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Improving Car Fuel Efficiency by Optimising Transmission Parameters

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ABSTRACT

The theoretical and experimental studies of the car power unit functioning at unsteady modes have been conducted. The experimental studies have been conducted using the test stand for measuring engine performance. In addition, the measurements of the parameters of the power unit of the category N_1 vehicle under natural conditions (the road tests in the modes of standardized driving cycles) have been made. The aim of the work is increasing the fuel efficiency of the vehicles under operating conditions. For this purpose, the transmission parameters have been optimised for the conditions of the vehicle driving in the standardized NEDC and WLTC driving cycles. The research results enable to give recommendations on improving both the design of the power unit and its operating modes in operation.

Keywords: Fuel efficiency; driving cycle, transient mode, optimisation; gear ratios, torque.

INTRODUCTION

The dynamic properties and integrated safety are the important components of the modern automobile. The specific indicator of the fuel efficiency (the travel fuel consumption in the standardized driving cycle) combines those properties. Some traction-speed properties of the vehicle beyond the driving cycle can degrade dramatically if considering fuel consumption as a complex criterion for energy efficiency. This will occur even under high available power of the vehicle. Thereby, the considered complex criterion of the specific fuel efficiency must contain additional adjustments in accordance with the category of the vehicle and other requirements of traction dynamics [1,2,3].

The development of this criterion and the measures to reduce fuel consumption is the priority task in designing and operating automobiles. This is due to the rising prices for motor fuel and the tightening of environmental standards. The fuel consumption depends on two groups of factors [4] such as the mass and the shape of the vehicle body (the inertial and dissipative factors) and; the characteristics of the engine, the transmission, and the chassis. The first group determines the energy consumption for the transport work. Reducing fuel consumption due to the factors in this group entails a significant increase in the cost of the vehicle. It requires using structural materials with greater specific strength and the appropriate technologies for their processing. The overall dimensions and the body shape are taken in accordance with the prevailing modes of the movement of the vehicle and its purpose. The size and the shape of the vehicle affect the fuel consumption on highways significantly. Thus, dealing with the factors of the first group does not give an unambiguous reduction in fuel consumption. The second group of factors determines the efficiency of creating mechanical power in the engine and the efficiency of its transfer to the drive wheels. The total energy loss in the power unit (the engine and the transmission) is significant [5]. For example, the overall energy efficiency of the power unit (PU) for various driving conditions varies from 0.05 to 0.35 [4, 6] for vehicles of the N_1 category. Thereby, the way of increasing the specific fuel efficiency by improving the design and the control modes of the PU (working on the factors of the second group) is rational. This direction is associated with the working processes of the units with large energy losses [3].

This problem is solved effectively in two ways. The first way is the traditional one. It implies using more advanced and more expensive components and assemblies of greater energy efficiency [6]. The second one is the increasing of the average value of the overall engine and transmission efficiency. This is achieved by means of matching the parameters of the PU. It provides rationale modes of the operation of the engine and the transmission. This direction is the task of the optimisation. Such tasks are multiparameter ones. To solve them the methods of the numerical simulation of the vehicle movement process are used. Such formulation of the problem requires the mathematical definition of the objective function of the complex optimality criterion and choosing the optimisation parameters. In addition, it requires the system of restrictions and the mathematical models of the characteristics of the vehicle and the modes of its movement. There is also a need for the algorithm for correcting results.

This optimisation problem is solved relatively simply for vehicles with manual discreet gearboxes. The optimisation parameters are the parameters of the transmission. Those are the gear ratios, the gear range and the density of the range. This reduces the cost of the subsequent introducing of the optimisation results in the production process. The paper proposes the theory and the results of the computational and experimental studies of the improving of the fuel efficiency of vehicles of various categories. The results are given for vehicles with discreet manual transmissions. The studies have been carried out in the complex of non-stationary driving modes on the example of standardised driving cycles.

PRINCIPLES OF CHOOSING POWER UNIT PARAMETERS

The proposed structural diagram of the method of the rational selection of the power unit parameters is shown in Figure 1. This process consists of the sequential execution of four blocks (modules).

Formulation of Optimisation Problem

The formulation of the optimisation problem is carried out in the first module. The basic criterion is the specific indicator of fuel efficiency, the fuel consumption in the driving cycle:

$$m = \int_0^{t_C} G(t)\ dt, \text{ g/driving cycle,} \tag{1}$$

where *G(t)* is a function of mass fuel consumption of the engine depending on the time of the movement in the driving cycle, g/s; t_C is the general time of the driving cycle, s.

The objective of the optimisation function is represented by the Eq. (1). The expression takes into account the mode of the operation of the internal combustion engine as a function of the optimisation parameters, transmission ratios ($U_{(m)}$):

$$m_{(m)} = f(U_{(m)}) = \sum_{1}^{n_C} \int_{t_n}^{t_{n+1}} G(t,U_{(m)})dt + \sum_{1}^{s_C} \int_{t_s}^{t_{s+1}} G_{FI}(t,U_{(m)})dt, \text{ g/driving cycle,} \qquad (2)$$

where $G(t,U_{(m)})$, $G_{FI}(t,U_{(m)})$ are the functions of the mass fuel consumption when the vehicle is moving in the traction mode (acceleration and constant speed) and in the mode of the forced idling (FI) of the internal combustion engine (ICE); t_n is the start time of the n_C sections of the driving cycle (driving in traction mode), t_s is the start time of the s_c sections of the driving cycle in which the engine works in forced idling mode, s.

The above definition of the criterion is acceptable for vehicles with a low power rating. The main purpose of those vehicles is carrying out the transport work (the vehicles of the N and M categories with the power supply of less than 30 W/kg). Such vehicles are usually designed based on passenger cars. There are additional requirements for passenger cars: ensuring the specified traction and speed properties like acceleration time up to 100 km/h, etc. It is rational to supplement the objective criterion function for such cars with private conditions at the stage of correcting the results (module 4, Figure 1).

Optimisation parameters

The optimisation parameters are the parameters of the discreet mechanical transmission. They have some connection with each other. They are only relatively independent values. For example, an increase in the number of gears leads to an increase in the density of the gear range. At the same time, the rationality of optimising each gear ratio is reduced, i.e. the properties of the transmission approach those of the continuously variable transmission. The rational choosing the gear ratios for each gear does not require an adjustment of the final gear ratio as the calculations use the total gear ratios of the power unit. Thereby, the simplest and the most inexpensive solution is accepting the gear ratio of the final gear as the optimisation parameter. But this solution provides the least beneficial effect from the upgrade. This is due to the presence of the previous patterns between the gear ratios of the gearbox.

