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## OSCILLATION PROCESSES IN A TRANSMISSION WITH A DUAL-MASS FLYWHEEL WHILE MOVING A CAR FROM REST

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**Abstract.** *Problem statement.* A wide range of modern engines, gearboxes, and mass-dimension characteristics of a car requires the development of efficient algorithms and methods of designing dual mass flywheels (DMFs) for the given transmission parameters. Improper selection of the design parameters of the DMF can lead to a deterioration of its properties and, consequently, to the increase of vibrations, reduction of the lifetime of the gearboxes, etc. Equally important is the problem of the DMF durability, which depends on many factors, in particular, the character of driving the car by the driver. The solution of the two mutually contradictory tasks formulated above is closely linked to the creation of simulation models of the car drive with a DMF, which will allow simulating different modes of the transmission operation, in order to optimize the parameters of the DMF to the specific design parameters of the car drive. *The purpose.* Taking into account the design features of the transmission and DMF, to develop and substantiate dynamic, mathematical, and stimulating models of the car drive with a DMF and to study the oscillation processes while moving a car from rest. *Research methods.* To construct a mathematical model of a car drive with a DMF, the Lagrange method of the 2nd kind was applied. To solve the differential equations of the car motion during acceleration and simulate oscillation processes in the transmission and DMF sections, we used numerical methods with their implementation in the MatLab Simulink environment. *Results.* A simulation model of a front-wheel car drive with a dual-mass flywheel has been developed in the Simulink environment. The oscillation processes in the transmission and elastic elements of the DMF by the case of moving the car of category M1 from rest were investigated. Recommendations were made to increase the lifetime of the DMF. *The novelty.* It has been found that the use of a DMF contributes to reducing the dynamic load applied to the drive sections, absorbing the vibration energy generated by the engine. The amplitude of torque oscillations in the transmission sections, semi-axles, and tires is approximately 1.4 times smaller in the DMF drive than in the single-mass flywheel drive. Increasing the energy dissipation coefficient in the DMF from 4 to 20 N·s·m leads to a decrease in the torque amplitude in the drive sections by 1.3–1.6 times at the beginning of moving the car from rest, reducing the duration of transient processes from 2 to 0.75 s, which eliminates the overloading of DMF elastic sections, increasing their lifetime. *The practical value.* A simulation model of a car drive with a DMF during the period of moving the car from rest has been developed, which makes it possible to simulate the influence of the design parameters of the drive and DMF on the loading of the elastic sections and to calculate their strength under long-term cyclic loading. *Areas for further research.* To investigate oscillation processes in the DMF and transmission sections under different modes of the car movement in order to determine ways to increase the DMF lifetime by reducing the influence of cyclic loading on their elastic sections.

**Keywords:** dual-mass flywheel, car drive, transmission, oscillation process, dynamic loading.

## **Introduction**

The rapid development of cars in recent years has led to the creation of powerful and high-torque engines. The requirements for driver and passenger comfort have also increased. However, equipping modern cars with high-power low-speed diesel engines has led to increased vibration and noise. Traditional flywheels with cylindrical springs do not provide effective damping of the torsional vibrations in the transmission over a wide range of speeds. Already with the existing clutch mounting space, and even more so when it is reduced, the classic damper in the clutch disc is unable to transmit increasing torques and therefore loses its purpose. The way out of this situation was the introduction of a dual-mass flywheel (DMF), which is structurally divided into two parts, one of which is rigidly connected to the crankshaft and the other to the input shaft of the gearbox (GB). Both parts of the DMF are interconnected by means of thrust and radial bearings with the possibility of relative rotation. An elastic damping system is installed between the housings to absorb the vibrations generated by the engine and to smoothly move a car from rest.

## **Problem Statement**

A wide range of modern engines, gearboxes, and mass-dimension characteristics of a car requires the development of efficient algorithms and methods of designing dual mass flywheels for the given transmission parameters. Improper selection of the design parameters of the dual-mass flywheel can lead to a deterioration of its properties and, consequently, to the increase of vibrations, reduction of the lifetime of the gearboxes, etc. Equally important is the problem of the DMF durability, which depends on many factors, in particular, the character of driving the car by the driver [1]. Intensive acceleration or deceleration by the engine leads to alternating cyclic loading of the DMF elastic elements, and thus accelerates their destruction and failure of the flywheel as a whole. The solution of the two mutually contradictory tasks formulated above is closely linked to the creation of simulation models of the car drive with a DMF, which will allow simulating different modes of the transmission operation, in order to optimize the parameters of the DMF to the specific design parameters of the car drive.

## **Review of Information Sources**

A number of papers [1–10] are devoted to the study of DMFs, in particular, they relate to the analysis of design features, causes of faults, the influence of cyclic loading on elastic elements, modeling of systems with a DMF using various software systems and calculation of the main structural elements of a dual-mass flywheel.

In the work [11], a model of friction clutch with a hydropneumatic actuator was developed, which contains the following structural Simscape models: the friction clutch model with a membrane spring of a petal-type, the pneumatic clutch booster model, the first gear transmission model. They are made in the software complex MatLab Simulink. In the work [12], a structural diagram and a mathematical model for the calculation of dynamic loads arising in the transmission of a car with a hybrid power unit are proposed. Moving a car from rest was investigated and found that the highest loads of the transmission elements, in this case, occur in the semi-axles, tires and elastic suspension elements. In the work [13], a developed mathematical model of the motion when moving a car from rest is divided into three stages, each of which is described by a corresponding system of ordinary differential equations, depending on the drive structure.

The analysis of the literature sources shows that insufficient attention is paid to the simulation of oscillation processes in the transmissions of cars with dual-mass flywheels, in particular, the influence of modes of the car movement on the loading of elements of the transmission and DMF. Therefore, the development of simulation models of the car drive with a DMF and the study of transients in the transmission is an urgent task and is of great practical importance.

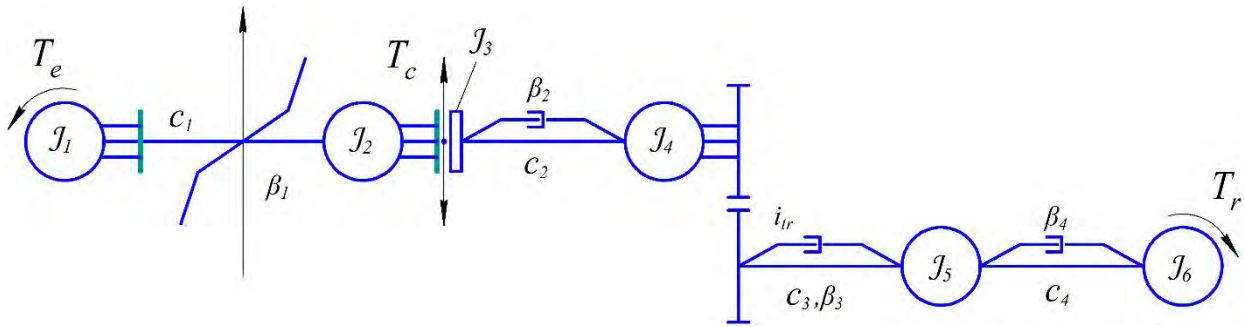
## **Objectives of the Research**

Taking into account the design features of the transmission and DMF, to develop and substantiate dynamic, mathematical, and stimulating models of the car drive with a DMF and to study the oscillation processes while moving a car from rest.

