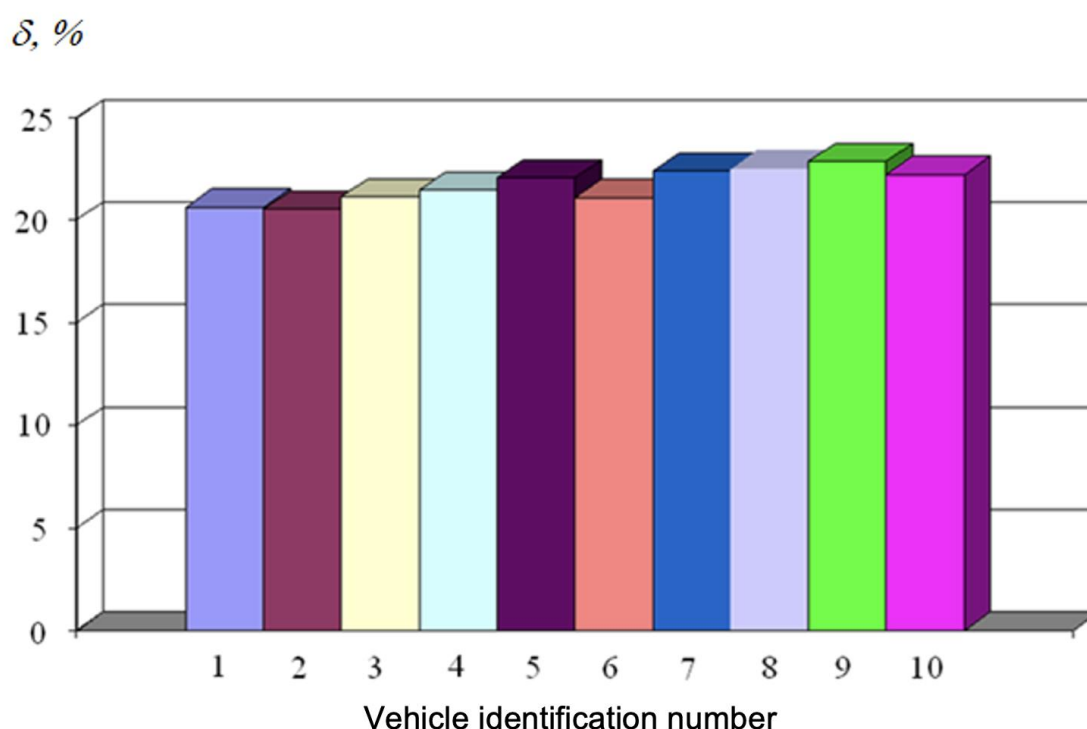


Fig. 5. Histogram of the wear ratio of rotors in drum and disc brake mechanisms (manufacturer 2)

Thus, an analysis of the histograms showing the distribution of wear ratios for the rotors of drum and disc brake mechanisms indicates that the coefficients characterizing the wear rate of the friction material from Manufacturer 2 and from Manufacturer 1 do not differ.

Prospects for further research. The authors consider promising areas of research to be

those related to the study of wear in disc and drum brake mechanisms of passenger cars in service.

3. Conclusions:

A methodology has been developed to determine the wear rate of the friction surfaces of disc and drum brakes in passenger cars. However, this is possible only for known values of the coefficient characterizing the wear rate of the friction material used and the braking distance of the vehicle over a specific mileage.

Based on experimental values of brake friction surface wear for the studied vehicles, it was established that the ratio of drum rotor wear to disc rotor wear in braking mechanisms using friction material from Manufacturer 1 ranges from 17.5% to 33.8%, whereas for vehicles using friction material from Manufacturer 2, it ranges from 12.9% to 25.0%.

Based on a comparison of the ratios of the corresponding minimum wear and maximum wear for the friction materials of both manufacturers, it can be concluded that the wear rate of the friction surfaces of brake rotors from Manufacturer 2, compared to the wear rate of rotors from Manufacturer 1, and thus their service life, is the same.

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ВЛИЯНИЕТО НА АДАПТИВНОТО ОКАЧВАНЕ ВЪРХУ ЕКСПЛОАТАЦИОННИЯ СРОК И ФУНКЦИОНАЛНАТА БЕЗОПАСНОСТ НА ПРЕВОЗНОТО СРЕДСТВО

THE IMPACT OF ADAPTIVE SUSPENSION ON VEHICLE SERVICE LIFE AND FUNCTIONAL SAFETY

Vyacheslav Pavlenko¹, Vitaliy Pavlenko²

Abstract. The article investigates the influence of active suspension systems on the active safety and operational reliability of modern vehicles. A comparative analysis of passive and active suspension systems was conducted using mathematical modeling of dynamic loads. The author substantiates that the implementation of model predictive control (MPC) algorithms allows for a 47% reduction in vertical body accelerations and a 54.2% reduction in pitch angular accelerations. Special attention is paid to the correlation between dynamic stress amplitude and the fatigue life of vehicle structural components. It is proved that reduced vibration loading contributes to an exponential increase in component resource. Recommendations are formulated for transitioning to proactive maintenance strategies for active systems to ensure a consistently high level of traffic safety.

Keywords: active suspension, traffic safety, operational reliability, fatigue life, model predictive control (MPC), mathematical modeling, technical diagnostics.

1. Introduction:

The current stage of automotive development is characterized by stringent requirements for active safety and operational reliability of vehicles. The key component determining these characteristics is the suspension, which provides a flexible connection between the wheels and the body. Traditional passive systems always represent a compromise between ride comfort and handling, which limits a vehicle's capabilities in extreme driving conditions.

An active suspension, equipped with an onboard computer and actuators, is capable of controlling the vertical movement of each wheel individually in real time (Fig. 1). This allows not only to neutralize the impact of road irregularities but also to dynamically adjust stiffness and damping depending on speed, steering angle, and road surface conditions [1].

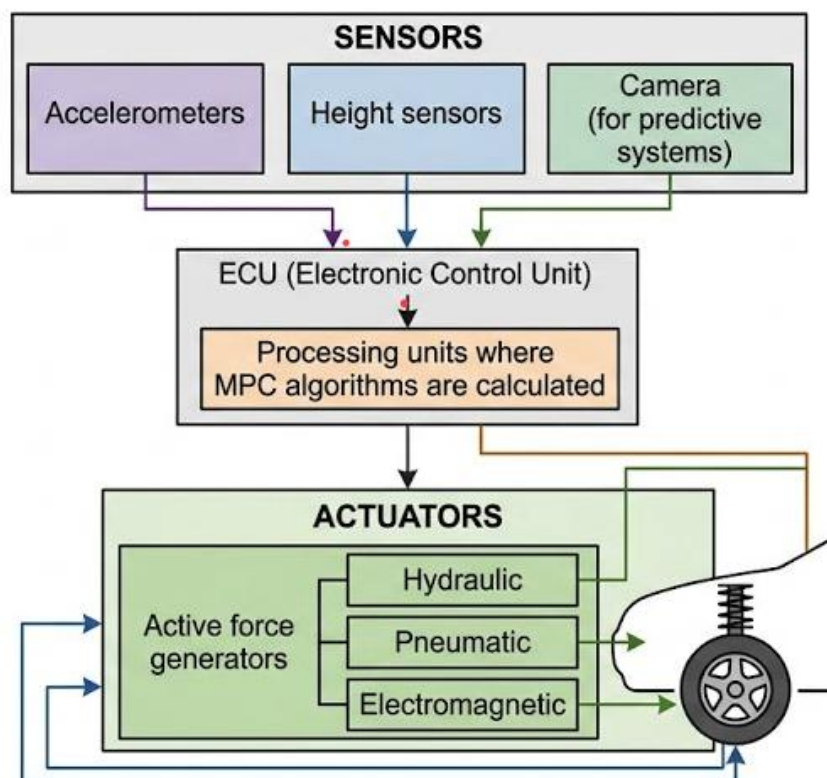


Fig. 1. Active suspension system architecture

The relevance of this study stems from the need for a scientific justification of how the transition to active systems affects the service life of the vehicle body and traffic safety. The implementation of intelligent control algorithms minimizes roll and vertical acceleration, which directly correlates with a reduction in fatigue stresses in the load-bearing structural elements.

The application of mathematical reliability models and dynamic load analysis makes it possible to evaluate the effectiveness of such systems not only in terms of passenger comfort but also as a means of extending the vehicle's service life. The aim of this work is to analyze the relationship between active suspension control parameters and indicators of the vehicle's technical condition under conditions of intensive use.

2. Description:

The assessment of the operational reliability of an active suspension is based on an analysis of the probability that components will remain failure-free over a specified time t . For complex technical systems [2], the exponential reliability model is the most appropriate:

$$P(t) = e^{-\lambda t}, \quad (1)$$

where: λt is the failure rate, which, for active systems, depends not only on mechanical wear but also on the reliability of the electronic control components.

To study dynamic stability, the differential equation of motion for the sprung mass of the vehicle body is used; a conceptual diagram of this equation is shown in Fig. 2. In an active system, unlike a passive one, an additional control force vector $F_{control}(t)$ is introduced, which allows for real-time compensation of external disturbances:

$$m \cdot \ddot{z} + c \cdot (\dot{z} - \dot{q}) + k \cdot (z - q) = F_{control}(t), \quad (2)$$

where: m — vehicle body mass; z and q — vertical coordinates of the vehicle body and the road profile; k , c — stiffness and damping coefficients.

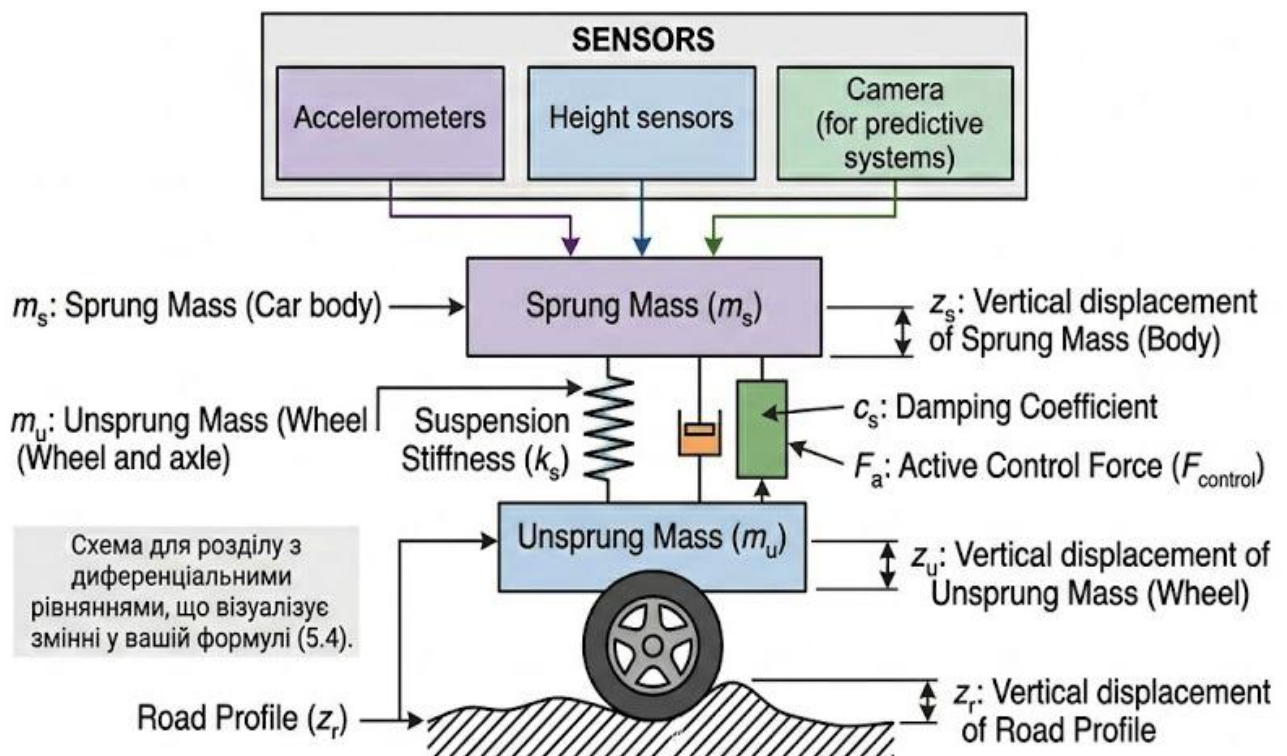


Fig. 2. Quarter-car mathematical model

Reducing the amplitude of dynamic loads directly affects the fatigue life of metal structures. The relationship between the stress amplitude σ_a and the number of cycles to failure N is described by the Veller curve equation:

$$\sigma_a^m \cdot N = \text{const} , \quad (3)$$

The use of active suspension allows for a reduction in the value of σ_a , which, due to the exponent m , leads to an exponential increase in the service life N of the vehicle's load-bearing components.