From the authors' point of view, the most rational approach is choosing the gear ratios of the gearbox as the optimisation parameters. For the fixed number of gears in the gearbox, this allows effective coordinating the engine operation with the driving conditions of the vehicle according to the selected criterion. For the trucks of the N_2 and N_3 categories, it is efficient to vary both the number of gears and the values of the individual gear ratios in the presence of the transmission divider and the multiplexer. It is also possible to consider the moments of the gear shifting as the optimisation parameters of the power unit. The conditions for the shifting from one gear to another are regulated to varying degrees by the driving cycles (for example, in NEDC). This issue is more freely addressed in WLTP.

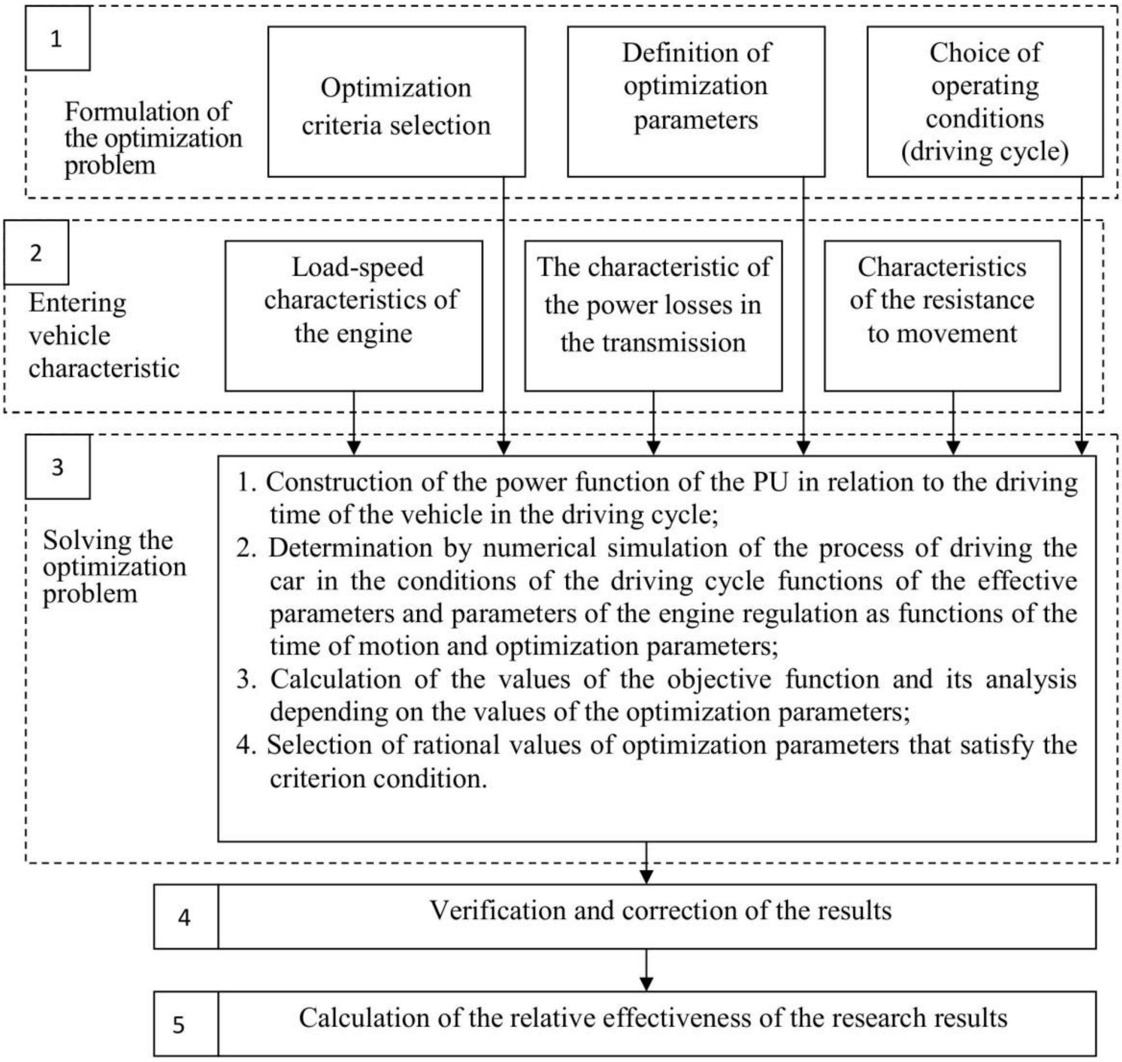

Figure 1 Scheme of choosing the rational parameters of the power unit.

Limitations

Conditions for limiting the optimisation process are the observability of the power balance in the transmission at each time point, the possibility of vehicle's overcoming the required road resistance, meeting the reliability and durability requirements, ensuring the movement at maximum speed. The gear shift points and the number of gears may also be limitations. This is relevant if these parameters are not varied by the condition of the specific optimisation problem.

Optimisation procedure

The optimisation procedure represents the extremum test of the functions of the (2) form. Their number corresponds to the number of the gear ratios as variable parameters. The direct differential analysis implemented is applied for this purpose. Despite the fact that the problem is multiparameter, the study of each function (2) is a one-parameter optimisation problem. The results of the optimisation search with considering the system of constraints are shown in heading «The results of solving the optimisation problems».

MATHEMATICAL DESCRIPTION OF VEHICLE CHARACTERISTICS

The reliability, the scientific and practical value of the research results is determined by the quality of mathematical models of the characteristics of the vehicle. The results are obtained by the method of the numerical simulation of the process of movement of the vehicle.

