Vol. 3, No. 2, 2022

Bohdan Kindratskyy, Roman Litvin
Lviv Polytechnic National University
12, S. Bandery Str., Lviv, 79013, Ukraine

https://doi.org/10.23939/tt2022.02.072

THE INFLUENCE OF FRICTION BETWEEN ELEMENTS OF DUAL-MASS FLYWHEEL ON OSCILLATORY PHENOMENA IN A CAR TRANSMISSION

Summery. *Automobile manufacturers, when designing new cars, are increasingly faced with the need to reduce the weight of components in order to achieve the required level of fuel consumption and environmental standards. As a result, internal combustion engines (ICEs) with a small number of cylinders are designed and manufactured, which allows to achieve an increase in output power due to increased pressure in the cylinder and more efficient fuel combustion. As a result of this, torsional vibrations occur on the crankshaft, which are transmitted and negatively affect the transmission, causing it to fail prematurely. The damping properties of dual-mass flywheel (DMF) in a straight line depend on their structure and design parameters. All modern DMF contain a certain amount of thick lubricant, which in one way or another improves its characteristics. But in addition to parts, flywheels that constantly work in an environment with lubricant, there are also elements between which dry friction occurs, which also affects the damping characteristics of the flywheel. Therefore, it can be assumed that its presence affects the elastic-damping properties of the DMF. The purpose of the work is to develop simulation models and study the effect of friction between DMF elements on oscillatory processes in the car transmission and to develop recommendations for reducing the load on DMF elements and transmission links. The effect of dry and viscous friction between the elements of a DMF on the damping of oscillations in the links of its elastic-damping system and the drive links of the car was studied. It is shown that an increase in the coefficient of dry friction between DMF elements from 0 to 0.3 does not provide a noticeable damping of oscillations in the drive links and tension in the DMF springs. The coefficient of viscous friction between the links of the DMF has a significant influence on the amount of tension in the springs of the DMF. To increase the resource of the DMF, it is advisable to install separators made of polymer material between the elastic links with a small coefficient of friction between it and the steel body of the DMF.*

Key words: *transmission, dual-mass flywheel, simulation model, oscillatory phenomena, stress, torque.*

1. INTRODUCTION

Automobile manufacturers, when designing new cars, are increasingly faced with the need to reduce the weight of components in order to achieve the required level of fuel consumption and environmental standards. As a result, internal combustion engines with a small number of cylinders are designed and manufactured, which makes it possible to achieve a decrease in output power due to an increase in pressure in the cylinder and more efficient fuel combustion [1]. As a result of this, torsional vibrations occur on the crankshaft, which are transmitted and negatively affect the transmission, causing it to fail prematurely.

Today, dual-mass flywheels (DMF) can be found in the transmissions of cars and commercial cars, as well as trucks, this is explained primarily by their high efficiency. To date, DMFs of various designs have been developed and successfully proven: with a pendulum damper, with planetary gears, with springs of different stiffness, etc. [2], one of the latest design solutions was implemented by Valeo (VBladeTM DMF technology), which involves replacing arc springs with plates [3].

2. RESEARCH RELEVANCE

The durability of DMF depends on many factors, in particular, the main ones are: operating conditions of the car (urban or suburban cycles), the style and nature of driving the car (sporty manner, constant acceleration and deceleration, sharp braking) – negatively affect the durability of DMF. The most vulnerable structural element of the DMF is the arc spring, which perceives constant cyclic loads, which over time leads to its destruction [4]. Since the DMF is not repairable, in case of destruction of the arc spring, it must be replaced. The grease inside the DMF, which lubricates the arc springs, can burn out, leak, dry, etc. over time, which gradually changes its initial damping properties, dry friction between the structural elements of the DMF, in turn, also leads to their rapid wear, so research is aimed at elucidating the effect of friction between DMF elements on its durability is relevant and has important practical significance.

3. PROBLEM FORMULATION

The damping properties of DMF in a straight line depend on their structure and design parameters. All modern dual-mass flywheels contain a certain amount of thick lubricant, which in one way or another improves its characteristics. But in addition to parts, flywheels that constantly work in an environment with lubricant, there are also elements between which dry friction occurs, which also affects the damping characteristics of the flywheel. Therefore, it can be assumed that its presence affects the elastic-damping properties of the dual-mass flywheel.

4. AIMS AND OBJECTIVES OF THE STADY

The purpose of the work is to develop simulation models and study the effect of friction between DMF elements on oscillatory processes in the car transmission and to develop recommendations for reducing the load on DMF elements and transmission links.

5. ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

A number of works [1, 2, 4–12] are devoted to the study of DMF, they cover the issue of absorption of torsional vibrations generated by internal combustion engines and their further influence on the transmission, constructive improvement of DMF, development of experimental installations for the study of oscillatory processes inside DMF.

