MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE
NATIONAL TECHNICAL UNIVERSITY
KHARKIV POLYTECHNICAL INSTITUTE

TRACTION AND DYNAMIC, FUEL AND ECONOMIC CALCULATION OF THE VEHICLE

Methodological instructive regulations
for calculation, course works and diploma theses
of bachelor and master degrees
of specialty
274 "Automobile transport"
full-time education

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INTRODUCTION

Traction and dynamic, fuel and economic calculation of the vehicle is an important and integral part of a complex approach to vehicle design. According to the results of traction calculation, the main dynamic parameters of the vehicle are determined, the assessment of dynamic qualities is given by comparing them with similar indicators of other existing construction. During the traction calculation, such important parameters as the maximum power of the internal combustion engine, the gear ratios of the transmission elements, etc. are determined.

There are two main methods for traction and dynamic calculation. The first method is a design traction calculation. It is performed when designing a new vehicle. The second method is a test traction calculation, which is performed for an existing vehicle. The test calculation is performed during the diploma. It is related with the selection of the base vehicle for a specialized vehicle or tractor for a road train.

The purpose of this work is to help students determine the main dynamic and economic parameters of the vehicle and engine when changing the load and speed modes of its operation; dynamic performance during acceleration, to calculate the fuel efficiency of the vehicle. The proposed method of calculation will help to form students' knowledge, skills and abilities to analyze and select the parameters of the vehicle, which ensure the implementation of the specified indicators of its performance properties.

The goal is achieved as a result of calculations and plotting of detailed tractionand speed characteristics of the vehicle, which characterize its consumer properties, followed by analysis of the parameters of vehicles.

The peculiarity of this publication is to provide examples of determining the parameters of dynamic, fuel and economic characteristics of the vehicle, which allow to implement the concept of individual approach in the learning process.

In the methodological instructive regulations the technique of check tractioncalculation on an example of the car Opel Astra Classic is resulted.

1 INITIAL DATA AND MAIN CHARACTERISTICS OF A VEHICLE

During performing a test traction calculation of the vehicle as the initial data, the values of its parameters specified by the manufacturer in the technical data sheet of the vehicle are used.

1.1 Mass characteristics of the vehicle

Total mass M_a (fully loaded) and mass of equipped vehicle M_o (fully filled and equipped with appropriate equipment and tools) are accepted according to technical data sheet.

The weight of the vehicle, which is accounted for by the front axle M_1 and rear M_2, is also taken according to the technical data sheet characteristics. But if M_1 and M_2 are not given, they can be determined when the percentage of load distribution on the axles of the vehicle is known.

Approximately for a vehicle with front-wheel drive, the axle load distribution is 60 % and 40 % (where 60 % of the load is on the front axle).

For a truck, the load distribution on the axle is approximately 30 % and 70% (where 30 % of the load is on the front axle).

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Mass characteristics of the vehicle

$Mo := 1320$ *(kg) – mass of the equipped vehicle*

$Ma := 1850$ *(kg) – total mass of the vehicle*

$M1 := 955$ *(kg) – load on the front axle of the vehicle*

$M2 := 895$ *(kg) – load on the rear axle of the vehicle*

1.2 The selection of the vehicle tire and calculation of static radius of wheel rolling

Tires greatly affect the dynamic and economic factors, traction and braking properties, stability, handling, smoothness of the vehicle.

Tires are divided on purpose, the form of a profile, the form of tire tread pattern, the principle of sealing, a design (radial, diagonal, wide-profile, low-profile, etc.).

In marking of the tire its sizes, a design, loading capacity, date of manufacturing are specified.

Under the action of a vertical load, as a result of deformation of the elastic tire rim, the distance from the wheel axis to the bearing surface decreases. This distance is called the static radius r_{st} of the wheel. If the wheel is also under the action of torque, this distance becomes even smaller due to the action of tangential deformation of the tire, it is called the dynamic radius r_k. Due to the small difference in size between static and dynamic radii the static radius given in standard at movement of the vehicle without slipping and sliding in practical calculations is equal to dynamic.

The static radius of the wheel is determined by the formula

$$r_k = 0,5 \cdot d \cdot 0,0254 + k_{sh} \cdot 10^{-2} \cdot B \cdot 10^{-3} \cdot \lambda_{sm}, \qquad (1.1)$$

where d – nominal landing diameter of a wheel rim, in inches (in the formula it is converted to meters);

k_{sh} – tire series – the nominal ratio of the height of the tire profile to its width "*H*/*B*" in %;

B – nominal width of the tire profile in mm (in the formula it is converted to meters);

λ_{sm} – the coefficient, which takes into account the collapse of the tire under load, is in the range: 0,8÷0,85 – for cars [18, p. 10], and 0,86÷0,935 – for trucks[14, p. 8].

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of static wheel rolling radius

In the calculation the tires are selected for the technical data sheet characteristics of the vehicle – 185/65 R15.

$d := 15$ *(inch) – nominal diameter at wheel rim seat*

$ksh := 65$ *(%) – nominal ratio of profile height to its width*

$B := 185$ *(mm) – nominal width of the tire profile*

$\lambda sm := 0.825$ *– coefficient of collapse of the tire under load*

$$rk := 0.5 \cdot d \cdot 0.0254 + ksh \cdot 10^{-2} \cdot B \cdot 10^{-3} \cdot \lambda sm$$

$rk = 0.290$ *(m)*

2 SELECTION OF THE ENGINE AND CONSTRUCTION OF ITS EXTERNAL (SPEED) CHARACTERISTICS

For carrying out traction calculation the information on the form of a curve of power and torque of the engine of the vehicle according to the graph of external speed characteristic is necessary.

The external speed characteristics of the engine is the dependence of the effective power N_e, torque M_k (Fig. 2.1) and other indicators of the engine from the crankshaft speed when the throttle valve fully open in the gasoline engine or at the maximum (set by the manufacturer) cycle feed fuel in a diesel engine.

If the actual external speed characteristic of the vehicle engine is absent, but there are two points of this characteristic:

$M_{\max}$, $n_{M\max}$ – maximum engine torque and crankshaft speed at maximum torque, respectively, and

$N_{\max}$, $n_{N\max}$ – maximum engine power and crankshaft speed corresponding to it, which are usually given in the technical characteristics,

then the curve of the effective power of the engine is usually described using the Leiderman formula, which is a polynomial of 3rd degree

$$N_e = N_{\max} \cdot \left[a \cdot \frac{n}{n_{N\max}} + b \cdot \left(\frac{n}{n_{N\max}} \right)^2 - c \cdot \left(\frac{n}{n_{N\max}} \right)^3 \right], \qquad (2.1)$$

where $N_{\max}$ – maximum vehicle engine power, W;

$n_{N\max}$ – crankshaft revolution corresponding to $N_{\max}$;

n – the current value of the engine crankshaft revolution in the range from $n_{\min}$ to $n_{\max}$;

a, b and c – approximation coefficients.

The current value of torque is determined by the formula

$$M_k = \frac{N_e}{\omega}, \tag{2.2}$$

where ω – current value of angular speed of rotation of a crankshaft of the vehicle engine, rad/s.

$$\omega = \frac{n \cdot \pi}{30}, \tag{2.3}$$

where n – the current value of the engine crankshaft revolution in the range from $n_{\min}$ to $n_{\max}$.

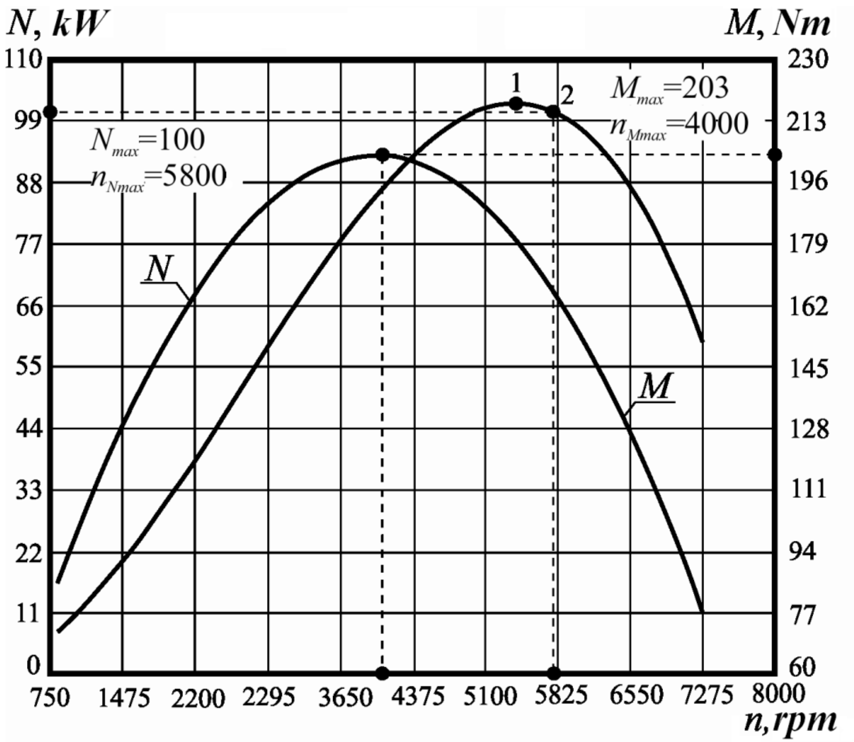


Figure 2.1 – External (speed) characteristics of the vehicle engine

2.1 Calculation of coefficients *a, b, c* which characterize the type of engine

The approximation coefficients characterize a specific vehicle engine and can be calculated by the following formulas

$$a = 1 - \frac{M_z}{100} \cdot \frac{K_\omega \cdot (2 - K_\omega)}{(K_\omega - 1)^2}; \tag{2.4}$$

$$b = 2 \cdot \frac{M_z}{100} \cdot \frac{K_\omega}{(K_\omega - 1)^2}; \tag{2.5}$$

$$c = \frac{M_z}{100} \cdot \left(\frac{K_\omega}{K_\omega - 1} \right)^2, \tag{2.6}$$

where M_z – the magnitude of the torque reserve;

K_ω – the coefficient of adaptability of the engine by angular speed of rotation.

The following formula is used to check the correctness of the calculation of the approximation coefficients

$$a + b - c = 1. \tag{2.7}$$

The amount of torque reserve is determined by the formula

$$M_z = (K_\mathrm{M} - 1) \cdot 100, \tag{2.8}$$

where K_M – the coefficient of adaptability of the engine by torque.

The coefficient of adaptability of the engine by torque is the ratio of the value of the maximum torque to the value of the torque at maximum power

$$K_\mathrm{M} = \frac{M_{k\max}}{M_{kN\max}}, \tag{2.9}$$

where $M_{k\max}$ – maximum torque of engine;

$M_{kN\max}$ – value of the torque at maximum power.

$$M_{kN\max} = \frac{30 \cdot N_{\max}}{\pi \cdot n_{N\max}}, \tag{2.10}$$

where $N_{\max}$ – maximum power of engine;

$n_{N\max}$ – value of the engine crankshaft speed at the maximum power.

The coefficient of adaptability of the engine by torque is the ratio of the value of the maximum torque to the value of the torque at maximum power

$$K_{\omega} = \frac{n_{N\max}}{n_{M\max}}, \tag{2.11}$$

where $n_{N\max}$ – value of the engine crankshaft revolution at the maximum power;

$n_{M\max}$ – value of the engine crankshaft revolution at the maximum torque.

The values of the coefficients K_{ω} and K_{M} determine the ability of the engine to automatically adapt to changes in load and range of steady engine operation. The dependence $M_k=f(n)$ has a maximum at the revolution $n_{M\max} < n_{N\max}$. If $n > n_{M\max}$, then the increase in load on the engine causes a decrease in crankshaft speed n, which leads to an increase in torque M_k, i.e. the engine automatically adapts to changes in load. This ability is estimated not only by K_{ω} and K_M, but also by the value of the torque reserve M_z in percentage.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of coefficients a, b, c

$\mathrm{Nmax} := 100000$ *(W) – maximum power of engine*

$\mathrm{nNmax} := 5800$ *(rpm) – engine crankshaft revolution at maximum power*

$\mathrm{Mmax} := 203$ *(N·m) – maximum torque of engine*

$\mathrm{nMmax} := 4000$ *(rpm) – engine crankshaft revolution at maximum torque*

$\mathrm{M\kappa Nmax} := \dfrac{30 \cdot \mathrm{Nmax}}{\pi \cdot \mathrm{nNmax}}$ *– engine torque at maximum power*

$\mathrm{M\kappa Nmax} = 164.643$ *(N·m)*

$\mathrm{K\text{м}} := \dfrac{\mathrm{Mmax}}{\mathrm{M\kappa Nmax}}$ *– the coefficient of adaptability of the engine by torque*

$\mathrm{K\text{м}} = 1.233$

$\mathrm{Kw} := \dfrac{\mathrm{nNmax}}{\mathrm{nMmax}}$ *– the coefficient of adaptability of the engine by angular speed of rotation*

$\mathrm{Kw} = 1.450$

$\mathrm{Mz} := (\mathrm{K\text{м}} - 1) \cdot 100$ *– the magnitude of the torque reserve*

$\mathrm{Mz} = 23.297$ (%)

$$a := 1 - \frac{\mathrm{Mz}}{100} \cdot \frac{\mathrm{Kw} \cdot (2 - \mathrm{Kw})}{(\mathrm{Kw} - 1)^2} \quad b := 2 \cdot \frac{\mathrm{Mz}}{100} \cdot \frac{\mathrm{Kw}}{(\mathrm{Kw} - 1)^2} \quad c := \frac{\mathrm{Mz}}{100} \cdot \left(\frac{\mathrm{Kw}}{\mathrm{Kw} - 1} \right)^2$$

$a = 0.082 \quad b = 3.336 \quad c = 2.419$

$a + b - c = 1.000$ *– check: the sum of the coefficients a, b and c must be equal to one*

2.2 The initial data for the construction of the external characteristics of the engine

The maximum engine power is selected according to the technical data sheet of the vehicle, or determined by the formula

$$N_{\max} = \frac{N_{V\max}}{\left[a \cdot \frac{n}{n_{N\max}} + b \cdot \left(\frac{n}{n_{N\max}}\right)^2 - c \cdot \left(\frac{n}{n_{N\max}}\right)^3\right]}, \tag{2.12}$$

where $N_{V\max}$ – the maximum engine power, which provides a given maximum speed $V_{\max}$ of the vehicle.

Engine power $N_{V\max}$ is determined from the condition of the possibility of moving the vehicle on the road with a given maximum speed $V_{\max}$ without a trailer in direct gear (i_g= 1) on a horizontal road with a hard surface. That is, two conditions must be met: the power must be sufficient to accelerate the vehicle to a given maximum speed, and in direct gear at angular speed of the engine crankshaft, which corresponds to the maximum torque, the dynamic factor D_a must be not less than given.

This power is determined based on the equation of power balance, which has the following form

$$N_{V\max} = \frac{G_a \cdot f \cdot V_{\max} + k_w \cdot F \cdot V_{\max}^3}{\eta_{tr}}, \tag{2.13}$$

where G_a – total vehicle weight, N;

f – coefficient of rolling resistance of vehicle wheels;

$V_{\max}$ – given maximum speed of the vehicle movement, m/s;

k_w – coefficient of air resistant, N·s^2/m^4;

F – the area of frontal resistance of the vehicle, м^2;

η_{tr} – efficiency of the vehicle transmission.

The total weight of the vehicle is determined by the formula

$$G_a = M_a \cdot g, \tag{2.14}$$

where g – acceleration of gravity, m/s^2.

The coefficient of rolling resistance of the wheels of the vehicle when driving on a horizontal road with a hard surface is determined by the formula

$$f = f_o \cdot \left(1 + A_f \cdot V_a^2\right), \tag{2.15}$$

where f_o – coefficient of rolling resistance of tires when driving a vehicle at low speed, the approximate value of which is recommended to take [2, p. 11, 11, p. 9]:

0,005…0,008 – for the tires with road tread pattern,

0,008…0,012 – for the tires with universal tread pattern,

0,008…0,014 – for off-road tires;

A_f – coefficient that takes into account the effect of speed on the rolling resistance of the vehicle wheels when driving on a road with asphalt-concrete pavement; the approximate value of which is recommended to take [1, 17, p. 11]:

(2,5…7,5)·10^{-4}s^2/m^2 – for cars,

(3,5…10,5)·10^{-4}s^2/m^2 – for trucks;

V_a – the current value of the vehicle speed, m/s.

The coefficient of air resistant is determined by the formula

$$k_w = 0{,}5 \cdot C_x \cdot \rho_в, \tag{2.16}$$

where C_x – coefficient of frontal drag, determined experimentally;

$\rho_в$ – air density under normal conditions, i.e. at temperature 20^{o}C and pressure 101325 Pa, is 1,189 kg/m^3.

The area of frontal resistance of the vehicle is determined by the formula

$$F = \beta \cdot B \cdot H, \qquad (2.17)$$

where β – the coefficient of filling the area, it is recommended to

take [19, p. 9]:

0,78…0,8 – for cars,

0,75…0,9 – for trucks;

B – overall width of the vehicle, m;

H – overall height of the vehicle, m.

During determining the efficiency, the transmission takes into account the hydraulic losses caused by shaking and spraying of oil in the crankcase and drive axle, and mechanical losses associated with friction between the teeth of the gears, in the bearing units and in the universal joints.

The efficiency of the transmission is determined as the product of the efficiency of all pairs of gears that are simultaneously in the catching and the cardans that transmit torque by the formula

$$\eta_{tr} = \eta_c^n \cdot \eta_{pb}^m \cdot \eta_{uj}^k, \qquad (2.18)$$

where $\eta_{ц}$ – the efficiency of a pair of cylindrical gears, approximately in the range of 0,97÷0,98;

η_{pb} – the efficiency of a pair of bevel gears, approximately in the range 0,96÷0,97;

η_{uj} – the efficiency of the universal joint, approximately assumed to be equal to 0,99;

n – the number of pairs of cylindrical gears;

m – the number of pairs of bevel gears;

k – the number of the universal joint, which transmit the torque.

In order to simplify the calculation of the efficiency of the transmission is taken constant.

It is recommended to take the approximate values of vehicles transmission efficiency [9]:

η_{tr} = 0,90...0,92 – for cars;

η_{tr} = 0,88...0,90 – for trucks of small loading capacity;

η_{tr} = 0,86...0,88 – for trucks of medium loading capacity;

η_{tr} = 0,84...0,86 – for trucks of big loading capacity.

Larger values of transmission efficiency refer to the direct gearin the gearbox of the vehicle.

In off-road vehicles, the transmission efficiency is reduced by 0.02, respectively.

In the presence of experimental data on the vehicle it is necessary to use them for a selection of η_{tr}.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

The initial data for the construction of the external characteristics of the engine

ORIGIN := 1

ki := 150 – *the number of points on which the revolution range is divided*

nmin := 800 *(rpm) – minimum steady no-load engine revolution*

nVmaxD := nNmax – *the crankshaft revolution of the diesel engine, which corresponds to the maximum speed of the vehicle*

nVmaxD = 5800 *(rpm)*

nVmaxB := nNmax · 1.25 – *crankshaft revolution of the gasoline engine, which corresponds to the maximum speed of the vehicle*

$nVmaxB = 7250$ *(rpm)*

$nVmax := nVmaxB$ – "***nVmaxD***" *corresponds to the diesel engine and gasoline one with a speed limiter, and*
"***nVmaxB***" *corresponds to a gasoline engine without a speed limiter*

$nmax := nVmax$ – *maximum engine revolution, which corresponds to the revolution at maximum speed of the vehicle*

$nmax = 7250$ *(rpm)*

$i := 1..ki \quad j := 1..ki \quad kj := \dfrac{nmax - nmin}{ki}$

$n_{i,j} := nmin + j \cdot k_j$

n =

	1	2
1	843.000	886.000
2	843.000	...

(rpm)

$\rho b := 1.189$ *(kg/m³)* – *air density*

$Cx := 0.33$ – *the coefficient of frontal drag*

$k := 0.5 \cdot Cx \cdot \rho b$ – *the coefficient of air resistant*

$k = 0.196$ $(N \cdot s^2/m^4)$

$\beta := 0.8$ *(m)* – *the coefficient of filling the area*

$B := 1.753$ *(m)* – *overall width of the vehicle*

$H1 := 1.458$ *(m)* – *overall height of the vehicle*

$F := \beta \cdot B \cdot H1$ – cross-sectional area (frontal resistance) of the vehicle

$F = 2.045$ (m^2)

$\eta_{tr} := 0.9$ – *transmission efficiency*

$Vmin := 0$ *(m/s)*

$Vmax := \dfrac{183}{3.6}$ – *maximum speed of the vehicle*

$Vmax = 50.833$ *(m/s)*

$$kj := \frac{Vmax - Vmin}{ki}$$

$$Va_{i,j} := Vmin + j \cdot kj$$

$$Va1 := Va^T$$

$f0 := 0.011$ – *coefficient of rolling resistance of the tire when the vehicle is moving at low speed*

$Af := 5.5 \cdot 10^{-4}$ *(s^2/m^2) – coefficient that takes into account the effect of speed on the rolling resistance of the vehicle wheels*

$$f1_{i,j} := f0 \cdot \left[1 + Af \cdot \left(Va1_{i,1}\right)^2\right]$$

$$f := max(f1)$$

$$f = 0.027$$

$g := 9.81$ *(m/s^2) – gravity coefficient*

2.3 Plotting of the external characteristics of the engine

Plotting of the external characteristics of the vehicle engine is carried out according to the previously given formulas (2.1) and (2.2). According to the results of calculations, the graph of the dependence of the effective power N_e and torque M_k on the revolution of the engine crankshaft n is built

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of the angular velocity of the engine crankshaft

$$w_{i,j} := \frac{n_{i,j} \cdot \pi}{30}$$

Calculation of engine power according to Leiderman's formula

$$N_{i,j} := Nmax \cdot \left[a \cdot \frac{n_{i,j}}{nNmax} + b \cdot \left(\frac{n_{i,j}}{nNmax} \right)^2 - c \cdot \left(\frac{n_{i,j}}{nNmax} \right)^3 \right]$$

Calculation of engine torque

$$M_{i,j} := \frac{N_{i,j}}{w_{i,j}}$$

Calculation results

$w_{i,j} =$

	1
1	88.279
2	88.279
3	...