The mathematical model of the load-speed characteristics of the effective indicators of the engine is of particular importance as the engine determines fuel-economic, traction-speed and environmental properties of the automobile. The effective power N_e and the effective specific fuel consumption in the steady-state mode of the internal combustion engine g_e are functions of the crankshaft rotation speed n and the power factor k: $N_e=f(n, k)$, $g_e=f(n, k)$. The values of the effective indicators and the parameters of the regulation of the ICE in the steady-state mode do not change with time (permissible change is 0.2% in 30 seconds). The coefficient k is determined by the ratio between the current developed effective power to the maximum power achieved at the current rotational speed of the engine crankshaft with the full fuel supply: $k=N_{e(n)}/N_{eSC(n)}=M_{e(n)}/M_{eSC(n)}$.

The automotive engine runs in the unsteady mode most of the time. Consider fragments of the ECE-15 NEDC driving cycle. The engine operates in the traction mode 47.7% of the total travel time. 38.7% of this time is accompanied by the acceleration processes [8]. Slowing down and operating the engine without a load with the clutch disengaged takes 13.3% and 39% of the driving cycle time, respectively. There are no sections of the movement in the steady-state engine mode under WLTC cycle conditions.

The effective performance indicators of the ICE in the unsteady mode may differ significantly from the values in the corresponding steady-state modes. During acceleration the power decreases and the specific fuel consumption increases. This deterioration for the automotive engines may be over 40% in the operating conditions. This figure depends on the intensity of the unsteady mode. The functional definition of the mathematical model of the load-speed characteristics of the ICE is adopted in accordance with [5]. The parameters of the regulation of the ICE and their first derivatives are arguments in $N_e=f(n, \lambda, \varepsilon, \delta)$, $g_e=f(n, \lambda, \varepsilon, \delta)$, where ε is the crankshaft angular acceleration, rad/s^2 ($\varepsilon=d(\pi \cdot n \cdot 30^{-1})/dt$); δ is the load change rate ($\delta = d\lambda/dt$). The load factor λ is associated only with the position of the regulator of the load (i.e. the fuel supply) of the engine. This model is more relevant for sports cars and light vehicles (categories M_1 and N_1). This is partly due to the maximum positive longitudinal accelerations in driving cycles. For NEDC (categories M_1 and N_1) it's 1.04 m/s^2; for driving cycles according to GOST R 54810-2011 it's 0.85 m/s^2 (N_2 with a gross mass up to 7.5 tons), 0.83 m/s^2 (N_2 with a total mass from 7.5 to 12 tons), 0.65 m/s^2 (N_3 with a gross mass up to 21 tons), 0.64 m/s^2 (N_3 with a gross mass from 21 to 32 tons). More modern driving cycles within the same category (for example, M_1 and N_1) have an increased traffic intensity as compared to NEDC. The WLTC cycle for a vehicle with the power capacity of 40.5 W/kg (subgroup b) has a maximum acceleration of 1.637 m/s^2. Determining the effective performance of the ICE in the unsteady operating modes by the calculation method is a relatively difficult task as it is necessary to take into account the influence of the mechanical, thermal, and hydraulic inertia in the systems and aggregates. These issues require additional research. Therefore, the experimental determination of the load-speed characteristics of the ICE with regard to the unsteady modes has been performed in the present work.

The operation of the engine in PU is associated with the additional loss of the developed power in the gearbox. This loss must be considered when calculating fuel consumption. The range of variation of the efficiency of the manual gearbox is small relative to other types of transmissions. This allows taking efficiency as a constant. The function of the power needed for overcoming the environmental resistance is determined by the power balance equation:

$$(N_C + N_R + N_A + N_I + N_{I\varepsilon}) / \eta_t = N_T + N_{ET} = N_{PU} = N_Q - N_{GB} - N_{ICE}, \quad (3)$$

where η_t is the transmission efficiency (minus the gearbox); N_C, N_R, N_A, N_I, $N_{I\varepsilon}$ are the powers needed for overcoming (respectively) the lifting resistance, the rolling resistance, the air resistance, the inertia force, and the inertia of the rotating masses (minus the flight masses of PU); N_T is the power traction ($N_T = N_C + N_R + N_A + N_I + N_{I\varepsilon}$); N_{ET} is the total loss of power in the transmission units (except for gearbox) and the chassis; N_{PU} is the power developed by PU; N_Q is the thermal power (it is primary and enters the engine with the fuel, $N_Q = N_e / \eta_e = H_u \cdot G$, where η_e is the effective efficiency of the ICE, H_U is the net calorific value of fuel, MJ/kg); N_{GB} and N_{ICE} are the power losses in SU determined by the value of η_e and the gearbox efficiency. The balance (3) is represented in Figure 2.

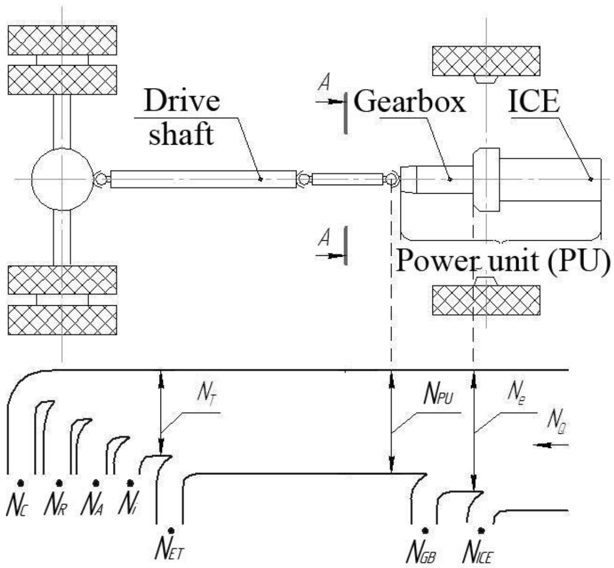

Figure 2. Diagram of power components with power unit and transmission.

The left side of the expression (3) can be calculated in accordance with the equation of the vehicle straight motion:

$$N_{PU} = f(V, j) = (V \cdot (\psi \cdot m_a \cdot g + k_A \cdot F \cdot V^2 + m_a \cdot j) + J \cdot j \cdot V / r_R^2) / (1000 \cdot \eta_T), \text{ kW} \quad (4)$$

where V is vehicle speed (m/s), j is longitudinal acceleration (m/s^2), ψ is running resistance coefficient, m_a is mass of vehicle (kg), k_A is air resistance coefficient (kg/m^3), F is area of midsection of vehicle (m^2)m, J is total moment of inertia of the rotating masses of vehicle relative to drive wheel (kg$\cdot m^2$) and; r_R is the drive wheel rolling radius (m).