DMF improves car comfort indicators, in particular by reducing vibration and noise. In work [5], the authors, based on the created model of torsional oscillations, simulated the torsional characteristics under various conditions using the LMS AMESim software product. The calculation formula for the angular stiffness of the arc spiral spring is obtained, the angular stiffness of the spring is optimized using arithmetic averaging, which shows an adjusted angular stiffness of 12.8 Nm/°. The results of the tests established that DMF can achieve effective vibration isolation of about 85 %.

In work [6], a mathematical and simulation model of a car drive with DMF during the period of starting from a standstill was developed with the help of the Matlab Simulink software product. This model takes into account the dependence of the torque and power of the internal combustion engine on the frequency of rotation of the crankshaft and the unevenness of its rotation, road resistance, as well as stiffness and inertia parameters of the vehicle drive.

The authors [7] did not take into account the magnitude of the change in torque depending on the angular velocity, when studying the process of moving the car from a standstill, while the magnitude and nature of the change in the torque of the internal combustion engine were set by a harmonic function.

In the paper [8], the authors used the ADAMS software product, in particular the ADAMS/Vibration module, for a household virtual vehicle model. The simulation results show that the impact factor of vibration at a speed of 1600 rpm is the stiffness of the clutch rotation, and the impact factor of gear shifting impact is the inertial mass of the DMF.

The author [9] considers the problem of weight-vibration optimization of DMF according to Pareto in order to investigate the feasibility of its use in power units of heavy-duty trucks. The obtained results indicate that a solution to the considered optimization problem exists; parameters of mass inertia, stiffness and damping of DMF, optimized in the working range of engine revolutions of 600-2000 rpm, provide the best absorption of torque fluctuations on the input shaft of the transmission, which indicates the feasibility of using DMF optimized for weight and vibration parameters in the transmission of heavy trucks.

In [10], the influence of forces and moments between the primary mass, the spring seat, the spring and the secondary mass, as well as the dynamic analysis model of the DMF rotor system, taking into account the influence of the clearance and friction between its elements, was analyzed. The vibrational response of the DMF was investigated by a numerical method. Analyzing the bifurcation diagram, time history, phase trajectories, Poincaré plot and frequency domain relative to the angular displacement, the change in the shape of system oscillations at different excitation frequencies is investigated. The influence of the load, the amplitude of the rotation of the primary mass and the stiffness of the spring on the vibration of the system was analyzed.

A simple model of a centrifugal pendulum is described in [1], since a more complex dynamic model of the DMF involves friction between the components of the spring. A multi-element DMF model using a CAD model is considered. The dynamic model consists of a torsion spring and two bodies. The presented experimental setup allows simulating the torque of the internal combustion engine and determining its effect on the spring-damping system of the DMF.

To study the dynamic characteristics of DMF, including torque and stiffness, taking into account the influence of centrifugal force and friction, the discrete method analyzes the mechanical effects on the structural elements of DMF. The dynamic characteristics of DMF are obtained by analysis and calculation. The influence of speed, friction coefficient, mass of the spring and mass of the spring seat on the moment and stiffness characteristics was analyzed [11].

In [12], an analysis of the influence of the coefficient of moment of inertia, torsional stiffness and damping on the reduction of vibration of DMF was carried out by means of simulation. The amplitude of speed fluctuations on the input shaft of the transmission and the natural frequency of the vehicle are taken for the purpose of optimization.

6. PRESENTING MAIN MATERIAL

Dynamic model of the car drive with DMF. The dynamic model of the drive of a front-wheel drive car with DMF is shown in Fig. 1. Unlike the existing ones, it takes into account, in addition to the stiffness of the elastic links of the DMF and viscous friction, dry friction between the moving links of the DMF. The creation of mathematical and simulation models based on it will provide an opportunity to investigate the effect of dry friction between DMF elements on the vibration phenomena in the drive of the car and the load on its links.