### Main Material Presentation

**1. A dynamic model of a front-wheel drive car with a DMF.** Dual-mass flywheels are used in transmissions of lorries, cars, and commercial vehicles with front- or rear-wheel drive. The elastic-damping properties of such flywheels are determined by their structure [2] and design parameters. Therefore, to build a dynamic model of the car drive with a DMF it is necessary to carry out the distribution and consolidation of masses and stiffness of the drive elements.

Given the structure of the drive of the front-wheel drive car, we will present its generalized dynamic model in the form shown in Fig. 1.



**Fig. 1.** A dynamic model of the drive of the front-wheel drive car with a DMF when slipping the clutch

Here  $J_1$  is the consolidated moment of inertia of the moving sections of an internal combustion engine (ICE): crankshaft, pistons, connecting rods and the DMF primary mass,  $J_2$  is the consolidated moment of inertia of the DMF secondary mass and the clutch masses coupled to it,  $J_3$  is the consolidated moment of inertia of the clutch disc and the masses of the primary GB shaft sections,  $J_4$  is the consolidated moment of inertia of the secondary GB shaft and the main gear sections,  $J_5$  is the consolidated moment of inertia of the hubs and rims of the wheels,  $J_6$  is the moment of inertia of the mass of the car consolidated to the wheel;  $c_1$ ,  $c_2$ ,  $c_3$ , and  $c_4$  are consolidated stiffness of the DMF elastic elements, GB sections, semi-axles, and tires respectively;  $b_1$ ,  $b_2$ ,  $b_3$ , and  $b_4$  are energy dissipation coefficients in the DMF, GB, transmission, and tires respectively;  $j_1$ ,  $j_2$ ,  $j_3$ ,  $j_4$ ,  $j_5$ , and  $j_6$  are the angles of rotation of the masses  $J_1$ ,  $J_2$ ,  $J_3$ ,  $J_4$ ,  $J_5$ , and  $J_6$  respectively;  $T_e$  is torque of the ICE;  $T_c$  is the friction moment of the clutch;  $T_r$  is the moment of resistance to the car motion, consolidated to the wheels.

The consolidated stiffness of DMF elastic elements will be presented in the form of the stiffness of an equivalent torsion shaft [3]:

$$c_1 = 2c_{sp}R^2, \quad (1)$$

where  $c_{sp}$  is the linear stiffness of DMF springs mounted on a circle of radius  $R$ .

**2. Mathematical model of a car drive with a DMF.** The process of moving a car from rest involves two stages. The first one is the attachment of the transmission to the running engine using a slipping clutch, and the second one is the acceleration of the car with the clutch closed. Applying the Lagrange equation of the 2nd kind, after the corresponding transformations we obtain the following system of differential equations of the masses motion for the dynamic model of the car drive with a DMF (Fig. 1) when moving a car from rest in the first stage:

$$\begin{cases} \ddot{j}_1 = T_e - c_1(j_1 - j_2) - b_1(\dot{j}_1 - \dot{j}_2); \\ \ddot{j}_2 = c_1(j_1 - j_2) + b_1(\dot{j}_1 - \dot{j}_2) - T_c; \\ \ddot{j}_3 = T_c - c_2(j_3 - j_4) - b_2(\dot{j}_3 - \dot{j}_4); \\ \ddot{j}_4 = c_2(j_3 - j_4) + b_2(\dot{j}_3 - \dot{j}_4) - c_3(j_4/i_{tr} - j_5) - b_3(\dot{j}_4/i_{tr} - \dot{j}_5); \\ \ddot{j}_5 = c_3(j_4/i_{tr} - j_5) + b_3(\dot{j}_4/i_{tr} - \dot{j}_5) - c_4(j_5 - j_6) - b_4(\dot{j}_5 - \dot{j}_6); \\ \ddot{j}_6 = c_4(j_5 - j_6) + b_4(\dot{j}_5 - \dot{j}_6) - T_r. \end{cases} \quad (2)$$

The torque of an ICE on a steady-state of operation will be presented in the form of a periodic function [10]:

$$T_e = T_{const} + T_{var} \times \sin(\omega t + \mathbf{a}), \quad (3)$$

where  $T_{const}$  is a constant component of the ICE torque;  $T_{var}$  is a variable component of the ICE torque;  $\omega$  is the circular oscillation frequency of the variable component of the ICE torque;  $\mathbf{a}$  is the phase shift.

The friction moment of the clutch during slipping:

$$T_c = f \times P_{cl} \times R_T \times \text{sign}(j_1 - j_2), \quad (4)$$

where  $f$  is the coefficient of friction of the clutch;  $P_{cl}$  is the clamping force applied to the friction surfaces of the clutch by a pressure device;  $R_T$  is the friction radius.

The moment that transmits the clutch in the closed state,  $T_{cl\ max} = \beta \cdot T_{e\ max}$ , where  $\beta$  is the clutch reserve factor.

The moment of resistance to the car motion, consolidated to the wheels, in general form will be given by a function:

$$T_r = G_a \times r \times \left( f_0 + k_f (j_{\omega} \times r)^2 \right) \cos \mathbf{g} + \sin \mathbf{g} \dot{\mathbf{g}}, \quad (5)$$

where  $G_a$  is the car weight,  $r$  is the dynamic radius of the wheel,  $f_0$  is the rolling resistance coefficient at low motion speed,  $k_f$  is the factor that takes into account the increase in rolling resistance with increasing the car speed,  $\mathbf{g}$  is the road slope angle,  $j_{\omega}$  is the angular velocity of the car wheels.

In the second stage of moving a car from rest, when the friction moment in the clutch reaches its maximum value  $T_{cl\ max}$ , the drive structure changes (Fig. 2).