The accurate determination of the values of η_t, ψ, k_A, F, J for a particular vehicle model is the daunting task and requires special research. In the present paper, the $N_{PU}=f(V,j)$ function is determined experimentally in the course of conducting the road tests. It is necessary to increase the reliability of the numerical simulation results. The transmission dynamometer has been applied in the tests. The device has been installed in place of the intermediate cardan shaft in section A–A, Figure 2.

EXPERIMENTAL SETUP

The laboratory setup has been used for the experimental determination of the load-speed characteristics of the automotive ICE taking into account the unsteady operating modes. The scheme of the setup is shown in Figure 3.

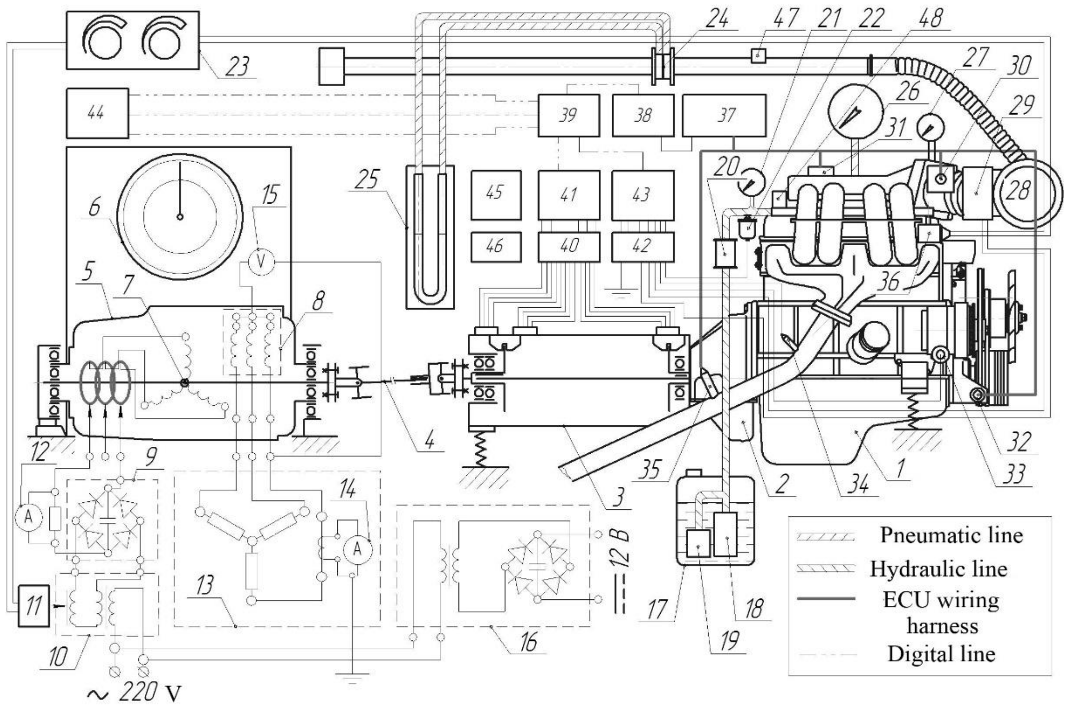

Figure 3 Scheme of the stand for studying the effective performance of automotive ICE taking into account unsteady operating modes:

(1 – ICE; 2 – clutch coupling; 3 – torsion dynamometer; 4 – cardan transmission; 5 – electric machine; 6 – balance dynamometer; 7 – phase rotor; 8 – stator winding; 9 – rectifier; 10 – adjustable transformer; 11 – brake load control drive; 12 – rotor ammeter; 13 – resistors; 14 – ammeter of the stator circuit (of the load); 15 – stator voltmeter; 16 –the source of electricity for the auxiliary devices (12 V); 17 – fuel tank; 18 – fuel pump; 19 – pressure reducer; 20 – fuel filter; 21 – fuel pressure gauge; 22 – fuel pressure sensor; 23 –the control unit of the engine and brakes load; 24 – air flow meter; 25 – differential gauge of the air flow meter; 26 – vacuum gauge; 27 – gauge lubrication system; 28 – air filter; 29 – air flow sensor; 30 – throttle position sensor; 31 – TMAP sensor; 32 – crankshaft position sensor; 33 – oil temperature sensor; 34 – exhaust temperature sensor; 35 – oxygen sensor; 36 – drive throttle; 37 – ECU; 38 – diagnostic scanner; 39 – digital amplifier; 40 – torsion dynamometer block; 41 – automated data acquisition system; 42 – additional mounting block; 43 – additional automated data acquisition system; 44 – personal computer; 45 – aneroid barometer; 46 – psychrometric hygrometer; 47 – air flow meter thermometer, 48 – fuel thermometer)

The upgraded break-stand KI-5543 GOSNITI is the base of the setup as shown in Figure 4. The torque on the unsteady mode of the ICE is measured using a non-contact electric torsional dynamometer (position 3 of Figure 3) [9]. Its graphic model and longitudinal section are shown in Figure 5. The torque is calculated in the LabVIEW environment. The asynchronous machine has been converted to the synchronous brake.

This is necessary to extend the rotation speed range. The "L-Card" company's E-14-140-M module is used as an automated data collection system.

(a) (b)

Figure 4. Stand for testing the ICE taking into account the unsteady modes: (a) side view of ICE and dynamometer, (b) front view of the brake-stand.