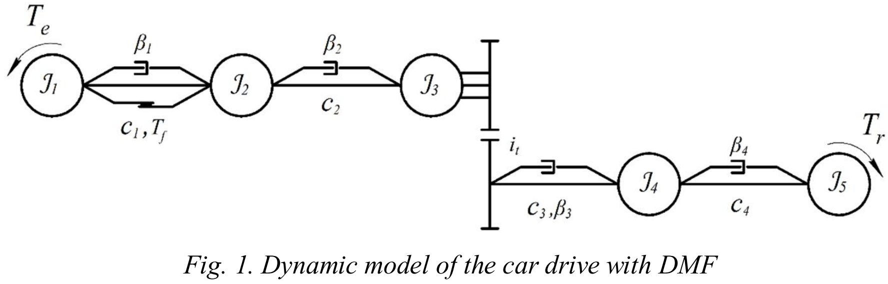


Fig. 1. Dynamic model of the car drive with DMF

In Fig. 1 marked: J_1 – combined moment of inertia of the crankshaft, links of the connecting rod-piston group and the primary mass of the DMF; J_2 – the total moment of inertia of the secondary mass of the DMF and the parts of the clutch and the primary shaft of the gearbox; J_3 – combined moment of inertia of the secondary shaft of the gearbox and the main gear; J_4 – moment of inertia of hubs and wheel rims; J_5 is the moment of inertia of the vehicle mass reduced to the wheel. The combined stiffnesses of the parts of the dual-mass flywheel, gearbox elements, half-axles, and tires are indicated, respectively: c_1, c_2, c_3, c_4. The coefficients of energy dissipation in DMF links, gearboxes, shaft lines and tires are marked β_1, β_2, β_3, β_4, respectively. Generalized coordinates are selected angles of rotation of the combined masses of the dynamic model of the drive: φ_1, φ_2, φ_3, φ_4, φ_5. T_e is engine torque. T_f is the moment of dry friction between the elements of the DMF. T_r is the moment of resistance to the movement of the car.

Mathematical model of a car drive with a dual-mass flywheel. The oscillatory motion of the links of the dynamic model (see Fig. 1), taking into account the external forces acting on it, is presented in the following form:

$$\begin{cases} J_1\ddot{\varphi}_1 = T_e - c_1(\varphi_1-\varphi_2) - \beta_1(\dot{\varphi}_1-\dot{\varphi}_2) - T_f; \\ J_2\ddot{\varphi}_2 = c_1(\varphi_1-\varphi_2) + \beta_1(\dot{\varphi}_1-\dot{\varphi}_2) - c_2(\varphi_2-\varphi_3) - \beta_2(\dot{\varphi}_2-\dot{\varphi}_3) + T_f; \\ J_3\ddot{\varphi}_3 = c_2(\varphi_2-\varphi_3) + \beta_2(\dot{\varphi}_2-\dot{\varphi}_3) - c_3\left(\varphi_3\dfrac{1}{i_t}-\varphi_4\right) - \beta_3\left(\dot{\varphi}_3\dfrac{1}{i_t}-\dot{\varphi}_4\right); \\ J_4\ddot{\varphi}_4 = c_3\left(\varphi_3\dfrac{1}{i_t}-\varphi_4\right) + \beta_3\left(\dot{\varphi}_3\dfrac{1}{i_t}-\dot{\varphi}_4\right) - c_4(\varphi_5-\varphi_6) - \beta_4(\dot{\varphi}_5-\dot{\varphi}_6); \\ J_5\ddot{\varphi}_5 = c_4(\varphi_5-\varphi_6) + \beta_4(\dot{\varphi}_5-\dot{\varphi}_6) - T_r. \end{cases} \tag{1}$$

We present the change in the torque of the internal combustion engine in the form of a harmonic function:

$$T_e = T_0 + T_A \sin(\omega t + \alpha), \tag{2}$$

where T_0 – the constant component of the engine torque; T_A – the variable component of the torque; ω – the frequency of oscillations of the variable component; α – the phase shift.

The moment of dry friction forces between the moving links of the DMF is calculated using the formula:

$$T_f = N f_{FR} R_{FR}\,\mathrm{sign}(\dot{\varphi}_1 - \dot{\varphi}_2), \tag{3}$$

where N – the pressing force of the friction elements in the DMF between themselves; R_{FR} – radius of application of dry friction force; f_{FR} – the coefficient of dry friction between DMF elements.

The total resistance to movement of the car on the road surface:

$$T_r = G_a \cdot r\left[\left(f_0 + k_f(\dot{\varphi}_6 \cdot r)^2\right)\cos\gamma + \sin\gamma\right], \tag{4}$$

where G_a – the mass of the car, r is the dynamic radius of the wheel; f_0 – the coefficient of resistance at low speed; k_f – the coefficient that takes into account the increase in rolling resistance with an increase in the speed of the car; γ – the angle of inclination of the road; $\dot{\varphi}_6$ – angular speed of the wheels of the car.

The tension in the coils of the DMF springs is determined by the formula

$$\tau_{\max} = \frac{16\cdot(\varphi_1-\varphi_2)\cdot c_{isp}\cdot R\cdot R_m}{\pi\cdot d_i^3}\left(\frac{4m-1}{4m-4}+\frac{0.615}{m}\right), \tag{5}$$

where c_{isp} – the linear stiffness of the spring; R – the spring installation radius; R_m – the average radius of the spring; d_i – the diameter of the spring wire; $m = 2R_m/d_i$ – the correction factor.