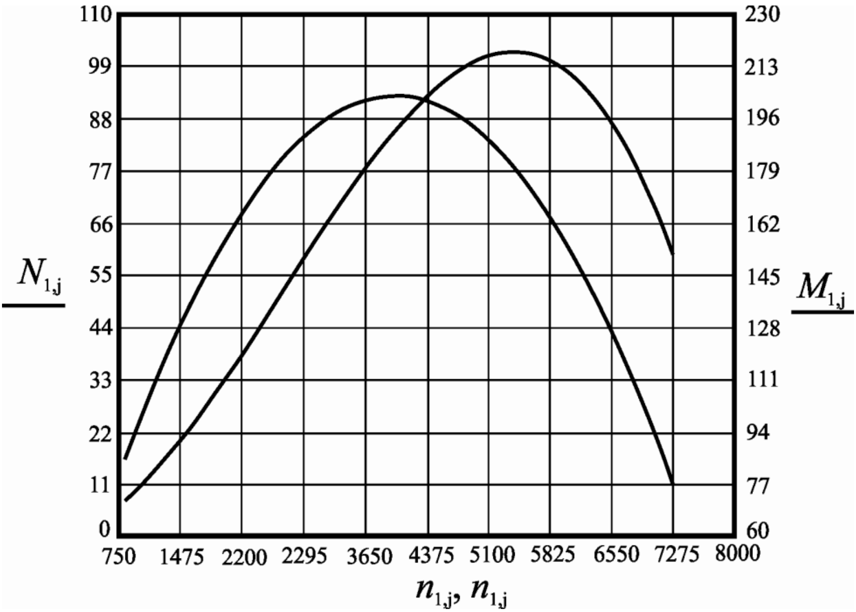
$N_{i,j} =$

	1
1	7504.503
2	7504.503
3	...

$M_{i,j} =$

	1
1	85.009
2	85.009
3	...

External (speed) characteristics of the vehicle engine



3 CALCULATION OF GEAR RATIOS OF POWER TRANSMISSION

In order to best adapt to different operating conditions, modern vehicles are available in different configurations: engines with different power, different gearboxes, different main gears.

When the vehicle is moving, the engine crankshaft develops up to 5000-6000 rpm, and the drive wheels rotate at a speed of not more than 1300 rpm.

So, even under favorable road conditions, the vehicle's wheels rotate four and a half times slower than the crankshaft. And under adverse road conditions, when the resistance of the vehicle increases and it is necessary to move at low speed, this ratio increases.

When operating the vehicle there is a need to change not only the speed and torque supplied to the drive wheels, but also to maneuver, stop, drive in reverse.

All these steps are made possible by the fact that the torque developed by the engine is fed to the drive wheels through the mechanisms that make up the transmission of the vehicle. And it is the transmission of the vehicle allows to vary widely by changing the torque transmitted from the engine to the drive wheels of the vehicle.

3.1 Calculation of the gear ratio of the main transmission

The calculation of the transmission of a road vehicle begins with the main transmission for the following reasons. In the transmission, the angular velocity can be changed in several places: in an additional box (trucks), gearbox, transfer box (off-road vehicles), main gear, wheel reducer (rarely used in cars). However, only in the last two units of modern vehicles, the gear ratio is always constant, while in gearboxes the gear ratio can change widely. But, as a rule, in all boxes there is a so-called "direct gear" at which inclusion angular speeds of input and output shafts coincide. At the same time intermediate shafts remain not involved that promotes reduction of losses in transmission and achievement of the maximum efficiency of the unit. When driving in direct gear, the vehicle becomes the most economical. On a direct gear the majority of vehicles develop the maximum speed (exceptions are, for example, BMW vehicles which reach the maximum speed on an accelerating transfer in gearbox [1]). Therefore, the total gear ratio of the constant gears (main gear) is determined from the ratio of the angular velocities of the engine and the drive wheels at maximum speed, assuming that the engine will run at maximum power.

$$i_0 = \frac{n_{V\max} \cdot \pi}{V_{\max} \cdot 30} \cdot r_k, \tag{3.1}$$

where $n_{V\max}$ – engine crankshaft revolution at maximum vehicle speed $V_{\max}$;

$V_{\max}$ – the set maximum speed of movement of the vehicle, m/s;

r_k – static wheel radius, m.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of the gear ratio of the main transmission

$$io := \frac{nmax \cdot \pi}{Vmax \cdot 30} \cdot rk$$

$io = 4.327$

$io := 3.95$ – *the gear ratio of the main transmission can be selected according to the technical data sheet characteristics of the vehicle*

3.2 Calculation of the gear ratio of the first gear of the gearbox

The gear ratio of the first gear i_1 is determined with the condition that all the traction force is spent on overcoming the specified maximum drag according to the formula

$$i_1 = \frac{M_a \cdot g \cdot \psi_{\max} \cdot r_k}{M_{k\max} \cdot i_0 \cdot \eta_{tr}}, \tag{3.2}$$

where M_a – total weight of the vehicle, kg;

g – gravity, m/s^2;

$\psi_{\max}$ – the maximum total coefficient of resistance of the vehicle is taken [19, Tabl. 1]:

0,35...0,40 – for single cars and trucks (for vehicles with front wheel drive not more than 0,35),

0,20...0,24 – for road trains,

0,22...0,25 – for city and intercity buses,

0,28...0,32 – for buses of general purpose,

0,60...0,70 – for off-road vehicles;

r_k – static wheel radius, m;

$M_{k\max}$ – maximum torque of the vehicle engine, N·m;

i_0 – gear ratio of the main transmission;

η_{tr} – transmission efficiency.

The obtained values of the gear ratio of the first gear should be checked in the absence of slippage by formulas:

– if all vehicle wheels are driving

$$i_{\varphi 1} = \frac{M_a \cdot g \cdot \varphi \cdot r_k}{M_{k\max} \cdot i_0 \cdot \eta_{tr}}, \tag{3.3}$$

where φ –the coefficient of adhesion of the wheels to the bearing surface.

– if only the front wheels of the vehicle are driving

$$i_{\varphi 1}=\frac{M_1\cdot g\cdot m_{g1}\cdot\varphi\cdot r_k}{M_{k\max}\cdot i_0\cdot\eta_{tr}},\tag{3.4}$$

where m_{g1} – the coefficient of dynamic redistribution of vertical reactions on the front drive axle.

$$m_{g1}=\frac{L\cdot\ \ L_2-\varphi\cdot h_g}{L_2\cdot\ \ L-\varphi\cdot h_g},\tag{3.5}$$

where L – longitudinal base of the vehicle, m;

L_2 – the distance from the center of mass of the vehicle to the rear axle, m;

h_g – height of the center of mass of the vehicle, m.

– if only the rear wheels of the vehicle are driving

$$i_1=\frac{M_2\cdot g\cdot m_{g2}\cdot\varphi\cdot r_k}{M_{k\max}\cdot i_0\cdot\eta_{tr}},\tag{3.6}$$

where m_{g2} – the coefficient of dynamic redistribution of vertical reactions on the rear drive axle.

$$m_{g2}=\frac{L}{L-\varphi\cdot h_g}.\tag{3.7}$$

To determine the distance from the center of mass of the vehicle to the front axle L_1 and the distance to the rear axle L_2it is necessary to calculate the partial weight distribution of the vehicle on the axes according to the formulas

$$n_1 = \frac{M_1}{M_a}, \tag{3.8}$$

$$n_2 = \frac{M_2}{M_a}. \tag{3.9}$$

Then the distances from the center of mass to the front and rear axles of the vehicle are determined by formulas

$$L_1 = L \cdot n_2, \tag{3.10}$$

$$L_2 = L \cdot n_1. \tag{3.11}$$

The obtained value of the gear ratio of the first gear$i_1$by formula (3.2) is compared with the value of$i_{\varphi 1}$calculated by formulas (3.3), (3.4) and (3.6) according to the engine scheme.

If

$$i_1 \le i_{\varphi 1}, \tag{3.12}$$

then this result allows to provide overcoming by the vehicle of the maximum resistance of the movement, which is set in the technical task, and as far as possible to move away from slipping conditions.

If the test result is negative, it is necessary to recalculate the value$i_1$byformula (3.2) changing ψ_{max}, or indicate on which roads the vehicle can be operated.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of the gear ratio of the first gear

$\psi max := 0.35$ – *the maximum total coefficient of resistance of the vehicle*

$$ik1 := \frac{Ma \cdot g \cdot \psi max \cdot rk}{Mmax \cdot io \cdot \eta_{tr}}$$

$ik1 = 2.550$

<u>*Determination of h, distances L_1, L_2 and coefficients m_1 and m_2*</u>

$\phi := 0.7$ – *the coefficient of adhesion of the wheels to the bearing surface*

$L := 2.806$ *(m) – longitudinal base of the vehicle*

$h := 0.39 \cdot H1$ – *height of the center of mass of the vehicle*

$h = 0.569$ *(m)*

$n1 := \frac{M1}{Ma}$ – *the proportion of mass that falls on the front axle of the vehicle*

$n1 = 0.516$

$n2 := \frac{M2}{Ma}$ – *the proportion of mass that falls on the rear axle of the vehicle*

$n2 = 0.484$

$L1 := L \cdot n2$ – *the distance from the center of mass of the vehicle to the front axle*

$L1 = 1.357$ *(m)*

$L2 := L \cdot n1$ – *the distance from the center of mass of the vehicle to the rear axle*

$L2 = 1.449$ *(m)*

$m1 := \frac{L \cdot (L2 - \phi \cdot h)}{L2 \cdot (L - \phi \cdot h)}$ – *the coefficient of dynamic redistribution of vertical reactions on the front drive axle*

$m1 = 0.845$

$m2 := \frac{L}{L - \phi \cdot h}$ – *the coefficient of dynamic redistribution of vertical reactions on the rear drive axle*

$m2 = 1.165$

$i11 := \frac{Ma \cdot g \cdot \phi \cdot rk}{Mmax \cdot io \cdot \eta_{tr}}$ – *if all vehicle wheels are driving*

$i11 = 5.100$

$$i11 := \frac{M1 \cdot g \cdot \phi \cdot m1 \cdot rk}{Mmax \cdot io \cdot \eta_{tr}}$$ – *if only the front wheels of the vehicle are driving*

$i11 = 2.225$

$$i11 := \frac{M2 \cdot g \cdot \phi \cdot m2 \cdot rk}{Mmax \cdot io \cdot \eta_{tr}}$$ – *if only the rear wheels of the vehicle are driving*

$i11 = 2.875$

<u>*Checking the first gear ratio for slippage conditions*</u>

$i11 := 2.225$ – *substitute the value of i11 to the corresponding engine formula*

$$s1 := \left| \begin{array}{l} s1 \leftarrow \text{"condition is met"} \quad \text{if } ik1 \le i11 \\ s1 \leftarrow \text{"condition is not met"} \quad \text{if } ik1 > i11 \\ s1 \end{array} \right.$$

$s1 =$ "condition is not met"

3.3 Calculation of gear ratios "2, 3, ..., n" gearbox transmission

The gear ratios of the second, third and fifth gears are determined by different progressions: the gear ratios of vehicles are most often carried out using a hyperbolic series, which provides the best dynamics of the vehicle at high speeds (higher gears, such as 3 and 4, are close)

$$i_k = \frac{i_1}{1 + (np - 1) \cdot \frac{i_1 - 1}{k_{pr} - 1}}, \qquad (3.13)$$

where np – the number of gear ratio, which is calculated;

k_{pr} – the number of direct gear.

For off-road vehicles, trucks and buses, for which traction characteristics are important in the whole range of speeds, the calculation of gearbox is usually carried out by geometric progression, which allows to switch gears without shocks, driving at low speeds in difficult road conditions and when maneuvering, overcoming given the largest upgrades.

$$i_k = \sqrt[k_{pr}-1]{i_1^{k_{pr}-np}}, \tag{3.14}$$

where np – the number of gear ratio, which is calculated;

k_{np} – the number of direct gear.

At intensive acceleration of the vehicle (the engine "spins" to turns of the maximum power) and if gearbox is calculated on a geometrical number at the next gear change the angular speed of the engine always falls to the same size.

If the gearbox is calculated on a hyperbolic series, then as the gear changes, the minimum angular speed of the engine increases. Such a vehicle will have an advantage during accelerating, especially at high speeds.

If the vehicle is more often operated on hard-surface roads, though it belongs to off-road vehicles, it is possible to calculate gearbox of such vehicle on both progressions depending on wishes of the customer.

Also, the value of acceleration (economic) transmission can be obtained by solving the cubic equation of power balance for optimal engine load in terms of power and angular velocity. To use Cardano's formula

$$k_2 = M_a \cdot g \cdot f_o \cdot A_f + k_w \cdot F, \tag{3.15}$$

$$k_1 = \frac{M_a \cdot g \cdot f_o}{k_2}, \tag{3.16}$$

$$k_0 = -\left(\frac{0,75 \cdot N_{\max} \cdot \eta_{tr}}{k_2}\right), \tag{3.17}$$

$$D_r = \left(\frac{k_1}{3}\right)^3 + \left(\frac{k_0}{2}\right)^2, \tag{3.18}$$

$$V_{ek} = \sqrt[3]{-\frac{k_0}{2} + \sqrt{D_r}} + \sqrt[3]{-\frac{k_0}{2} - \sqrt{D_r}}, \tag{3.19}$$

$$i_{ek} = \frac{0,75 \cdot \omega_{N\max} \cdot r_k}{V_{ek} \cdot i_0}, \qquad (3.20)$$

where $\omega_{N\max}$ – angular velocity of the engine crankshaft at maximum power, rad/s.

$$\omega_{N\max} = \frac{n_{N\max} \cdot \pi}{30}, \qquad (3.21)$$

where $n_{N\max}$ – engine crankshaft revolution at maximum power, rpm.

The reversing of vehicles is used for maneuvering and is not involved in the acceleration of the vehicle. The gear ratio is determined by the layout of the gearbox, usually by the formula

$$i_{zx} = (1,0 \ldots 1,3) \cdot i_1. \qquad (3.22)$$

<u>Program listing:</u>

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

ik1 = 2.550 *– the calculated value of the gear ratio of the first gear*

kpr := 4 *– the number of direct gear*

<u>*Calculation of the value of the gear ratios of the gearbox by hyperbolic progression*</u>

np := 2 *– second gear*

$$ik2 := \frac{ik1}{1 + (np - 1) \cdot \frac{ik1 - 1}{kpr - 1}}$$

ik2 = 1.681

$np := 3$ – *third gear*

$$ik3 := \frac{ik1}{1 + (np - 1) \cdot \frac{ik1 - 1}{kpr - 1}}$$

$ik3 = 1.254$

$np := 4$ – *forth gear*

$$ik4 := \frac{ik1}{1 + (np - 1) \cdot \frac{ik1 - 1}{kpr - 1}}$$

$ik4 = 1.000$

$np := 5$ – *fifth gear*

$$ik5 := \frac{ik1}{1 + (np - 1) \cdot \frac{ik1 - 1}{kpr - 1}}$$

$ik5 = 0.832$

Calculation of the value of the gear ratios of the gearbox by geometric progression

$np := 2$ – *second gear*

$$ik2 := \sqrt[kpr-1]{ik1^{kpr-np}}$$

$ik2 = 1.866$

$np := 3$ – *third gear*

$$ik3 := \sqrt[kpr-1]{ik1^{kpr-np}}$$

$ik3 = 1.366$

$np := 4$ – *forth gear*

$$ik4 := \sqrt[kpr-1]{ik1^{kpr-np}}$$

$ik4 = 1.000$

$np := 5$ – *fifth gear*

$$ik5 := \sqrt[kpr-1]{ik1^{kpr-np}}$$

$ik5 = 0.732$

Calculation of the gear ratio of the accelerator transmission

$$\omega Nmax := \frac{nNmax \cdot \pi}{30}$$ *– angular velocity of the engine crankshaft at maximum power*

$\omega Nmax = 607.375$ *(rad/s)*

$$k2 := Ma \cdot g \cdot f0 \cdot Af + k \cdot F$$

$$k1 := \frac{Ma \cdot g \cdot f0}{k2}$$

$$k0 := -\left(\frac{0.75 \cdot Nmax \cdot \eta_{tr}}{k2}\right)$$

$$Dr := \left(\frac{k1}{3}\right)^3 + \left(\frac{k0}{2}\right)^2$$

$$Vek := \sqrt[3]{-\left(\frac{k0}{2}\right) + \sqrt{Dr}} + \sqrt[3]{-\left(\frac{k0}{2}\right) - \sqrt{Dr}}$$

$Vek = 48.374$ *(m/s)*

$$iek := \frac{0.75 \cdot \omega Nmax \cdot rk}{Vek \cdot io}$$

$iek = 0.691$ *– the gear ratio of the accelerator transmission according to the Cardano formula*

Calculation of the gear ratio of gear of the reversing of vehicles

$izx := 1.15 \cdot ik1$

$izx = 4.117$

Also, the gear ratios of the gearbox can be selected according to the technical data sheet characteristics of the vehicle

$ik1 := 3.58$ $ik2 := 2.02$ $ik3 := 1.35$ $ik4 := 0.98$ $ik5 := 0.81$

3.4 Calculation of the gear ratio of the reduction of the transfer case

The calculation of the gear ratio of the reduced transmission of the distributor box i_{pk} can be done based on the requirements of the absence of wheel slipping. To do this, use the formula

$$i_{\text{рк}} = \frac{M_a \cdot g \cdot \varphi \cdot r_k}{M_{k\max} \cdot i_1 \cdot i_0 \cdot \eta_{tr}}. \tag{3.23}$$

As a rule, the reduction of the downshift is when overcoming difficult road conditions. The dynamics of acceleration of the vehicle to maximum speed is not considered. Therefore, for further calculation of the dynamic characteristics of the vehicle, the gear ratio of the distributor box can be taken equal to the gear ratio of the highest gear according to the technical data sheet characteristics of the vehicle (if this number is not known then its value is accepted equal to one).

If the transmission of the vehicle does not have a distributor box, then its gear ratio is taken equal to one.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of the gear ratio of the downshift of the distributor box

$$\text{irk} := \frac{\text{Ma} \cdot \text{g} \cdot \phi \cdot \text{rk}}{\text{Mmax} \cdot \text{ik1} \cdot \text{io} \cdot \eta_{\text{tr}}}$$

$\text{irk} = 1.425$

$\text{irk} := 1$ – *in the transmission of the vehicle there is no distributor box therefore its gear ratio is accepted equal to **one***

3.5 Speed character rustics of the vehicle

The gear ratios are one of the main parameters of the gearbox, which shows the acceleration-traction ratio/maximum speed of the vehicle in each gear. For example, if the gear ratio of the 1st gear of the gearbox is 4, and the 2nd gear is 2 (2 times the difference), it means that the maximum speed in the 2nd gear will be twice as high, and the acceleration-traction twice as weak as on the first at the same engine crankshaft revolution.

When the gear ratios are high, such a gearbox is called "short", i.e. the maximum speed in each gear is not so high, and it is necessary to switch more often (the gears are short). Such gear ratios provide, other things being equal, better acceleration and traction, so they are typical for supercars and sports cars with the most efficient acceleration dynamics or for SUVs (sports utility vehicles) with two-speed transfer cases for maximum traction in difficult road conditions at low speeds.

The situation with low gear ratios is reversed. Such boxes are called "long" and they provide greater maximum speed in each gear and better efficiency in exchange for less traction and acceleration. It is important that "long" gearboxes with low gear ratios are not synonymous with greater top speed of the vehicle as a whole. The final maximum speed of the vehicle equipped with such box, as a rule, is reached on lower gears (for example, on the 6th speed, and the following 7th and the 8th gears are necessary only for increase of economy and cannot increase the maximum speed reached on the 6th gear. Whereas in vehicles with "short" gearboxes, the final maximum speed is reached in higher gear, and there are no additional economic gears.