The implementation of active suspension systems with modern Model Predictive Control (MPC) algorithms [3] allows for a significant optimization of a vehicle's dynamic performance. The control is based on an optimization model that minimizes the cost function, whose parameters are the state vector x_k , the control input u_k , and the weight matrices Q , R , and Q_f . The matrix Q accounts for the energy cost of the vehicle's state deviation, while the matrix R limits excessive use of drive energy. As a result, the algorithm enables the system to predict changes in the body's position in advance and generate the optimal control force to ensure maximum stability.

Simulation studies fully confirm the effectiveness of this approach. As shown in the comparative frequency response graph (Fig. 3), the use of MPC algorithms reduces the vertical acceleration of the vehicle body by 47 % compared to a passive suspension. At the same time, there is a 54.2% reduction in pitch angular acceleration and a 15.5% reduction in roll angular acceleration during cornering.

Reducing the amplitude of these accelerations is a key factor in improving operational reliability. Passive suspensions often operate in resonance modes, leading to an uncontrolled increase in vibrations. In contrast, an active suspension effectively suppresses resonance phenomena, significantly reducing the amplitude of vibrations and inertial loads on structural components. This smooths out shock impacts and minimizes microdeformations in materials, preventing the rapid accumulation of fatigue damage.

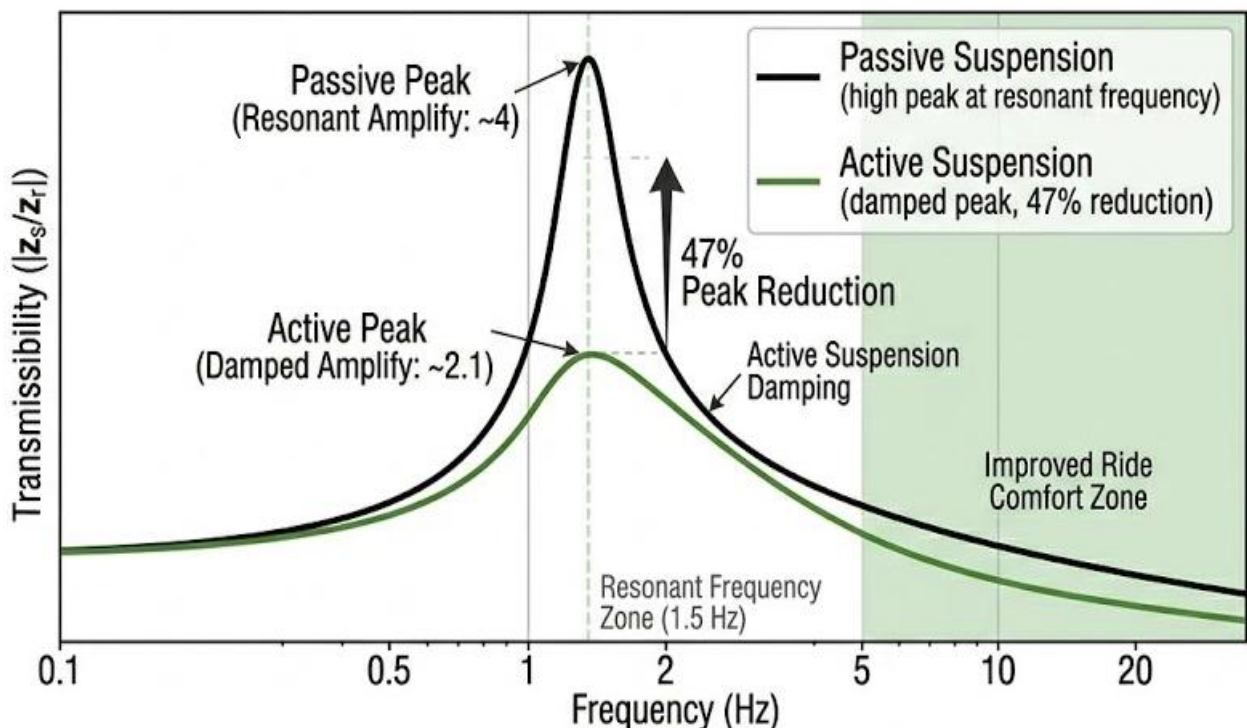


Fig. 3. Amplitude-Frequency Response Comparison

A comparison table has been developed to comprehensively assess the discrepancies (table 1).

Table 1. Comparison of the performance characteristics of passive and active suspensions

<i>Description</i>	<i>Passive suspension</i>	<i>Active suspension</i>
How it works	Conversion of the kinetic energy of vibrations into heat using springs and shock absorbers.	The use of additional energy sources and actuators to generate counteracting forces.
Adaptability	It represents a fixed compromise and is unable to adapt to changing driving conditions.	It adjusts stiffness and damping properties in real time, compensating for roll and pitch.
Impact on reliability	It causes peak stresses and accelerated accumulation of fatigue damage.	Significantly dampens impact forces, minimizing the amplitude of cyclic stresses in critical components.
Service	For the most part, it involves replacing worn parts only after obvious symptoms have appeared.	This requires a proactive approach and comprehensive monitoring of fluids and electronics using specialized tools.

3. Conclusions:

This study confirms that the implementation of active suspension is an effective means of improving a vehicle's active safety and operational reliability. Thanks to the use of model predictive control (MPC) algorithms, it was possible to reduce body vertical accelerations by 47 % and pitch accelerations by 54.2 %, which significantly improves the vehicle's dynamic stability.

Mathematical modeling using the exponential reliability model $P(t) = e^{-\lambda t}$ and analysis using the Veller curve demonstrated that a reduction in the amplitude of dynamic stresses σ_a directly leads to an increase in the fatigue life of the body's load-bearing elements. Thus, the active suspension acts as a damper not only for passenger comfort but also to preserve the integrity of the vehicle's metal structures.

It has been established that the transition to intelligent suspension systems requires a paradigm shift in maintenance – from reactive repair to proactive diagnostics. The effectiveness of such systems critically depends on the stability of electronic components and the cleanliness of operating media (hydraulic fluids or air), which underscores the importance of using specialized diagnostic tools.

Prospects for further research lie in the deeper integration of suspension systems with V2X (Vehicle-to-Everything) technologies and cloud services to obtain preliminary data on road surface conditions, which will allow for even greater optimization of vehicle energy efficiency and durability.

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ОБОСНОВКА И ИЗБОР НА КРІ ЗА СИСТЕМИ ЗА УПРАВЛЕНИЕ НА ПЪТНОТО ДВИЖЕНИЕ В ГРАДСКИ МРЕЖИ

JUSTIFICATION AND SELECTION OF KPIS FOR TRAFFIC MANAGEMENT SYSTEMS IN URBAN NETWORKS

Hennadii Ptytsia¹, Liudmyla Abramova²

Abstract. This publication presents the results of the selection of criteria and methods for the quantitative evaluation of traffic management solutions in urban street and road networks. A comprehensive approach is substantiated, integrating key performance indicators such as delay, capacity, environmental safety, and level of service (LOS). Particular attention is given to the role of LOS as a universal criterion for assessing traffic conditions across different network elements. The study highlights the effectiveness of traffic control measures based on technical regulation tools and emphasizes the importance of analyzing critical traffic volumes to identify transitions to congested conditions. The findings demonstrate that improving the performance of urban road networks requires a system-based approach that combines infrastructure development, traffic management strategies, and socio-economic planning.

Keywords: traffic management; level of service (LOS); traffic delay; traffic flow; traffic safety; urban transport systems.

1. Introduction:

Road traffic, due to a combination of factors, is associated with significant negative social, economic, and environmental impacts. Every year, hundreds of thousands of people are killed in road traffic accidents, while tens of thousands are injured. During peak periods, traffic speeds decrease to 5–10 km/h. This effectively undermines the fundamental concept of the automobile, which is based on saving time in the transportation of passengers and goods through higher travel speeds.

The recognition of the need for urgent intervention in this evolving process has made it possible to formulate general directions for solving the problem. These directions involve identifying cause-and-effect relationships between transport systems, the environment, and society. This approach enables the development and implementation of measures aimed at improving the efficiency of the transport process while minimizing its negative impacts. The primary requirement for traffic management systems and road safety management systems is to minimize travel time while ensuring traffic safety.

The rapid growth of the vehicle fleet and transportation volumes leads to an increase in traffic intensity. In cities with historically developed urban structures, this results in the aggravation of transport problems and an increase in the number of road traffic accidents. Under such conditions, enhancing the role of public transport becomes one of the key approaches to improving the overall efficiency of the transport system.

At the same time, the effective functioning of public transport requires the proper development of infrastructure. It is necessary to create appropriate conditions for the movement of both public transport vehicles and the general traffic flow. These conditions should comply with the principles of efficiency and rationality.

Special attention should be paid to the organization of public transport stops and their types, depending on planning and operational solutions. These locations are characterized by complex vehicle maneuvering conditions. In addition, it is essential to ensure safe and convenient conditions for pedestrian movement. This significantly affects both safety and traffic speed.

Safety at public transport stops is one of the key priorities of their functioning within the urban street and road network. These locations involve interactions between vehicles, pedestrians, and passengers, who are the most vulnerable road users. At the same time, the operation of stops should not significantly reduce the capacity of the street and road network. Moreover, they must ensure a high quality of transport services.

This study is devoted to the substantiation of criteria for improving traffic efficiency on sections of the street and road network in medium-sized and large cities that include public transport stops. The application of theoretically grounded criteria makes it possible to select effective organizational and planning measures. In turn, this ensures an improvement in traffic performance and contributes to the development of efficient traffic management systems.

2. Selection of Criteria and Methods for Quantitative Evaluation of the Effectiveness of Traffic Management Solutions:

Various criteria for evaluating the efficiency of transport system operation have been proposed by the authors of studies [1–6]. These criteria include economic indicators, road safety indicators, environmental safety indicators, as well as indicators of the sustainability of the street and road network functioning.

The wide range of practical tasks and diverse situations encountered in the design of street and road networks necessitates the use of an extensive set of quantitative indicators. A generalized set of indicators commonly applied in traffic management practice is presented in [4].

In accordance with the objective of developing a methodology for evaluating the effectiveness of a set of measures aimed at improving the performance of the street and road network, it is advisable to apply a system of criteria and corresponding indicators (Table 1).

Table 1. Set of Criteria and Indicators Used for Evaluating the Effectiveness of Design Solutions in the Street and Road Network

Criterion	Quantitative Indicators
Delay and Queue Length	Average Delay
	Total Vehicle Delay
	Maximum Vehicle Delay
	Queue Length
Capacity	Road Network Density
	Road Network Capacity
Vehicle-Kilometers Traveled	Total System Travel Time
Environmental Safety	Carbon Monoxide Emission Rate
	NOx Emission Rate
	Road Traffic Noise Level
Road Safety	Reduction in Road Traffic Injuries and Fatalities
	Crash Risk Based on Total Traffic Conflict Load
Economic Effect	Cumulative Economic Effect - Cumulative Effect over the Analysis Period
	Profitability Index – Benefit-Cost Ratio;
	Payback Period - Time Required for the Cumulative Effect to Become Non-Negative
Operational Stability of the Street and Road Network	Stability Coefficient
Network Level of Service	Level of Service - Volume-to-Capacity Ratio
	Service Volume

Average delay is most commonly used as a criterion for optimizing traffic control at the local level. Such applications include individual intersections, public transport stops, and other elements of the street and road network. The value of average delay is closely related to traffic volume, signal control parameters, and queue length. The accuracy of its calculation method is of fundamental importance. This is because queue length and total delay are determined based

on average delay. In the specialized literature, Webster's formula is recommended for practical estimation of average delay.

Total delay is defined as the delay experienced by all vehicles over a given time period. This indicator is used to assess the economic efficiency of the street and road network at the city level or within individual urban districts. In contrast, average delay primarily reflects the level of service for an individual vehicle.

At the level of the entire urban street and road network, approaches to efficiency assessment remain ambiguous. A wide range of viewpoints and methods exists. In urban planning practice, road network density is most commonly used as a key indicator. It is applied in combination with regulatory requirements for streets of different categories. It is generally assumed that compliance with these density standards ensures sufficient capacity.

Some design organizations use the concept of road network capacity. This indicator represents the maximum number of vehicles that can simultaneously operate within a given area.

Traffic engineering specialists often associate capacity exhaustion with the occurrence of congestion. However, in practice, this approach does not provide a clear calculation method. This is due to the need to define acceptable congestion parameters, including the number of congested locations, their length, duration, and other characteristics.

Environmental safety indicators typically include total emissions of carbon monoxide and nitrogen oxides per unit time. They also include the equivalent traffic noise level measured at a distance of 7.5 m from the edge of the roadway. Most modern transport models allow the estimation of total vehicle emissions both in urban environments and on interurban roads.