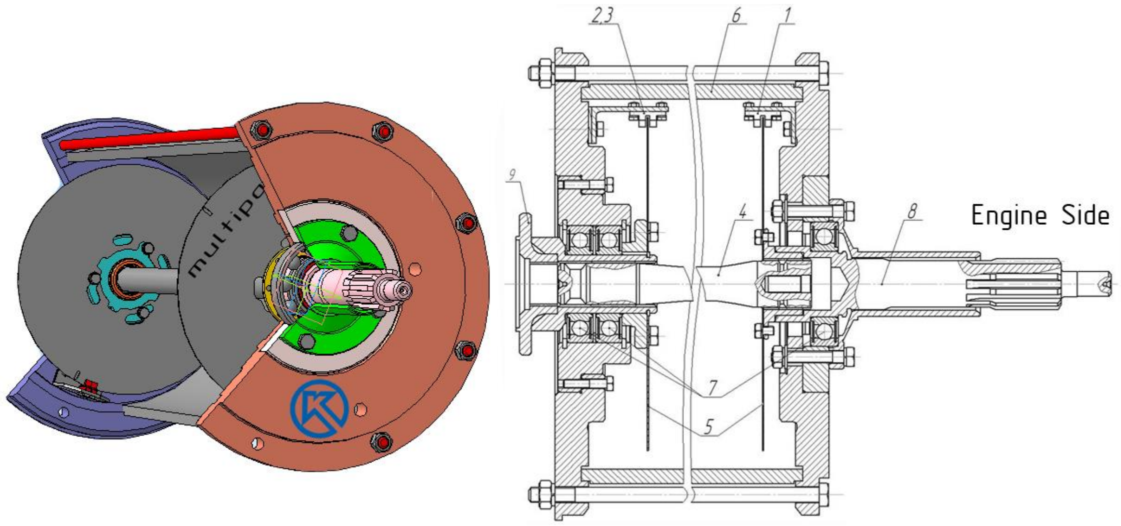

Figure 5 Device for changing the torque of the ICE at the unsteady mode:
(with 1,2,3 - optical sensors; 4 - torsion shaft; 5 - measuring discs; 6 - housing, 7 - bearings, 8 - drive shaft, 9 - cardan shaft flange).

The fuel consumption is determined by the dependency [11]:

$$G = f(\tau_C, \Delta P) = 3600^{-1} \cdot \rho_F \cdot n \cdot 30 \cdot 10^{-6} \cdot \sum(\tau_C \cdot Q_a(\tau_C, \Delta P)),\ \text{g/s}, \tag{5}$$

where τ_C is the electrical nozzle opening time, ms; ΔP is the pressure drop across the nozzle, bar; ρ_F is the density of the fuel, kg/m^3; Q_a is the function of the volumetric performance of the electromechanical nozzle, ml/s. The $Q_a = f(\tau_C, \Delta P)$ dependency has been obtained for each nozzle using the NA4 200W stand.

RESULTS AND DISCUSSION

The vehicle GAZ-3302 of the N_1 category has been tested as an example. The resulting load-speed characteristics of the engine considering the unsteady modes are shown in Figure 6. The results are shown as a family of diagrams in the coordinates *n* and *k* [12].

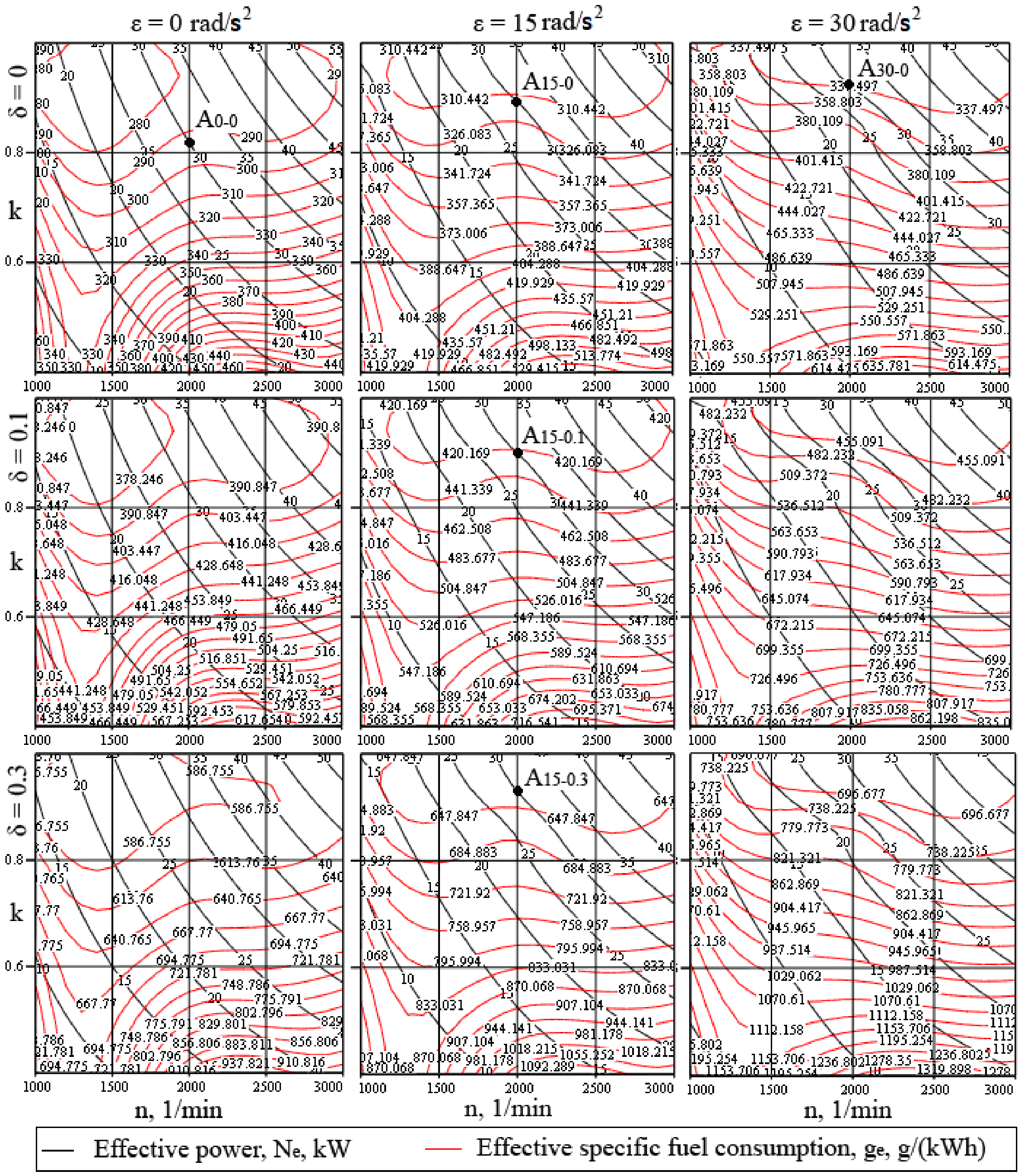

Figure 6. Universal dynamic characteristics of UMZ-4216 ICE.