Simulation results and their analysis. To build a stimulation model of a car drive in the MathLab Simulink environment, we present the system of differential equations of motion of the combined masses of the dynamic model (1) in a form convenient for integration:

$$\begin{cases} \ddot{\varphi}_1 = \dfrac{1}{J_1}\left[T_e - c_1(\varphi_1-\varphi_2) - \beta_1(\dot{\varphi}_1-\dot{\varphi}_2) - Nf_{TR}R_{TR}\,\mathrm{sign}(\dot{\varphi}_1-\dot{\varphi}_2)\right]; \\ \ddot{\varphi}_2 = \dfrac{1}{J_2}\left[c_1(\varphi_1-\varphi_2) + \beta_1(\dot{\varphi}_1-\dot{\varphi}_2) - c_2(\varphi_2-\varphi_3) - \beta_2(\dot{\varphi}_2-\dot{\varphi}_3) + Nf_{TR}R_{TR}\,\mathrm{sign}(\dot{\varphi}_1-\dot{\varphi}_2)\right]; \\ \ddot{\varphi}_3 = \dfrac{1}{J_3}\left[c_2(\varphi_2-\varphi_3) + \beta_2(\dot{\varphi}_2-\dot{\varphi}_3) - c_3\left(\varphi_3\dfrac{1}{i_t}-\varphi_4\right) - \beta_3\left(\dot{\varphi}_3\dfrac{1}{i_t}-\dot{\varphi}_4\right)\right]; \\ \ddot{\varphi}_4 = \dfrac{1}{J_4}\left[c_3\left(\varphi_3\dfrac{1}{i_t}-\varphi_4\right) + \beta_3\left(\dot{\varphi}_3\dfrac{1}{i_t}-\dot{\varphi}_4\right) - c_4(\varphi_5-\varphi_6) - \beta_4(\dot{\varphi}_5-\dot{\varphi}_6)\right]; \\ \ddot{\varphi}_5 = \dfrac{1}{J_5}\left[c_4(\varphi_5-\varphi_6) + \beta_4(\dot{\varphi}_5-\dot{\varphi}_6) - T_r\right]. \end{cases} \tag{6}$$

The simulation model that implements the developed mathematical model (2) – (6) of the car drive with DMF in the Simulink environment is shown in Fig. 2.