The effectiveness of road safety improvement measures is primarily reflected in the reduction of losses caused by road traffic accidents. These losses are typically divided into three groups. The first group includes losses associated with fatalities and injuries. The second group covers material damage to vehicles, road infrastructure, and cargo. The third group consists of societal costs. These include costs related to traffic disruption, as well as expenditures of control authorities, courts, and law enforcement agencies.

The operational stability of the street and road network is defined as its ability to maintain capacity when individual elements are partially or completely removed from the transport process. Such disruptions may be caused by traffic accidents, failures of utility systems, mass events, and other factors.

Level of service indicators are determined using methods of queuing theory. Their calculation is typically complex and requires significant computational effort. Therefore, simplified indicators, such as various forms of service coefficients, are often used in practice:

$$k = N / P \quad (1)$$

where: N – Arrival Rate; P – Service Rate.

This indicator meets the requirement that a criterion should be understandable to a wide audience and should be determined both through field observations and analytical calculations. The indicator is referred to as the "Level of Service" and is defined as "a qualitative measure that reflects such combined factors as speed, travel time, freedom to maneuver, safety, and driving comfort" [1]. These definitions are presented with minimal changes in the editions of the Highway Capacity Manual [4, 7] (Table 2).

Level of Service (LOS) is the primary criterion for evaluating the quality of traffic management in European countries and in the United States. This indicator has been incorporated into regulatory documents. At present, a comprehensive system of LOS indicators has been established.

The Highway Capacity Manual [4] provides LOS measures for most elements of the street and road network. For each mode of movement, including vehicular traffic and pedestrians, as well as for each type of network element, specific indicators and corresponding evaluation

methods are applied. These elements include road segments, intersections, sidewalks, pedestrian crossings, and others.

Thus, traffic conditions for vehicles, pedestrians, and other users can be assessed across different elements of the street and road network using a unified criterion. This criterion is the Level of Service (LOS), also referred to as the level of comfort.

Table 2. Levels of Service and Comfort Level Classifications

Level of Service (LOS)	Volume-to-Capacity Ratio	Traffic Operating Conditions	Level of Comfort	Volume-to-Capacity Ratio	Traffic Operating Conditions
A	<0,1	Free-Flow Traffic	A	<0,2	Free-Flow Traffic
B	>0,1	Steady Traffic Flow	Б	0,2-0,45	Partially Constrained Traffic Flow
C	>0,3	Steady Traffic Flow	B	0,45-0,7	Constrained Traffic Flow
D	>0,7	Approach to Unstable Traffic Flow	Г-a	0,7-1,0	Saturated Traffic Flow
E	>1,0	Unstable Traffic Flow	Г-6	>1,0	Highly Saturated Traffic Flow

The main areas of application of LOS indicators include [8]:

- operational analysis, which involves determining the level of service under given road and traffic conditions, as well as developing traffic management measures and design solutions;
- design-stage analysis, which includes the evaluation of transport service quality based on medium- and long-term forecasts, as well as the determination of design parameters of transport infrastructure to ensure a specified level of service;
- planning-stage analysis, which involves assessing service quality in the context of long-term forecasts and strategic transport system development.

According to [9], the street and road network can be considered as a queuing system. In this case, the evaluation of the network and its individual elements can be reduced to two key indicators. The first is capacity, defined as the maximum number of demands that the network can serve. The second is the Level of Service, which reflects the quality of service provided.

The factors that determine the level of network loading are complex in nature. They result from the combined influence of multiple, often conflicting, causes. Accordingly, the measures aimed at regulating these processes must also be comprehensive. This requires a systems approach. Such an approach encompasses socio-economic development planning, urban and transport planning, traffic management and control, as well as road construction (Fig. 1)

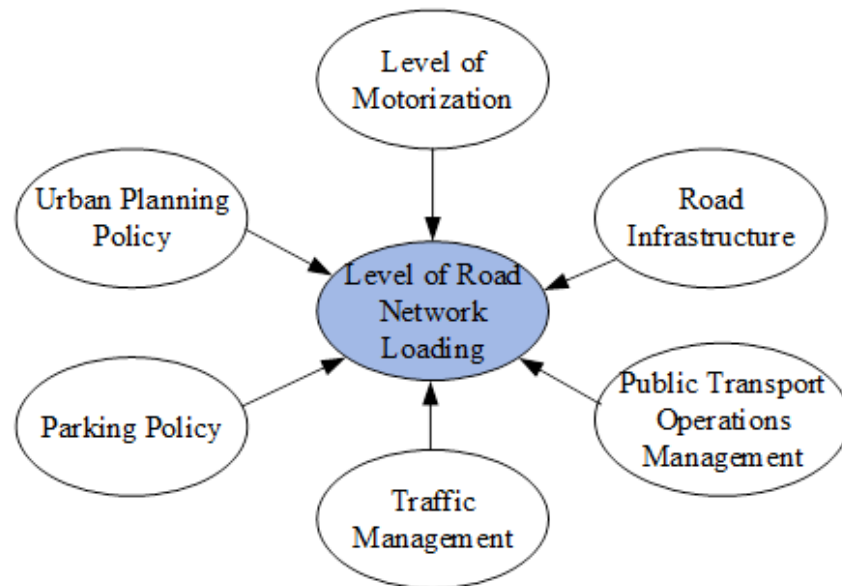


Figure 1. Factors Influencing Road Network Load

3. Conclusions:

The proposed approach makes it possible to develop recommendations for improving the operational efficiency of a network segment. These recommendations are based on the analysis of critical traffic volumes at which traffic flow conditions rapidly approach congestion.

It has been established that the evaluation of the operational efficiency of the street and road network should be based on a comprehensive approach that integrates indicators of delay, capacity, environmental safety, and level of service.

It has been determined that the Level of Service (LOS) is a universal criterion for assessing the quality of traffic management across various elements of the transport network.

It has been found that the analysis of critical traffic volumes makes it possible to identify, in a timely manner, the conditions under which traffic flow transitions to a congested state.

Based on the conducted review and research, it has been concluded that improving the efficiency of street and road network segments requires a comprehensive and system-based approach that takes into account infrastructure, organizational, and socio-economic factors.

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ЕФЕКТИВНОСТ НА ЛОГИСТИЧНИТЕ СИСТЕМИ ПРИ ДОСТАВКА НА ПОСЛЕДНАТА МИЛЯ

EFFICIENCY OF LOGISTICS SYSTEMS IN LAST-MILE DISTRIBUTION

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Abstract. The article examines the modern features of the functioning of logistics systems for cargo delivery in the context of increasing complexity of flow processes, the development of e-commerce and increased requirements for the level of service. It is established that the increase in the volume of batch deliveries leads to an increase in the number of operations, the complication of route networks and an increase in the load on the logistics infrastructure. Based on the analysis of modern research, it is determined that a significant share of logistics costs is formed at the last mile stage and at the points of interaction of elements of the logistics chain, where losses of time and resources occur. The aim of the study is to determine the impact of logistics system parameters on cargo delivery costs and develop practical recommendations for improving the efficiency of its operation. The work uses mathematical modeling and regression analysis methods to assess the impact of key factors, including the density of the retail network, service radius, market share, turnover volumes, and vehicle load factor. The obtained parameter values show the nature of the influence and indicate the presence of rational areas of their values, within which the minimization of logistics costs is achieved. The existence of optimal relationships between the number of points on the route and the radius of half demand is proven. The practical significance of the results is that they can be used to improve the efficiency of logistics planning, optimize delivery routes, and reduce last-mile costs. The results of the study can be applied by retailers and logistics operators in conditions of uncertainty and limited resources.

Keywords: logistics system, logistics costs, last mile; cargo delivery, optimization, logistics network, efficiency, delivery system.

1. Introduction:

Modern logistics systems, particularly in the fields of distribution and product delivery, are undergoing continuous transformation under the influence of the external environment and digitalization. This is driven by the rapid growth of e-commerce and increasing requirements for service speed and reliability. Under such conditions, the issue of improving the efficiency of operational processes within logistics systems becomes especially relevant, as these processes directly determine the performance of transport and warehousing complexes.

The growth in small-batch shipments, characteristic of both B2B and B2C segments, leads to increased complexity of flow movement, a higher number of handling operations, and greater pressure on distribution infrastructure. According to [1–2], global expenditures on logistics operations exceed €300 billion annually and continue to grow due to the increasing complexity of logistics systems. This indicates significant potential for efficiency improvement specifically at the level of operational process organization.

At the same time, a key problem remains the presence of so-called “bottlenecks” at the interaction points of logistics chain elements (handover points), where delays, time losses, and information gaps occur. According to recent estimates, up to 13–19% of logistics costs arise precisely from the inefficiency of such operations [1–2]. This confirms the need for a systematic revision of approaches to organizing processes in logistics systems.

An additional factor exacerbating this issue is the operation of transport and logistics systems under conditions of instability, particularly in Ukraine, where military actions significantly affect infrastructure, routes, and resource availability. In such conditions, the issues of adaptability, flexibility, and resource efficiency of logistics processes become especially critical. Thus, there is a need to develop effective approaches to organizing and managing operational processes in logistics systems that ensure increased productivity, cost reduction, and resilience under conditions of high uncertainty [2, 3].

2. Analysis of literary data and statement of the problem:

Theoretical and applied aspects of logistics process organization are widely represented in the works of leading scholars. A significant contribution to the development of supply chain

management concepts was made by Martin Christopher, Donald Bowersox, and David Closs, who substantiated the necessity of an integrated approach to managing material flows.

Modern research focuses on improving the operational efficiency of logistics systems through [3–4]:

- optimization of warehouse processes;
- digital modeling and the use of digital twins;
- implementation of analytical management tools;
- automation and robotization of individual operations.

The use of digital models of logistics systems (digital twins) makes it possible to increase the efficiency of warehouse operations by 20–25% even before the actual implementation of changes [3, 5]. This indicates the significant potential of analytical approaches to optimization.

A separate research direction is related to improving the productivity of cargo handling operations. According to recent studies, high-quality planning and automation of individual processes can increase productivity several times; however, their effectiveness depends on the structure of cargo flows and the level of their standardization [5, 6].

At the same time, the analysis of the literature shows that existing studies mainly consider individual aspects of optimization, whereas the issue of comprehensive improvement of operational processes in logistics systems, taking into account network expansion, remains insufficiently explored. This is particularly important under conditions of instability and limited resources.

3. Purpose and Objectives of the Study:

The purpose of this study is to determine the influence of logistics system parameters and to develop practical recommendations for improving its efficiency in terms of logistics costs by enhancing the organization of operational processes for cargo delivery.

4. Research results:

The functioning of logistics systems in modern conditions is characterized by increasing complexity, driven by growing demand variability, shorter product life cycles, and the need to ensure a high level of service. Under such conditions, system efficiency is determined not only by the parameters of transport or warehousing operations separately, but also by the level of their integration and coordination.

A key characteristic of modern logistics systems is the presence of a significant number of interaction points between elements (logistics nodes), where material flows are transferred. These points are often sources of inefficiency due to delays, information asymmetry, and lack of coordination among process participants. It has been established that a substantial share of losses in logistics systems occurs precisely at these stages, which necessitates their systematic analysis and optimization.

An important direction for improving efficiency is the application of the process approach, which involves considering a logistics system as a set of interconnected processes. This makes it possible to identify bottlenecks, determine critical operations, and optimize resource flows. One of the effective tools is the use of digital modeling, which allows the creation of virtual replicas of logistics systems and testing of various operational scenarios. This provides the ability to assess the impact of changes without risk to the real system and supports informed managerial decision-making.

Particular attention should be paid to resource management issues. Modern logistics systems operate under conditions of labor shortages and high costs of workforce engagement. This necessitates increasing labor productivity through process optimization and the introduction of flexible forms of work organization. An important aspect is also ensuring the adaptability of logistics systems. Conditions of uncertainty require systems to respond quickly to changes in the external environment, which can be achieved through flexible planning, resource diversification, and the use of scenario-based management approaches.

Improving the efficiency of logistics systems requires a comprehensive approach that combines organizational, technological, and analytical tools.

A significant reserve for efficiency improvement is also the optimization of the spatial organization of logistics facilities. Rational allocation of resources and flows reduces non-productive movements, directly affecting system performance. As studies show [7], inefficient planning of a logistics network leads to increased operational costs and reduced cargo handling speed. The increase in the number of logistics facilities, the acceleration of product flows throughout the supply chain, and changes in retail formats necessitate the application of a systemic approach to organizing last-mile delivery processes.

With the increasing frequency of deliveries and the reduction in shipment sizes, the efficiency of organizing grocery cargo delivery at the last-mile stage becomes particularly important. This stage is the most cost-intensive element of the logistics chain, the share of which, according to recent estimates, can reach 40–60% of total logistics costs.

The results of experimental studies presented in [8] confirm the existence of an extremum in the function describing the share of logistics costs for batch cargo delivery depending on the parameters of the delivery system. The obtained regression models demonstrate a sufficient level of adequacy and statistical significance, which allows them to be used as a tool for optimization.