The upper left diagram corresponds to the steady-state mode (ε=0; δ=0). The constant rate of the load change corresponds to each horizontal row of diagrams δ=0; δ=0,1 and δ=0,3. Vertical columns of the diagrams have a constant acceleration of the crankshaft: ε=0 rad/s^2; ε=15 rad/s^2; ε=30 rad/s^2. The experimental dependency of power factor *k* on the load factor λ for the tested ICE model is shown in Figure 7. λ is the relative area of the throttle pipe. This parameter is necessary to indicate the degree of completeness of the engine load. This makes it easier to recreate the typical unsteady mode in the course of the testing. The characteristic of the resistance to movement is shown in Figure 8 with $N_{PU} = f\ (V,\ j)$. The two-factor dependency is obtained by the approximation of the experimental data determined during road tests of GAZ-3302 vehicle. The experiments have been carried out according to GOST R 54810-2011.

A special modification of the dynamometer was used for changing the torque in the transmission (shown in Figure 2). Consider the example of the influence of the unsteady mode on the performance of the engine (from Figure 6). The engine develops the effective power N_e = 30 kW at the speed of 2000 min^{-1}, where g_e= 294.3 g/kWh (point «A_{0-0}») is the steady-state mode. When giving the angular acceleration of 15 rad/s^2 or 30 rad/s^2 (points «A_{15-0}» and «A_{30-0}») g_e is 314.3 g/kWh and 337.5 g/kWh. The specific fuel consumption increases with an increase in the load change rate δ at the selected mode. At ε=15 rad/s^2 and δ=0,1 (point «$A_{15-0.1}$») g_e= 423 g/kWh; at ε=15 rad/s^2 and δ=0,3 (point «$A_{15-0.3}$») g_e= 630 g/kWh. Thereby, the intensity of the unsteady modes has a significant impact on the effective performance of automotive ICE.

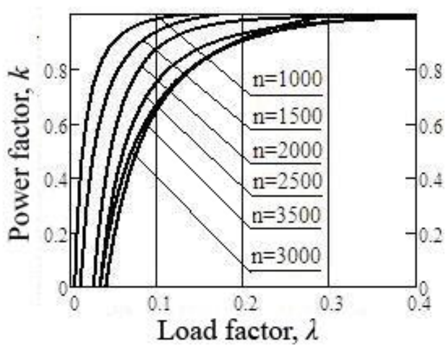

Figure 7. The relation of power factor k and load factor λ.

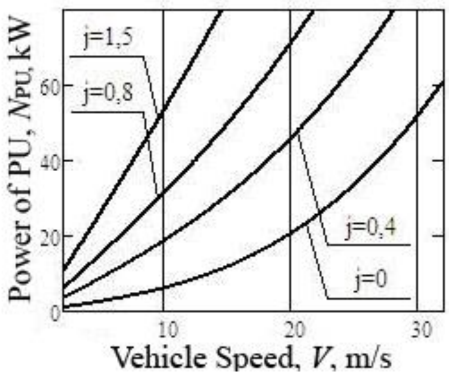

Figure 8. Characteristic of power of the movement resistance for GAZ-3302 vehicle with test weight of 2750 kg.

Optimisation

The procedure of the numerical simulation of the process of movement in the driving cycle has been carried out on the basis of equality of the power balance in the transmission. The kinematic parameters of the load-speed characteristics (n and ε) are consistent with the parameters of the driving mode (V and j) through the dependencies $n = V \cdot U_0 \cdot U_{(m)} / (0{,}105 \cdot r_R)$, min^{-1}; $\varepsilon = j \cdot U_0 \cdot U_{(m)} / r_R$, rad/s^2, where U_0 is the final gear ratio. The amount of power in the transmission N_{PU} is consistent with the parameters λ and δ by means of the iterative algorithm. The calculation has been made in the Mathcad software environment.

The calculation of fuel efficiency according to Eq. (2) is shown in Figure 9. The optimisation parameters are the gear ratios of the higher gears of the gearbox $U_{(m)}$. The gear ratio of the first gear is selected according to the condition of the overcoming the maximum road resistance.

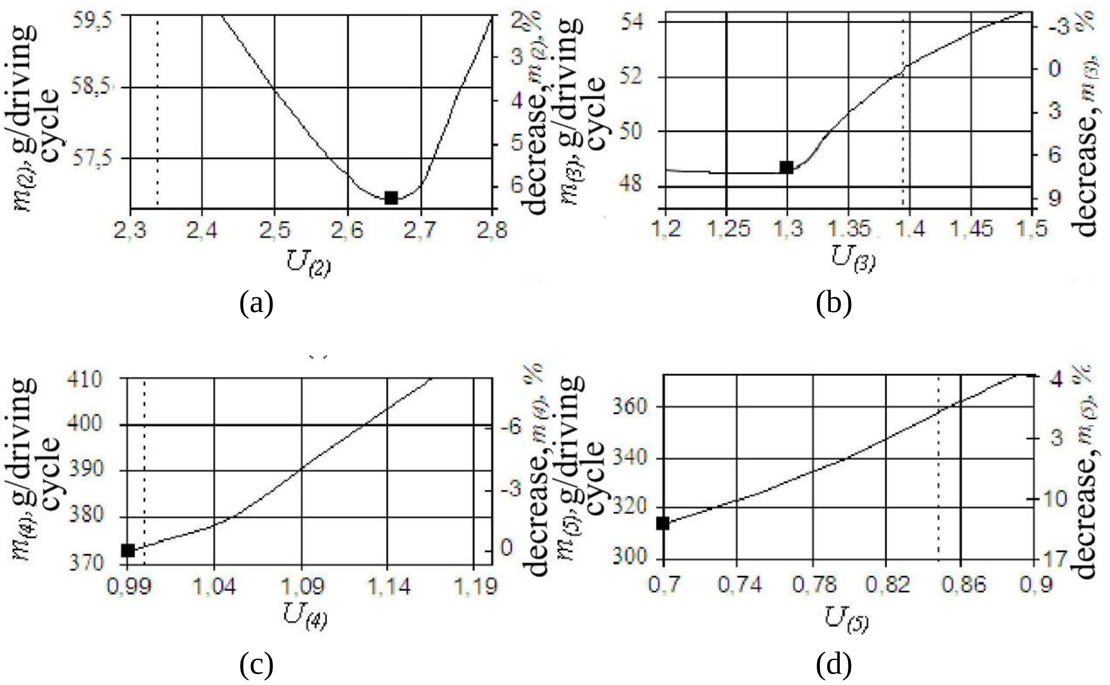

Figure 9. Values of the target optimisation function for the higher gears of GAZ-3302 vehicle gearbox in the NEDC low power vehicle at (a) second gear ratio, (b) third gear ratio, (c) fourth gear ratio and; (d) fifth gear ratio.