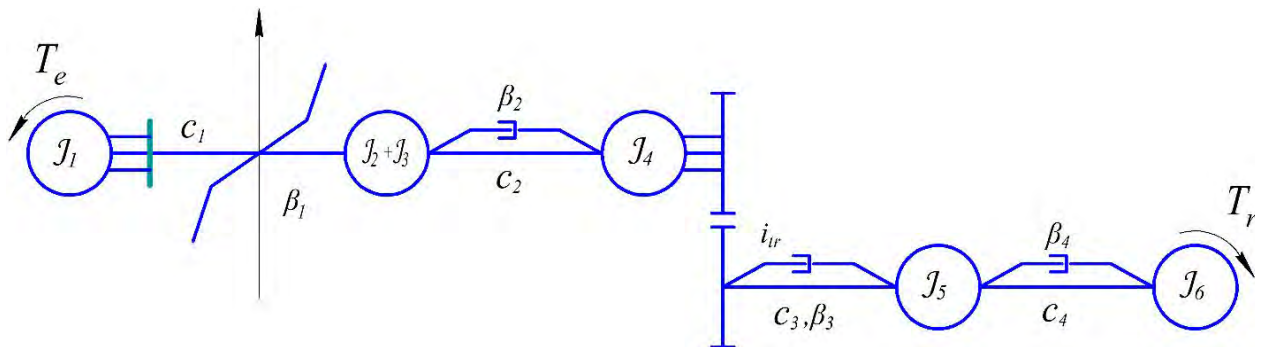


Fig. 2. A dynamic model of the front-wheel drive car with a DMF after the clutch is closed

The differential equations that describe the car acceleration after the clutch is closed will look like:

$$\begin{cases} J_1 \dot{j}_1 = T_e - c_1(j_1 - j_2) - b_1(j_1 - j_2); \\ (J_2 + J_3) \dot{j}_2 = c_1(j_1 - j_2) + b_1(j_1 - j_2) - c_2(j_2 - j_3) - b_2(j_2 - j_3); \\ J_4 \dot{j}_4 = c_2(j_3 - j_4) + b_2(j_3 - j_4) - c_3(j_4/i_{tr} - j_5) - b_3(j_4/i_{tr} - j_5); \\ J_5 \dot{j}_5 = c_3(j_4/i_{tr} - j_5) + b_3(j_4/i_{tr} - j_5) - c_4(j_5 - j_6) - b_4(j_5 - j_6); \\ J_6 \dot{j}_6 = c_4(j_5 - j_6) + b_4(j_5 - j_6) - T_r. \end{cases} \quad (6)$$

The stress in the turns of DMF springs in the course of dynamic loading during car acceleration will be determined by the formula [14]:

$$t_{\max} = \frac{16 \times (j_1 - j_2) \times c_{is} \times R \times R_{iav}}{\rho \times d_i^3} \times \frac{4m-1}{8m-4} + \frac{0,615 \ddot{\theta}}{m \times \varnothing} \quad (7)$$

where  $c_{is}$  is the linear stiffness of the  $i$ -th spring,  $R$  is the radius of mounting of the  $i$ -th spring,  $R_{iav}$  is the average radius of the  $i$ -th spring,  $d_i$  is the wire diameter of the  $i$ -th spring,  $m = 2R_{iav} / d_i$  is the correction factor for the  $i$ -th spring.

The above dependences (1) – (7) display a mathematical model of the car drive with a DMF in the course of moving a car from rest, the implementation of which allows to study the change in the values of dynamic loads in the DMF elastic elements and drive shafts under different acceleration modes.

**3. Simulation of oscillation processes in a car drive with a DMF and their analysis.** As an example of the implementation of a mathematical model of a car drive with a DMF in the period of moving a car from rest, the car of category M1 of a class Volkswagen Caddy is considered. The brief technical specification of the car is given in Table 1.

Table 1

Technical specification of Volkswagen Caddy car

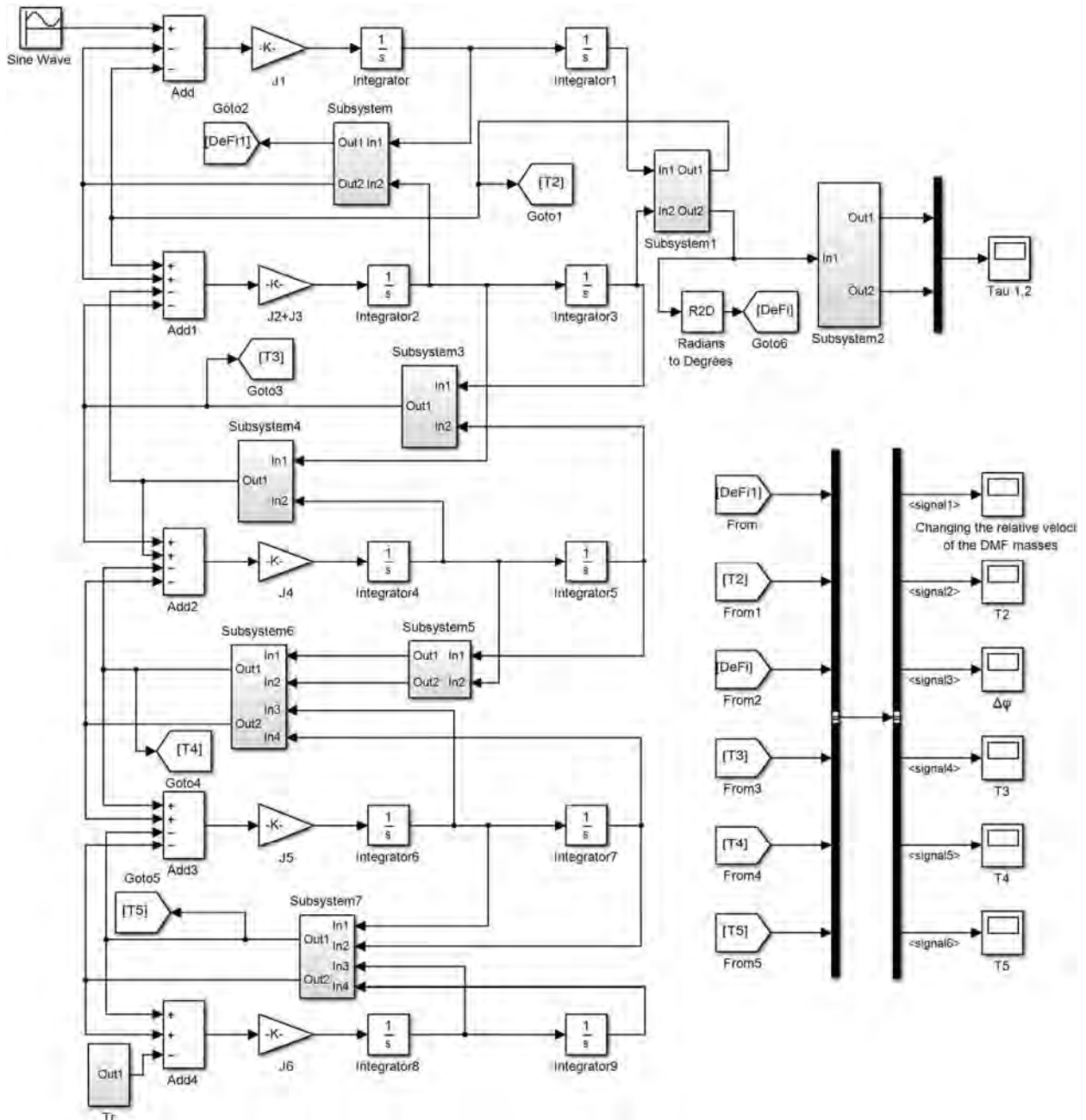
Parameter	Value
Equipped weight, kg	1336
Maximum weight, kg	2066
Maximum weight of luggage, kg	730
Engine power, kW/ rpm	75/4000
Maximum torque, N·m/rpm	210/1900
Tire size	195/65R15
Mechanical gearbox type	a five-gear
The gear ratio of the main gear	3.94
The gear ratio of the first gear of GB	3.78