It has been established that the key factors determining the efficiency of the grocery delivery system are:

- the density of retail outlet locations;
- the average service radius;
- the market share of the network;
- the value of retail turnover in the region;
- the static load utilization factor of vehicles.

By adopting three levels of variation (minimum, average, and maximum) for the analyzed factors, and using the mathematical model proposed in [7, 8], calculations of the share of logistics costs for cargo delivery were performed for the three most common retail formats.

The analysis of the obtained dependencies [8] showed that the influence of most factors is monotonic but nonlinear. In particular, an increase in the density of retail outlets requiring delivery leads to a growth in the share of logistics costs. This is explained by the increased complexity of the routing network, a higher number of stops, and rising operational costs. Similar conclusions are consistent with studies in the field of last-mile logistics, which emphasize that demand fragmentation is a key driver of cost growth.

An increase in the average service radius leads to higher transportation costs due to longer routes and delivery times. At the same time, threshold values have been identified, beyond which costs grow at an accelerated rate, indicating the nonlinear nature of this relationship.

Conversely, an increase in the network's market share contributes to a reduction in logistics costs due to economies of scale, manifested in more efficient vehicle utilization, consolidation of cargo flows, and optimization of delivery routes. This is consistent with modern logistics concepts of scale efficiency.

From the perspective of improving the efficiency of delivery operations, particular attention should be paid to the static load utilization factor, which is one of the most controllable parameters of the system. The research results indicate that increasing the vehicle load factor within the range of 0,7–0,85 significantly reduces logistics delivery costs. Thus, optimizing order picking and cargo consolidation processes is an important direction for improving efficiency.

A relationship has been established between changes in logistics delivery costs and the number of stops on a route, as well as the half-demand radius, for the main retail formats: “Hypermarket,” “Supermarket,” and “Convenience Store” (Fig. 1). The analysis reveals the existence of specific combinations of the number of stops and the half-demand radius at which the share of logistics costs for delivery is minimized.

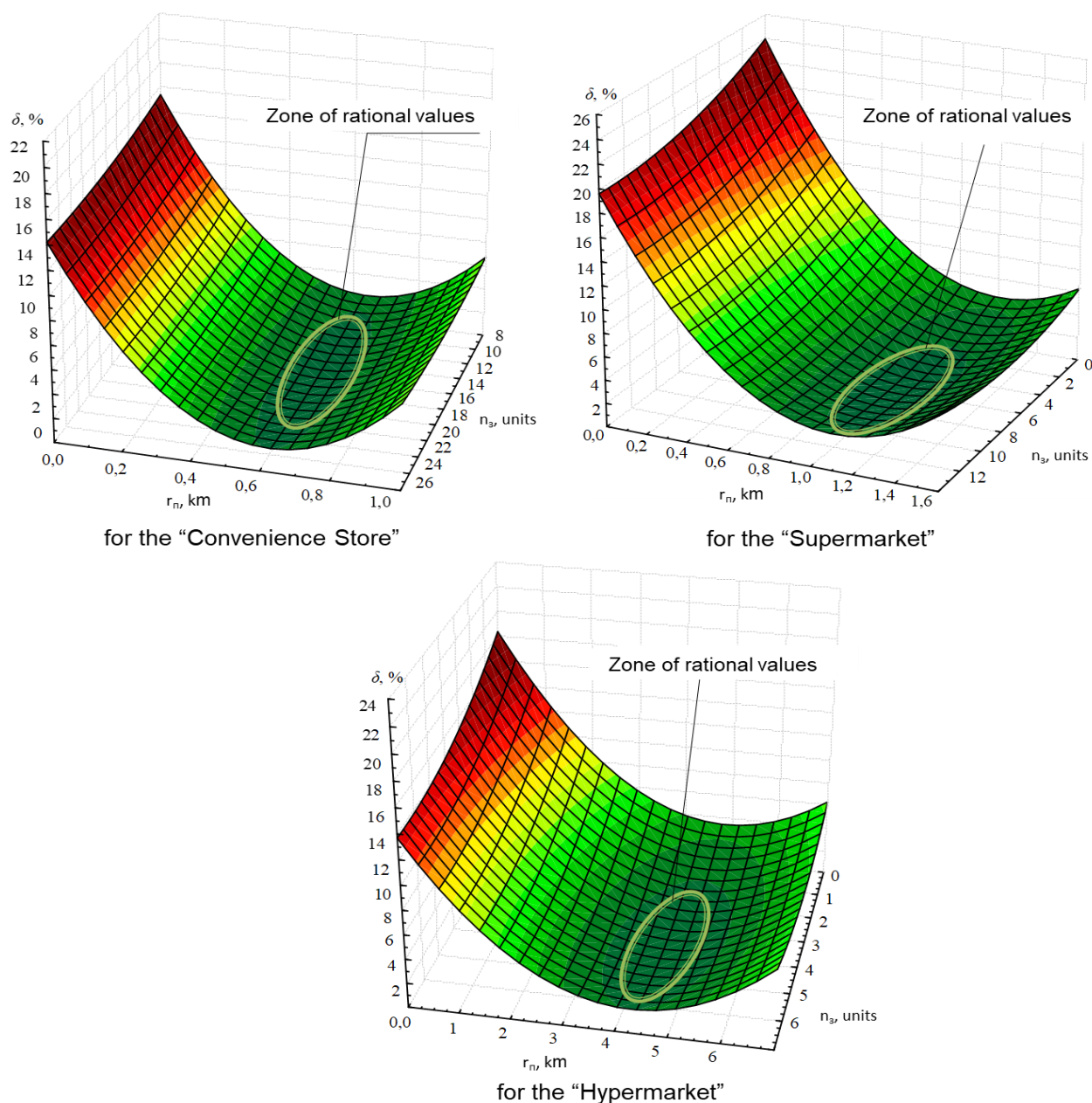


Fig.1. Dependence of changes in logistics delivery costs ($\delta, \%$) on the number of stops on the route (n_3, units) and the half-demand radius (r_n, km)

The identified ranges of rational values for different retail formats indicate the influence of store format on logistics costs in last-mile delivery, which is explained by differences in sales volumes. Therefore, this factor should be taken into account when planning delivery processes.

6. Conclusions:

As a result of the conducted study, it has been established that the efficiency of logistics systems under modern conditions is determined by the level of organization of operational processes and the degree of their integration. It has been formulated that the greatest effect is achieved through the comprehensive application of the considered approaches, which makes it possible not only to reduce costs but also to increase the resilience of logistics systems under conditions of uncertainty.

It has been determined that there exists a range of rational values of logistics parameters for retail networks that influence logistics delivery costs at the last-mile stage. The obtained results are of practical significance and can be used to improve the efficiency of logistics

systems under modern conditions, particularly in the context of planning and ensuring effective cargo delivery processes.

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ДИАГНОСТИЧНИ ХАРАКТЕРИСТИКИ НА ЦИЛИНДРОВО-БУТАЛНАТА ГРУПА НА БЕНЗИНОВИТЕ ДВИГАТЕЛИ НА АВТОМОБИЛ

DIAGNOSTIC FEATURES OF THE CYLINDER PISTON GROUP OF GASOLINE ENGINES OF A CAR

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Abstract. The task of diagnosing the engine is reduced to establishing the belonging of the technical condition to one of two subsets - serviceable and serviceable or faulty, but serviceable. It is necessary to recognize and localize the fault location. The analysis of the diagnostic and structural parameters of the engine's CPG is carried out. A typical range of diagnostic parameters has been established for most modern gasoline engines.

Keywords: car, engine, diagnostics, cylinder, piston, clearance, compression.

1. Introduction:

When diagnosing a cylinder-piston group (CPG) of a gasoline engine without disassembling it, there is a problem of correct diagnosis. First of all, this is due to both the complexity of the design of the internal combustion engine (ICE) and the complexity of the processes that accompany the conversion of thermal energy into mechanical work. The most loaded elements that perceive both mechanical and thermal loads in the internal combustion engine are the CPG and the valve group.

2. Statement:

The CPG belongs to the crank mechanism (KShM) and ensures the tightness of the engine cycles. CPG includes cylinders, pistons and piston rings. The main factors under the influence of which a change in the technical condition of the engine CPG occurs are: high pressure of 3-5 MPa; high gas temperature up to 2500oK inside the cylinder; relocation of the piston in the vicinity of its top dead center (TDC) (figure 1).

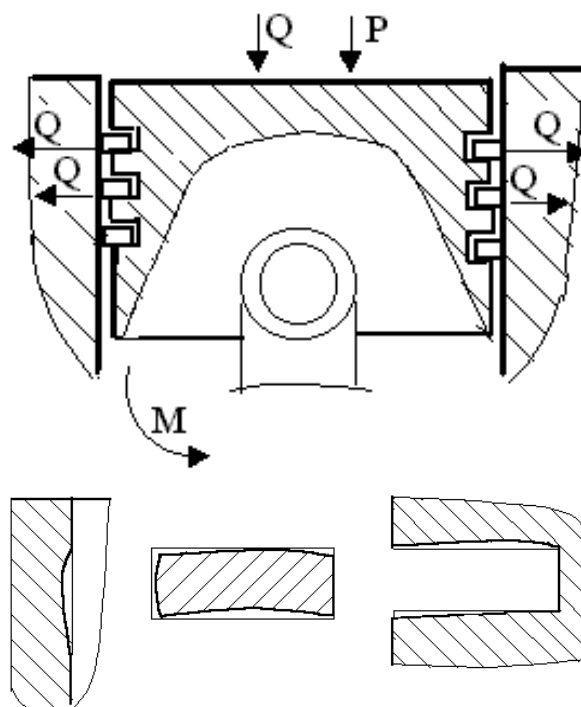


Fig. 1. The main factors affecting the wear of engine CPG parts: P – pressure; Q – heat load flows; M is the moment causing the piston shift

The greatest load in this case is perceived by the upper compression ring, the piston partition under this ring and the upper belt of the cylinder. The increase in pressure during

combustion leads to a significant increase in the force of pressing the ring against the surface of the cylinder and the lower surface of the piston groove. In this case, up to 50-60% of the total heat released is removed through the upper compression ring. In order to ensure heat transfer through the ring, the ring needs to fit snugly against the piston groove and against the cylinder surface. In addition, the upper compression ring operates under conditions of greater oil starvation than other rings, which is also a significant factor that increases the wear of both the ring itself and the upper cylinder belt.

As the piston approaches TDC, under the action of the gas pressure force, the friction force between the piston bosses and the piston pin increases. This causes the piston to rotate along with the connecting rod, that is, the piston moves from one wall of the sleeve to the other, the so-called piston shifting. The piston shifting near TDC, depending on the degree of wear and the ignition timing, occurs with a strong shock impulse, which additionally loads the CPG. Structurally, the moment of the beginning of the shift and the speed of the piston shift depend on the displacement of the axis of the piston pin. It has been established that the displacement of the piston pin relative to the piston axis by the optimal value (1-2.5 mm) leads to a decrease in the wear of the cylinder liner in the upper zone by 20-23% and a decrease in ring wear by 28%.

However, the problem of wear and hence control of this wear remains. As a result of thermal and mechanical loads, the upper compression ring acquires a characteristic cross-sectional profile with an asymmetric barrel shape of the outer surface and a slight concavity at the ends. The lower surface of the piston groove becomes conical with a rounded edge, and a characteristic wear appears on the cylinder wall, in the vicinity of TDC.

If the above listed malfunctions are not eliminated in time, a stronger destruction of the CPG occurs with a fracture of the piston rings and the formation of scoring on the surface of the cylinder. The type of technical condition of an object is characterized by the conformity or non-compliance of the quality of the object with certain technical requirements established by the technical documentation for this object. There are the following types of technical condition: serviceability and malfunction, operability and inoperability, correct functioning and incorrect functioning.

The operation algorithm of the CPG is reduced to ensuring the tightness of engine cycles. In this case, excess pressure from the over-piston space should not penetrate into the engine crankcase, and oil should not flow from the crankcase into the combustion chamber. Consequently, the CPG of the engine, which carries high thermal and mechanical loads, under the influence of which the type of its technical condition changes, is serviceable if all the elements included in this group are operational and function correctly: pistons, piston rings, cylinders, and faulty if incorrectly at least one element from this group is functioning, which leads to a violation of the tightness of the combustion chamber, oil entering it, a drop in compression pressure in it and a breakthrough of gases into the engine crankcase.

Modern car engines, with rare exceptions, are multi-cylinder. The failure of a cylinder does not mean a complete loss of performance of the entire engine. However, such a faulty, but operable state of the engine leads to a deterioration in its design and operational characteristics: power loss, reduced torque, increased consumption of fuels and lubricants.