The gear ratio of the gearbox, designed to achieve the maximum speed of the vehicle, is selected so that when driving at maximum speed, the engine of the vehicle operates at maximum power. This is the only way to ensure maximum torque on the wheels to most effectively overcome the aerodynamic drag from the air flow. If the gear ratios of the gearbox are calculated geometrically, the acceleration in each gear begins at the same engine revolution n_1 and ends at the same revolution n_2.

Namely

$$\frac{i_1}{i_2} = \frac{i_2}{i_3} = \frac{i_3}{i_4} = \frac{i_4}{i_5} = \frac{n_{\max}}{n_{\min}}. \tag{3.24}$$

This makes it possible to use the same average engine power for acceleration in all gears. When designing the gearing of a real gearbox, its gear ratios are inevitably "deformed". At the same time, it is desirable that higher gears converge

$$\frac{i_1}{i_2} > \frac{i_2}{i_3} > \frac{i_3}{i_4} > \frac{i_4}{i_5}. \tag{3.25}$$

If the gear ratios of the gearbox are calculated in a hyperbolic series, then as the gear changes, the minimum angular speed of the engine increases. Such a vehicle will have advantages in acceleration, especially at high speeds. Plotting of the speed characteristic of the vehicle is the dependence of the speed on a particular gear on the number of revolutions of the engine crankshaft (Fig. 3.1). To do this, the following formula is used

$$V_{np} = \frac{n_{np} \cdot \pi}{30 \cdot i_{np} \cdot i_0 \cdot i_{rk}} \cdot r_k \tag{3.26}$$

where np – the gear number on which the speed of the vehicle is calculated.

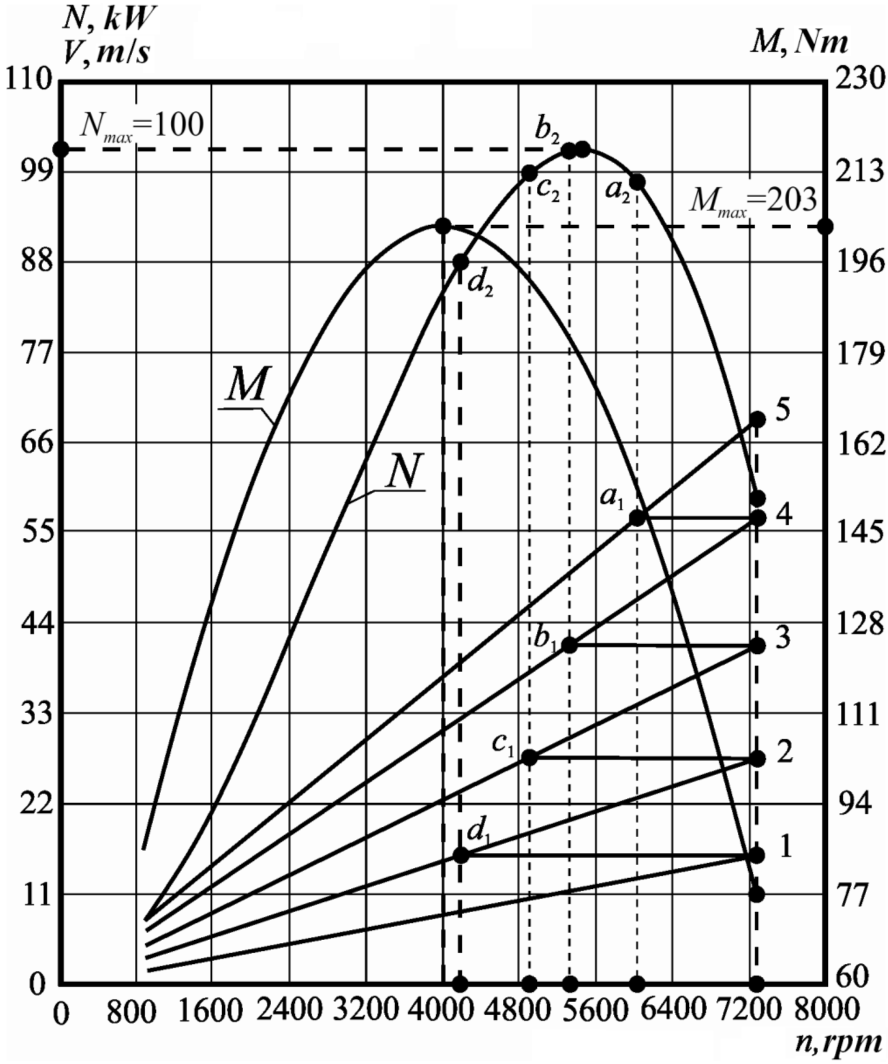


Figure 3.1 – Speed characteristics of the vehicle:
1 – first gear speed; 2 – second gear speed; 3 – third gear speed; 4 – forth gear speed; 5 – fifth gear speed; *N* – engine power; *M* − torque of the vehicle engine

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of a vehicle speed in 1st gear

$$V1_{i,j} := \frac{n_{i,j} \cdot \pi \cdot rk}{30ik1 \cdot io \cdot irk}$$

Calculation of a vehicle speed in 2nd gear

$$V2_{i,j} := \frac{n_{i,j} \cdot \pi \cdot rk}{30 \cdot ik2 \cdot io \cdot irk}$$

$$n2min := \frac{max(n)}{ik1} \cdot ik2$$

$n2min = 4090.782$ *(rpm)*

$$n12 := \begin{pmatrix} max(n) \\ n2min \end{pmatrix} \qquad V1max := \begin{pmatrix} max(V1) \\ max(V1) \end{pmatrix}$$

Calculation of a vehicle speed in 3rd gear

$$V3_{i,j} := \frac{n_{i,j} \cdot \pi \cdot rk}{30 \cdot ik3 \cdot io \cdot irk}$$

$$n3min := \frac{max(n)}{ik2} \cdot ik3$$

$n3min = 4845.297$ *(rpm)*

$$n23 := \begin{pmatrix} max(n) \\ n3min \end{pmatrix} \qquad V2max := \begin{pmatrix} max(V2) \\ max(V2) \end{pmatrix}$$

Calculation of a vehicle speed in 4th gear

$$V4_{i,j} := \frac{n_{i,j} \cdot \pi \cdot rk}{30 \cdot ik4 \cdot io \cdot irk}$$

$$n4min := \frac{max(n)}{ik3} \cdot ik4$$

$n4min = 5262.963$ *(rpm)*

$$n34 := \begin{pmatrix} \max(n) \\ n4min \end{pmatrix} \qquad V3max := \begin{pmatrix} \max(V3) \\ \max(V3) \end{pmatrix}$$

Calculation of a vehicle speed in 5th gear

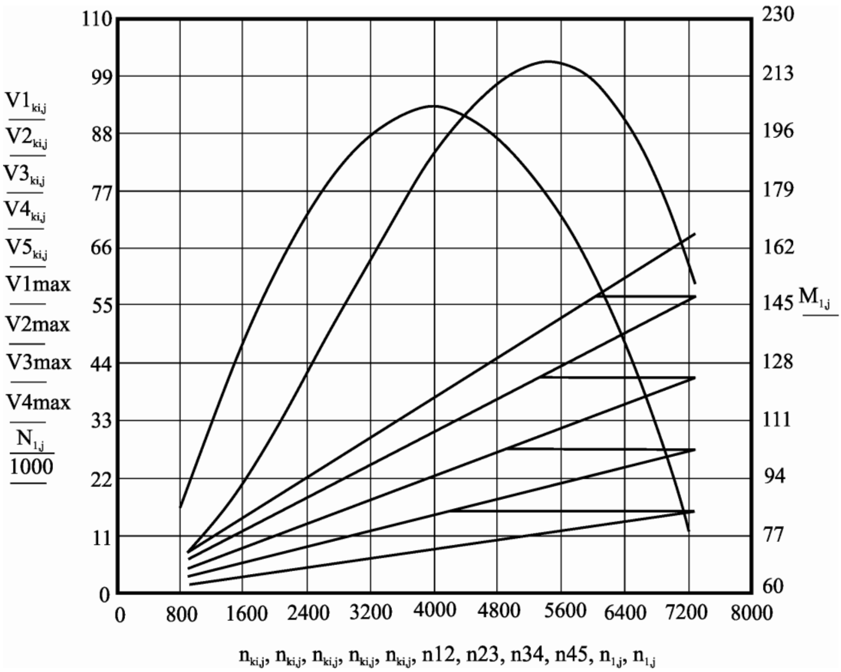
$$V5_{i,j} := \frac{n_{i,j} \cdot \pi \cdot rk}{30 \cdot ik5 \cdot io \cdot irk}$$

$$n5min := \frac{\max(n)}{ik4} \cdot ik5$$

$n5min = 5992.347$ *(rpm)*

$$n45 := \begin{pmatrix} \max(n) \\ n5min \end{pmatrix} \qquad V4max := \begin{pmatrix} \max(V4) \\ \max(V4) \end{pmatrix}$$

Speed characteristics of Opel Astra Classic



4 PLOTTING OF UNIVERSAL DYNAMIC CHARACTERISTICS OF THE VEHICLE

Productivity of the vehicle, is characterized by average speed of movement and depends on its traction and speed qualities. These qualities are determined by all longitudinal forces acting on the vehicle, the balance of which in the case of linear non-uniform movement allows to determine the traction force required to move the vehicle in a particular gear at a certain speed in a given road conditions.

4.1 Traction characteristics of the vehicle – power (traction) balance

For uniform or accelerated linear movement of the vehicle it is necessary that the total tangential reaction on all driving wheels was not less than the sum of all forces of external resistances to its translational movement.

$$P_k \geq P_\psi + P_j + P_w, \tag{4.1}$$

where P_k – total circumferential force on all driving wheels at uniform movement, is called as traction force of the vehicle, N;

P_ψ – the force of the total road resistance of vehicle movement, N;

P_j – total force of inertia of masses that move and rotate progressively, N;

P_w – the drag force, N.

The force of the total road resistance of vehicle movement is determine by the formula

$$P_\psi = P_f + P_\alpha, \tag{4.2}$$

where P_f – total rolling resistance force of the vehicle wheels, N;

P_a – the component of the weight force of the vehicle that is parallel to the bearing surface or the force of resistance of the vehicle to rise, N.

The traction force, which is supplied to the driving wheels of the vehicle when it is moving in any gear, is generally determined by the formula

$$P_{knp} = M_k \cdot i_{np} \cdot \frac{i_0 \cdot i_{rk} \cdot \eta_{tr}}{r_k}. \tag{4.3}$$

Total rolling resistance force of the vehicle wheels is determined as product of the gravity force of the vehicle and the coefficient of rolling resistance

$$P_f = G_a \cdot f. \tag{4.4}$$

When moving on the rise there is a component of weight force, which is directed towards the movement of the vehicle

$$P_\alpha = G_a \cdot \sin\alpha, \tag{4.5}$$

where α – angle of rise.

When changing the speed of the vehicle (during transients) there are forces of inertia, the value of which depends on the magnitude of the moving masses and accelerations. When the vehicle accelerates, the forces of inertia resist the movement; when the movement slows down, the forces of inertia prevent the deceleration. The force of inertia of the vehicle masses, which are progressively moving and rotating, is determined by the formula

$$P_j = M_a \cdot \delta_{np}, \tag{4.6}$$

where δ_{np} – the rotational inertia coefficient. It shows how many times the force required to accelerate with a given acceleration of the translational masses and the rotational masses of the vehicle, more than the force required to accelerate only its translational masses.

The rotational inertia coefficient

$$\delta_{np} = 1 + \delta_k + \delta_{tr} \cdot i_{np}^2, \tag{4.7}$$

where δ_k – the rotation al inertia coefficient of vehicle wheels, approximately accepted equal0,03÷0,05[6];

δ_{tr} – the rotational inertia coefficient of transmission and engine units, approximately accepted equal0,04÷0,06 [6].

Since the vehicle moves in an air environment that has a certain density, there are drag forces of the vehicle. To determine the drag force use expressions from aerodynamics

$$P_w = k_w \cdot F \cdot V_{np}^{\;2}, \tag{4.8}$$

where k_w – air drag coefficient, N·s^2/m^4;

F – the area of frontal drag of vehicle, m^2;

V_{np} – the current value of the vehicle speed at a particular gear, m/s.

At uniform movement on a horizontal surface forces P_a and P_j are equal to zero. In the case of an inclination and deceleration they become negative.

Taking into account the above, formula (4.1) takes the form

$$P_k \geq P_f + P_w. \tag{4.9}$$

This equation of movement is called the traction balance of the vehicle. It is used in the design of new and to assess the traction and speed characteristics of existing vehicle models. Traction and speed characteristics of the vehicle with the help of equations of its movement is most conveniently determined by graph analysis. To do this, the traction balance of the vehicle should be presented in the form of a graph (Fig. 4.1).On this graph in the coordinate system $P - V_a$ the traction-speed characteristic and resistance forces of the movement P_f , P_w and total resistance force of the vehicle movement $P_f + P_w$ are put.

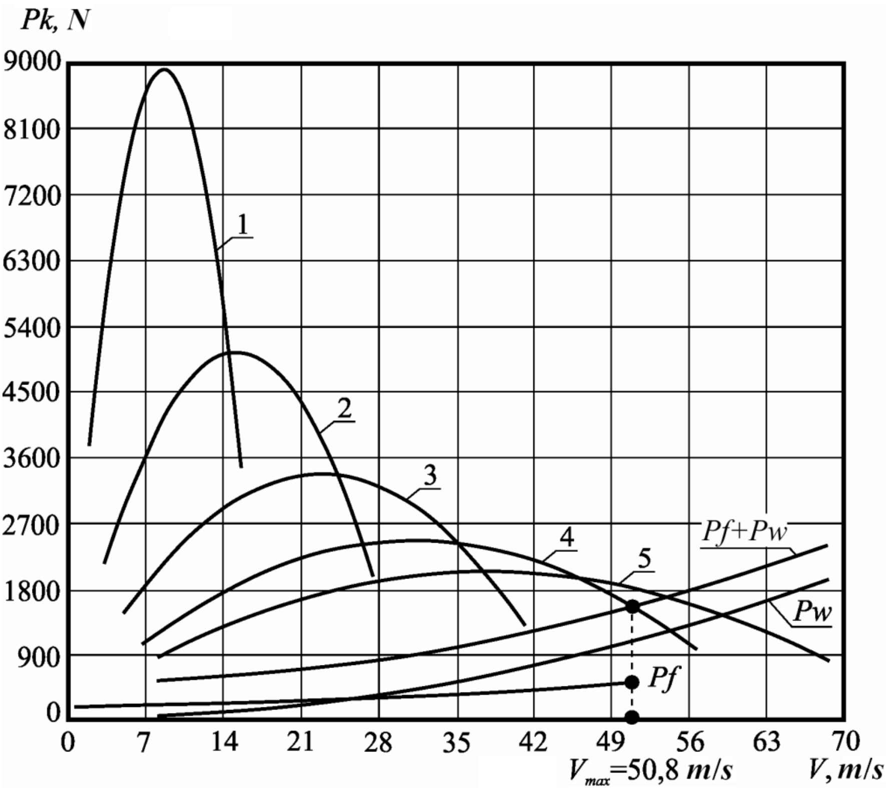


Figure 4.1 – Vehicle power (traction) balance:

1 – tangential traction force when driving in 1st gear; 2 – tangential traction force when driving in 2nd gear; 3 – tangential traction force when driving in 3rd gear; 4 – tangential traction force when driving in 4th gear; 5 – tangential traction force when driving in 5th gear; P_f – total rolling resistance force of the vehicle wheels; P_w – the drag force; $P_f + P_w$ – total resistance force of the vehicle movement

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Data

$Mamin := Mo + 2 \cdot 80 + 20$ – *weight of the equipped vehicle with the driver (80 kg), the passenger (80 kg) and luggage (20 kg).*

$Mamin = 1500$ *(kg)*

$Mamax := Ma$ – *total vehicle mass*

$$kj := \frac{Mamax - Mamin}{ki}$$

$$Ma_{i,j} := Mamin + j \cdot kj$$

$$Ma1 := Ma^{T}$$

Traction force when driving in 1st gear and drag force

$$Pk1_{i,j} := M_{i,j} \cdot ik1 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw1_{i,j} := k \cdot F \cdot \left| V1_{i,j} \right|^{2}$$

Traction force when driving in 2nd gear and drag force

$$Pk2_{i,j} := M_{i,j} \cdot ik2 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw2_{i,j} := k \cdot F \cdot \left| V2_{i,j} \right|^{2}$$

Traction force when driving in 3rd gear and drag force

$$Pk3_{i,j} := M_{i,j} \cdot ik3 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw3_{i,j} := k \cdot F \cdot \left| V3_{i,j} \right|^{2}$$

Traction force when driving in 4th gear and drag force

$$Pk4_{i,j} := M_{i,j} \cdot ik4 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw4_{i,j} := k \cdot F \cdot \left| V4_{i,j} \right|^2$$

Traction force when driving in 5th gear and drag force

$$Pk5_{i,j} := M_{i,j} \cdot ik5 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw5_{i,j} := k \cdot F \cdot \left| V5_{i,j} \right|^2$$

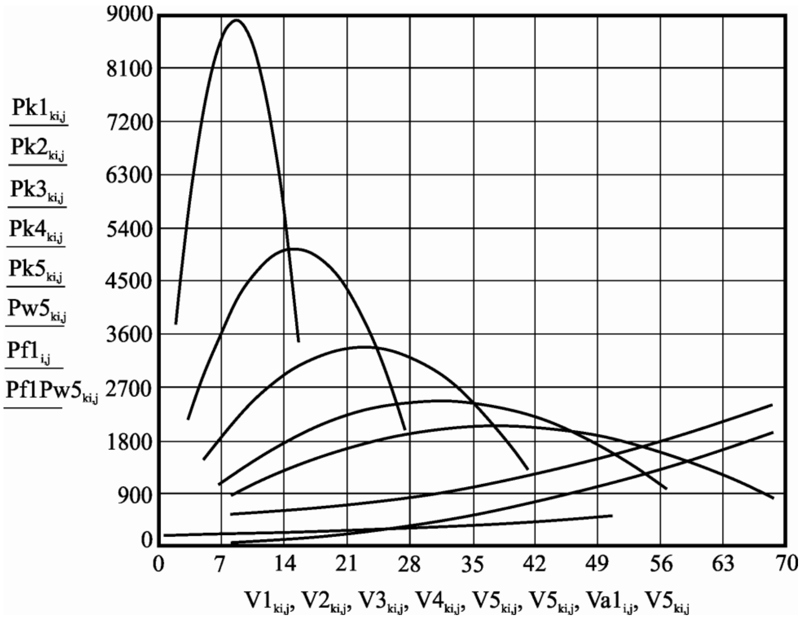
Total rolling resistance force of the vehicle wheels

$$Pf1_{i,j} := Ma1_{i,j} \cdot g \cdot f1_{i,j}$$

Total resistance force of the vehicle movement

$$Pf1Pw5_{i,j} := Pf1_{i,j} + Pw5_{i,j}$$

Vehicle power (traction) balance of Opel Astra Classic



4.2 Characteristics of vehicle power – power balance

To analyze the dynamic characteristics of the vehicle instead of the balance of forces, it is possible to use the balance of power supplied to the drive wheels, with the power needed to overcome the total drag.

By analogy with the equation of traction (power) balance (4.1), the equation of power balance can be written as

$$N_k \geq N_f + N_\alpha + N_j + N_w, \tag{4.10}$$

where N_k – power supplied to the driving wheels of the vehicle, W;

N_f – power consumed to overcome the total rolling resistance of the vehicle wheels, W;

N_a –power consumed to overcome the resistance of the vehicle to rise, W;

N_j – power consumed to accelerate the vehicle, W;

N_w – power consumed to overcome the drag force, W.

The power supplied to the driving wheels of the vehicle is determined by the formula

$$N_k = P_k \cdot V_{np}. \tag{4.11}$$

The power consumed to overcome the total rolling resistance of the vehicle wheels is determined by the formula

$$N_f = P_f \cdot V_{np}. \tag{4.12}$$

The power consumed to overcome the resistance of the vehicle to rise is determined by the formula

$$N_\alpha = P_\alpha \cdot V_{np}. \tag{4.13}$$

The power consumed to accelerate the vehicle is determined by the formula

$$N_j = P_j \cdot V_{np}. \tag{4.14}$$

The power consumed to overcome the drag force is determined by the formula

$$N_w = P_w \cdot V_{np}. \tag{4.15}$$

The graph of power balance is plotted for the given vehicle in coordinates $N - V_a$ (Fig. 4.2). On the basis of its analysis the same tasks, as at use of the graph of power balance are solved.