The parameters of the main parts of the DMF of the car, measured in the Research laboratory of the study of automobiles operating properties of the Department of operation and repair of automotive vehicles at Lviv Polytechnic National University, are as follows: the total weight of the flywheel assembled is 12 kg, the weight of the leading part of the flywheel is 6.4 kg, and the weight of the driven one is 5.6 kg; the diameter of a large spring  $D_1 = 25$  mm; diameter of a large spring wire  $d_1 = 4$  mm; the diameter of small spring  $D_2 = 16$  mm; the diameter of a small spring wire  $d_2 = 3$  mm; the number of turns of a large spring  $n_1 = 47$ ; the number of turns of a small spring  $n_2 = 66$ ; the diameter of the leading disk (primary mass of the DMF)  $D_0 = 280$  mm; the diameter of the driven disk (secondary mass of the DMF)  $d_0 = 270$  mm; the radius of the circle on which the large and small springs are mounted,  $R = 120$  mm.

As a result of the calculations, it is found that the stiffness of the large spring  $c_{ls} = 5881$  N/m and the small one  $c_{ss} = 5586$  N/m. The consolidated moment of inertia of the moving sections of the ICE  $J_1 = 1.2$  kg·m<sup>2</sup>; the consolidated moment of inertia of the DMF secondary mass and the clutch masses coupled to it  $J_2 = 0.1$  kg·m<sup>2</sup>; the consolidated moment of inertia of the GB moving sections  $J_3 = 0.1$  kg·m<sup>2</sup>, the consolidated moment of inertia of the secondary GB shaft and the main gear sections  $J_4 = 0.2$  kg·m<sup>2</sup>; the consolidated moment of inertia of the hubs and rims of the wheels  $J_5 = 0.18$  kg·m<sup>2</sup>; the moment of inertia of the mass of the car consolidated to the wheel  $J_6 = 173.8$  kg·m<sup>2</sup>. Total torsional stiffness: DMF elastic elements  $c_1 = 330$  N·m/rad; GB sections  $c_2 = 120000$  N·m/rad; semi-axles  $c_3 = 84000$  N·m/rad; tires  $c_4 = 27000$  N·m/rad. Energy dissipation coefficients: DMF sections  $b_1 = 4$  N·s·m; GB sections  $b_2 = 0.02$  N·s·m; semi-axles  $b_3 = 0.01$  N·s·m; tires  $b_4 = 800$  N·s·m. The constant component of the engine torque  $T_{const} = 120$  N·m; the variable component of the engine torque  $T_{var} = 60$  N·m; the circular oscillation frequency of the variable component of the torque ICE  $\omega = 100$  s<sup>-1</sup>. The road slope angle  $g = 0^\circ$ . The gear ratio of the transmission  $i_{tr} = 14.89$ .

**3.1. Simulation model.** An appropriate simulation model was developed in the Simulink environment to calculate the dynamic loads applied to the DMF elastic sections and the drive shafts while moving a car from rest (Fig. 3).

Using the developed simulation model, the processes of emergent move a car from rest on a dry asphalt road were investigated with the sharp release of the clutch after engaging the first gear and various design parameters of a DMF, as well as with the traditional single-mass flywheel and similar other parameters of the model. To evaluate the strength of the DMF elastic elements, a change in the stresses in the DMF spring turns was calculated.

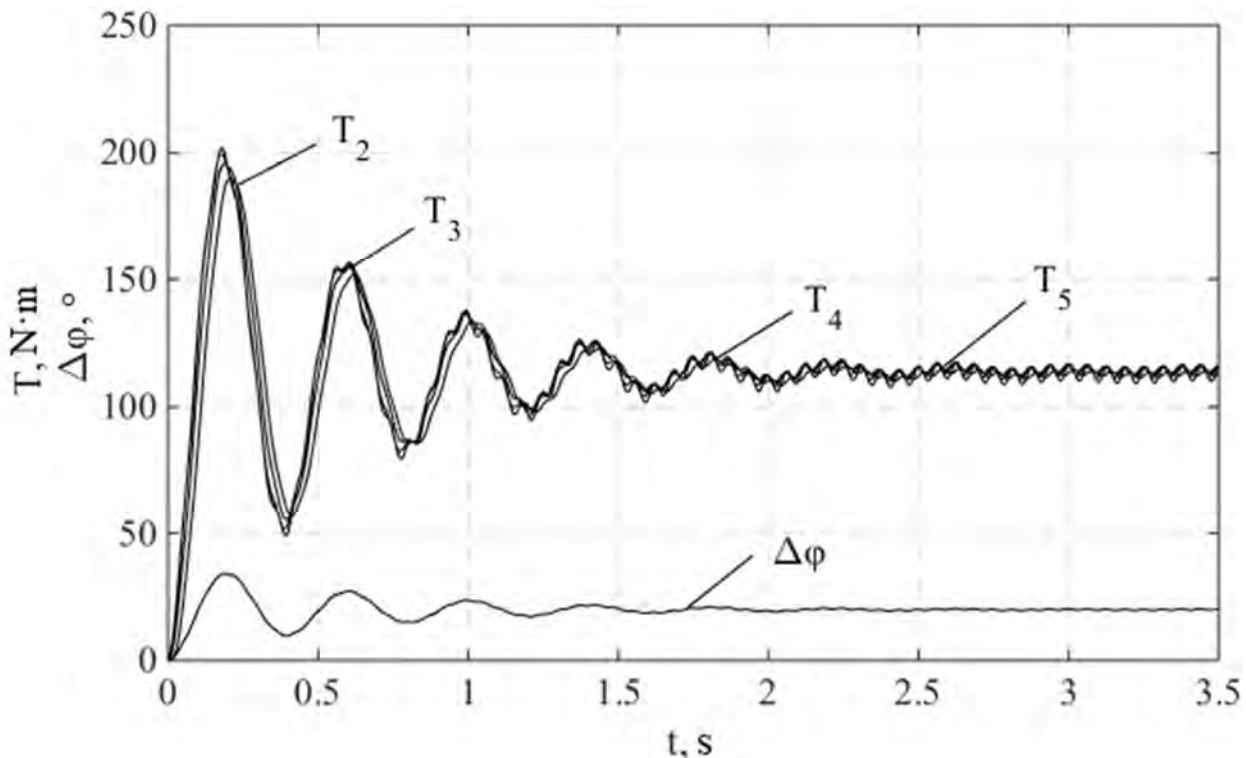


**Fig. 3.** A simulation model in the Simulink environment for calculating dynamic loads applied to the car drive sections