The consumed amount of fuel in the parts of the driving cycle when driving on gears $U_{(m)}$ is given on the axis of ordinates. Gear ratios of the serial gearbox are shown by the vertical dashed line. The minimum abscissa axis values are limited by the condition of the motion in the driving cycle. The maximum abscissa values are limited by the crankshaft rotational speed. The markers indicate the minimum values of fuel consumption while the corresponding gear ratios are rational (optimal) for use. The given graphs correspond to the movement of the GAZ-3302 vehicle with the UMZ-4216 engine in NEDC conditions.

Table 1 shows the ranges of gear ratios of the gearbox. The serial range is $U_{(m)S}$, the optimised range is $U_{(m)O}$, and the recommended range is $U_{(m)R}$. The recommended range differs from the optimised range by the possibility of the practical application in the serial gearbox. At the same time, compliance with the requirements of durability, reliability, durability, and driving smoothness are ensured. The results of the calculations of the relative reduction in fuel consumption in the driving cycle are shown in Table 2. The consumption depends on the various ranges of the gear ratios in the gearbox. The proposed method for choosing the parameters of the transmission allows increase of fuel efficiency of the vehicle in terms of the driving cycle. The reduction in fuel consumption for the GAZ-3302 vehicle is as follows: UDC driving cycle: 4.395%; EUDC low power vehicle: 4.447%; NEDC: 4.424%.

Table 1. Series of transmission ratios of the GAZ-3302 gearbox.

m	1	2	3	4	5
$U_{(m)C}$	4.05	2.34	1.395	1	0.849
$U_{(m)O}$	4.05	2.65	1.3	0.99	0.7
$U_{(m)P}$	4.05	2.63	1.305	1	0.738

The values given are valid with the final drive ratio of 4.3.

Table 2. Reducing the fuel consumption of GAZ-3302 vehicle when applying different gear ratios in the transmission.

Driving modes		Driving cycle (fragment)		
		ECE-15	EUDC	NEDC
		Relative reduction of fuel consumption (%) at optimised/recommended series of gear ratios		
Movement on sections of cycle, with m gear is engaged	m = 2	6.288 / 5.744	-10.119 / - 9.487	4.924 / 4.492
	m = 3	6.957 / 6.481	4.138 / 3.91	6.71 / 6.257
	m = 4	-	0.759 / 0	0.759 / 0
	m = 5	-	13.956 / 10.841	13.956 / 10.841
Traction mode		5.861 / 5.409	5.815 / 4.567	5.834 / 4.898
Overclocking Modes		-3.971 / -3.686	-1.074 / -0.995	-2.579 / -2.392
Constant speed		16.372 / 14.987	8.391 / 6.59	10.833 / 9.151
For the entire driving cycle		4.758 / 4.395	5.660 / 4.447	5.265 / 4.424

Gear Ratio

A similar optimisation has been performed on a car from N_1 category GAZ-2752. The power supply of the vehicle is 40.5 W/kg and the engine model is UMZ-4216. WLTC (subgroup b) is selected as the driving cycle. The optimisation parameter is the gear ratio of final gear U_0. The gear shifting moments are taken by the minimum travel fuel consumption.

Results are shown by line 2 plot in Figure 10 where line 1 plot is shown for comparison, which corresponds to the fuel consumption in the driving cycle when using the simplified mathematical model of the engine (without taking the unsteady operation modes into account): $N_e=f(n,k)$, $g_e=f(n,k)$. "A" and "B" markers correspond to the values of the fuel consumption for serial transmission. The fuel efficiency indicators in WLTC driving cycle and the relevant traction-speed properties are summarised in Table 3. Thereby, the final gear ratio 4.1 allows us to increase fuel efficiency in WLTC conditions by 6%. At the same time, the acceleration time to 100 km/h increases by 1.9% when compared to the serial value of the gear ratio of 4.3. The travel times at 400 m and 1000 m distances were increased by 0.6 and 2.45% respectively.

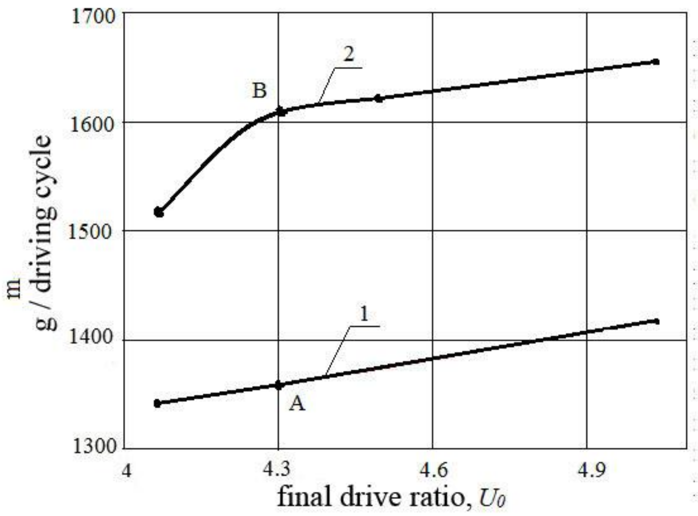

Figure 10. Fuel consumption per driving cycle of GAZ-2752 with UMZ-4216 engine, depending on the final gear ratio.

Table 3. Indicators of traction-speed properties and fuel efficiency in the WLTC driving cycle with different final gear ratios.