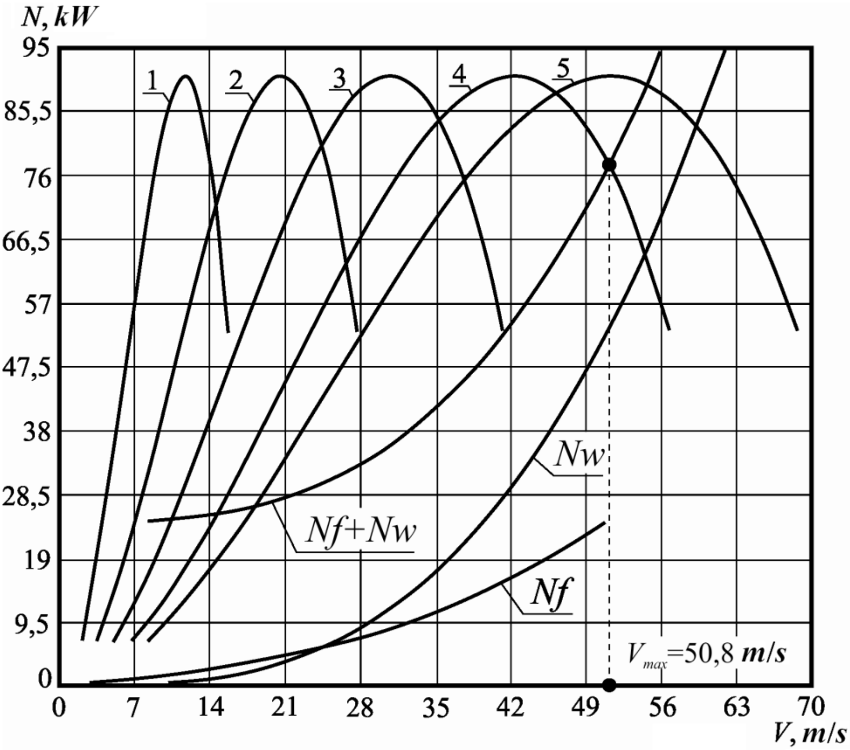


Figure 4.2 – The power balance of the vehicle:
1 – power supplied to the wheels when the vehicle is in 1st gear; 2 – power supplied to the wheels when the vehicle is in 2nd gear; 3 – power supplied to the wheels when the vehicle is in 3rd gear; 4 – power supplied to the wheels when the vehicle is in 4th gear; 5 – power supplied to the wheels when the vehicle is in 5th gear

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

The values of N_k and N_w when the vehicle is in 1st gear

$$Nk1_{i,j} := Pk1_{i,j} \cdot V1_{i,j}$$

$$Nw1_{i,j} := Pw1_{i,j} \cdot V1_{i,j}$$

The values of N_k and N_w when the vehicle is in 2nd gear

$$Nk2_{i,j} := Pk2_{i,j} \cdot V2_{i,j}$$

$$Nw2_{i,j} := Pw2_{i,j} \cdot V2_{i,j}$$

The values of N_k and N_w when the vehicle is in 3rd gear

$$Nk3_{i,j} := Pk3_{i,j} \cdot V3_{i,j}$$

$$Nw3_{i,j} := Pw3_{i,j} \cdot V3_{i,j}$$

The values of N_k and N_w when the vehicle is in 4th gear

$$Nk4_{i,j} := Pk4_{i,j} \cdot V4_{i,j}$$

$$Nw4_{i,j} := Pw4_{i,j} \cdot V4_{i,j}$$

The values of N_k and N_w when the vehicle is in 5th gear

$$Nk5_{i,j} := Pk5_{i,j} \cdot V5_{i,j}$$

$$Nw5_{i,j} := Pw5_{i,j} \cdot V5_{i,j}$$

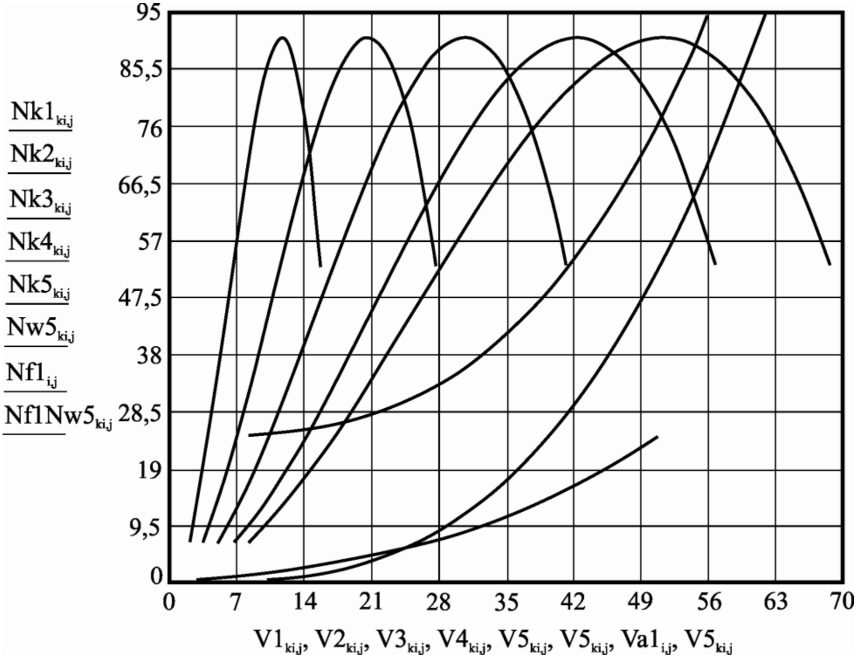
The power consumed to overcome the rolling resistance

$$Nf1_{i,j} := Pf1_{i,j} \cdot Va1_{i,j}$$

The value of the sum of the power consumed by the engine to overcome the forces of rolling resistance and drag

$$Nf1Nw5_{i,j} := Nf1_{i,j} + Nw5_{i,j}$$

The power balance of Opel Astra Classic



4.3 Dynamic characteristics of the vehicle – a dynamic factor

The methods of traction balance and power balance are difficult to apply when comparing the traction and dynamic characteristics of vehicles with different equipped masses and load capacity, because when moving them under the same conditions, the force and power required to overcome total road resistance are different. The method of solving the equation of movement by means of a dynamic characteristic was proposed by Academician E. A. Chudakov [16].

The dynamic characteristic allows to solve a number of problems of vehicle movement taking into account design and operational parameters which allow to estimate efficiency of vehicle use.

For this purpose, use a dimensionless factor D_a (dynamic traction factor), which is equal to the ratio of free traction $P_k - P_w$ to the weight of the vehicle G_a

$$D_a = \frac{P_k - P_w}{G_a} = \frac{P_{ft}}{G_a}. \qquad (4.16)$$

where P_{tr} – free traction that is independent of road conditions and accelerations and is a function of vehicle speed V_a.

Thus, expression (4.16) is an equation of force balance in dimensionless form. This ratio eliminates the influence of vehicle weight on the assessment of its properties. The values of the dynamic traction factor allow to estimate the traction and speed characteristics of a particular vehicle at different loads and to compare the traction and speed characteristics of different vehicles regardless of their weight. Thus the more dynamic factor on traction, the better traction-speed characteristics and higher passability of the vehicle. A vehicle is capable to develop bigger accelerations, to overcome steeper rises and to tow trailers of bigger weight.

In uniform movement, when the acceleration of the vehicle is zero, the dynamic factor must be numerically equal in magnitude to the coefficient of total road resistance, i.e. $D_a = \psi$. On the horizontal section, the value of the coefficient of resistance of the road can be equal to the coefficient of rolling resistance $\psi = f$. In all other cases, the coefficient ψ includes other parameters that affect the resistance of the vehicle.

It should also be noted that in low gears the dynamic factor is greater than in higher. This is due to an increase in the force P_k and a decrease in the force P_w.

Graphic representation of the dependence of the dynamic factor on the vehicle speed in different gears in the gearbox and at full load on the vehicle is called <u>the dynamic characteristic of the vehicle</u>, i.e. $D_a = \psi(V_a)$ (Fig. 4.3).

This characteristic can be plotted for the given vehicle and the given movement conditions.

If the dependence $f = f(V_a)$ is applied to the dynamic characteristic, then with the help of the obtained graph it is possible to solve the same problems as with the help of the graph of power balance or traction balance.

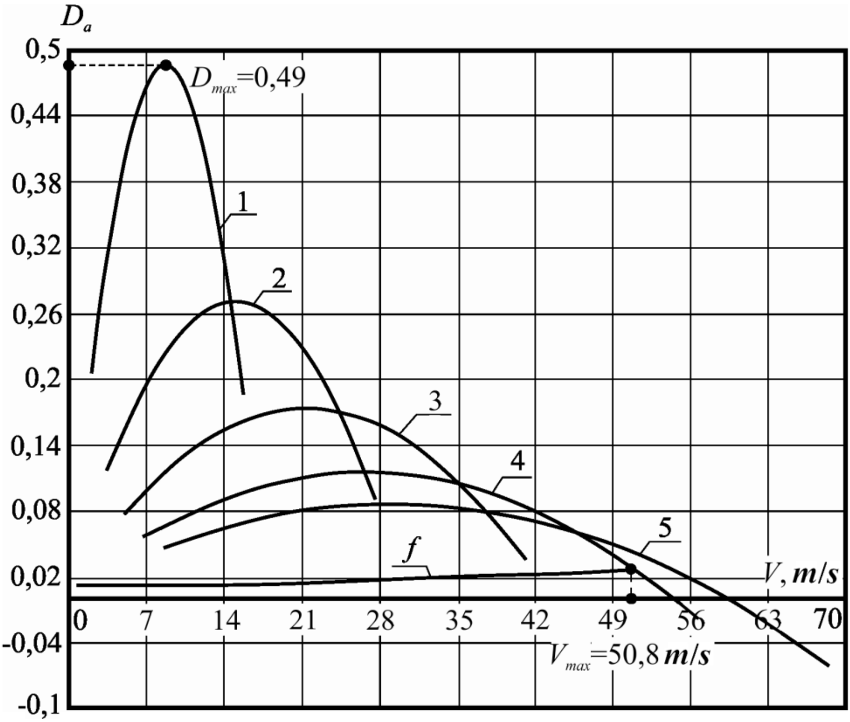


Figure 4.3 – The graph of the dependence of the dynamic factor on the speed of the vehicle:

1 – the dynamic factor when moving the vehicle in 1st gear;
2 – the dynamic factor when moving the vehicle in 2nd gear;
3 – the dynamic factor when moving the vehicle in 3rd gear;
4 – the dynamic factor when moving the vehicle in 4th gear;
5 – the dynamic factor when moving the vehicle in 5th gear;
f – the coefficient of rolling resistance of vehicle wheels

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

The dynamic factor when moving the vehicle in 1st gear

$$D1_{i,j} := \frac{\left| Pk1_{i,j} - Pw1_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

The dynamic factor when moving the vehicle in 2nd gear

$$D2_{i,j} := \frac{\left| Pk2_{i,j} - Pw2_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

The dynamic factor when moving the vehicle in 3rd gear

$$D3_{i,j} := \frac{\left| Pk3_{i,j} - Pw3_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

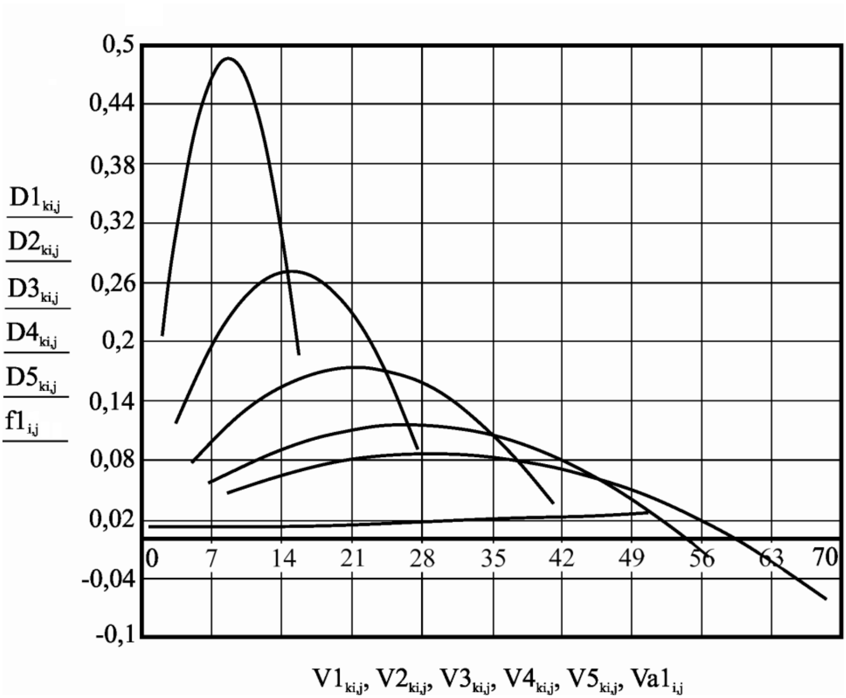
The dynamic factor when moving the vehicle in 4th gear

$$D4_{i,j} := \frac{\left| Pk4_{i,j} - Pw4_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

The dynamic factor when moving the vehicle in 5th gear

$$D5_{i,j} := \frac{\left| Pk5_{i,j} - Pw5_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

The graph of the dependence of the dynamic factor on the vehicle speed of Opel Astra Classic



4.4 Dynamic passport of vehicle

The dynamic characteristics of the vehicle greatly depends on its weight. Part of the time during operation, the vehicle is not fully loaded, but only partially, so of particular interest is the impact of vehicle load on its dynamic characteristics. Thus the universal dynamic characteristic allows to solve a number of problems of movement of the vehicle, including at change of its weight and road conditions.

In order not to determine the dynamic characteristic every time, the **load nomogram** (Fig. 4.4) complements it. For this purpose the

X-axis continues to the left and on it the scale of payload in units or in percent is put

$$H = \frac{G_{x\,ch} \cdot 100}{G_{a\,ch}}, \quad (4.17)$$

where $G_{x\,ch}$ – changed payload of the vehicle;

$G_{a\,ch}$ – maximum payload of the vehicle.

$$G_{a\,ch} = G_a - G_o, \quad (4.18)$$

where G_a – total vehicle weight, N;

G_o – the weight of the equipped vehicle, N;

The number of passengers may be specified for passenger vehicles. A vertical line parallel to the Y-axis is drawn through the beginning of the load scale, and the values of the dynamic factor of the vehicle without load (for example, a vehicle in the equipped state) are plotted on it.

The dynamic factor increases with decreasing weight. The new value of the dynamic factor D_x of the vehicle, the weight of which is changed, can be determined by the formula

$$D_x = \frac{D_a \cdot G_a}{G_x}, \quad (4.19)$$

where D_a – the dynamic factor of a fully loaded vehicle;

G_a – total vehicle weight;

G_x – changed vehicle weight.

In this form, the graph of the universal dynamic characteristics allows to solve the following problems of vehicle dynamics:

1. To determine the maximum value of road resistance ψ that can overcome a vehicle with a given load H as a percentage of the total payload $G_{a\,ch}$ at a steady movement at a given speed V in top gear.

2. To determine the speed *V* that the vehicle can develop if the load *H* and the total road resistance are known.

3. To determine the load *H* that can carry the vehicle when driving at a given speed *V* on the road with a known resistance ψ.

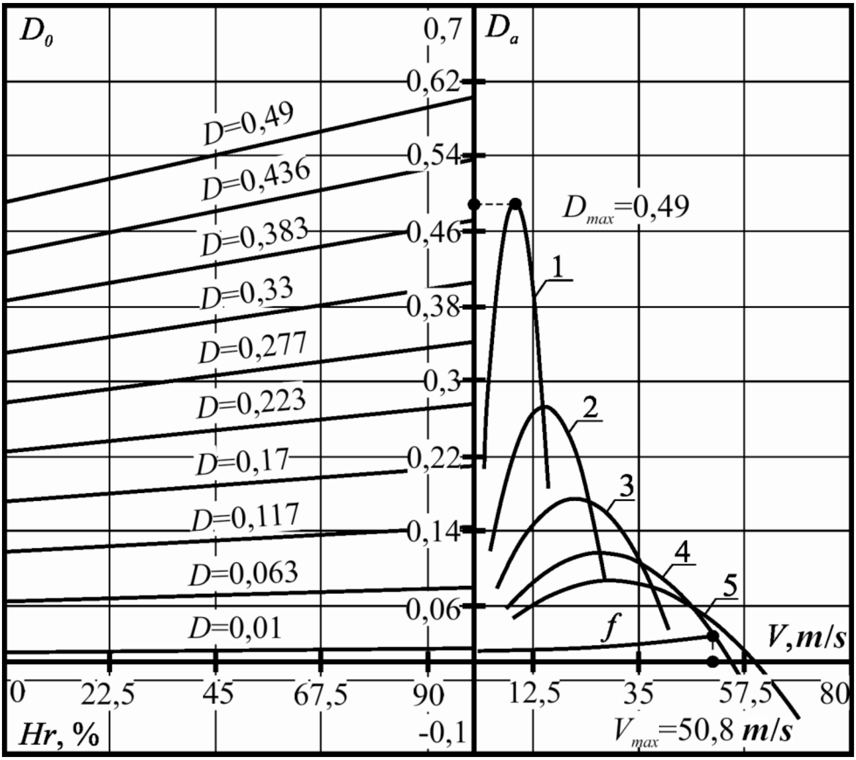


Figure 4.4 – Dynamic vehicle passport:

1 – the dynamic factor when moving the vehicle in 1st gear;

2 – the dynamic factor when moving the vehicle in 2nd gear;

3 – the dynamic factor when moving the vehicle in 3rd gear;

4 – the dynamic factor when moving the vehicle in 4th gear;

5 – the dynamic factor when moving the vehicle in 5th gear

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

The graph of dynamic factor with load nomogram

$Damin := 0.01$ – *the minimum value of D on the load nomogram*

$$Damax := \left| \begin{array}{l} Damax \leftarrow 0 \\ \text{for } i \in ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad Damax \leftarrow D1_{ki,j} \ \text{ if } \ D1_{ki,j} \geq Damax \\ Damax \end{array} \right.$$

$Damax = 0.490$ – *the maximum calculated value of D in first gear for a fully loaded vehicle.*

$$Dx := \frac{Damax - Damin}{9}$$

$$Dm1 := Damin \qquad Dm2 := Damin + Dx$$

$$Da1 := \begin{pmatrix} Dm1 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm1 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix} \qquad Da2 := \begin{pmatrix} Dm2 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm2 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix}$$

$$Dm3 := Damin + 2 \cdot Dx \qquad Dm4 := Damin + 3 \cdot Dx$$

$$Da3 := \begin{pmatrix} Dm3 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm3 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix} \qquad Da4 := \begin{pmatrix} Dm4 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm4 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix}$$

$$Dm5 := Damin + 4 \cdot Dx \qquad Dm6 := Damin + 5 \cdot Dx$$

$$Da5 := \begin{pmatrix} Dm5 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm5 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix} \qquad Da6 := \begin{pmatrix} Dm6 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm6 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix}$$

$$Dm7 := Damin + 6 \cdot Dx \qquad Dm8 := Damin + 7 \cdot Dx$$

$$Da7 := \begin{pmatrix} Dm7 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm7 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix} \qquad Da8 := \begin{pmatrix} Dm8 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm8 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix}$$

$$Dm9 := Damin + 8 \cdot Dx \qquad Dm10 := Damax$$

$$Da9 := \begin{pmatrix} Dm9 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm9 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix} \qquad Da10 := \begin{pmatrix} Dm10 \cdot \dfrac{\max(Ma1)}{\min(Ma1)} \\ Dm10 \cdot \dfrac{\max(Ma1)}{\max(Ma1)} \end{pmatrix}$$

$$Hr1 := \begin{pmatrix} 0 \\ -100 \end{pmatrix}$$

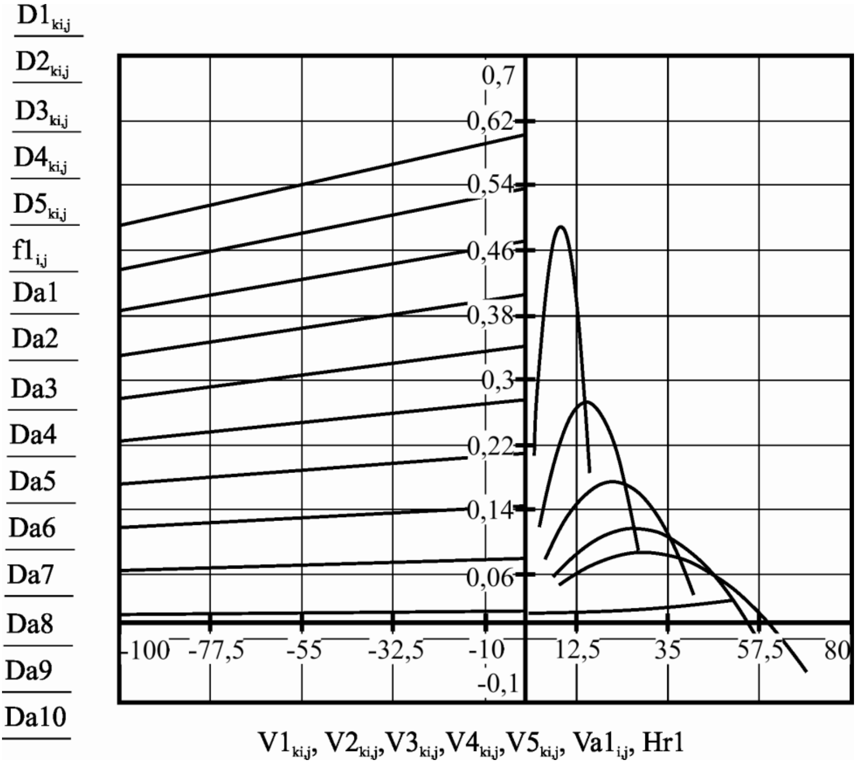
Each line on the load nomogram (left part of the graph) corresponds to a certain fixed value of the dynamic factor Da, respectively:

$Dm1 = 0.010 \quad Dm2 = 0.063 \quad Dm3 = 0.117 \quad Dm4 = 0.170 \quad Dm5 = 0.223$

$Dm6 = 0.277 \quad Dm7 = 0.330 \quad Dm8 = 0.383 \quad Dm9 = 0.436 \quad Dm10 = 0.490$

On this graph, the load scale Hr1 (left part of the graph) must be reversed, i.e. -100 = 0%, and 0 = 100%, this is due to the limited capabilities of MathCad guring plotting graphs.