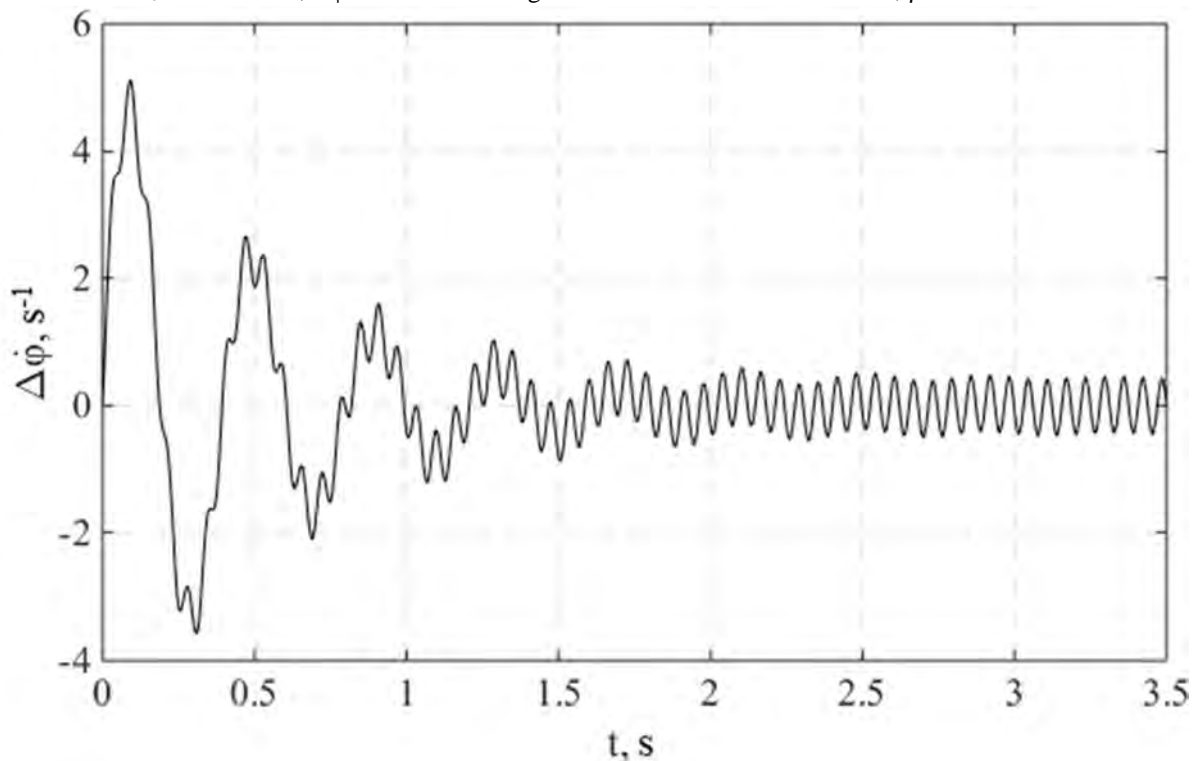
**3.2. Simulation results and analysis.** The results of the simulation of oscillation processes in the car transmission while moving a car from rest are shown in Fig. 4–11. The simulation was performed for two types of car drive: with a DMF and with a single-mass flywheel. The rest of the drive parameters were considered the same.

From Fig. 4–9 we see that in the car drive with a DMF, when the clutch is suddenly engaged, the torque in all the sections of the transmission in the initial stage of acceleration increases sharply and has an oscillation character. Thus, when the energy dissipation coefficient in a DMF is  $\beta_1 = 4 \text{ N}\cdot\text{s}\cdot\text{m}$ , the torques in the DMF and GB sections and semi-axes and tires reach 200 N·m, and the relative turning angle of the primary and secondary masses in a DMF is 34 degrees. After 2.5 s, the torque value in the drive sections stabilizes at 115 N·m and the relative turning angle of the DMF masses is at the level of 20 degrees. Further oscillations of the considered parameters are insignificant. However, as can be seen from Fig. 5, the relative velocity of the DMF mass oscillations has an alternating character, resulting in cyclic loading of the DMF

springs and their fatigue over the car operating life. The actual stresses in the small DMF spring initially exceed the permissible stress for the spring material, whereas relatively smaller stresses occur in the turns of the large spring (Fig. 6). After 1.5 s, the stresses in the turns of both springs decrease and subsequently change cyclically with a small amplitude.



**Fig. 4.** Changing torque in time:  $T_2$  – in DMF sections,  $T_3$  – in GB sections,  $T_4$  – in the semi-axes,  $T_5$  – in the tires;  $\Delta\varphi$  is the relative angle of rotation of the DMF masses;  $\beta_1 = 4 \text{ N}\cdot\text{s}\cdot\text{m}$