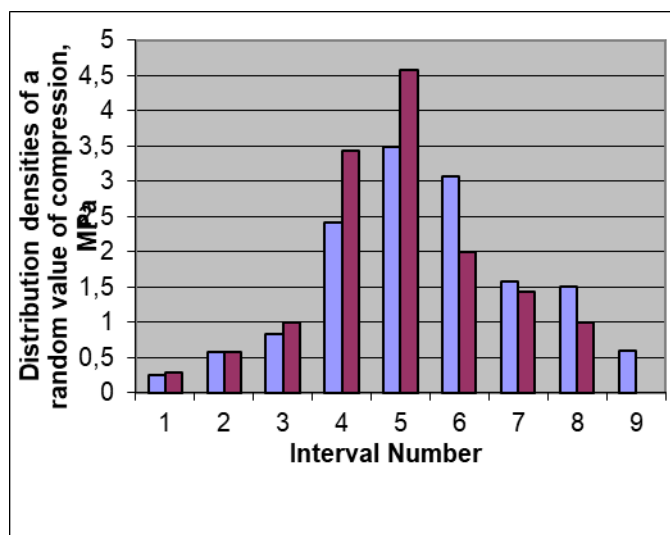
The CPG of an engine, not without reason, can be attributed to a complex mechanical object of diagnostics, in which the number of mating parts is proportional to the number of cylinders, and the complexity of the processes accompanying the conversion of thermal energy into mechanical work reduces the possibility of using accurate analytical dependencies. Direct access to the structural parameters of the CPG is limited. The task of diagnosing the CPG is reduced to establishing that the technical condition belongs to one of two subsets - serviceable and operable or faulty, but operable, as well as recognizing and localizing the fault location.

The analysis of diagnostic and structural parameters of the CPG of the engine was carried out [1-3]. A characteristic range of diagnostic parameters has been established for most modern gasoline engines (Table 1.) [4,5].

Table 1. Characteristics of the structural parameters of the CPG of modern gasoline engines of passenger cars

Names of controlled parameters	Nominal values of the parameter, mm	Permissible parameter values, mm	Maximum possible values of the parameter, mm
Piston ring gap	0,3-0,4 for upper	0,7-0,8	0,9-1,0
	0,2-0,3 for medium and oil scraper	0,7-0,8	0,9-1,0
	0,4-0,5 for type-setting oil scraper	1,2-1,4	1,5
Gap between piston and ring at groove height	0,015-0,02	0,03-0,09	0,1
clearance between cylinder and piston in the upper chord	0,02-0,05	0,06-0,09	0,12- 0,15

Based on the compiled tables, we construct histograms of the distribution density of a random compression value. The appearance of these histograms for both statistical series will correspond to the normal distribution law (Figure 2).

**Fig. 2.** Distribution density of random compression

Using the methods of mathematical statistics there were established: the mathematical expectation (arithmetic mean) of a random value of compression for the first variation series will be $\bar{x} = 1,29$ MPa, for the second series will be $\bar{x} = 1,256$ MPa; the mode for both variational series is $M_0 = 1.265$ MPa; for the first variation series, the dispersion range is 0.561 MPa, for the second row, 0.539 MPa; standard deviation for the first variation series $\sigma = 0.123$ MPa, for the second series - $\sigma = 0.107$ MPa; the variance for the first variation series will be $\sigma^2 = 0.015$, for the second series it will be $\sigma^2 = 0.0115$; the coefficient of variation for the first and second rows, respectively, will be $V = 9.5\%$, $V = 8.5\%$.

When diagnosing, such influencing parameters as engine speed, engine coolant temperature, and the angle of rotation of the throttle valve were taken into account and stabilized. We investigated as obviously well-functioning engines - new, after major repairs or with low mileage, as faulty engines. Diagnosis of faulty engines was accompanied by subsequent disassembly and measurement of structural parameters using special tools (micrometers).

In the course of the experimental studies, the following malfunctions were most often encountered: burnout of the exhaust valve; a loose fit of the valve into the seat due to the

deformation of the stem or the absence of the provided thermal clearance; scuffs on the piston, which correspond to scorings on the cylinder wall; occurrence or breakage of compression rings; deformation and kink of the piston partitions. CPG malfunctions associated with cylinder wear in the vicinity of top dead center, piston ring, and piston grooves were also encountered.

3. Conclusion:

To date, the most common, accessible and reliable are the methods of technical diagnostics of the CPG, which give a general assessment of the tightness of the over-piston space. However, these and other well-known methods do not provide the desired information about the technical condition of the CPG and the tightness of the engine valves, due to the inaccuracy and inconsistency of the information obtained, the reliability of which is 35-75%. At the same time, the reliability of the diagnostic conclusion is largely determined by the experience of the operator and, therefore, is subjective. The analysis of diagnostic and structural parameters of the CPG of the engine has been carried out. A characteristic range of diagnostic parameters has been established for most modern gasoline engines. The appearance of these histograms for both statistical series will correspond to the normal distribution law. Using the methods of mathematical statistics there were established: the mathematical expectation (arithmetic mean) of a random value of compression for the first variation series will be $\bar{x}=1,29$ MPa, the dispersion range is 0.561 MPa, for the second row, 0.539 MPa; standard deviation for the first variation series $\sigma = 0.123$ MPa, for the second series - $\sigma = 0.107$ MPa; the variance for the first variation series will be $\sigma^2 = 0.015$, for the second series it will be $\sigma^2 = 0.0115$.

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ОСОБЕНОСТИ НА ДИАГНОСТИЦИРАНЕТО НА МЕХАТРОННИТЕ СИСТЕМИ НА АВТОМОБИЛИТЕ

FEATURES OF DIAGNOSING AUTOMOTIVE MECHATRONIC SYSTEMS

Vladimir Volkov¹, Tetiana Volkova², Maksym Levchuk³, Ruslan Shulha⁴

Abstract. Mechatronics in transport represents the further development of process control automation, the refinement of vehicle subsystems and components, and the implementation of flexible computerised systems within the transport sector. Self-diagnostics is a standard feature of all microprocessor-based control systems. During normal operation, self-test functions are performed in parallel with other functions, such as fuel injection and ignition; suspension control; ABS/EBS operation, etc. Self-diagnosis involves meeting several requirements; using the example of a mechatronic engine and transmission control system, the specific features of the technology for diagnosing these systems are demonstrated.

Keywords: car, mechatronics, diagnostics, research methods, design principles

1. Introduction:

Mechatronics emerged as an interdisciplinary field through the convergence of mechanics and microelectronics. It can be defined as the science concerned with the analysis and synthesis of complex systems in which mechanical and electronic control devices are utilised to an equal extent. All mechatronic systems in cars are divided into three main groups according to their functional purpose [1]:

- engine control systems;
- transmission and chassis control systems;
- interior equipment control systems.

The engine control system is subdivided into petrol and diesel engine control systems. Depending on their purpose, they can be either single-function or integrated. In single-function systems, the electronic control unit (ECU) sends signals only to the injection system. Injection can be continuous or pulsed (multi-phase injection). With continuous fuel supply, the fuel quantity is varied by changing the pressure in the fuel line, whilst with pulsed injection, it is varied by the pulse duration and frequency. In integrated systems, a single ECU controls several subsystems: fuel injection, ignition, valve timing, self-diagnostics, etc.

The electronic control system for a diesel engine regulates the amount of fuel injected, the injection start time, the glow plug current, and other parameters.

In a vehicle's electronic transmission control system, the primary focus of control is the automatic transmission. Based on signals from the throttle position sensor and the vehicle's speed sensor, the ECU selects the optimal gear ratio, thereby improving fuel efficiency and handling.

Chassis control encompasses the management of vehicle motion, trajectory changes and braking. These systems act on the suspension, steering and braking systems to maintain the set speed.

Control of the interior equipment is designed to enhance the vehicle's comfort and consumer value. To this end, the following are used: air conditioning, an electronic instrument panel, a multifunctional information system, a compass, headlights, intermittent windscreen wipers, a bulb failure indicator, a rear-view obstacle detection system, anti-theft devices, communication equipment, central door locking, electric windows, adjustable seats, a safety mode, and more.

2. Aim and objectives of the work:

The crucial role played by electronic systems in a vehicle necessitates greater attention being paid to issues relating to their maintenance. The solution to these problems lies in incorporating self-diagnostic functions into the electronic system [2]. The implementation of these functions is based on the capabilities of the electronic systems already in use in the vehicle for continuous monitoring and fault detection, with the aim of storing this information and facilitating diagnosis.

3. Self-diagnosis of automotive mechatronic systems:

Advances in electronic engine and transmission control systems have led to improvements in a vehicle's performance. The control algorithms (programmes) for gear shifting in an automatic transmission ensure the optimal transfer of engine power to the vehicle's wheels, taking into account the required traction and speed characteristics as well as fuel economy [2]. However, the programmes designed to achieve optimal traction and speed characteristics and minimum fuel consumption differ from one another, as it is not always possible to achieve both objectives simultaneously. Therefore, depending on driving conditions and the driver's preference, a special switch can be used to select the 'economy' programme to reduce fuel consumption, the 'power' programme to improve traction and speed characteristics, or the 'manual' programme to switch to driver-controlled gear changes. Figure 1 shows a block diagram of the self-diagnostic system for automotive mechatronic systems [3].

The Electronic Control Unit (ECU) performs a self-test of its operation as follows: the programmable memory chips are loaded with test patterns that can be recalled and used for comparison. For memory devices, a comparison is made with the final test data to ensure that all data and programmes are stored correctly within these devices. Self-diagnostic capabilities include: system and ECU identification; recognition, storage and retrieval of information on static and single faults; retrieval of current real-time data, including environmental conditions and specifications; simulation of system functions; and programming of system parameters. Separate programmes for the test unit are stored in plug-in modules, whilst data correction and transfer within the system are carried out via a data interface.

The on-board monitoring and diagnostic system (OMDS), for example in a lorry [3], is designed to:

- enable the diagnosis of on-board electronic systems via the ISO 9141 interface directly on the vehicle;
- monitoring the axle load and operating modes of the vehicle;
- measuring the current time and, accordingly, displaying the monitored parameters and the current time on the LCD display (screen) of the control unit installed on the instrument panel without the use of external devices.

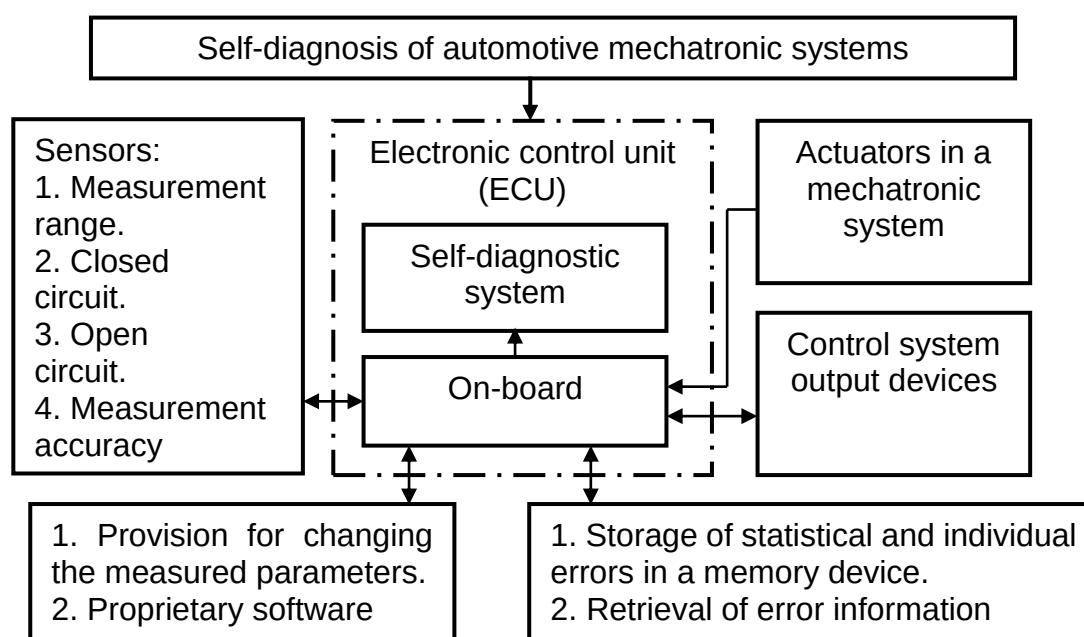


Fig. 1. Block diagram of the self-diagnostic system for automotive mechatronic systems

The BSKD system enables the diagnosis of a vehicle's electronic systems (ABS/ASR, ECAS, engine EDC) in accordance with the ISO 9141 protocol. Diagnostics includes reading data from the non-volatile memory of electronic control units: control unit serial number; control unit software version; control unit date of manufacture; reading the fault memory; clearing the fault memory.

The diagnostic process begins with the initialisation of the systems – detecting them within the vehicle's electrical system. Upon successful initialisation, it is possible to: read the fault memory; clear the fault memory; view data for the next detected system or exit to the main menu; change the reading of the selected parameter; correct the current time; correct the current date; and perform a number of additional functions.