The dynamic passport Opel Astra Classic



4.5 Calculation of values of limit accelerations of the vehicle

The time of uniform movement of the vehicle is usually small in comparison with the total time of its operation. At operation in the cities vehicles move evenly only 15-20 % of time, 40-45 % − accelerated and 30-40 % − decelerated. Acceleration is often used when driving outside the city on highways and even off-road.

An indicator of the dynamic characteristics of the vehicle during acceleration is the intensity of acceleration or capacity of the vehicle.

Acceptability (acceleration intensity) of the vehicle characterizes its ability to move quickly from a place and to increase speed of movement at acceleration. This property of the vehicle is especially important in urban traffic with frequent stops and starts, and also characterizes the speed of overtaking in suburban traffic. The acceleration intensity of the vehicle is measured by the magnitude of its acceleration.

The acceleration of the vehicle is determined experimentally or calculated according to a horizontal road with a good quality hard surface, provided that the maximum use of engine power (throttle is fully open) and the absence of wheel slippage.

The movement of the vehicle from a place is short-lived and is defined mainly by individual features of the driver. Therefore, it is believed that acceleration begins with the minimum speed $V_{\min}$ (Fig. 4.5) on the gear, from which the vehicle starts moving.

The minimum speed value $V_{\min}$ corresponds to the minimum stable revolution of the engine crankshaft $n_{\min}$. In the speed range 0 - $V_{\min}$ the vehicle starts from a place at slippage of connection and gradual increase in fuel supply.

To determine the acceleration of the vehicle, we use the equation of force balance (4.1) and convert it to a dimensionless form.

To group in the left part of this equation the forces which depend on speed of movement of the vehicle and do not depend on its weight. In the right part of the equation to leave the forces that depend on the weight of the vehicle and the condition of the road. Then, dividing both parts of the equation by the weight of the vehicle, to obtain

$$\frac{P_k - P_w}{G_a} = \frac{P_\psi + P_j}{G_a}. \tag{4.20}$$

Substituting in this equation the calculation formulas for the forces included in it and making the appropriate transformations, to obtain

$$D_a = \psi + \frac{J}{g} \cdot \delta_{np}. \tag{4.21}$$

Solving equation (4.21) with respect to the acceleration *J*, to obtain

$$J = \; D_a - \psi \; \cdot \frac{g}{\delta_{np}}. \tag{4.22}$$

Since the acceleration is proportional to the difference between the dynamic factor of the vehicle and the coefficient of road resistant ψ, to determine the acceleration as a function of the speed of the vehicle, i.e. $J = f(V)$, it is necessary to use the dynamic characteristics of the vehicle. Based on the results of the calculation, a graph of vehicle accelerations on gears is built depending on the speed of movement (Fig. 4.5).

In vehicles at maximum speed $V_{\max}$, the acceleration is usually zero, because there is no power reserve. Trucks with $V_{\max}$ have a power reserve, but it is not used for acceleration, because the engine speed limiter is triggered. For trucks and buses, the maximum acceleration $J_{\max}$ in 1st gear may be lower than in 2nd or approximately the same. This is due to the large value of the gear ratio of the transmission in these gears, resulting the coefficient of taking into account the rotational mass of the vehicle δ_{np} increases rapidly.

Acceleration curves depending on the speed allow not only to estimate the acceleration intensity and select the optimal modes of gear shifting, but also to determine the time and path of acceleration of the vehicle.

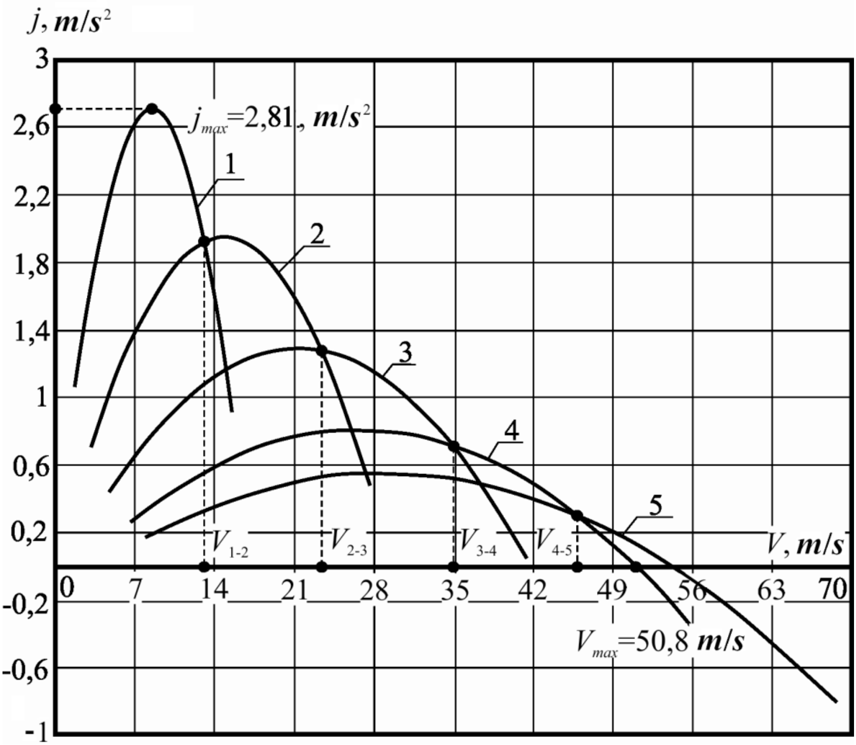


Figure 4.5 – The graph of the dependence of the limiting accelerations on the vehicle speed:

1 – acceleration of the vehicle when moving the vehicle in 1st gear;
2 – acceleration of the vehicle when moving the vehicle in 2nd gear;
3 – acceleration of the vehicle when moving the vehicle in 3rd gear;
4 – acceleration of the vehicle when moving the vehicle in 4th gear;
5 – acceleration of the vehicle when moving the vehicle in 5th gear

<u>Program listing:</u>

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Calculation of the coefficient of the given masses

$\sigma 1 := 0.03$

$\sigma 2 := 0.05$

$\delta 1 := 1 + \sigma 1 + \sigma 2 \cdot ik1^2$ $\delta 2 := 1 + \sigma 1 + \sigma 2 \cdot ik2^2$ $\delta 3 := 1 + \sigma 1 + \sigma 2 \cdot ik3^2$

$\delta 1 = 1.671$ $\delta 2 = 1.234$ $\delta 3 = 1.121$

$\delta 4 := 1 + \sigma 1 + \sigma 2 \cdot ik4^2$ $\delta 5 := 1 + \sigma 1 + \sigma 2 \cdot ik5^2$

$\delta 4 = 1.078$ $\delta 5 = 1.063$

Acceleration of the vehicle when moving the vehicle in 1st gear

$$\psi 1_{i,j} := f0 \cdot \left[1 + Af \cdot \left(V1_{i,j}\right)^2\right]$$

$$J1_{i,j} := \left(D1_{i,j} - \psi 1_{i,j}\right) \cdot \frac{g}{\delta 1}$$

Acceleration of the vehicle when moving the vehicle in 2nd gear

$$\psi 2_{i,j} := f0 \cdot \left[1 + Af \cdot \left(V2_{i,j}\right)^2\right]$$

$$J2_{i,j} := \left(D2_{i,j} - \psi 2_{i,j}\right) \cdot \frac{g}{\delta 2}$$

Acceleration of the vehicle when moving the vehicle in 3rd gear

$$\psi 3_{i,j} := f0 \cdot \left[1 + Af \cdot \left(V3_{i,j}\right)^2\right]$$

$$J3_{i,j} := \left(D3_{i,j} - \psi 3_{i,j}\right) \cdot \frac{g}{\delta 3}$$

Acceleration of the vehicle when moving the vehicle in 4th gear

$$\psi 4_{i,j} := f0 \cdot \left[1 + Af \cdot \left(V4_{i,j}\right)^2\right]$$

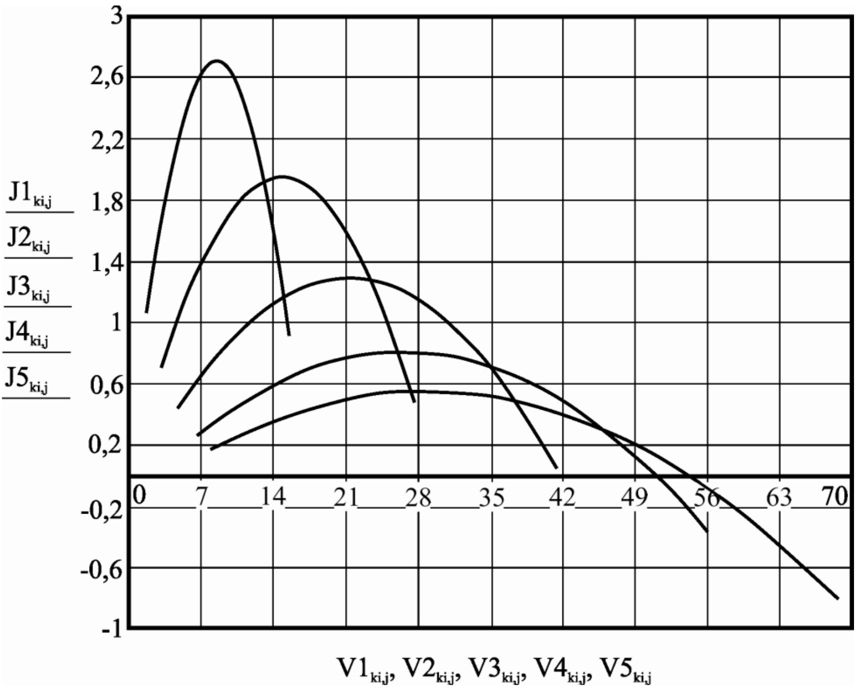
$$J4_{i,j} := \left(D4_{i,j} - \psi 4_{i,j}\right) \cdot \frac{g}{\delta 4}$$

Acceleration of the vehicle when moving the vehicle in 5th gear

$$\psi5_{i,j} := f0 \cdot \left[1 + Af \cdot \left(V5_{i,j}\right)^2\right]$$

$$J5_{i,j} := \left(D5_{i,j} - \psi5_{i,j}\right) \cdot \frac{g}{\delta5}$$

The graph of the dependence of the limiting accelerations on the vehicle speed of Opel Astra Classic



5 DETERMINATION OF ACCELERATION CHARACTERISTICS OF THE VEHICLE

More convenient and clear estimating measures of intensity of acceleration of the vehicle are time *t* and a path *S* of acceleration of the vehicle in the given speed range. These parameters can be determined experimentally or by calculation.

Some assumptions are taken into account in the calculation. So, at the vehicle with mechanical transmission at moving from a place and gear switching some time transfer of a torque from the engine to driving wheels occurs with slippage of catching. In the calculations, this process is despised, and it is believed that after the transmission to the wheels immediately transmits engine power, which corresponds to the full supply of fuel. In addition, it is believed that at each time the wheels are supplied with power determined by the external speed characteristic of the engine for the revolution that corresponds to the speed of the vehicle during acceleration.

The time and path of acceleration of the vehicle is calculated with the assumption that it accelerates on a flat horizontal road with full fuel supply on a section of length 2000 m [5].

5.1 Determining the moment (points) of gear shift

Starting from a place begins on the gear which provides the maximum acceleration. To determine the most intense acceleration in the calculation is introduced acceleration corresponding to the maximum allowable speed of the vehicle in this gear.

The points of intersection of the acceleration curves on two adjacent gears (Fig. 4.5) determine the speeds $V_{1\text{-}2}$, $V_{2\text{-}3}$, $V_{3\text{-}4}$ and $V_{4\text{-}5}$, at which it is necessary to switch in order to maximize the acceleration intensity.

If the lines on the graph do not intersect, then the switching point is the point of the corresponding maximum speed of the vehicle in this gear.

If shifting from one gear to another occurs earlier or later when these speeds are reached, the vehicle will accelerate less intensely over time.

When shifting gears, there is a break in the power flow and the vehicle moves in coasting, there is a loss of speed. The shift time depends on the driver's qualifications, the design of the gearbox and the type of engine.

In the process of shifting gears, the speed of the vehicle decreases. The magnitude of the decrease in velocity ΔV_n during the movement of the vehicle in the process of shifting, can be found by solving the equation of power balance (4.9)

Since the engine and transmission are separated, when the vehicle moves in coasting, the traction force on the drive wheels is zero ($P_k = 0$).To take the drag force approximately equal to zero when shifting gears ($P_w = 0$). Then the dynamic factor D_a will be equal to zero

$$D_a = \frac{P_k - P_w}{G_a} \cong 0. \qquad (5.1)$$

After the appropriate transformations of expression (4.22) to take into account the influence of inertial rotational masses, the loss of speed during gear shift can be determined by the formula

$$\Delta V_n = \frac{\psi \cdot g \cdot t_n}{\delta_{np}}, \qquad (5.2)$$

where t_n – gear shift time depends on the driver's qualifications, gearbox design and engine type, is approximately accepted [13, c. 31]:

0,5 …1,5 sec – for vehicles with a gasoline engine,

0,8…2,5 sec – for vehicles with a diesel engine.

To simplify the calculation during the shift period, the movement resistance is assumed to be constant.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Determining the moment of gear shift

$tn := 1$ *(s) – gear shift time from lower to higher for all gears is assumed to be the same*

Switching point from 1st gear to 2nd one

$V12max := 13.179$ *(m/s) –the speed at which the transmission is switched is determined by the graph of limiting accelerations using the "Trace" command.*

$$J1max := \left| \begin{array}{l} J1max \leftarrow 0 \\ \text{for } \ i \in 1..ki \\ \quad \text{for } \ j \in 1..ki \\ \quad\quad J1max \leftarrow J1_{ki,j} \ \text{ if } \ V1_{ki,j} \le V12max \\ J1max \end{array} \right.$$

$J1max = 2.027$ *(m/s²)*

$$V1max := \left| \begin{array}{l} V1max \leftarrow 0 \\ \text{for } \ i \in 1..ki \\ \quad \text{for } \ j \in 1..ki \\ \quad\quad V1max \leftarrow V1_{ki,j} \ \text{ if } \ J1_{ki,j} = J1max \\ V1max \end{array} \right.$$

$V1max = 13.156$ *(m/s)*

$$n1max := \left| \begin{array}{l} n1max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n1max \leftarrow n_{ki,j} \ \text{ if } \ V1_{ki,j} = V1max \\ n1max \end{array} \right.$$

$n1max = 6132.000$ *(rpm)*

<u>Decrease in speed when shifting from 1st gear to 2nd one</u>

$$\psi 1 := f0 \cdot \left[1 + Af \cdot (V1max)^2 \right]$$

$\psi 1 = 0.012$

$$V12n := \frac{g \cdot \psi 1 \cdot tn}{\delta 1}$$

$V12n = 0.071$ *(m/s)*

<u>Acceleration start point in 2nd gear</u>

$$V23min := V1max - V12n$$

$V23min = 13.085$ *(m/s)*

$$V2min := \left| \begin{array}{l} V2min \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad V2min \leftarrow V2_{ki,j} \ \text{ if } \ V2_{ki,j} \le V23min \\ V2min \end{array} \right.$$

$V2min = 13.015$ *(m/s)*

$$n2min := \left| \begin{array}{l} n2min \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n2min \leftarrow n_{ki,j} \ \text{ if } \ V2_{ki,j} = V2min \\ n2min \end{array} \right.$$

$n2min = 3423.000$ *(rpm)*

Switching point from 2nd gear to 3rd one

$V23max := 23.479$ *(m/s) – the speed at which the transmission is switched is determined by the graph of limiting accelerations using the "Trace" command.*

$$J2max := \left| \begin{array}{l} J2max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad J2max \leftarrow J2_{ki,j} \text{ if } V2_{ki,j} \le V23max \\ J2max \end{array} \right.$$

$J2max = 1.385$ *(m/s²)*

$$V2max := \left| \begin{array}{l} V2max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad V2max \leftarrow V2_{ki,j} \text{ if } J2_{ki,j} = J2max \\ V2max \end{array} \right.$$

$V2max = 23.479$ *(m/s)*

$$n2max := \left| \begin{array}{l} n2max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n2max \leftarrow n_{ki,j} \text{ if } V2_{ki,j} = V2max \\ n2max \end{array} \right.$$

$n2max = 6175.000$ *(rpm)*

Decrease in speed when shifting from 2nd gear to 3rd one

$$\psi 2 := f0 \cdot \left[1 + Af \cdot (V2max)^2 \right]$$

$\psi 2 = 0.014$

$$V23n := \frac{g \cdot \psi 2 \cdot tn}{\delta 2}$$

$V23n = 0.114$ *(m/s)*

Acceleration start point in 3rd gear

$V34min := V2max - V23n$

$V34min = 23.365$ *(m/s)*

$$V3min := \left| \begin{array}{l} V3min \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad V3min \leftarrow V3_{ki,j} \ \text{ if } \ V3_{ki,j} \le V34min \\ V3min \end{array} \right.$$

$V3min = 23.144$ *(m/s)*

$$n3min := \left| \begin{array}{l} n3min \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n3min \leftarrow n_{ki,j} \ \text{ if } \ V3_{ki,j} = V3min \\ n3min \end{array} \right.$$

$n3min = 4068.000$ *(rpm)*

Switching point from 3rd gear to 4th one

$V34max := 34.887$ *(m/s) – the speed at which the transmission is switched is determined by the graph of limiting accelerations using the "Trace" command.*

$$J3max := \left| \begin{array}{l} J3max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad J3max \leftarrow J3_{ki,j} \ \text{ if } \ V3_{ki,j} \le V34max \\ J3max \end{array} \right.$$

$J3max = 0.795$ *(m/s²)*

$$V3max := \left| \begin{array}{l} V3max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad V3max \leftarrow V3_{ki,j} \ \text{ if } J3_{ki,j} = J3max \\ V3max \end{array} \right.$$

$V3max = 34.887$ *(m/s)*

$$n3max := \left| \begin{array}{l} n3max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n3max \leftarrow n_{ki,j} \ \text{ if } V3_{ki,j} = V3max \\ n3max \end{array} \right.$$

$n3max = 6132.000$ *(rpm)*

Decrease in speed when shifting from 3rd gear to 4th one

$$\psi 3 := f0 \cdot \left[1 + Af \cdot (V3max)^2 \right]$$

$\psi 3 = 0.018$

$$V34n := \frac{g \cdot \psi 3 \cdot tn}{\delta 3}$$

$V34n = 0.161$ *(m/s)*

Acceleration start point in 4th gear

$$V45min := V3max - V34n$$

$V45min = 34.726$ *(m/s)*

$$V4min := \left| \begin{array}{l} V4min \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad V4min \leftarrow V4_{ki,j} \ \text{ if } V4_{ki,j} \le V45min \\ V4min \end{array} \right.$$

$V4min = 34.578$ *(m/s)*

$$n4min := \left| \begin{array}{l} n4min \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n4min \leftarrow n_{ki,j} \text{ if } V4_{ki,j} = V4min \\ n4min \end{array} \right.$$

$n4min = 4412.000$ *(rpm)*

<u>*Acceleration in 4th gear is done to speed 0,95 V_{max}*</u>

$$V45max := 0.95 \cdot Vmax$$

$$J4max := \left| \begin{array}{l} J4max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad J4max \leftarrow J4_{ki,j} \text{ if } V4_{ki,j} \le V45max \\ J4max \end{array} \right.$$

$J4max = 0.208$ *(m/s^2)*

$$V4max := \left| \begin{array}{l} V4max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad V4max \leftarrow V4_{ki,j} \text{ if } J4_{ki,j} = J4max \\ V4max \end{array} \right.$$

$V4max = 48.058$ *(m/s)*

$$n4max := \left| \begin{array}{l} n4max \leftarrow 0 \\ \text{for } i \in 1..ki \\ \quad \text{for } j \in 1..ki \\ \quad\quad n4max \leftarrow n_{ki,j} \text{ if } V4_{ki,j} = V4max \\ n4max \end{array} \right.$$

$n4max = 6132.000$ *(rpm)*

Decrease in speed when shifting from 4th gear to 5th one

$$\psi4 := f0 \cdot \left[1 + Af \cdot (V4max)^2\right]$$

$$\psi4 = 0.025$$

$$V45n := \frac{g \cdot \psi4 \cdot tn}{\delta4}$$

$V45n = 0.227$ *(m/s)*

5.2 Determining the acceleration time of the vehicle

Acceleration curves depending on speed allow not only to estimate intensity of acceleration and to select optimum modes of gear shift, but also to define time and a path of acceleration of the vehicle.