**Fig. 5.** Changing the relative velocity of the DMF masses in time;  $\beta_1 = 4 \text{ N}\cdot\text{s}\cdot\text{m}$

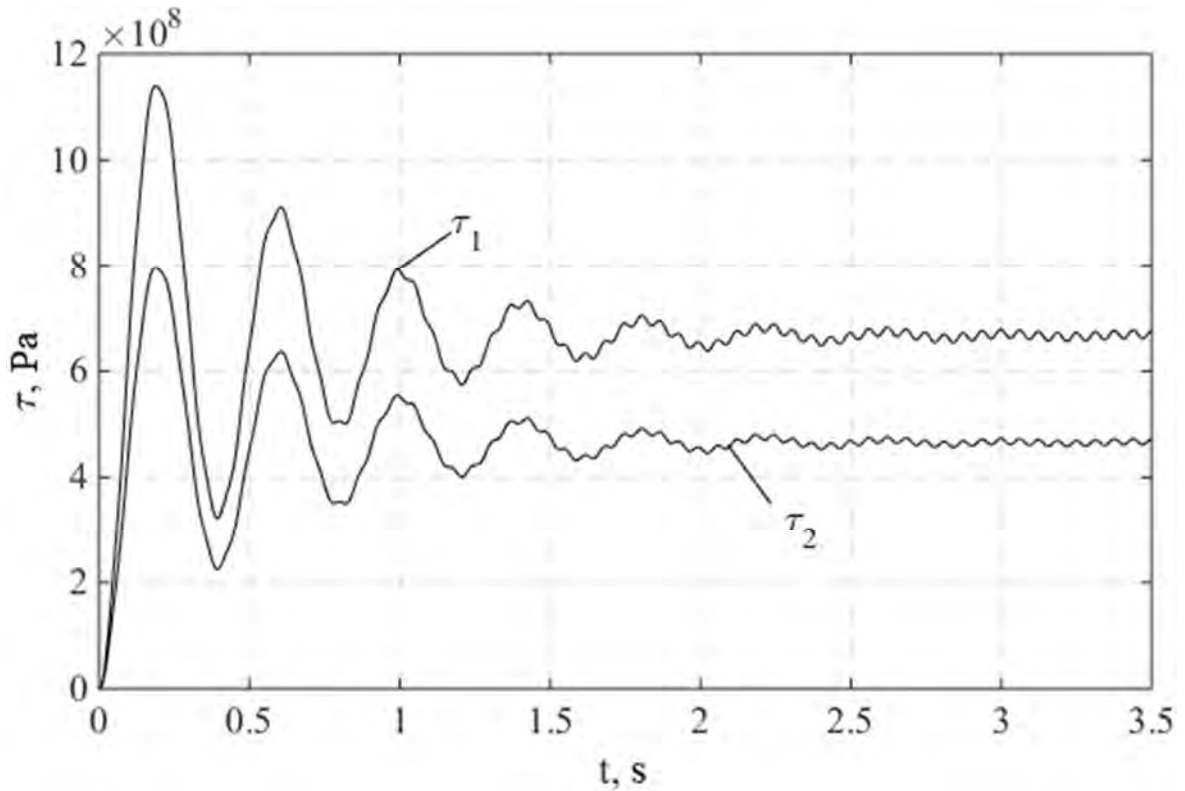


Fig. 6. Changing the stresses in DMF sections in time:  $\tau_1$  and  $\tau_2$  are the stresses in the small and large springs respectively;  $\beta_1 = 4 \text{ N}\cdot\text{s}\cdot\text{m}$

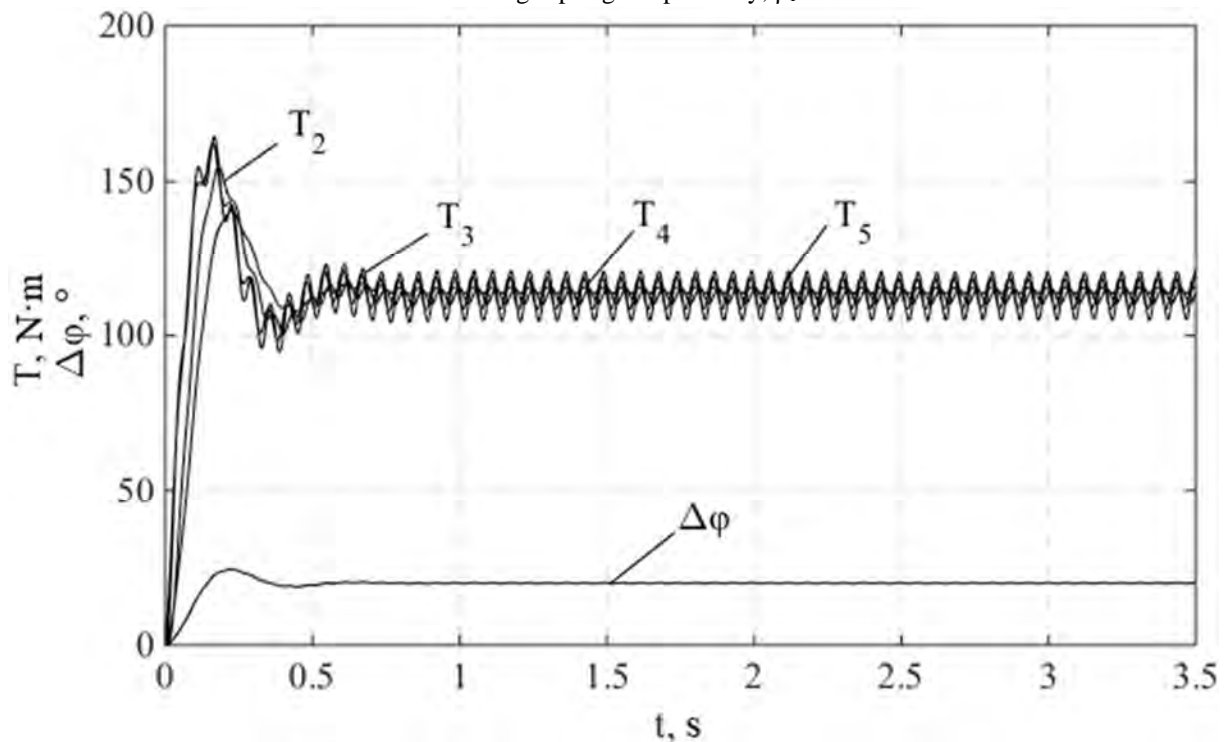


Fig. 7. Changing torque in time:  $T_2$  – in DMF sections,  $T_3$  – in GB sections,  $T_4$  – in the semi-axes,  $T_5$  – in the tires;  $\Delta\varphi$  is the relative turning angle of the DMF masses;  $\beta_1 = 20 \text{ N}\cdot\text{s}\cdot\text{m}$

With the increase of the energy dissipation coefficient in a DMF to  $\beta_1 = 20 \text{ N}\cdot\text{s}\cdot\text{m}$ , the character of the oscillation processes in the DMF and transmission sections does not change. But this leads to a decrease in torque in the drive sections at the initial moment (Fig. 7) and fast damping of oscillations after 0.5 s. The



relative turning angle of the masses in the DMF thus remains at 20 degrees, and the stresses in the turns of the springs do not exceed the permissible values (Fig. 9). However, the amplitude of the torque change in the DMF and drive sections when entering steady-state at  $\beta_1 = 20 \text{ N}\cdot\text{s}\cdot\text{m}$  increases (see Fig. 9) as compared to  $\beta_1 = 4 \text{ N}\cdot\text{s}\cdot\text{m}$  (see Fig. 4). Therefore, an unreasonable increase in the damping capacity of the DMF leads to an increase in cyclic loading of the DMF springs, which reduces its lifetime.