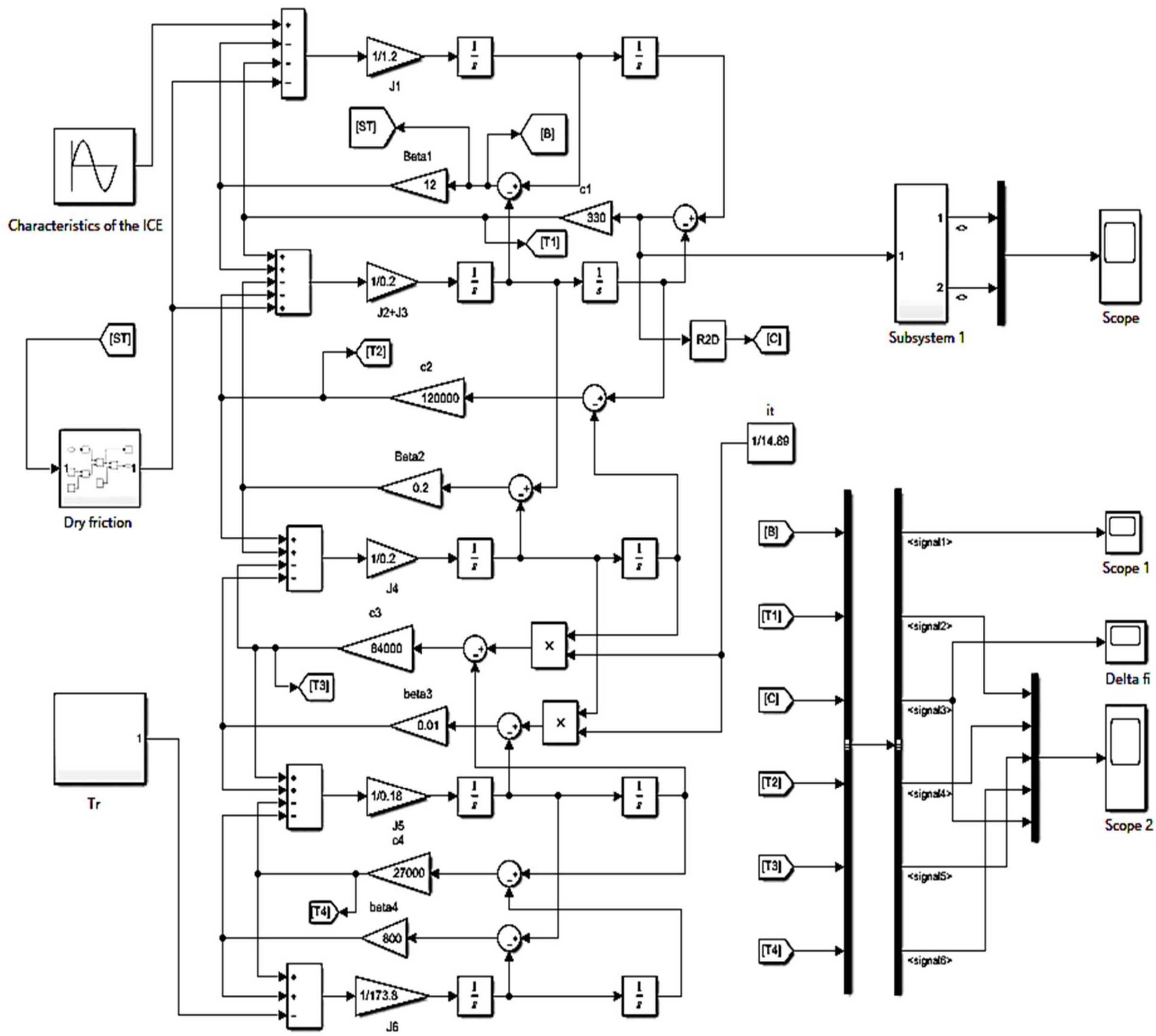


Fig. 2. Simulation model of a car drive with a dual-mass flywheel in the MatLab Simulink environment

Simulation results and their analysis. The analysis of the effect of dry friction between the links of the DMF of the car on the damping of vibrations in the drive links was carried out on the example of a passenger car, the parameters and characteristics of which are given in the Table. 1.

Table 1

Technical characteristics of the car

Parameter	Value
Curb mass	1336
Maximum total weight, kg	2066
Engine power, kW/rpm	75/4000
Maximum torque, N·m/rev/min	210/1900
Wheel parameters	195/65R15
Main gear ratio	3.94
First gear ratio	3.78

The calculations were carried out with the following parameters of the dynamic model of the drive: T_{do}=80 N·m; T_0=60 N·m; ω =63 s^{-1}; α=0; J_1=1.2 kg·m^2; J_2=0.4 kg·m^2; J_3=0.1 kg·m^2; β_1=25 N·s·m; β_2=0.12 N·s·m; c_1=330 N·m/rad; c_2=2475 N·m/rad; the total stiffness of two DMFs installed on a circle of the same radius of a large spring c_{bs}=5881 N/m and a small spring c_{ls}=5586 N/m was c_{pr}=11467 N/m.

In order to find out the effect of dry friction between the DMF links, calculations were carried out at the value of the coefficient of friction between them f_{TR}=0 and f_{TR}=0.3. The value of the pressing force of the moving links of the DMF between themselves N was assumed to be equal to 100 N. The radius of application of the dry friction force in the DMF R_{TR} = 0.13 m. The results of computer simulation of the change in torques in the drive shafts and stresses in the DMM elements when the car starts moving are shown in Fig. 3–9.