The acceleration time of the vehicle on the transmission in the speed range from V_{min} to V_{max} (Fig. 5.1) is based on the following ratios

$$J = \frac{dV}{dt}, \tag{5.3}$$

$$dt = \frac{1}{J} dV, \tag{5.4}$$

$$t = \int_{V\,min}^{V\,max} \frac{1}{J} dV, \tag{5.5}$$

where J – acceleration of the vehicle during supplemental motion, m/s^2.

When accelerating the vehicle from a place calculation needs to be conducted from speed which corresponds to the minimum steady revolution of a cranked shaft of the engine n_{min} at full supply of fuel on the first gear.

As the speed approaches the maximum, the acceleration of the vehicle approaches zero, as a result of which the value of $1/J$ asymptotically approaches the ordinate, which corresponds to V_{max}. This means that the vehicle accelerates to maximum speed, theoretically approaching infinity. In fact, the acceleration of the vehicle becomes almost imperceptible when reaching a speed (0,9 ÷0.95) V_{max}.

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

Acceleration time when the vehicle is in 1st gear

$$i := 1..ki \quad j := 1..ki \quad kj := \frac{n1max - nmin}{ki}$$

$$n1_{i,j} := nmin + j \cdot kj$$

$$N1_{i,j} := Nmax \cdot \left[a \cdot \frac{n1_{i,j}}{nNmax} + b \cdot \left(\frac{n1_{i,j}}{nNmax} \right)^2 - c \cdot \left(\frac{n1_{i,j}}{nNmax} \right)^3 \right]$$

$$w1_{i,j} := \frac{n1_{i,j} \cdot \pi}{30}$$

$$M1_{i,j} := \frac{N1_{i,j}}{w1_{i,j}}$$

$$V1_{i,j} := \frac{w1_{i,j} \cdot rk}{ik1 \cdot io \cdot irk} \qquad Pk1_{i,j} := M1_{i,j} \cdot ik1 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw1_{i,j} := k \cdot F \cdot \left| V1_{i,j} \right|^2$$

$$D1_{i,j} := \frac{\left| Pk1_{i,j} - Pw1_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

$$\psi1_{i,j} := f0 \cdot \left[1 + Af \cdot \left| V1_{i,j} \right|^2 \right]$$

$$J1_{i,j} := \left| D1_{i,j} - \psi1_{i,j} \right| \cdot \frac{g}{\delta1}$$

$$t1_{ki,j} := \int_{V1_{ki,1}}^{V1_{ki,j}} \frac{1}{J1_{ki,j}}\, dV1$$

$t1_{ki,j} =$

	1
1	0.000
2	0.065
3	0.126
4	...

$\max(t1) = 5.605$ *(s)*

Acceleration time when the vehicle is in 2nd gear

$$i := 1..ki \qquad j := 1..ki \qquad kj := \frac{n2max - n2min}{ki}$$

$$n2_{i,j} := n2min + j \cdot k$$

$$N2_{i,j} := Nmax \cdot \left[a \cdot \frac{n2_{i,j}}{nNmax} + b \cdot \left(\frac{n2_{i,j}}{nNmax} \right)^2 - c \cdot \left(\frac{n2_{i,j}}{nNmax} \right)^3 \right]$$

$$w2_{i,j} := \frac{n2_{i,j} \cdot \pi}{30}$$

$$M2_{i,j} := \frac{N2_{i,j}}{w2_{i,j}}$$

$$V2_{i,j} := \frac{w2_{i,j} \cdot rk}{ik2 \cdot io \cdot irk} \qquad Pk2_{i,j} := M2_{i,j} \cdot ik2 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw2_{i,j} := k \cdot F \cdot \left| V2_{i,j} \right|^2$$

$$D2_{i,j} := \frac{\left| Pk2_{i,j} - Pw2_{i,j} \right|}{Ma1_{i,j} \cdot g}$$

$$\psi 2_{i,j} := f0 \cdot \left[1 + Af \cdot \left(\left| V2_{i,j} \right| \right)^{2} \right]$$

$$J2_{i,j} := \left(\left| D2_{i,j} - \psi 2_{i,j} \right| \right) \cdot \frac{g}{\delta 2}$$

$$t2_{ki,j} := \int_{V2_{ki,1}}^{V2_{ki,j}} \frac{1}{J2_{ki,j}} \, dV2 + \max(t1) + tn$$

$t2_{ki,j} =$

	1
1	6.605
2	6.640
3	6.674
4	...

$\max(t2) = 14.109$ *(s)* $\quad t12 := \begin{pmatrix} t1_{ki,ki} \\ t2_{ki,1} \end{pmatrix} \quad V12 := \begin{pmatrix} V1_{ki,ki} \\ V2_{ki,1} \end{pmatrix}$

<u>*Acceleration time when the vehicle is in 3rd gear*</u>

$i := 1..ki \qquad j := 1..ki \qquad kj := \dfrac{n3max - n3min}{ki}$

$$n3_{i,j} := n3min + j \cdot kj$$

$$N3_{i,j} := Nmax \cdot \left[a \cdot \frac{n3_{i,j}}{nNmax} + b \cdot \left(\frac{n3_{i,j}}{nNmax} \right)^{2} - c \cdot \left(\frac{n3_{i,j}}{nNmax} \right)^{3} \right]$$

$$w3_{i,j} := \frac{n3_{i,j} \cdot \pi}{30}$$

$$M3_{i,j} := \frac{N3_{i,j}}{w3_{i,j}}$$

$V3_{i,j} := \dfrac{w3_{i,j} \cdot rk}{ik3 \cdot io \cdot irk} \qquad Pk3_{i,j} := M3_{i,j} \cdot ik3 \cdot \dfrac{io \cdot \eta_{tr}}{rk} \cdot irk$

$$Pw3_{i,j} := k \cdot F \cdot \left(\left| V3_{i,j} \right| \right)^{2}$$

$$D3_{i,j} := \frac{\left|Pk3_{i,j} - Pw3_{i,j}\right|}{Ma1_{i,j} \cdot g}$$

$$\psi 3_{i,j} := f0 \cdot \left[1 + Af \cdot \left|V3_{i,j}\right|^{2}\right]$$

$$J3_{i,j} := \left|D3_{i,j} - \psi 3_{i,j}\right| \cdot \frac{g}{\delta 3}$$

$$t3_{ki,j} := \int_{V3_{ki,1}}^{V3_{ki,j}} \frac{1}{J3_{ki,j}}\, dV3 + \max(t2) + tn$$

$t3_{ki,j} =$

	1
1	15.109
2	15.165
3	15.222
4	...

$\max(t3) = 29.772$ *(c)* $\quad t23 := \begin{pmatrix} t2_{ki,ki} \\ t3_{ki,1} \end{pmatrix} \quad V23 := \begin{pmatrix} V2_{ki,ki} \\ V3_{ki,1} \end{pmatrix}$

Acceleration time when the vehicle is in 4th gear

$$i := 1..ki \qquad j := 1..ki \qquad kj := \frac{n4max - n4min}{ki}$$

$$n4_{i,j} := n4min + j \cdot kj$$

$$N4_{i,j} := Nmax \cdot \left[a \cdot \frac{n4_{i,j}}{nNmax} + b \cdot \left(\frac{n4_{i,j}}{nNmax}\right)^{2} - c \cdot \left(\frac{n4_{i,j}}{nNmax}\right)^{3}\right]$$

$$w4_{i,j} := \frac{n4_{i,j} \cdot \pi}{30}$$

$$M4_{i,j} := \frac{N4_{i,j}}{w4_{i,j}}$$

$$V4_{i,j} := \frac{w4_{i,j} \cdot rk}{ik4 \cdot io \cdot irk} \qquad Pk4_{i,j} := M4_{i,j} \cdot ik4 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk$$

$$Pw4_{i,j} := k \cdot F \cdot \left(\left|V4_{i,j}\right|\right)^2$$

$$D4_{i,j} := \frac{\left|Pk4_{i,j} - Pw4_{i,j}\right|}{Ma1_{i,j} \cdot g}$$

$$\psi 4_{i,j} := f0 \cdot \left[1 + Af \cdot \left(\left|V4_{i,j}\right|\right)^2\right]$$

$$J4_{i,j} := \left|D4_{i,j} - \psi 4_{i,j}\right| \cdot \frac{g}{\delta 4}$$

$$t4_{ki,j} := \int_{V4_{ki,1}}^{V4_{ki,j}} \frac{1}{J4_{ki,j}}\, dV4 + \max(t3) + tn$$

$t4_{ki,j} =$

	1
1	30.772
2	30.884
3	30.997
4	...

$\max(t4) = 95.234$ *(s)* $\quad t34 := \begin{pmatrix} t3_{ki,ki} \\ t4_{ki,1} \end{pmatrix} \quad V34 := \begin{pmatrix} V3_{ki,ki} \\ V4_{ki,1} \end{pmatrix}$

5.3 Determining the speeding-up path of the vehicle

The acceleration path of the vehicle is determined from the fact that the speed is the first derivative of the path in time

$$V = \frac{dS}{dt}, \tag{5.6}$$

then

$$dS = Vdt. \tag{5.7}$$

Therefore, the path traveled by the vehicle during acceleration (Fig. 5.1) in the time range from $t_{\min}$ to $t_{\max}$ which corresponds to the speed range from $V_{\min}$ to $V_{\max}$ is determined by the expression

$$S = \int_{t_{\min}}^{t_{\max}} Vdt. \tag{5.8}$$

The path traveled by the vehicle during the shift t_n from lower to higher gear is determined by the formula

$$\Delta S_n = \ V_{\max} - 0{,}5 \cdot \Delta V_n \ \ \cdot t_n. \tag{5.9}$$

<u>Program listing:</u>

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

<u>*The path that the vehicle travels during acceleration in 1st gear*</u>

$$S1_{ki,j} := \int_{t1_{ki,1}}^{t1_{ki,j}} V1_{ki,j}\, dt1$$

$S1_{ki,j} =$

	1
1	0.000
2	0.122
3	0.245
4	...

$\max(S1) = 73.743$ *(m)*

<u>The path that the vehicle travels during acceleration in 2nd gear</u>

$S12n := \left| V1_{ki,ki} - 0.5 \cdot V12n \right| \cdot tn$ – *the path that the vehicle travels when gear shift*

$S12n = 13.120$ *(m)*

$$S2_{ki,j} := \int_{t2_{ki,1}}^{t2_{ki,j}} V2_{ki,j}\, dt2 + \max(S1) + S12n$$

$S2_{ki,j} =$

	1
1	86.863
2	87.313
3	87.767
4	...

$\max(S2) = 263.04$ *(m)* $S12 := \begin{pmatrix} S1_{ki,ki} \\ S2_{ki,1} \end{pmatrix}$ $V12 := \begin{pmatrix} V1_{ki,ki} \\ V2_{ki,1} \end{pmatrix}$

<u>The path that the vehicle travels during acceleration in 3rd gear</u>

$S23n := \left| V2_{ki,ki} - 0.5 \cdot V23n \right| \cdot tn$ – *the path that the vehicle travels when gear shift*

$S23n = 23.422$ *(m)*

$$S3_{ki,j} := \int_{t3_{ki,1}}^{t3_{ki,j}} V3_{ki,j}\, dt3 + \max(S2) + S23n$$

$S3_{ki,j} =$

	1
1	286.459
2	287.770
3	289.093
4	...

$\max(S3) = 798.01$ *(m)* $S23 := \begin{pmatrix} S2_{ki,ki} \\ S3_{ki,1} \end{pmatrix}$ $V23 := \begin{pmatrix} V2_{ki,ki} \\ V3_{ki,1} \end{pmatrix}$

The path that the vehicle travels during acceleration in 4th gear

$S34n := \left| V3_{ki,ki} - 0.5 \cdot V34n \right| \cdot tn$ – *the path that the vehicle travels when gear shift*

$S34n = 34.806$ *(m)*

$$S4_{ki,j} := \int_{t4_{ki,1}}^{t4_{ki,j}} V4_{ki,j}\, dt4 + \max(S3) + S34n$$

$S4_{ki,j} =$

	1
1	832.814
2	836.714
3	840.657
4	...

$\max(S4) = 3930.70$ *(m)* $S34 := \begin{pmatrix} S3_{ki,ki} \\ S4_{ki,1} \end{pmatrix}$ $V34 := \begin{pmatrix} V3_{ki,ki} \\ V4_{ki,1} \end{pmatrix}$

5.4 Determining the time and path that the vehicle travels at accelerating to a given speed

In accordance with standard [5] when determining the acceleration time from place to a given speed set the following values of the final acceleration speed:

100 km/h − for vehicles of all types with a total weight of up to 3,5 tons;

80 km/h − for trucks, buses (except city) with a total weight of more than 3,5 tons and road trains;

60 km/h − for city buses.

For vehicles that have a maximum speed below or above the given speed of not more than 5 km/h, the nearest lower speed, a multiple of ten, is accepted.

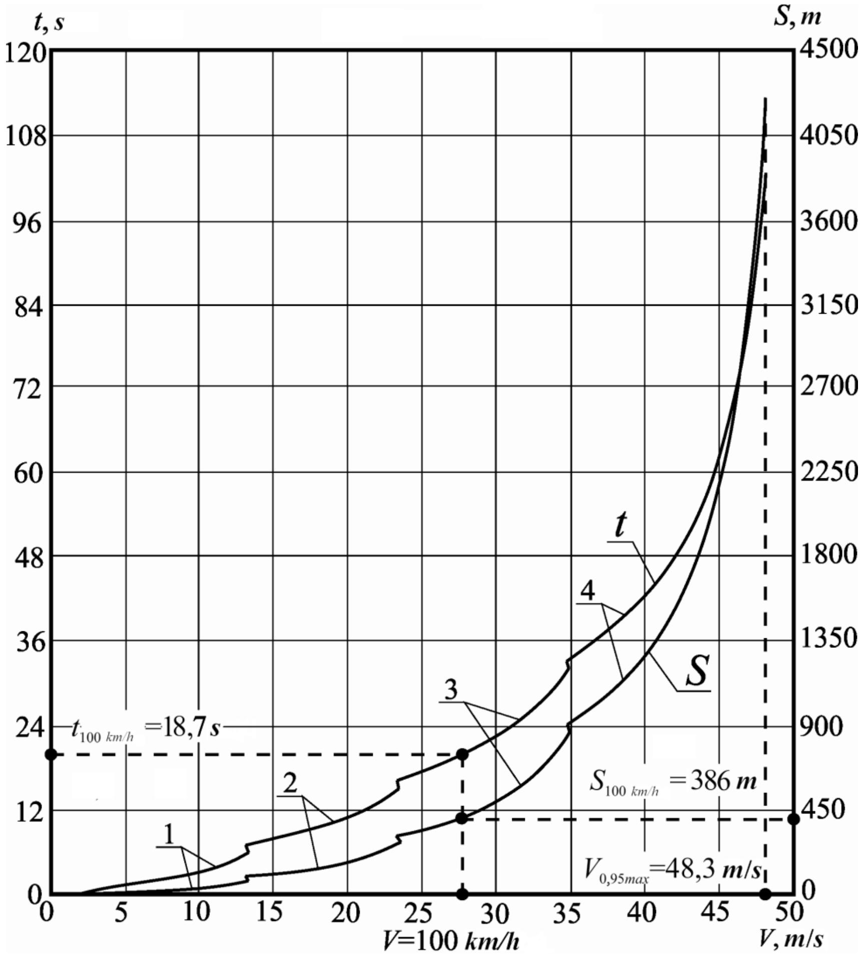


Figure 5.1 – Acceleration characteristics of the vehicle:

1 – time and path traveled when accelerating the vehicle in 1st gear;

2 – time and path traveled when accelerating the vehicle in 2nd gear;

3 – time and path traveled when accelerating the vehicle in 3rd gear;

4 – time and path traveled when accelerating the vehicle in 4th gear

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

*Determining the acceleration time **tr** for which the vehicle reaches speed **Vr***

$Vr := 100$ *(km/h) – the speed reached by the vehicle and at which it is necessary to determine the time and path of acceleration*

$$
tr := \left|
\begin{array}{l}
tr \leftarrow 0 \\
\text{for } i \in 1..ki \\
\quad \text{for } j \in 1..ki \\
\qquad tr \leftarrow t1_{ki,j} \ \text{ if } \ V1_{ki,j} \le \dfrac{Vr}{3.6} \\
\text{for } i \in 1..ki \\
\quad \text{for } j \in 1..ki \\
\qquad tr \leftarrow t2_{ki,j} \ \text{ if } \ V2_{ki,j} \le \dfrac{Vr}{3.6} \\
\text{for } i \in 1..ki \\
\quad \text{for } j \in 1..ki \\
\qquad tr \leftarrow t3_{ki,j} \ \text{ if } \ V3_{ki,j} \le \dfrac{Vr}{3.6} \\
\text{for } i \in 1..ki \\
\quad \text{for } j \in 1..ki \\
\qquad tr \leftarrow t4_{ki,j} \ \text{ if } \ V4_{ki,j} \le \dfrac{Vr}{3.6} \\
tr
\end{array}
\right.
$$

$tr = 18.71$ *(s) –acceleration time of the vehicle to the speed* $\boldsymbol{V_r}$

*<u>Determining the path **Sr** that the vehicle travels during acceleration to the speed **Vr**</u>*

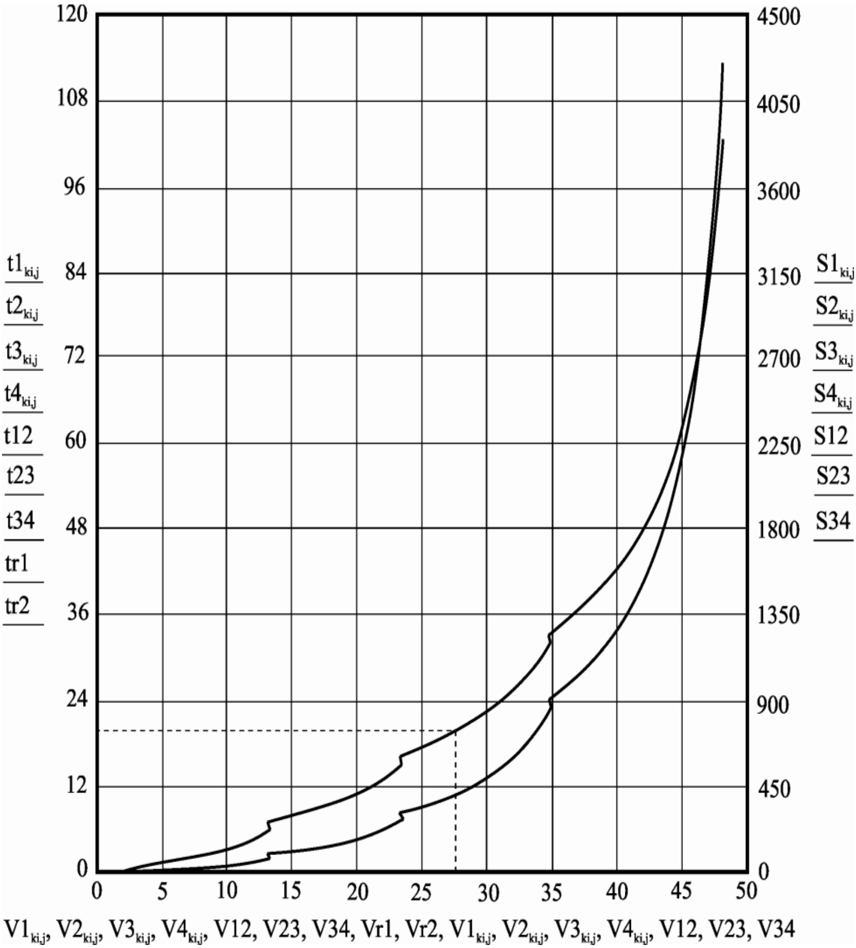
$$
\mathrm{Sr} := \left|
\begin{array}{l}
\mathrm{sr} \leftarrow 0 \\
\text{for } i \in 1..\mathrm{ki} \\
\quad \text{for } j \in 1..\mathrm{ki} \\
\quad\quad \mathrm{Sr} \leftarrow \mathrm{S1}_{\mathrm{ki},j} \ \text{ if } \ \mathrm{V1}_{\mathrm{ki},j} \le \dfrac{\mathrm{Vr}}{3.6} \\
\text{for } i \in 1..\mathrm{ki} \\
\quad \text{for } j \in 1..\mathrm{ki} \\
\quad\quad \mathrm{Sr} \leftarrow \mathrm{S2}_{\mathrm{ki},j} \ \text{ if } \ \mathrm{V2}_{\mathrm{ki},j} \le \dfrac{\mathrm{Vr}}{3.6} \\
\text{for } i \in 1..\mathrm{ki} \\
\quad \text{for } j \in 1..\mathrm{ki} \\
\quad\quad \mathrm{Sr} \leftarrow \mathrm{S3}_{\mathrm{ki},j} \ \text{ if } \ \mathrm{V3}_{\mathrm{ki},j} \le \dfrac{\mathrm{Vr}}{3.6} \\
\text{for } i \in 1..\mathrm{ki} \\
\quad \text{for } j \in 1..\mathrm{ki} \\
\quad\quad \mathrm{Sr} \leftarrow \mathrm{S4}_{\mathrm{ki},j} \ \text{ if } \ \mathrm{V4}_{\mathrm{ki},j} \le \dfrac{\mathrm{Vr}}{3.6} \\
\mathrm{Sr}
\end{array}
\right.
$$

$\mathrm{Sr} = 386.34$ *(m) – the path **Sr** that the vehicle travels during acceleration to the speed **Vr***

<u>Plotting of the acceleration characteristics of the vehicle</u>

$$tr1 := \begin{pmatrix} 0 \\ tr \end{pmatrix} \; tr2 := \begin{pmatrix} tr \\ tr \end{pmatrix} \; Vr1 := \begin{pmatrix} \frac{Vr}{3.6} \\ \frac{Vr}{3.6} \end{pmatrix} \; Vr2 := \begin{pmatrix} 0 \\ \frac{Vr}{3.6} \end{pmatrix} \; Sr1 := \begin{pmatrix} 0 \\ Sr \end{pmatrix} \; Sr2 := \begin{pmatrix} Sr \\ Sr \end{pmatrix}$$

Acceleration characteristics of Opel Astra Classic



6 CALCULATION OF FUEL AND ECONOMIC CHARACTERISTICS OF THE VEHICLE

6.1 Estimates of fuel efficiency

Fuel economy of the vehicle is called a set of properties that determine the fuel consumption when the vehicle operates in different conditions.