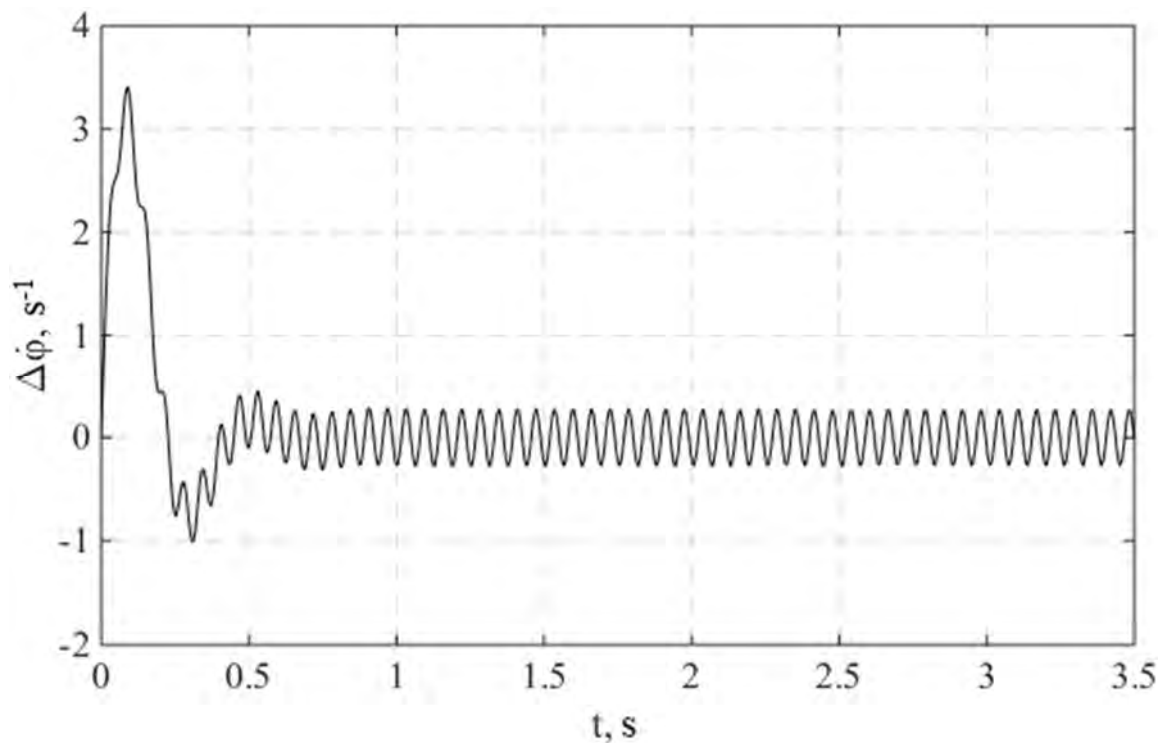


Fig. 8. Changing the relative velocity of the DMF masses in time;  $\beta_1 = 20 \text{ N}\cdot\text{s}\cdot\text{m}$

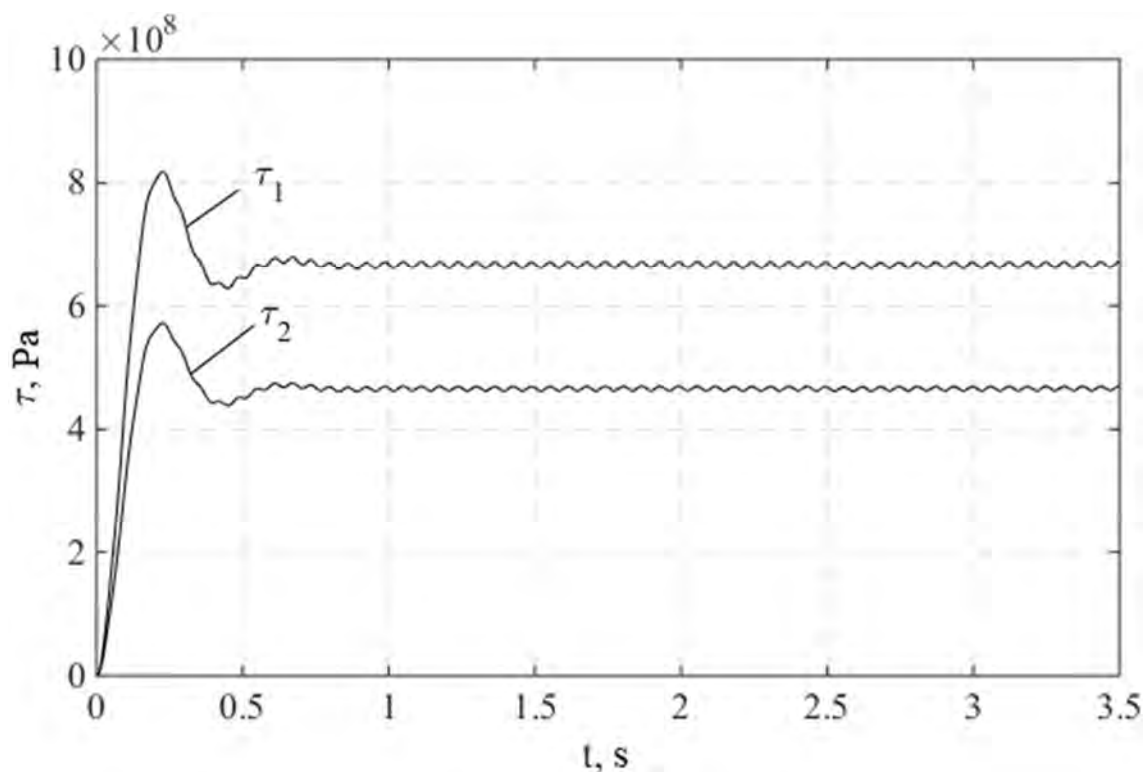
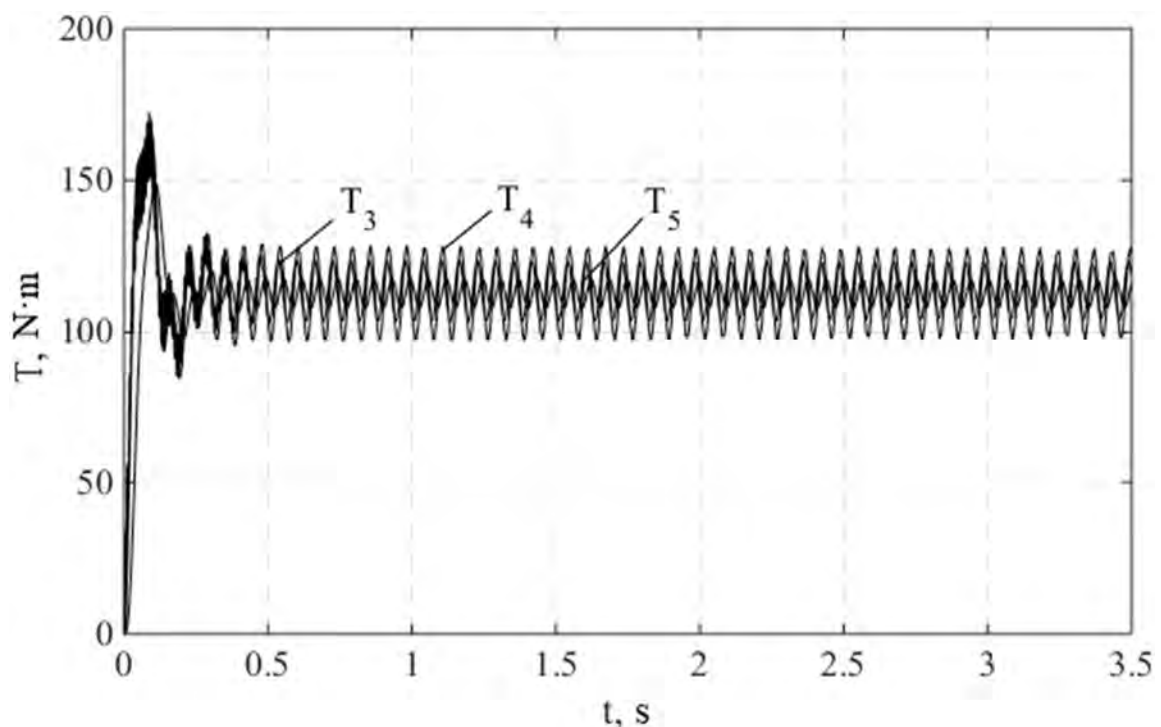