One of the key features of the MAN D2866-LF2 engine fitted to the MAZ vehicle is the inclusion of Electronic Diesel Control (EDC) – an electronic control unit (ECU) with a built-in diagnostic system. Every time the ignition switch is turned on or off, as well as throughout the entire period of engine operation, the ECU diagnoses the condition of the electrical circuits and the parameters of the built-in sensors (temperature, boost pressure, etc.) and the engine's fuel injection system. In the event of a fault or a deviation from the set parameters, a warning light illuminates, the engine operating mode is adjusted (for example, fuel supply is reduced, power is lowered) and a fault code is stored in the ECU's memory.

Fault codes are stored in two memory chips. One memory chip contains the codes, whilst the other contains fault information that can only be read and erased using the MAN-CATs computerised diagnostic system. Erasing the contents of the first memory chip does not affect the contents of the second memory chip. At any one time, the memory of a single chip can only contain information on five faults. Once a fault recorded in the memory of one microchip has been rectified (or has disappeared), the information relating to it is automatically erased from the memory of both microchips (provided that the fault has not recurred within 100 cycles of the starter and instrument lock being switched on or off).

Basic EDC fault diagnosis can be carried out by pressing the EDC diagnostic button on the instrument panel. In this case, fault codes are identified by the number and duration of flashes from the 'EDC' warning light, and the types of faults are determined using the code tables.

In-depth diagnostics and system parameter setting (configuration) can be carried out using the MAN-CATs computerised diagnostic system (CDS).

Simplified fault diagnosis using flashing light codes can be performed with the vehicle's engine either running or switched off. To enter diagnostic mode, press the EDC diagnostic button for 3 seconds (but no longer than 10 seconds) and then release it. While the button is pressed, the 'EDC' indicator light on the instrument panel illuminates; it goes out when the button is released (this also allows the lamp to be checked for faults).

If there is a fault in the system, the 'EDC' light will start flashing 3 seconds after the button is released, displaying a fault code using long (2 seconds) and short (0.5 seconds) flashes. Initially, only the code for the single (most recent) fault is displayed. To display the code for the next fault, press and release the EDC diagnostic button again. Continue this process until the code that was displayed first reappears.

A timing diagram (Fig. 2) for codes 2–4 (fault in the idle switch of the fuel pedal position sensor) is provided as an example of reading the light code.

After pressing and releasing the EDC diagnostic button, the following sequence occurs: a 3-second pause before the code is displayed; a long flash lasting 2 seconds; a 1-second pause between flashes; a long flash lasting 2 seconds; pause between flashes – 5 s; four short flashes of 0.5 s each with pauses of 0.5 s. After reading the codes and rectifying the faults, the fault memory (light codes) is cleared. In doing so, all codes for rectified faults are erased from the ECU memory. If any codes remain in the memory after clearing, this means that a fault has not been rectified and its code can only be cleared once the fault has been rectified. If problems arise in identifying the fault based on the flashing light codes, an analysis of probable causes and a comprehensive EDC diagnosis are carried out using the MAN-CATs diagnostic tool.

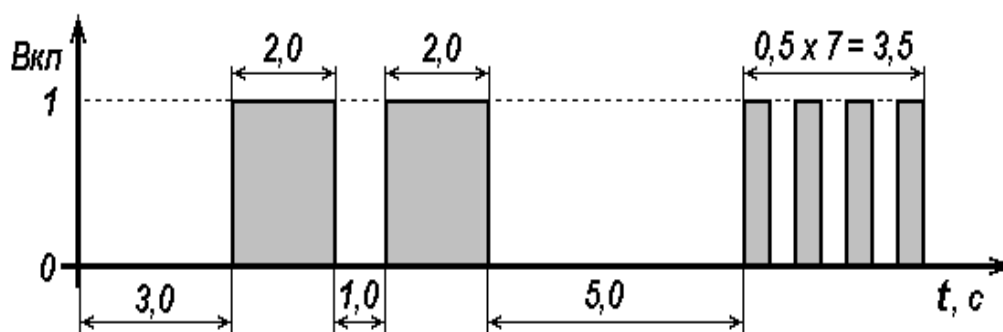


Fig. 2. Light code timing diagram 2 – 4

Computerised (in-depth) diagnostics of automotive electronic systems can also be carried out using a specialised computer diagnostic system based on a laptop. Computer diagnostics work can be carried out directly on the vehicle (in open areas), as well as in service centres and vehicle repair workshops, workshops, technical inspection departments and laboratories. To monitor the operation of vehicle systems under road conditions, for example during a test drive, a notebook computer is usually used.

The principle of computer diagnostics (Fig. 3) is based on the following [3, 4]:

- the generation and transmission, under the control of a personal computer via a diagnostic interface (adapter) compliant with the ISO 9141 standard, of electrical signals that establish communication with a specific electronic control unit;
- receiving information from the electronic control unit via the diagnostic channel;
- converting the information into RS 232C signals;
- processing this data according to a specified programme and displaying the results on the personal computer screen.

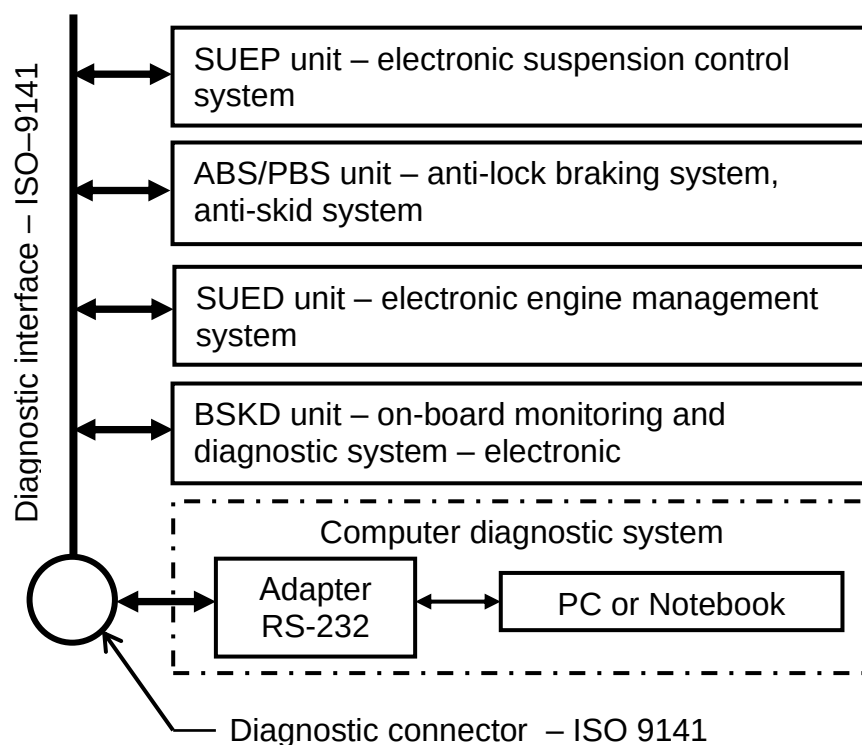


Fig. 3. Block diagram of computer diagnostics for the mechatronic systems of a MAZ vehicle with an engine MAN D2866-LF20

The ISO-9141 diagnostic interface line is used to transmit commands and data to the electronic control unit, whilst the line via the adapter is used to receive data from the electronic control unit, relay it, and transmit it via the RS-232C interface to the PC. When configuring the system, this line can also be used to transmit data to the electronic control unit.

The diagnostic software allows you to obtain information about current system faults (active faults), if any, as well as information about faults that have occurred but are not currently present (passive faults). In addition, information can be obtained regarding the electronic control unit's serial number, its manufacturer and date of manufacture, software version, and the date of the last parameter change. Data on the engine's total operating hours (total number of complete crankshaft revolutions, engine hours), including idling time, the vehicle's total mileage, mileage since the last service, etc., can be read from the engine management control unit.

4. Fault diagnosis and parameter setting using the diagnostic system MAN-CATs:

The MAN-CATs computerised diagnostic system (CDS) comprises: a notebook-type computer; an ISO-9141 interface unit; an adapter unit; a set of connection cables; and software. Information on the contents of the software versions is obtained in the same way after selecting the specified mode. The CDS software is compatible with the Windows 11 operating system. The software package includes an installation package and a suite of programmes for electronic systems and their modifications. Communication with the ECU is carried out via a channel in accordance with the ISO-9141 standard.

Once the software has finished downloading, the main menu of modes appears on the PC screen: diagnostics; parameter programming in EOL (end of line) mode; system installation; information on the contents of MAN-CATs software versions. After selecting the required operating mode, for example 'Mot', in 'Diagnostics' mode and the EDC version, you can then enter the dialogue mode with the PC, which allows you to perform system diagnostics or parameter programming in EOL mode. System installation takes place after selecting the specified mode and pressing the Enter key. At this point, information regarding the procedure for using the installation disks appears on the screen.

5. Conclusions:

Self-diagnosis is standard for all microprocessor-based control systems. During normal operation, self-test functions are performed in parallel with other functions, such as fuel injection and ignition; suspension control; ABS/EBS operation, etc. Self-diagnosis is characterised by the fulfilment of several requirements:

- monitoring the operation of complex systems and components. The increasingly complex design of the engine makes self-diagnostic capabilities extremely important for detecting and rectifying faults. The aim is to integrate the entire system into the diagnostic process;
- protection of assemblies and components that are particularly at risk in the event of faults. For example, one might cite the protection of the catalytic converter, which reacts to misfires in the engine. The system responds to a certain frequency of misfires by cutting off the fuel supply to the faulty cylinder to prevent the converter from overheating;
- operation in an emergency situation is carried out using 'default' values. For example, if the load sensor (which determines the mass air flow) fails, a signal is generated to replace it, based on the crankshaft speed and throttle position;
- driver notification of faults in the diagnostic system via warning lights, displays and audible warning devices. In the US, requirements for on-board vehicle diagnostics have been established. In Europe, similar standards are being developed to detect faults in exhaust emission control devices;
- Storage of accurate information. The system stores warning information and data on specific faults in the ECU. The memory unit also stores data on engine operating conditions at the time the fault was first detected. The type and completeness of the information are regulated by ISO standards;

- access to stored faults. Data stored in the self-diagnostic system's memory whilst the vehicle is in operation can be transmitted to a diagnostic bench with a display via a serially connected multi-channel input (port). The communication protocols required for this are specified in ISO-9141 and ISO-14230;

- The simplest method of transmitting fault data is a message in the form of a flashing code on the instrument panel of the measuring equipment. This helps service personnel to speed up diagnosis by narrowing down the range of possible sources of faults.

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СИСТЕМИ ЗА ЗАДВИЖВАНЕ И ТРАНСМИСИЯ НА ПРЕВОЗНИ СРЕДСТВА

VEHICLE DRIVE AND TRANSMISSION SYSTEMS

Yuriy Zybtshev¹, Pavlo Voroshyllov²

Abstract. Drawing on test data obtained by the independent Swedish research institute Rototest, this study presents a detailed statistical evaluation of torque losses in automotive transmission systems of different types and configurations. The analysis is based on a large dataset including more than 600 vehicles manufactured between 1993 and 2010, covering both manual and automatic transmissions as well as various drivetrain layouts (FWD, RWD, AWD). The measurements were carried out using hub dynamometers, which enable direct torque measurement at the wheel hubs and avoid common inaccuracies associated with conventional chassis dynamometers, such as tire slip and uncertainties in rolling radius.

The findings reveal a clear dependence of transmission efficiency on both transmission type and production period. For manual gearboxes, the average efficiency is approximately 0.91 for vehicles produced before 2004, while for models manufactured after 2005 it increases to about 0.96, indicating notable improvements in engineering accuracy, materials, and lubrication technologies. In contrast, automatic transmissions demonstrate relatively consistent efficiency values of around 0.89, generally exhibiting higher losses than manual systems, regardless of drivetrain configuration.

The study further examines both modal and mean values of power losses across different vehicle configurations. It is shown that front- and rear-wheel-drive vehicles equipped with manual transmissions tend to have lower losses compared to similar vehicles with automatic gearboxes. Although automatic transmissions have evolved through features such as torque converter lock-up systems and multi-gear designs, their practical efficiency improvements remain comparatively modest.

Overall, the results highlight the significance of independent, statistically robust testing in assessing drivetrain performance and offer valuable insights for manufacturers, engineers, and policymakers aiming to enhance vehicle energy efficiency.

Keywords: transmission system, automobile, manual gearbox, automatic gearbox, wheel, front-wheel drive, rear-wheel drive, all-wheel drive.

1. Introduction

The vehicle transmission is a fundamental component of the powertrain, responsible for transferring torque from the engine to the driving wheels. Its performance has a direct impact not only on the dynamic characteristics of a vehicle, such as acceleration, responsiveness, and drivability, but also on fuel consumption, energy efficiency, and the overall environmental footprint. As global regulations on fuel economy and emissions become increasingly stringent, improving the efficiency of drivetrain components—particularly the transmission—has emerged as a key focus for automotive engineers, researchers, and policymakers [1].