Fuel efficiency mainly depends on the design of the vehicle and its operating conditions. It is determined by the degree of perfection of the working process in the engine, efficiency and gear ratio, the ratio between the equipped and total weight of the vehicle, the intensity of its movement, as well as the resistance to the movement of the vehicle environment.

Thus, the fuel efficiency of the vehicle characterizes its ability to rationally use fuel energy. The lower the fuel consumption, the cheaper the operation of the vehicle.

The energy source for the movement of the vehicle is the engine installed on it. Therefore, the fuel efficiency of the vehicle is largely determined by such indicators of the engine as time fuel consumption G_t, kg/h (mass of fuel consumed per hour of engine operation) and specific fuel consumption g_s, g/(kW·h) (mass of fuel, which consumed per hour per unit of engine power).

The main indicator of fuel efficiency of the vehicle in the CIS countries and the majority of the European countries is a fuel consumption in l on 100 km of the passed way, or path(linear) fuel consumption Q_s, l/100 km.

To assess the efficiency of fuel use in the performance of transport work using fuel consumption in 1 l per unit of transport work (100 t·km) Q_w, l/(100 t·km), i.e. the ratio of actual fuel consumption to the performed transport work.

Accordingly standard "Vehicles. Fuel economy. Test methods" [4] estimates of fuel efficiency are:

− control fuel consumption;

− fuel consumption in the main cycle on the road;

− fuel consumption in the city cycle on the road;

− fuel consumption in the city cycle on the stand;

− fuel characteristics of steady movement;

− fuel-speed characteristic on the main-hilly road.

These estimates do not have standardized values, they are used in the comparative assessment of the level of fuel economy with foreign counterparts and indirect assessment of the technical condition of vehicles.

Usually, the fuel efficiency of vehicles is determined experimentally [4]. While the control fuel consumption is determined for all categories of vehicles when driving on a straight horizontal road with a length of 1000 m or more in higher gear with a given speed with an accuracy of ± 2 km/h.

Depending on the type of vehicle and the maximum speed, the following values of the speeds V are set:

40 and 60 km/h – for city buses and all-wheel drive cars with total weight of more than 3,5 tons;

60 and 80 km/h – for trucks (including all-wheel drive), special purpose buses, long-distance and international buses, road trains weighing more than 3,5 tons;

90 and 120 km/h – for cars (including all-wheel drive), buses and trucks with total weight of up to 3,5 tons.

If the maximum speed is less than 120 km/h, the fuel consumption at 120 km/h is not determined. If the maximum speed is less than the set speed or exceeds it by 5 km/h, then the speed should be set to the nearest smaller one multiple of 10.

6.2 Calculation of estimated indicators of fuel efficiency of stable vehicle movement

The specific effective fuel consumption q_e (Fig. 6.1, 6.2) is a variable, and depends on the speed and load modes of the engine and is determined experimentally. In the absence of experimental data, the specific fuel consumption at steady movement of the vehicle is determined by the formula proposed by I. S. Schlippe [8, 15].

$$q_e = q_{eN\max} \cdot K_n \cdot K_N, \tag{6.1}$$

where $q_{eN\max}$ – specific fuel consumption at maximum engine power according to the external speed characteristic, approximately can be taken[10]:

200...290 g/kW·h –gasoline internal combustion engines with electronic fuel injection,

230 ... 310 g/kW·h – carburetor internal combustion engines,

200 ... 325 g/kW·h – diesel internal combustion engines with undivided chambers,

200 ... 260 g/kW·h – diesel internal combustion engine chambers vortex and pre-chamber (pre-chamber),

223 ... 257 g/kW·h – two-stroke diesel internal combustion engines,

202 ... 235 g/kW·h – four-stroke diesel internal combustion engines without inflation,

188 ... 223 g/kW·h – four-stroke diesel internal combustion engines with inflatable,

234 ... 327 g/kW·h – four-stroke internal combustion engines with spark ignition without inflation;

K_n – the coefficient that takes into account the impact on the specific fuel consumption of the engine speed operation mode;

K_N − the coefficient that takes into account the impact on the specific fuel consumption of the engine load operation mode.

The coefficients K_n and K_N are defined by empirical formulas [3, 7, 12], wich are obtained by statistical processing of the results of experimental data conducted to build the load characteristics of internal combustion engines.

The coefficient K_n, according to average data, for both gasoline and diesel engines can be calculated by the same formula

$$K_n = 1,25 - 0,99 \cdot \frac{n}{n_{N\max}} + 0,98 \cdot \left(\frac{n}{n_{N\max}} \right)^2 - 0,24 \cdot \left(\frac{n}{n_{N\max}} \right)^3, \quad (6.2)$$

where $n_{N\max}$ – engine crankshaft revolutions that correspond to $N_{\max}$;

n – the current value of the engine crankshaft revolution in the range of from $n_{\min}$ to $n_{\max}$;

The coefficient K_N that takes into account the effect on the specific fuel consumption of the load engine operation mode can be calculated by the formula:

− for carburetor engines

$$K_N = 3,27 - 8,22 \cdot \left(\frac{N_\psi + N_w}{N_{\max} \cdot \eta_{tr}} \right) + 9,13 \cdot \left(\frac{N_\psi + N_w}{N_{\max} \cdot \eta_{tr}} \right)^2 - 3,18 \cdot \left(\frac{N_\psi + N_w}{N_{\max} \cdot \eta_{tr}} \right)^3, \quad (6.3)$$

where N_ψ – power consumed to overcome the total rolling resistance of the wheels of the vehicle, W (when driving on a horizontal section of road can be taken $N_\psi = N_f$);

N_w – power consumed to overcome the drag force, W;

$N_{\max}$ – maximum engine power of the vehicle, W;

η_{tr} – efficiency of vehicle transmission.

− for diesel and gasoline engines with fuel injection

$$K_N = 1,2 - 0,14 \cdot \left(\frac{N_\psi + N_w}{N_{\max} \cdot \eta_{\text{тр}}} \right) - 1,8 \cdot \left(\frac{N_\psi + N_w}{N_{\max} \cdot \eta_{\text{тр}}} \right)^2 + 1,46 \cdot \left(\frac{N_\psi + N_w}{N_{\max} \cdot \eta_{\text{тр}}} \right)^3. \quad (6.4)$$

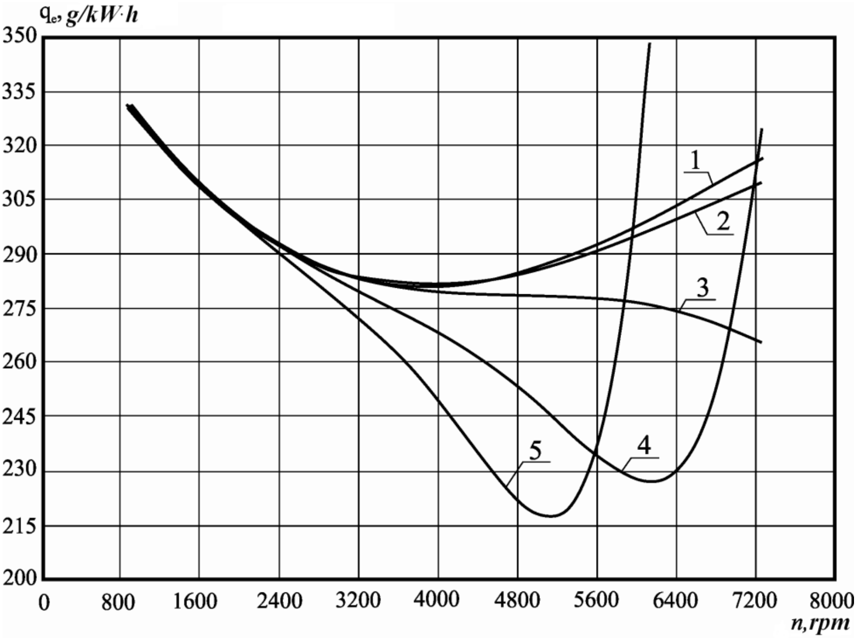


Figure 6.1 – The graph of the dependence of the specific fuel consumption on the engine crankshaft revolution:

1 – specific fuel consumption when moving the vehicle in 1st gear;

2 – specific fuel consumption when moving the vehicle in 2nd gear;

3 –specific fuel consumption when moving the vehicle in 3rd gear;

4 – specific fuel consumption when moving the vehicle in 4th gear;

5 – specific fuel consumption when moving the vehicle in 5th gear

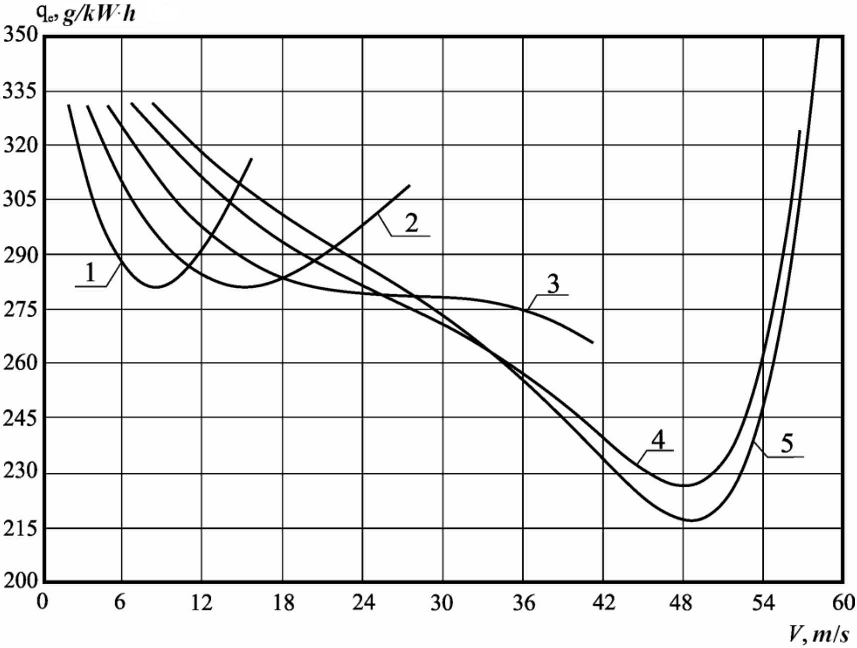


Figure 6.2 – The graph of the dependence of the specific fuel consumption on the speed of steady movement of the vehicle:

1 – specific fuel consumption when moving the vehicle in 1st gear;

2 – specific fuel consumption when moving the vehicle in 2nd gear;

3 –specific fuel consumption when moving the vehicle in 3rd gear;

4 – specific fuel consumption when moving the vehicle in 4th gear;

5 – specific fuel consumption when moving the vehicle in 5th gear

Fuel consumption in 1 *l* per 100 km of path, or path (linear) fuel consumption at constant speeds (Fig. 6.3, 6.4) is determined by the formula

$$Q_s = \frac{q_e \cdot 10^{-3} \cdot \left(N_\psi + N_w\right) \cdot 10^{-3}}{V_{np} \cdot 3{,}6 \cdot \rho_t \cdot \eta_{tr}} \cdot 100, \qquad (6.5)$$

where ρ_t – fuel density, approximately can be taken [10]:

0,725… 0,77 kg/l (g/cm^3) – gasoline,

0,825… 0,86 kg/l (g/cm^3) – diesel fuel.

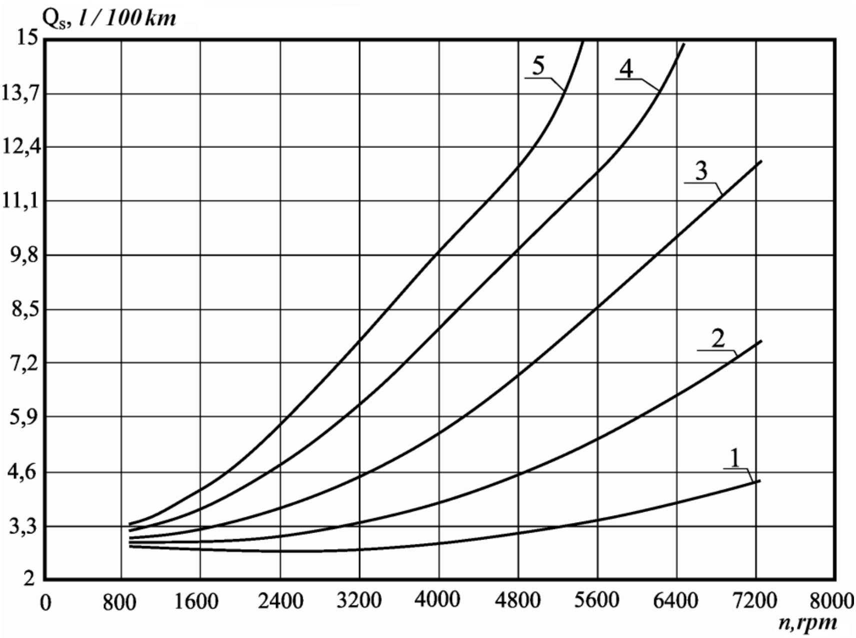


Figure 6.3 – The graph of the dependence of the path fuel consumption on the engine crankshaft revolution:

1 – path fuel consumption when moving the vehicle in 1st gear;

2 – path fuel consumption when moving the vehicle in 2nd gear;

3 –path fuel consumption when moving the vehicle in 3rd gear;

4 – path fuel consumption when moving the vehicle in 4th gear;

5 – path fuel consumption when moving the vehicle in 5th gear

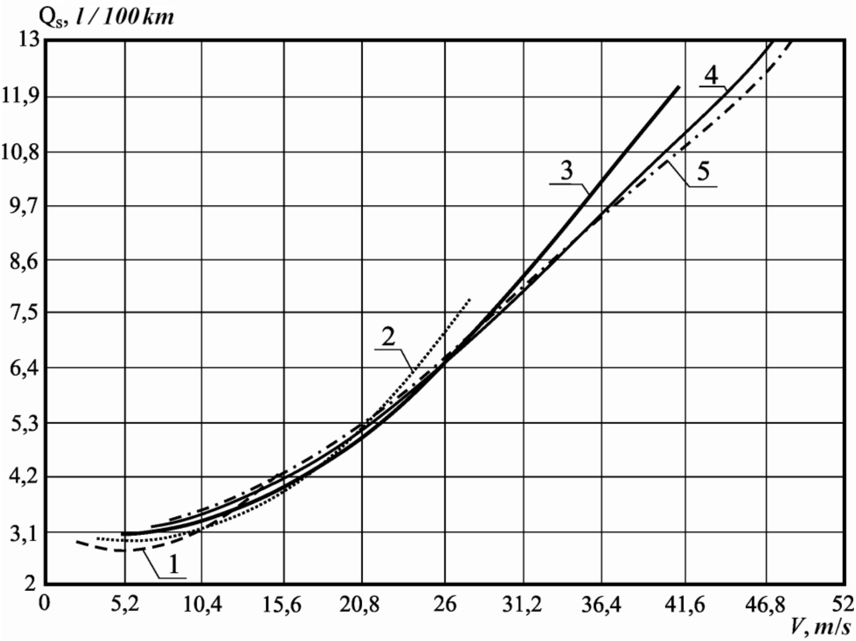


Figure 6.4 – The graph of the dependence of the path fuel consumption on the speed of steady movement of the vehicle:

1 – path fuel consumption when moving the vehicle in 1st gear;

2 – path fuel consumption when moving the vehicle in 2nd gear;

3 –path fuel consumption when moving the vehicle in 3rd gear;

4 – path fuel consumption when moving the vehicle in 4th gear;

5 – path fuel consumption when moving the vehicle in 5th gear

Program listing:

Gray fields are filled with the values of the parameters given in the technical data sheet characteristics of the vehicle or the results of preliminary calculations.

$qeNmax := 245$ *(g/kW·h) – specific fuel consumption at maximum engine power according to the external speed characteristic*

$\rho t := 0.725$ *(kg/l) – fuel density*

The coefficient that takes into account the effect on the specific fuel consumption of the speed engine operation mode

$$Kn_{i,j} := 1.25 - 0.99\frac{n_{i,j}}{nNmax} + 0.98\left(\frac{n_{i,j}}{nNmax}\right)^2 - 0.24\left(\frac{n_{i,j}}{nNmax}\right)^3$$

Estimates of fuel economy at steady movement of the vehicle on the 1st gear

$$V1_{i,j} := \frac{w_{i,j} \cdot rk}{ik1 \cdot io \cdot irk}$$

$$Pk1_{i,j} := M_{i,j} \cdot ik1 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk \qquad Pw1_{i,j} := k \cdot F \cdot \left|V1_{i,j}\right|^2$$

$$\psi 1_{i,j} := f0 \cdot \left[1 + Af \cdot \left|V1_{i,j}\right|^2\right]$$

$$P\psi 1_{i,j} := Ma1_{i,j} \cdot g \cdot \psi 1_{i,j}$$

$$N\psi 1_{i,j} := P\psi 1_{i,j} \cdot V1_{i,j} \qquad Nw1_{i,j} := Pw1_{i,j} \cdot V1_{i,j}$$

The coefficient that takes into account the effect on the specific fuel consumption of the load engine operation mode

$$KN1_{i,j} := 1.2 + 0.14 \cdot \left(\frac{N\psi 1_{i,j} + Nw1_{i,j}}{Nmax \cdot \eta_{tr}}\right) - 1.8\left(\frac{N\psi 1_{i,j} + Nw1_{i,j}}{Nmax \cdot \eta_{tr}}\right)^2 + 1.46 \cdot \left(\frac{N\psi 1_{i,j} + Nw1_{i,j}}{Nmax \cdot \eta_{tr}}\right)^3$$

Specific fuel consumption

$$qe1_{i,j} := qeNmax \cdot Kn_{i,j} \cdot KN1_{i,j}$$

Path (linear) fuel consumption

$$Qs1_{i,j} := \frac{qe1_{i,j} \cdot 10^{-3} \cdot \left| N\psi1_{i,j} + Nw1_{i,j} \right| \cdot 10^{-3}}{V1_{i,j} \cdot 3.6 \cdot \rho t \cdot \eta_{tr}} \cdot 100$$

<u>*Estimates of fuel economy at steady movement of the vehicle on the 2nd gear*</u>

$$V2_{i,j} := \frac{w_{i,j} \cdot rk}{ik2 \cdot io \cdot irk}$$

$$Pk2_{i,j} := M_{i,j} \cdot ik2 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk \qquad Pw2_{i,j} := k \cdot F \cdot \left| V2_{i,j} \right|^2$$

$$\psi2_{i,j} := f0 \cdot \left[1 + Af \cdot \left| V2_{i,j} \right|^2 \right]$$

$$P\psi2_{i,j} := Ma1_{i,j} \cdot g \cdot \psi2_{i,j}$$

$$N\psi2_{i,j} := P\psi2_{i,j} \cdot V2_{i,j} \qquad Nw2_{i,j} := Pw2_{i,j} \cdot V2_{i,j}$$

<u>*The coefficient that takes into account the effect on the specific fuel consumption of the load engine operation mode*</u>

$$KN2_{i,j} := 1.2 + 0.14 \cdot \left(\frac{N\psi2_{i,j} + Nw2_{i,j}}{Nmax \cdot \eta_{tr}} \right) - 1.8 \left(\frac{N\psi2_{i,j} + Nw2_{i,j}}{Nmax \cdot \eta_{tr}} \right)^2 + 1.46 \cdot \left(\frac{N\psi2_{i,j} + Nw2_{i,j}}{Nmax \cdot \eta_{tr}} \right)^3$$