Fig. 9. Changing the stresses in DMF sections in time:  $\tau_1$  and  $\tau_2$  are the stresses in the small and large springs respectively;  $\beta_1 = 20 \text{ N}\cdot\text{s}\cdot\text{m}$

A comparison of the simulation results makes it possible to draw the following conclusions: (1) increasing the viscosity of the lubricant between the primary and secondary DMF masses increases its effectiveness in reducing the load applied to the transmission sections; (2) leakage of the lubricant or decrease in its viscosity due to heat loads in the course of slipping the clutch leads to a deterioration of the damping capacity of the DMF and, consequently, an increase in the load applied to the drive sections.

Despite the relatively smaller amplitude values of the torques in the drive sections and stresses in the turns of the DMF springs, the load applied to the DMF springs changes cyclically after moving a car from rest, even though it decreases in magnitude. The cyclic frequency depends on the rotation speed of the engine crankshaft. This leads to fatigue of the spring material in time and the spring fracture after a certain run of the car. So, the more intensively the car is moving from rest and maneuvering in overtaking, etc., the faster the DMF elastic sections are destroyed and the DMF fails.

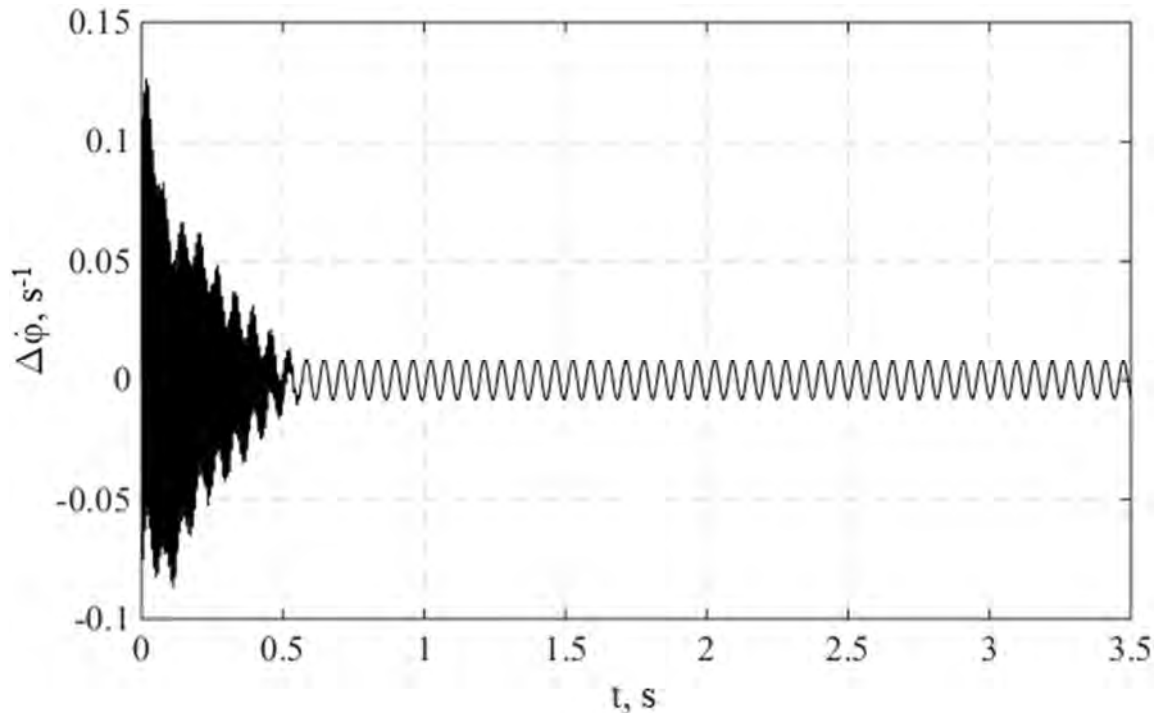
A similar simulation was carried out for a drive equipped with a single-mass flywheel in a car starting from rest with acceleration. The simulation results are shown in Fig. 10–11.



**Fig. 10.** Changing torque in time for a single-mass flywheel drive:  $T_3$  – in GB sections,  $T_4$  – in the semi-axes,  $T_5$  – in the tires

The graphs show that the values of torque in the sections of the drive with a single-mass flywheel and unchanged other parameters increase sharply at the beginning of acceleration (Fig. 10), remaining commensurate with similar parameters for the drive with a DMF. But after 0.25 s the oscillation processes in the drive sections stabilize, and the torque changes cyclically with a sufficiently large amplitude. This leads to the continuous loading of sections of the transmission by the pulsating torque and causes fatigue of its individual elements.

Comparison of the results of the simulation of moving a car from rest, the drives of which are equipped with a DMF and a single-mass flywheel, confirm the effectiveness of the use of a DMF to increase the lifetime of the transmission sections in unsteady modes of the car motion. However, being an absorber of oscillation processes in a car transmission, the DMF is subjected to intense loading, which reduces its lifetime. Therefore, it is urgent to develop design solutions aimed at increasing the DMF lifetime by reducing the susceptibility to the driving style of car drivers. One such design solution may be the use of an adaptive DMF [15].



**Fig. 11.** Changing the relative velocity of masses ( $J_2 + J_3$ ) and  $J_4$  in time for the model of a car drive with a single-mass flywheel

### Conclusions

1. Taking into account the design features of the transmission of the front-wheel drive car, the dynamic, mathematical, and simulation models of the drive with a DMF and a single-mass flywheel when moving a car from rest were developed.

2. The oscillation processes in the sections of the drive with dual- and single-mass flywheels during the period of moving the car from rest were investigated. It has been found that the use of a DMF contributes to reducing the dynamic load applied to the drive sections, absorbing the vibration energy generated by the engine. The amplitude of torque oscillations in the transmission sections, semi-axes, and tires is approximately 2 – 3.7 times smaller in the DMF drive than in the single-mass flywheel drive.

3. Increasing the energy dissipation coefficient in the DMF from 4 to 20 N·s·m leads to a decrease in the torque amplitude in the drive sections by 1.8 times at the beginning of moving the car from rest, reducing the duration of transient processes from 2 to 0.75 s, which eliminates the overloading of DMF elastic sections, increasing their lifetime.

4. To increase the DMF lifetime, design solutions should be developed to increase the lifetime of its elastic sections, reducing their susceptibility to the driving style of car drivers.

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