One of the primary metrics used to evaluate transmission performance is the efficiency coefficient (EC), which is defined as the ratio of useful mechanical power delivered to the wheels to the total power produced by the engine. This coefficient reflects the portion of engine energy that is effectively converted into vehicle propulsion, as opposed to being lost through internal friction, mechanical resistance, hydraulic drag, fluid interactions, and other parasitic effects inherent to the transmission system. Accurate determination of EC is crucial for both vehicle design and performance optimization, as it allows engineers to identify sources of inefficiency, compare different transmission architectures, and guide improvements in materials, lubrication systems, and control strategies [2].

Furthermore, the efficiency coefficient serves as a vital benchmark when assessing the impact of new technologies, such as multi-speed gearboxes, dual-clutch systems, continuously variable transmissions (CVTs), and torque converter lock-up mechanisms. By quantifying how much energy is preserved and effectively transmitted to the wheels under real-world operating conditions, EC provides a meaningful measure of how design choices, component quality, and maintenance affect both the operational performance of the vehicle and its long-term environmental impact. As automotive systems continue to evolve, particularly with the increasing adoption of hybrid and electrified drivetrains, understanding and optimizing transmission efficiency remains a central concern for achieving sustainable mobility goals.

Transmission efficiency depends on numerous factors, including:

- transmission type (manual, automatic, CVT, dual-clutch),
- drivetrain layout (front-wheel, rear-wheel, or all-wheel drive),
- design features and manufacturing precision of components,
- lubrication properties and oil viscosity,
- operating conditions such as load, gear ratios, and driving modes,
- temperature effects and component wear over time.

Traditionally, transmission efficiency values are derived from standardized assumptions or manufacturer-provided data, typically ranging from 85% to 97% depending on the system configuration [2]. However, such figures are not always fully transparent and may fail to represent real operating conditions. In some instances, manufacturer-reported values are optimized for presentation purposes and do not adequately account for factors such as aging, wear, or temperature-related losses in transmission fluids.

To overcome these limitations, researchers have increasingly adopted statistical analysis approaches, enabling the evaluation of extensive datasets obtained from both laboratory experiments and real-world vehicle measurements. Such methods offer a more objective and reliable assessment of transmission efficiency, as they make it possible to analyze distributions, determine average losses for different vehicle categories, and detect outliers that may indicate mechanical issues or inconsistencies in the data (Fig. 1).

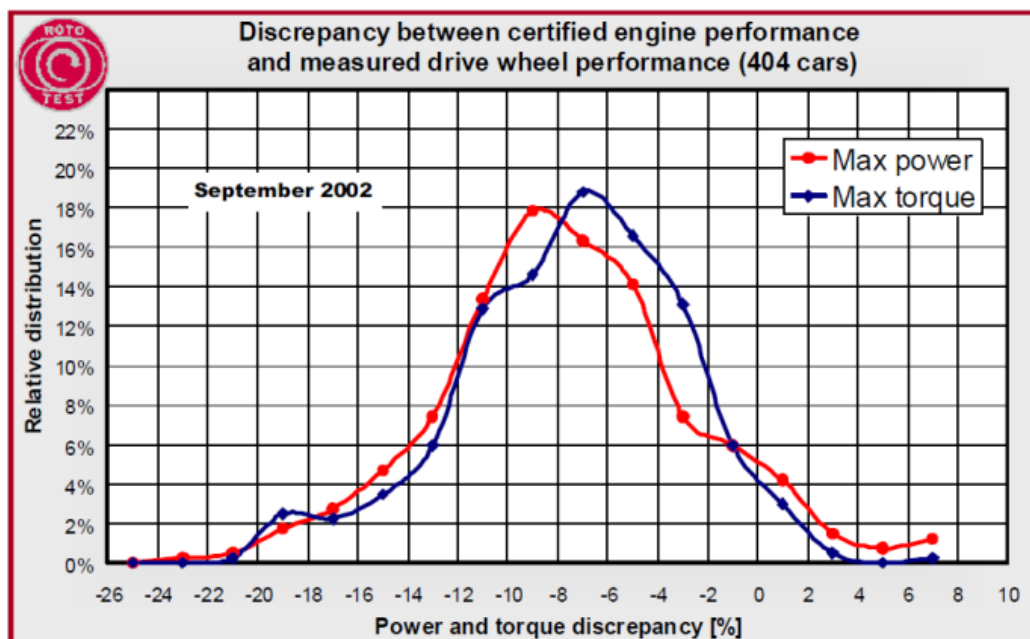


Fig. 1. Deviation between manufacturer-specified performance parameters and wheel-dependent values [4,11].

One of the key organizations making significant contributions to the independent study of transmission performance is the Rototest Research Institute in Sweden. This institute has developed highly advanced dynamometer systems capable of measuring power output directly at the wheel hubs [3]. By bypassing the use of tires and traditional rollers, this approach minimizes many of the uncertainties and measurement errors commonly associated with conventional chassis dynamometers, such as tire slippage, variable rolling radius, and drivetrain compliance. As a result, it allows for a highly accurate and focused assessment of transmission-specific losses, providing data that reflect the true performance of the powertrain without interference from external factors.

The objective of this study is to examine the principles and applications of statistical methods for evaluating transmission efficiency using publicly available datasets and results from independent experimental techniques. By applying these methods, it is possible to quantify average and modal losses, identify trends across different vehicle categories, and analyze the impact of various design parameters and operational conditions on drivetrain performance.

Furthermore, such analysis can reveal variations due to production year, transmission type, and drivetrain layout, offering insights into the factors that most significantly affect efficiency [5].

A comprehensive understanding of real-world transmission efficiency is essential not only for engineers and researchers engaged in powertrain optimization but also for policymakers, fleet operators, and environmentally conscious users who seek to reduce fuel consumption and emissions across the transportation sector. Reliable efficiency data support better decision-making in vehicle design, maintenance practices, and regulatory planning. In addition, they provide a scientific foundation for evaluating the effectiveness of new transmission technologies, such as multi-speed gearboxes, dual-clutch systems, and torque converter lock-up mechanisms, and for guiding the development of strategies aimed at improving energy efficiency and sustainability in modern vehicles.

2. Analysis of Literary Data and Problem Statement

The study of resistance forces in automotive transmissions has a long and well-documented history, spanning several decades of engineering research. As early as the 1930s and 1940s [1], researchers formulated the first generalized estimates of mechanical transmission efficiency. According to these pioneering studies, vehicles equipped with a single driven axle demonstrated efficiency values ranging from 0.90 to 0.95, while multi-axle vehicles—those with two or three driven axles—exhibited somewhat lower efficiencies, typically between 0.85 and 0.90. These early findings provided the foundation for subsequent research in transmission design and are still frequently cited in educational literature and applied in engineering calculations and preliminary design evaluations.

In more recent decades, particularly in publications from automotive manufacturers, efficiency values reported for modern transmission systems have often been significantly higher, sometimes reaching up to 0.97 for advanced designs such as dual-clutch transmissions. While these figures highlight the progress in transmission technologies, including improved materials, precision manufacturing, and lubrication techniques, they are often presented in a promotional context and may not always correspond to performance under real-world operating conditions. This discrepancy underscores the challenge of obtaining objective, reliable data on actual transmission efficiency, which is critical for both engineering assessment and policy decisions related to fuel consumption and emissions [6].

Among the few independent and reliable sources of data, the Rototest Research Institute in Sweden stands out. This organization has developed specialized dynamometer systems capable of directly measuring torque at the wheel hubs, without relying on tires or rollers. By removing common sources of error associated with traditional chassis dynamometers—such as tire slip, rolling radius uncertainties, and drivetrain compliance—these measurements provide a highly accurate estimate of losses that occur specifically within the transmission. This methodological precision allows researchers and engineers to isolate the impact of transmission design and technical condition on overall drivetrain efficiency, offering insights that are essential for improving both vehicle performance and energy efficiency.

Moreover, the availability of such independent datasets enables statistically robust analyses across a wide range of vehicles and drivetrain configurations. By comparing historical benchmarks with modern measurements, it is possible to track the evolution of transmission technologies, identify trends in efficiency improvements, and provide guidance for future design enhancements. This combination of historical perspective, modern technological evaluation, and independent testing forms a critical foundation for understanding real-world transmission performance in contemporary automobiles.

3. Aim and Tasks of the Research

The aim of this study is to perform a statistical assessment of automobile transmission efficiency using data obtained from bench testing.

To accomplish this objective, the following tasks were defined:

- to review relevant literature and assess existing information on transmission efficiency;

- to examine a dataset of publicly available results collected by the Rototest Research Institute in order
- to determine average and most representative values of transmission losses;
- to compare efficiency levels across different transmission types (manual and automatic) as well as drivetrain configurations (front-wheel, rear-wheel, and all-wheel drive);
- to investigate how transmission efficiency varies with the year of vehicle manufacture.

4. Research Results

In the statistical analysis of the dataset collected by Rototest, the characteristics of over 600 vehicles produced between 1995 and 2010 were examined (Fig. 2). The results represent actual drivetrain losses, calculated as the difference between the engine's rated output and the power delivered to the wheels.

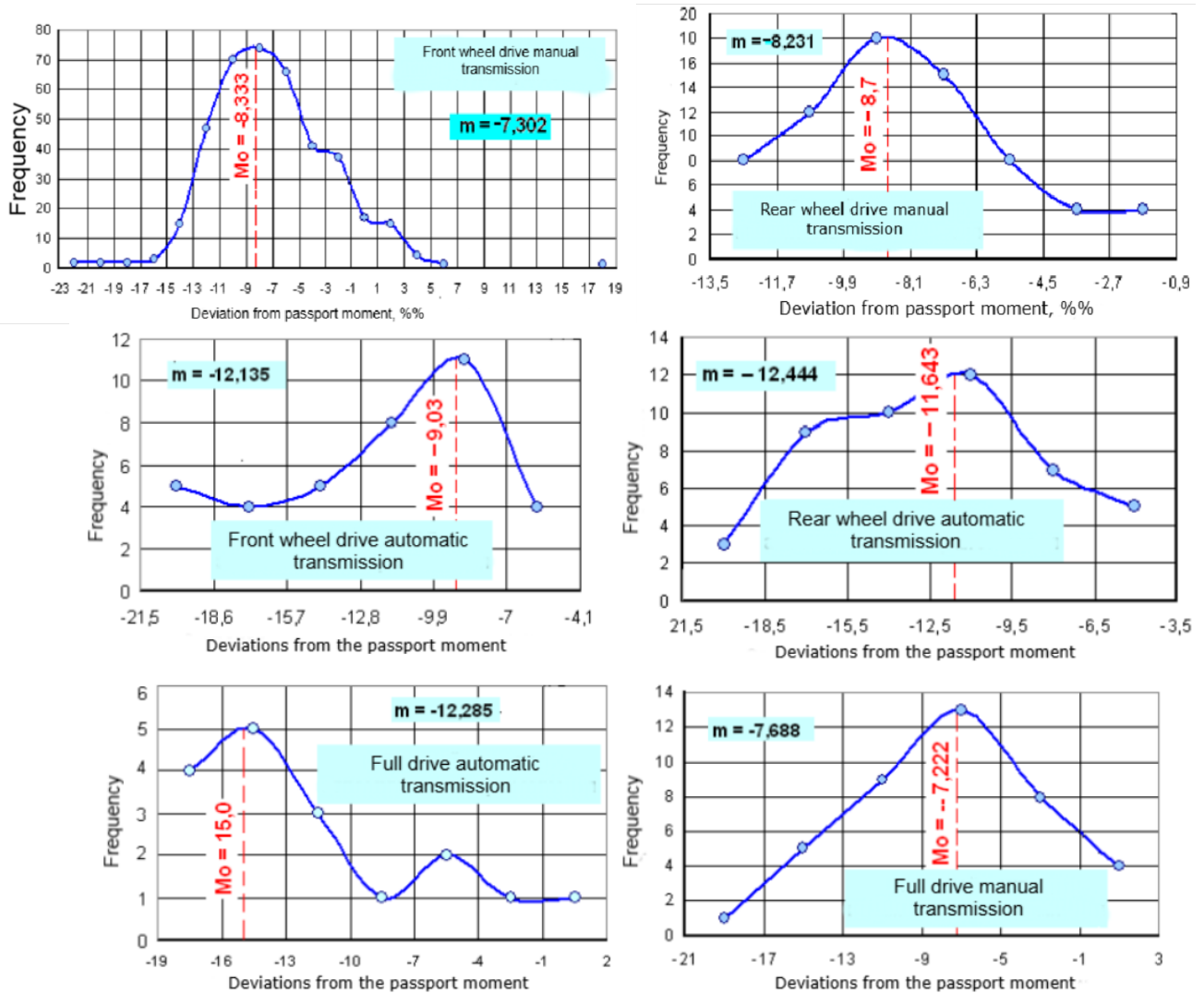


Fig. 2. Distribution of deviations between measured torque values and manufacturer-specified ratings [11].