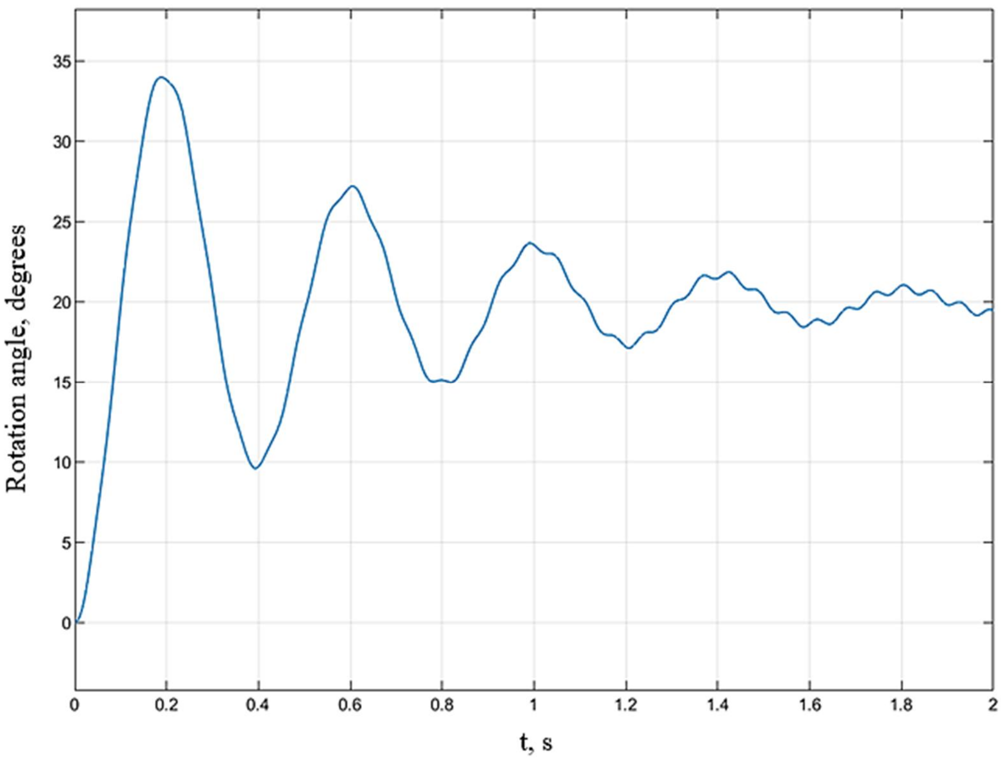


Fig. 3. Dependence of the relative rotation angle of DMF masses depending on time: f_{TR}=0; β_1=4 N·m·s/rad

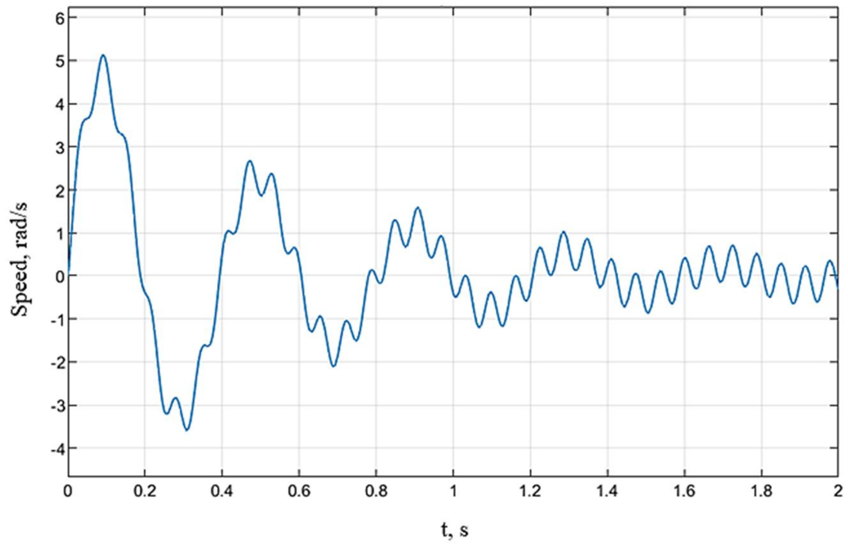


Fig. 4. Rate of damping of oscillations in DMF depending on time: f_{TR}=0; β_1=4 N·m·s/rad

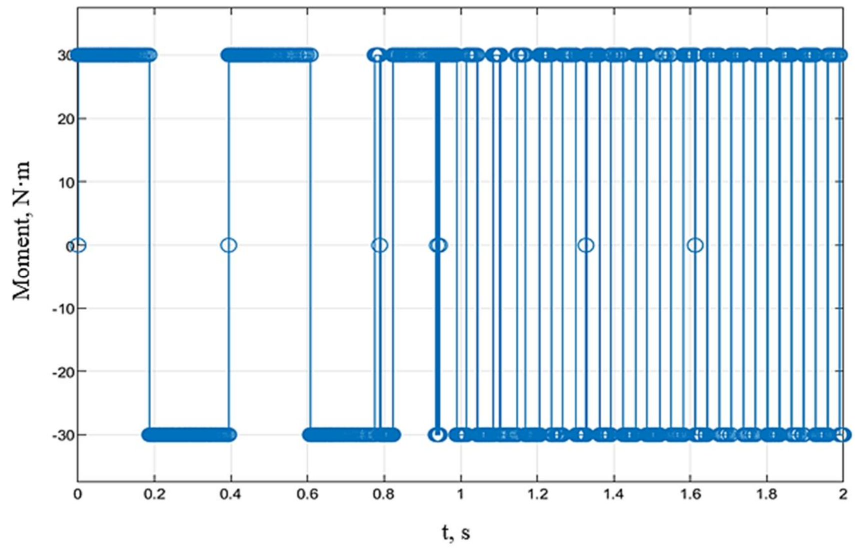


Fig. 5. Dependence of the moment of dry friction between DMF links on time: f_{TR}=0.3