Final drive ratio	Fuel consumption in driving cycle *m*, (g/driving cycle)	Acceleration time to 100 km/h (s)	Travel time at 400 m (s)	Travel time at 1000 m (s)
Values are derived from the calculation model – steady-state model [4,5]: $N_e=f(n, k)$, $g_e=f(n, k)$				
4.1	1341(1.7)	21.1(-0.95)	23.1(-0.26)	41,78(-0.19)
4.3*	1357(0)	20.9(0)	23,04(0)	41,7(0)
5.125	1418(-4.5)	21,3(-1,9)	23,8(-3,29)	44(-5,5)
Values are obtained on the basis of the experimental model - unsteady-state model of Figure 6 [5]: $N_e=f(n, \lambda, \varepsilon, \delta)$, $g_e=f(n, \lambda, \varepsilon, \delta)$				
4.1	1553(6)	30/(1.9)	29.4(0.6)	49.2(2.45)
4.3*	1654(0)	30.6(0)	29.57(0)	50.44(0)
4.556	1664(-0.6)	29.3(4.2)	29.4(0.6)	50.511(-0.14)

* - gear ratio of the serial transmission of the car GAZ 2752 "Sobol Business"

** - the percent difference of indicators in relation to the corresponding values obtained with the use of the serial transmission gear ratio (U_0=4.3) is given in parentheses

Figure 11 shows the results of the influence of the number of gears and the type of transmission ratio numbers on the fuel efficiency of trucks of categories N_2 and N_3 under the conditions of the sequential execution of urban and motorway driving cycles according to GOST R 54810-2011. The average value of fuel consumption in the driving cycle is defined as the criterion: L / 100km = $m \rho / S$, where m is the complex optimisation criterion (1), *g / s*; ρ is the fuel density, kg / dm^3; S is the length of the driving cycle, *m*. The number of gears of discrete series of transmission and its form has been considered as optimisation parameters: the geometric and hyperbolic series were considered as the most common ones. Mathematical models of the load-speed characteristics of the effective engine performance indicators, as well as those of the losses in the transmission and the power expenses for overcoming the total environmental resistance, have been compiled based on the well-known recommendations of the theory of the automobile [4]. The average values of the corresponding parameters for the four conditional groups shown in Figure 11(a) to 11(d), were obtained based on the study of the characteristics of cars production (used as initial data).

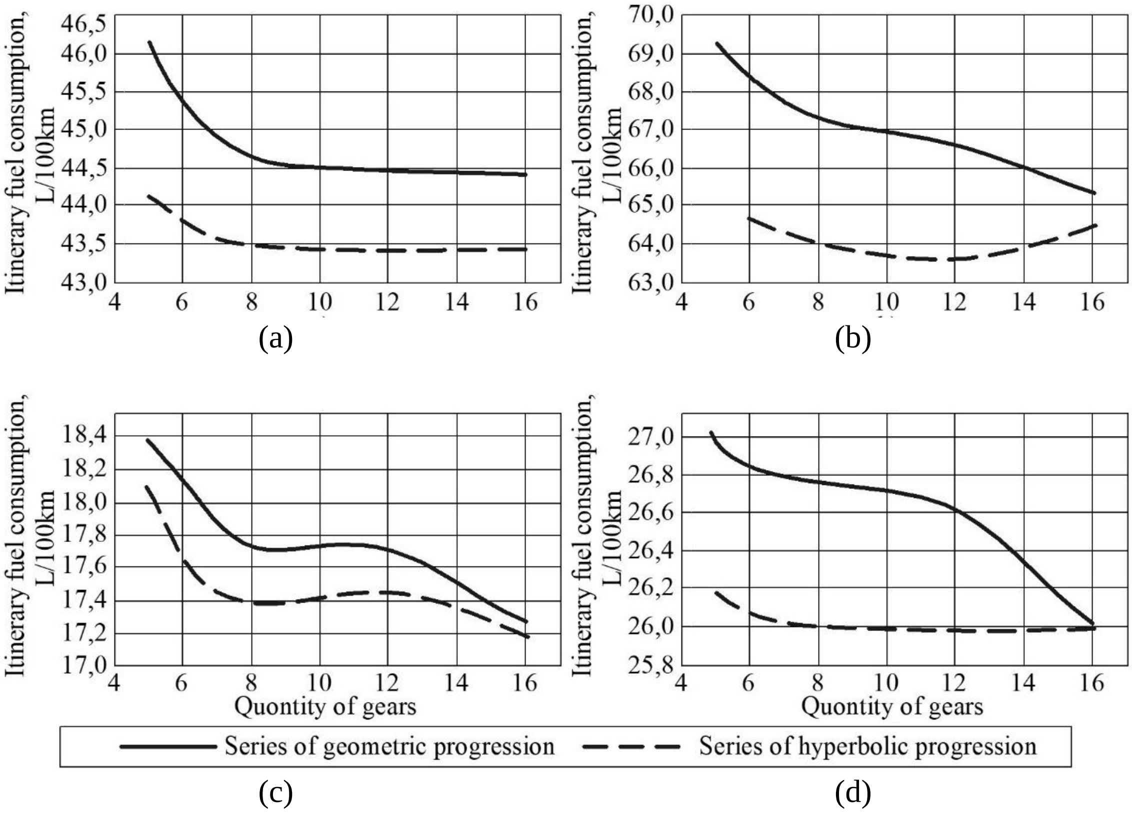

Figure 11. Fuel consumption for the driving cycle of the different category vehicle, depending on the quantity of gear in gearbox and series of gear ratios: N_2 category with gross mass of 7600 kg (a) and 12000 kg (b), N_3 category with gross mass 21000 kg (c) and; 32000 kg (d).

The results from numerical modelling of process of motion prove that fuel efficiency increases with increasing number of gears for trucks. In this case, in accordance with Figure 11, it is possible to recommend a rational limit: for trucks of categories N_2 and N_3 with a gross mass of 7600 kg and 21000 kg respectively - 8-10 gears; for trucks of categories N_2 and N_3 with a total mass of 12000 kg and 32000 kg respectively - 12-16 gears and more. The increase of fuel consumption with increasing number of gears in individual cases is explained by the influence of the gearshift processes in the gearbox and the lower efficiency of the unit itself.

CONCLUSION

The load characteristics of the automotive gasoline engine, when operated at unsteady mode, have been determined. The mathematical model of the characteristics of the gasoline ICE has been obtained based on the above results. The model allows calculating the indications of the ICE for the modes of the vehicle acceleration which are characteristic of the driving cycles. The characteristics of the vehicle's power unit when moving at the unsteady modes are determined not only by the engine parameters but also by the transmission parameters such as gear ratios in manual transmissions. The results obtained and the research methods proposed can be applied in the development of new car designs as well as in the recommendations for their efficient operation.

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