Key quantitative findings:

- The average drivetrain losses were approximately 7–9% of the engine's rated power.
- The most probable values (mode) of losses were 7% for torque and 9% for power.
- Vehicles with automatic transmissions exhibited an average loss of 12.3%, with a mode of 10.75%.
- Manual transmissions generally experienced lower losses, particularly in more recent models.
- Drivetrain layout (front- vs. rear-wheel drive) had a minor impact on efficiency, with differences falling within the margin of statistical error.

- All-wheel drive vehicles showed higher losses; however, the limited sample size prevents drawing definitive conclusions (Tab. 1).
- Analysis by vehicle production year indicates that reductions in drivetrain losses were observed only for manual transmissions.

Table 1. Average deviations between measured and manufacturer-specified torque values across different passenger car categories [7]

Transmission Type	Drive Type	Sample Size	Mean Loss (%)	Mode Loss (%)
Manual	Front-Wheel Drive (FWD)	396	-7.302	-8.333
Manual	Rear-Wheel Drive (RWD)	69	-8.23	-8.7
Manual	All-Wheel Drive (AWD)	40	-7.688	-7.222
Automatic	Front-Wheel Drive (FWD)	37	-12.135	-9.03
Automatic	Rear-Wheel Drive (RWD)	46	-12.444	-11.643
Automatic	All-Wheel Drive (AWD)	17	-12.285	-15.0

5. Discussion of the Research Results

The analysis results confirm that actual transmission losses are strongly influenced by both the design features and the technical condition of the system. The obtained data raise questions about the overly optimistic efficiency figures frequently cited in promotional materials and manufacturer specifications, which often fail to reflect real-world performance. These discrepancies highlight the importance of independent testing and statistically robust evaluation methods for obtaining reliable data on drivetrain performance.

Despite the widespread introduction of torque converter lock-up mechanisms, automatic transmissions generally exhibit lower efficiency than manual gearboxes. This reduced performance can be attributed to their inherently more complex design, which includes hydraulic systems, additional moving components, and higher internal friction. Furthermore, the efficiency of automatic transmissions is less sensitive to improvements over time, as the benefits of advanced control strategies, multi-speed designs, and lock-up features are partially offset by the unavoidable energy losses inherent in fluid coupling and torque conversion [8].

In contrast, manual transmissions have shown a clear and consistent trend of decreasing losses over the past decades. This improvement is likely due to several factors, including the adoption of advanced bearing technologies, higher manufacturing precision, improved surface finishes, low-viscosity transmission oils, and the development of next-generation synchronizers that reduce friction during gear changes. As a result, modern manual transmissions demonstrate substantially higher efficiency than their predecessors, particularly in vehicles produced after 2005.

It is also important to emphasize that variability in transmission losses among different car models can reach up to 25%. This variability arises not only from inherent design differences—such as the number of gears, gear ratios, and drivetrain layout—but also from the condition of individual transmission units, including wear, maintenance history, and operating environment. Such a wide range of potential losses underscores the necessity of a case-by-case assessment when evaluating the efficiency of a specific vehicle. It also illustrates that generalized efficiency values, while useful for broad comparisons, may not accurately represent the performance of any single automobile.

Overall, these findings demonstrate the critical interplay between transmission design, technological improvements, and maintenance in determining real-world efficiency. They highlight the need for ongoing research, independent testing, and the application of statistical evaluation methods to provide engineers, fleet managers, and policymakers with reliable insights for optimizing drivetrain performance and reducing energy consumption and emissions in modern automobiles [9].

6. Conclusions

Therefore, based on the analysis of modal values of transmission losses, the following efficiency figures can be regarded as the most probable: manual transmissions, both front- and rear-wheel drive, show an efficiency of approximately 0.91 for vehicles produced between 1995 and 2004, and 0.96 for those manufactured between 2005 and 2010. Automatic transmissions, regardless of drivetrain layout, exhibit a lower efficiency of around 0.89. These results confirm the significant influence of transmission type, design features, and production year on drivetrain performance [10].

The findings also underscore the effectiveness of long-term engineering improvements in manual transmissions, such as the use of advanced bearings, higher manufacturing precision, low-viscosity oils, and modern synchronizers, which have contributed to the gradual reduction of losses over the past decades. In contrast, the inherently more complex design of automatic transmissions, with hydraulic systems and torque converters, limits their practical efficiency gains despite technological advancements such as lock-up mechanisms and multi-speed designs.

Additionally, the observed variability of transmission losses between different models—sometimes reaching up to 25 % — highlights the necessity of considering both the technical condition of individual units and specific design characteristics when assessing real-world efficiency. Overall, these results provide a statistically robust foundation for evaluating transmission performance, offering valuable insights for vehicle manufacturers, engineers, fleet operators, and policymakers aiming to optimize energy efficiency and reduce emissions in modern automobiles.

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ЕКСПЕРИМЕНТАЛНО ИЗСЛЕДВАНЕ НА СПИРАЧНОТО ЗАКЪСНЕНИЕ НА ВЕЛОСИПЕД

EXPERIMENTAL STUDY OF BICYCLE BRAKING DECELERATION

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Abstract. This study investigates the braking deceleration characteristics of bicycles under various operating conditions. The research aims to quantify the relationship between braking force, speed, and road surface type, and to evaluate how these factors affect the overall stopping performance. Experimental tests were conducted using different brake systems, including rim and disc brakes. The braking distance and deceleration were measured with high-precision sensors and analyzed statistically. The results indicate that surface conditions and brake type significantly influence braking efficiency, with disc brakes providing more stable deceleration under variable conditions. The findings can contribute to improving the design of braking systems and enhancing cyclist safety.

Keywords: Bicycle dynamics, braking deceleration, braking performance, road safety.

1. Въведение:

Изследването на динамиката на спирането при велосипедите представлява съществен аспект от инженерните изследвания в областта на пътната безопасност и анализа на движението на превозните средства. Велосипедите, като леки и конструктивно опростени транспортни средства, проявяват колебания в характеристиките на спиране в зависимост от редица фактори, включително типа на спирачната система, взаимодействието на гумата с пътната настилка, разпределението на масата на водача и условията на околната среда. Въпреки нарастващото използване на велосипедите като устойчиво средство за градски транспорт, експерименталните данни за спирачното закъснение остават ограничени в сравнение с тези при моторните превозни средства.

Настоящата разработка има за цел да изследва експериментално спирачното закъснение при велосипеди в контролирани условия, като се анализира влиянието на типа спирачки, началната скорост и характеристиките на пътната настилка. Чрез използването на специализирана измервателна апаратура се цели да се определят характеристиките на спирачното закъснение, както и да се идентифицират основните параметри, които определят ефективността на спирането. Получените резултати имат практическо значение за усъвършенстване на стандартите за безопасност на велосипедите, оптимизация на конструкцията на спирачните системи и разработване на аналитични методи за реконструкция на пътнотранспортни произшествия.

2. Изложение:

Спирачното закъснение представлява основен параметър за оценка на ефективността на спирачната система на велосипеда и способността на водача да запази контрол по време на аварийно или контролирано спиране. То оказва пряко влияние върху спирачния път, устойчивостта и риска от инциденти. Различни изследвания разглеждат теоретични и симулационни модели на процеса на спиране, но точността на тези модели силно зависи от експериментално потвърдени входни данни, получени при реални експлоатационни условия.

Експерименталните изследвания са осъществени на хоризонтален пътен участък при суха асфалтова настилка, като са използвани различни модели хидравлични дискови спирачки. Всички измервания са осъществени с една и съща начална скорост на спиране от 20 км/ч, характерна за градски условия. Целта на изследването е да се определи ефективността на различните спирачни системи и да се направи сравнителен анализ на техните показатели на база на реални измервания на спирачно закъснение и спирачен път. Експериментите са проведени с велосипед “NS BIKES SNABB” (фиг.1), оборудван с 27.5”

гуми, предно и задно въздушно окачване, механична трансмисия “SRAM SX” и спиращи дискове с диаметър 180мм. Теглото на велосипеда е 16.3кг.



Фиг.1. Велосипед на марката “NS BIKES SNABB”.

На фиг.1 е показано мястото на закрепване на уреда върху велосипеда и местоположението на сензора за ускорение. Измервателната апаратура представлява система за регистрация на параметрите на спиращия процес. Уредът е предназначен за определяне на спиращото закъснение и динамиката на спиране. Системата позволява да се записва в реално време моментното ускорение, скоростта на превозното средство и силата на натиск върху спиращката от началото на задействане на спиращната система до пълното спиране.

В проведенето изследване са използвани осем модела хидравлични дискови спиращки, характеризиращи се с различна конструкция, материал на буталата, тип хидравлична течност и маса на спиращния апарат. Основната цел е да се установи влиянието на конструктивните и технологични параметри върху ефективността на спиращния процес. В таблица 1 са представени характеристиките на спиращките.

Таблица 1. Характеристики на спиращките

№	Модел	Брой бутала	Материал на буталата	Размер на буталата (mm)	Вид хидравлична течност	Тегло (g)
1	Shimano BRM	2	Стомана	22	Минерално масло	280
2	Magura MT2	2	Стомана	22	Минерално масло	245
3	Tectro	2	Стомана	20	Минерално масло	311
4	Sram Level	2	Пластмаса	21	DOT 4	330
5	Shimano deore XT	4	Керамика	2x17 2x15	Минерално масло	317
6	Magura MT5	4	Стомана	4x17	Минерално масло	255
7	TRP DHR	4	Стомана	4x16	Минерално масло	309
8	Sram Code RSC	4	Пластмаса	2x16 2x15	DOT 5.1	343

На фиг.2 и фиг. 3 са показани конструктивните разлики между част от двубуталните и четирибуталните спирачки.



а) Magura MT5

б) Magura MT2

Фиг.2. Двубутални а) и четирибутални б) спирачки на марката “Magura MT”.



а) Shimano BRM

б) Shimano deore XT

Фиг.3. Двубутални а) и четирибутални б) спирачки на марката “Shimano”.

Сравнителният анализ между моделите показва, че увеличаването на броя и диаметъра на буталата, както и използването на висококачествени материали, оказват пряко влияние върху ефективността на спиране. Резултатите от експериментите са показани на таблица 2.

Таблица 2. Експериментални резултати

№	Модел	Начална скорост на спиране (km/h)	Изминато разстояние (m)	Спирачен път (m)	Спирачно закъснение (m/s ²)
1	Shimano BRM	20,05	25,17	4,74	3,27
2	Magura MT2	21,36	29,71	3,27	5,38
3	Tectro	21,20	22,21	6,04	2,87
4	Sram Level	20,73	23,07	2,92	5,67
5	Shimano deore XT	21,21	20,18	2,61	6,66
6	Magura MT5	22,64	20,78	3,14	6,29
7	TRP DHR	20,57	18,13	2,75	5,94
8	Sram Code RSC	21,83	21,92	3,34	5,51

Настоящият анализ разглежда експериментално определените параметри на спирачния процес за осем модела хидравлични дискови спирачки, включително дробутални и четирибутални конструкции. Изпитаванията са проведени на начална скорост на движение от 20 км/ч, като спирането е извършено чрез едновременно въздействие върху предната и задната спирачка. За всеки модел са измерени началната скорост, изминатото разстояние, спирачния път и средното спирачно закъснение. Резултатите, представени в таблица 2, показват различия в поведението на отделните системи, което може да се свърже както с техните конструктивни особености, така и с изора на материали, тип хидравлична течност и геометрия на буталата. Сравнителният анализ показва, че съществува ясна нелинейна зависимост между спирачното закъснение и спирачния път. Резултатите, потвърждават, че четирибуталните хидравлични дискови спирачки осигуряват значително по-добри динамични показатели в сравнение с двубуталните. Техните конструктивни предимства – по-голяма ефективна площ, по-добро разпределение на налягането на хидравличната течност водят до по-високи стойности на спирачното закъснение и по-къс спирачен път.

3. Заключение

В резултат на проведението експериментални изследвания върху спирачните характеристики на различни модели хидравлични дискови спирачки е направена цялостна оценка на тяхната ефективност и динамично поведение при реални условия на експлоатация. Изпитванията са проведени при еднаква начална скорост на спиране от 20 км/ч. За измерване на основните параметри – спирачно закъснение, спирачен път и скорост е използвана специализирана апаратура, състояща се от сензор за ускорение и сигнала.

Анализът на получените резултати показва отчетлива зависимост между конструктивните особености на спирачните системи и тяхната ефективност. Четирибуталните спирачки демонстрират значително по-високи стойности на спирачно закъснение и по-къс спирачен път в сравнение с двубуталните системи. Това се дължи на по-голямата ефективна контактна площ и равномерното разпределение на налягането.

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