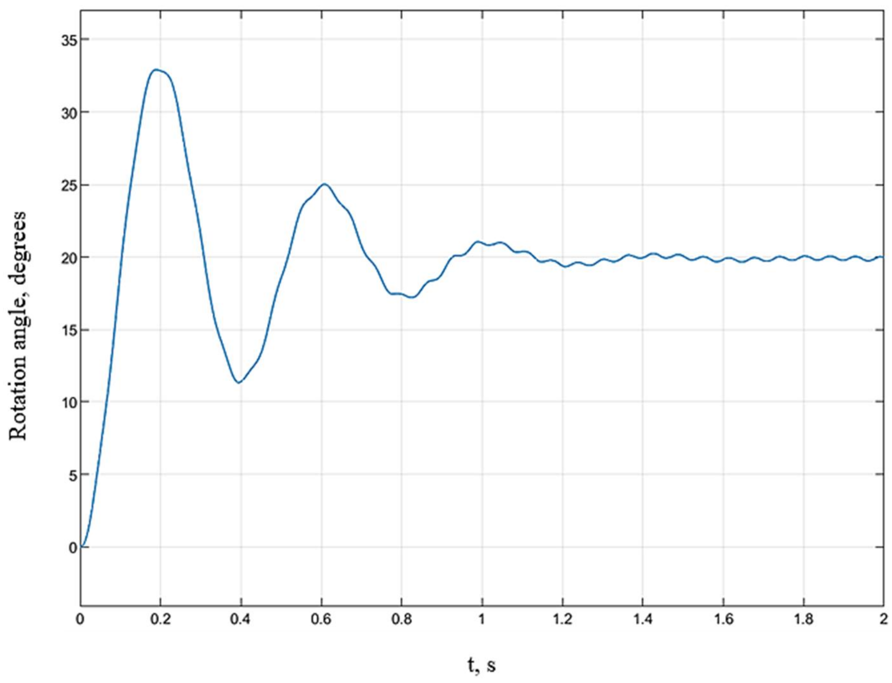


Fig. 6. Dependence of the relative rotation angle of DMF masses as a function on time: f_{TR}=0.3; β_1=4 N·m·s/rad

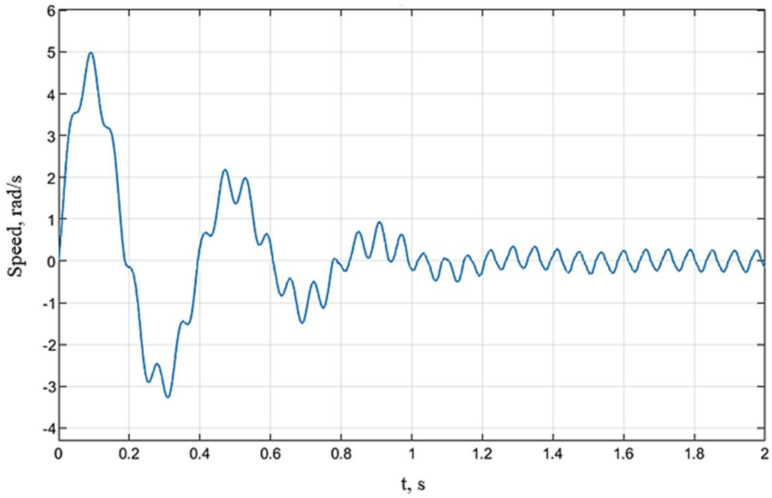


Fig. 7. Rate of damping of oscillations in DMF elements depending on time: f_{TR}=0.3; β_1=4 N·m·s/rad

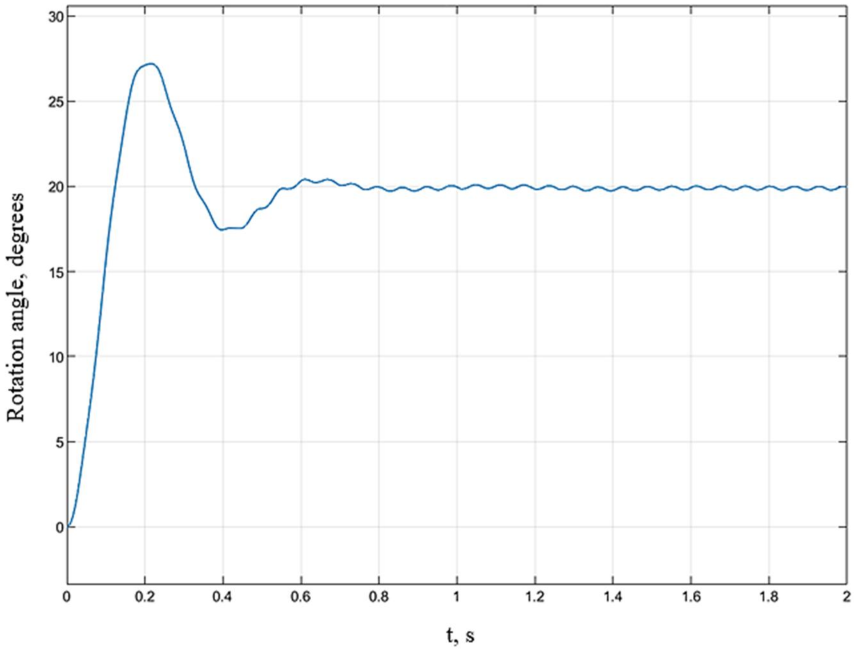


Fig. 8. Dependence of the relative rotation angle of DMF masses as a function on time: f_{TR}=0.3; β_1=12 N·m·s/rad