Specific fuel consumption

$$qe2_{i,j} := qeNmax \cdot Kn_{i,j} \cdot KN2_{i,j}$$

Path (linear) fuel consumption

$$Qs2_{i,j} := \frac{qe2_{i,j} \cdot 10^{-3} \cdot \left| N\psi2_{i,j} + Nw2_{i,j} \right| \cdot 10^{-3}}{V2_{i,j} \cdot 3.6 \cdot \rho t \cdot \eta_{tr}} \cdot 100$$

<u>*Estimates of fuel economy at steady movement of the vehicle on the 3rd gear*</u>

$$V3_{i,j} := \frac{w_{i,j} \cdot rk}{ik3 \cdot io \cdot irk}$$

$$Pk3_{i,j} := M_{i,j} \cdot ik3 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk \qquad Pw3_{i,j} := k \cdot F \cdot \left| V3_{i,j} \right|^2$$

$$\psi3_{i,j} := f0 \cdot \left[1 + Af \cdot \left| V3_{i,j} \right|^2 \right]$$

$$P\psi3_{i,j} := Ma1_{i,j} \cdot g \cdot \psi3_{i,j}$$

$$N\psi3_{i,j} := P\psi3_{i,j} \cdot V3_{i,j} \qquad Nw3_{i,j} := Pw3_{i,j} \cdot V3_{i,j}$$

<u>*The coefficient that takes into account the effect on the specific fuel consumption of the load engine operation mode*</u>

$$KN3_{i,j} := 1.2 + 0.14 \cdot \left(\frac{N\psi3_{i,j} + Nw3_{i,j}}{Nmax \cdot \eta_{tr}} \right) - 1.8 \left(\frac{N\psi3_{i,j} + Nw3_{i,j}}{Nmax \cdot \eta_{tr}} \right)^2 + 1.46 \cdot \left(\frac{N\psi3_{i,j} + Nw3_{i,j}}{Nmax \cdot \eta_{tr}} \right)^3$$

Specific fuel consumption

$$qe3_{i,j} := qeNmax \cdot Kn_{i,j} \cdot KN3_{i,j}$$

Path (linear) fuel consumption

$$Qs3_{i,j} := \frac{qe3_{i,j} \cdot 10^{-3} \cdot \left| N\psi3_{i,j} + Nw3_{i,j} \right| \cdot 10^{-3}}{V3_{i,j} \cdot 3.6 \cdot \rho t \cdot \eta_{tr}} \cdot 100$$

Estimates of fuel economy at steady movement of the vehicle on the 4th gear

$$V4_{i,j} := \frac{w_{i,j} \cdot rk}{ik4 \cdot io \cdot irk}$$

$$Pk4_{i,j} := M_{i,j} \cdot ik4 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk \qquad Pw4_{i,j} := k \cdot F \cdot \left| V4_{i,j} \right|^2$$

$$\psi4_{i,j} := f0 \cdot \left[1 + Af \cdot \left| V4_{i,j} \right|^2 \right]$$

$$P\psi4_{i,j} := Ma1_{i,j} \cdot g \cdot \psi4_{i,j}$$

$$N\psi4_{i,j} := P\psi4_{i,j} \cdot V4_{i,j} \qquad Nw4_{i,j} := Pw4_{i,j} \cdot V4_{i,j}$$

The coefficient that takes into account the effect on the specific fuel consumption of the load engine operation mode

$$KN4_{i,j} := 1.2 + 0.14 \cdot \left(\frac{N\psi4_{i,j} + Nw4_{i,j}}{Nmax \cdot \eta_{tr}} \right) - 1.8 \left(\frac{N\psi4_{i,j} + Nw4_{i,j}}{Nmax \cdot \eta_{tr}} \right)^2 + 1.46 \cdot \left(\frac{N\psi4_{i,j} + Nw4_{i,j}}{Nmax \cdot \eta_{tr}} \right)^3$$

Specific fuel consumption

$$qe4_{i,j} := qeNmax \cdot Kn_{i,j} \cdot KN4_{i,j}$$

Path (linear) fuel consumption

$$Qs4_{i,j} := \frac{qe4_{i,j} \cdot 10^{-3} \cdot \left|N\psi4_{i,j} + Nw4_{i,j}\right| \cdot 10^{-3}}{V4_{i,j} \cdot 3.6 \cdot \rho t \cdot \eta_{tr}} \cdot 100$$

<u>Estimates of fuel economy at steady movement of the vehicle on the 5th gear</u>

$$V5_{i,j} := \frac{w_{i,j} \cdot rk}{ik5 \cdot io \cdot irk}$$

$$Pk5_{i,j} := M_{i,j} \cdot ik5 \cdot \frac{io \cdot \eta_{tr}}{rk} \cdot irk \qquad Pw5_{i,j} := k \cdot F \cdot \left|V5_{i,j}\right|^{2}$$

$$\psi5_{i,j} := f0 \cdot \left[1 + Af \cdot \left|V5_{i,j}\right|^{2}\right]$$

$$P\psi5_{i,j} := Ma1_{i,j} \cdot g \cdot \psi5_{i,j}$$

$$N\psi5_{i,j} := P\psi5_{i,j} \cdot V5_{i,j} \qquad Nw5_{i,j} := Pw5_{i,j} \cdot V5_{i,j}$$

<u>The coefficient that takes into account the effect on the specific fuel consumption of the load engine operation mode</u>

$$KN5_{i,j} := 1.2 + 0.14 \cdot \left(\frac{N\psi5_{i,j} + Nw5_{i,j}}{Nmax \cdot \eta_{tr}}\right) - 1.8\left(\frac{N\psi5_{i,j} + Nw5_{i,j}}{Nmax \cdot \eta_{tr}}\right)^{2} + 1.46 \cdot \left(\frac{N\psi5_{i,j} + Nw5_{i,j}}{Nmax \cdot \eta_{tr}}\right)^{3}$$

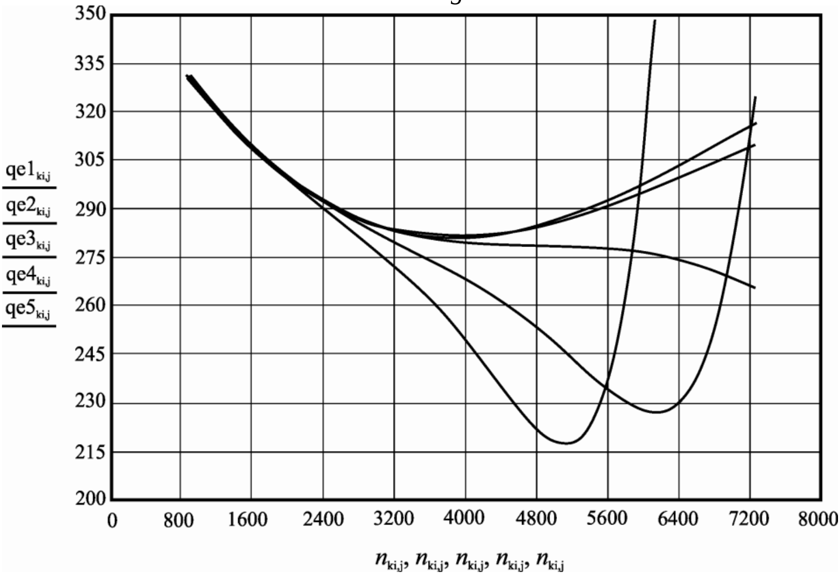
Specific fuel consumption

$$qe5_{i,j} := qeNmax \cdot Kn_{i,j} \cdot KN5_{i,j}$$

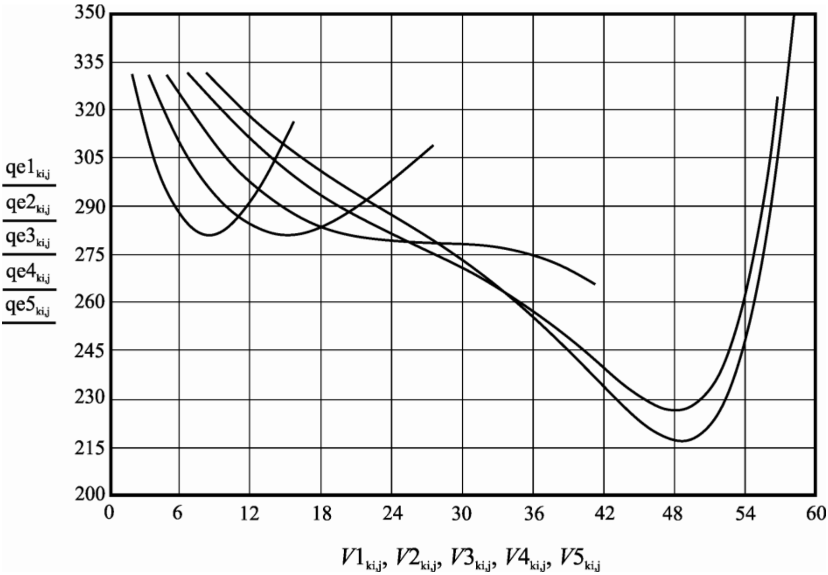
Path (linear) fuel consumption

$$Qs5_{i,j} := \frac{qe5_{i,j} \cdot 10^{-3} \cdot \left|N\psi5_{i,j} + Nw5_{i,j}\right| \cdot 10^{-3}}{V5_{i,j} \cdot 3.6 \cdot \rho t \cdot \eta_{tr}} \cdot 100$$

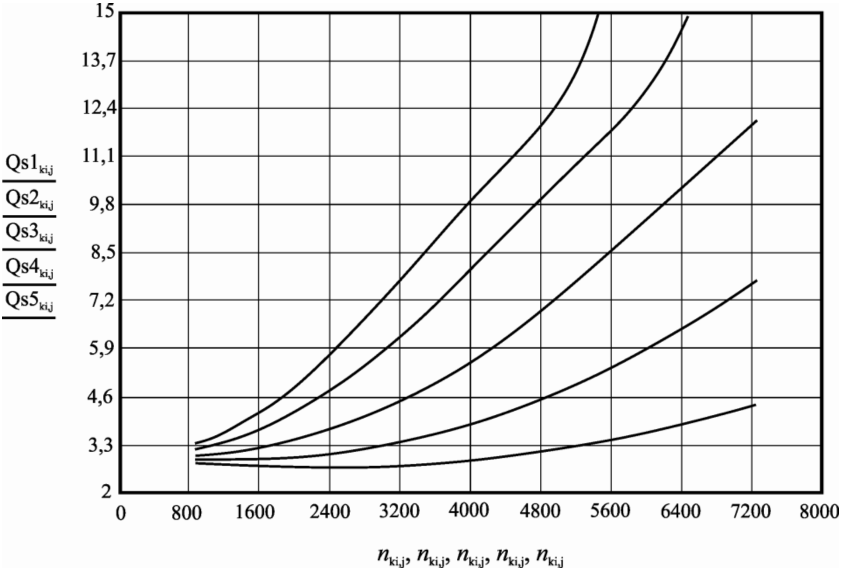
The graph of the dependence of q_e on the crankshaft revolution of Opel Astra Classic engine



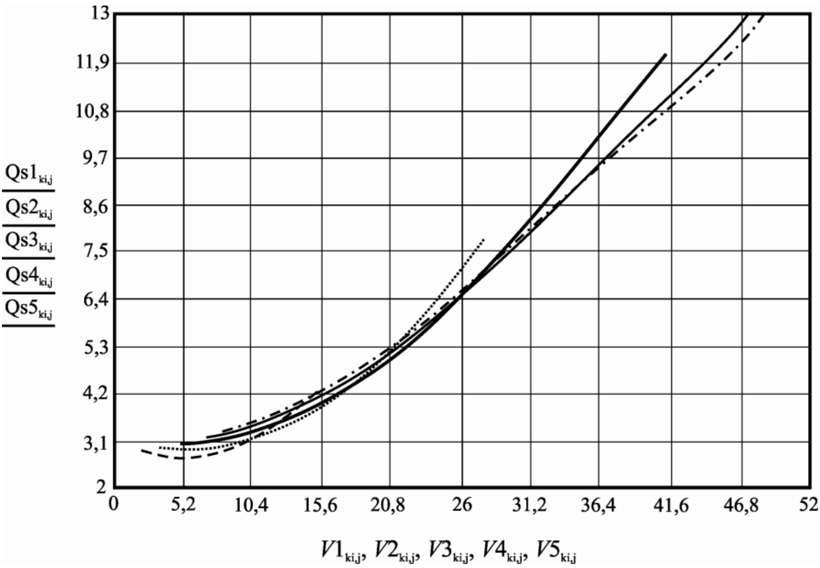
The graph of the dependence of q_e on the speed of Opel Astra Classic



The graph of the dependence of Q_s on the crankshaft revolution of Opel Astra Classic



The graph of the dependence of Q_s on the speed of Opel Astra Classic



REFERENCES

1. Avtomobil'nyj spravochnik [Automobile manual]. Bosch. – M. : ZAO KZHI «Za rulem», 2002. – 896 p.

2. Gaevskij V. V. Metodicheskie ukazaniya k prakticheskim zanyatiyam po discipline «Teoriya ekspluatacionnyh svojstv ATS» dlya studentov special'nosti «avtomobili i avtomobil'noe hozyajstvo» [Methodical instructive regulations for practical classes in the discipline "Theory of operational properties of ATE" for students of a specialty "Automobiles and automotive facilities"] / V. V. Gaevskij, A. M. Ivanov. – Moskva : Rotaprint MADI (GTU), 2007. – 54 p.

3. Gladkij P. P. Ekspluatacionnye svojstva avtotransportnyh sredstv. Metodicheskie ukazaniya. Laboratornye raboty. Napravlenie podgotovki 23.04.03 «Ekspluataciya transportno-tekhnologicheskih mashin i kompleksov» [Operational properties of vehicles. Methodical instructive regulations. Laboratory work. Direction of training 23.04.03 "Operation of transport and technological machines and complexes"] / P. P. Gladkij. – Stavropol' : Federal'noe gosudarstvennoe avtonomnoe obrazovatel'noe uchrezhdenie vysshego professional'nogo obrazovaniya «Severo-Kavkazskij federal'nyj universitet», 2016. – 125 p.

4. GOST 20306-90 Avtotransportnye sredstva. Toplivnaya ekonomichnost'. Metody ispytanij [Vehicles. Fuel efficiency. Test methods]. – Vved. 01.01.92. – M. : Izd-vo standartov, 1991. – 33 p.

5. GOST 22576-90 (ST SEV 6893-89). Avtotransportnye sredstva. Skorostnye svojstva. Metody ispytanij [Vehicles. Speed properties. Test methods]. – Vved. 01.01.92. – M. : Izd-vo standartov, 1991. – 14 p.

6. D'yakov I. F. Teoriya avtomobilya. Elementy raschyota tekhniko-ekspluatacionnyh svojstv avtomobilya. Uchebnoe posobie [Vehicle theory. Elements of calculation of technical and operational properties of the vehicle. Textbook] / I. F. D'yakov. – 2-e uzd., pererab. – Ul'yanovsk : UlGTU, 2002. – 99 p.

7. Litvin O. O. Tyagovij rozrahunok avtomobilya. Metodichni vkazivki dlya kursovogo proektu z disciplini "Avtomobili" dlya pidgotovki zdobuvachiv vishchoї osviti za special'nistyu 274 - "Avtomobil'nij transport" [Traction calculation of the vehicle. Methodical instructive regulations for the course project in the discipline "Vehicles" for the preparation of higher education students in the specialty 274 - "Automotive facilities"] / Ukl.: O. O. Litvin, Ya. V. Kuzhel'nij, V. M. Sklyar. – Chernigiv : CHNTU, 2019. − 48 p.

8. Litvinov A. N., Farobin Ya. E. Avtomobil': teoriya ekspluatacionnyh svojstv [Automobile: Theory of operation properties] / A. N. Litvinov, Ya. E. Farobin. – M. : Mashinostroenie, 1986. – 304 p.

9. Metodychni vkazivky do dyplomnoho ta kursovoho proektuvannia z kursu «Teoriia avtomobilia» dlia studentiv spetsialnostei «Avtomobili ta avtomobilne hospodarstvo» i «Kolisni ta husenychni transportni zasoby» vsikh form navchannia [Methodical instructive regulations for diploma and course works of the course "Vehicle Theory" for students of specialty "Automobiles and automotive facilities" and "Wheeled and Tracked Vehicles" of all forms of education] / uklad.: V. M. Velykodnyi, N. V. Pavlii, A. H. Mamontov. – Kharkiv : NTU "KhPI", 2009. – 40p.

10. Miroshnichenko A. N. Osnovy teorii avtomobilya i traktora [Fundamentals of vehicle and tractor theory] : uchebnoe posobie / A. N. Miroshnichenko. – Tomsk : Izd-vo Tom. gos. arhit.-stroit. un-ta, 2014. − 490 p.

11. Rodionov L. F. Proektirovochnyj tyagovyj raschyot avtomobilya [Design traction calculation of avehicle] ucheb.-metod. posobie / L. F. Rodionov, E. R. Shadyev. – Samara : Samar. gos. tekhn. un-t, 2008. – 58 p.

12. Rukteshel' O. S. Vybor parametrov i ocenka tyagovo-skorostnyh i toplivno-ekonomicheskih svojstv avtomobilya [Selection of parameters and assessment of traction and speed, fuel and economic properties of the vehicle] : uchebno-metodicheskoe posobie dlya studentov special'nostej 1-37 01 02 "Avtomobilestroenie", 1-37 01 06 "Tekhnicheskaya ekspluataciya

avtomobilej", 1-37 01 07 "Avtoservis", 1-44 01 02 "Organizaciya dorozhnogo dvizheniya" / O. S. Rukteshel'. – Minsk : BNTU, 2015. − 80 p.

13. Tyagovaya dinamika i toplivnaya ekonomichnost' avtomobilya s mekhanicheskoj transmissiej [Traction dynamics and fuel efficiency of a vehicle with a manual gearbox]. Uchebno-metodicheskoe posobie dlya studentov special'nostej T 04.02 «Ekspluataciya transportnyh sredstv», T 04.03 «Organizaciya dvizheniya i upravleniya na transporte», T 04.06 «Avtomobili» / O. S. Rukteshel', B. V. Bobrovskij, M. S. Lebedev, L. A. Moliboshko, S. G. Yakutovich. – Minsk : Izd-vo BGPA, 2000. – 68 p.

14. Tyagovyj i toplivno-ekonomicheskij raschyot avtomobilya [Traction, fueland economic calculation of the vehicle]: metodicheskie ukazaniya / V. M. Melisarov, A. V. Brusenkov, P. P. Bespal'ko. – Tambov : Izd-vo Tamb. gos. tekhn. un-ta, 2009. – 28 p.

15. Uchebnoe posobie po discipline «Konstrukciya i ekspluatacionnye svojstva avtomobilej» [Textbook on the discipline "Construction and operational properties of vehicles"] / A. M. Zhigalov, O. L. Kovalenko. – Arhangel'sk : Severnyj (Arkticheskij) federal'nyj universitet imeni M. V. Lomonosova, 2016. – 39 p.

16. Fal'kevich B. S. Teoriya avtomobilya [The theory of the vehicle]. Uchebnoe posobie / B. S. Fal'kevich. – 2-e izd., pererab. – M. : Mashgiz, 1963. – 240 p.

17. Husainov A. Sh. Teoriya avtomobilya [The theory of the vehicle]: konspekt lekcij / A. Sh. Husainov, V. V. Selifonov. – Ul'yanovsk : UlGTU, 2008. – 121 p.

18. Husainov A. Sh. Tyagovyj raschyot avtomobilya [Traction calculation of the vehicle]: uchebnoe posobie po discipline «Teoriya avtomobilya» dlya studentov obuchayushchihsya po special'nosti 19020165 – Avtomobile- i traktorostroenie / A. Sh. Husainov. – Ul'yanovsk : UlGTU, 2009. – 47 p.

19. Chernyshev V. A. Tyagovo-dinamicheskij i toplivno-ekonomicheskij raschyot avtomobilya [Traction and dynamic, fuel and economic calculation of the vehicle]: metodicheskie rekomendacii po vypolneniyu kursovoj raboty / V. A. Chernyshev. – M. : MGAU, 2002. – 39 p.

FOR NOTES

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Навчальне видання

ТЯГОВО-ДИНАМІЧНИЙ ТА ПАЛИВО-ЕКОНОМІЧНИЙ РОЗРАХУНОК АВТОМОБІЛЯ

Методичні вказівки
для виконання розрахункових, курсових
та дипломних робіт
кваліфікаційного рівня бакалавр та магістр
для студентів спеціальності
274 «Автомобільний транспорт»
всіх форм навчання

Укладачі: МАМОНТОВ Анатолій Геннадійович
ШЕВЦОВ Вадим Михайлович
РЕЗВАЯ Ксенія Сергіївна

Відповідальний за випуск доц. А. Г. Мамонтов
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