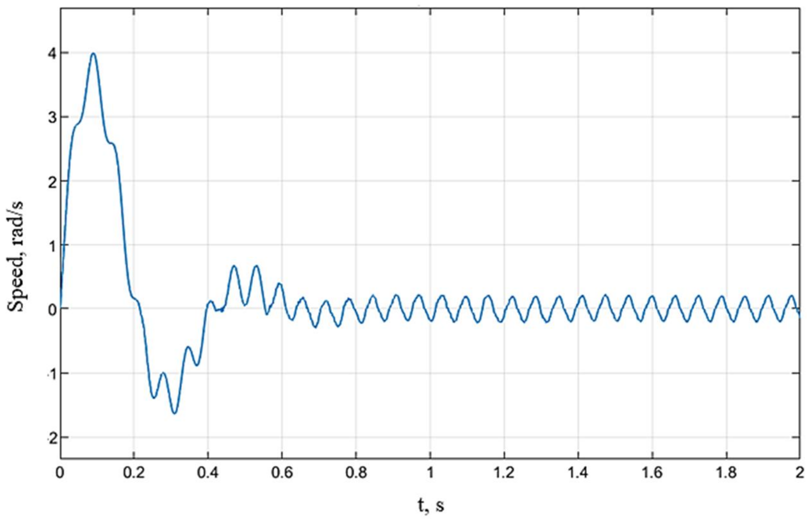


Fig. 9. Rate of damping of oscillations in DMF elements depending on time: f_{TR}=0.3; β_1=12 N·m·s/rad

From the results shown in the figures, it can be seen that with a zero value of the coefficient of friction between the elements of the DMF at the initial moment of time, it reaches 340 and within 2 s slowly decays, approaching 20° (Fig. 3), and the rate of change of the relative angle of scrolling of the primary and secondary masses of the DMF at the initial moment of time exceeds 5 s^{-1} and goes to zero after 2 s (Fig. 4).

When the value of the dry friction angle between the DMF links is 0.3, there is a sign-changing influence of the friction moment on the primary and secondary masses, respectively (see Fig. 5). At the same time, the angle of relative scrolling of the DMF masses is insignificantly reduced (less than 33°) and the oscillations decay faster (after 0.8 s), that is, almost twice as fast.

The rate of fluctuations of the relative angle of rotation of the DMF masses at the initial moment of time does not exceed 5 s^{-1}.

The value of the coefficient of viscous dissipation of energy in the DMF has a more significant influence on the process of damping of oscillations in the DMF elements. Thus, its increase to 12 N·m·s/rad leads to a significant decrease in the duration of oscillation damping (see Fig. 8–9), in fact, the oscillations are damped in 0.4 s.

From what has been said, it can be concluded that in DMF with cylindrical springs, dry friction between the elements does not significantly reduce the damping time of oscillations when the internal combustion engine torque increases. Therefore, it is advisable to equip such DMFs with separators made of polymer materials that have a small coefficient of friction with the steel body of the DMF, and to increase the coefficient of viscous damping due to the viscous lubricant inside the DMF.

The effect of friction between the elements of the dual-mass flywheel on the load on the drive links of the car. The results of computer simulation of the change in torques in the drive shafts and stresses in the DMF elements when the car starts moving are shown in Fig. 10–14. From the above graphs, we can see that at values of f_{TR}=0; β_1=4 N·m·s/rad (Fig. 10) the value of the torque in the drive links at the initial moment reaches 200 N·m and within 2 s decreases, approaching 120 N·m. The tension in the turns of the large DMF spring reaches 11.5×108 Pa, and in the turns of the small one – 8×108 Pa. After 1 s, the torque in the drive links and the tension in the DMM springs stabilize (Fig. 11). The effect of friction between the elements of the dual-mass flywheel on the load on the drive links of the car. The results of computer simulation of the change in torques in the drive shafts and stresses in the DMM elements whenthe car starts moving are shown in Fig. 10–14. From the above graphs, we can see that at values of f_{TR}=0; β_1=4 N·m·s/rad (Fig. 10)

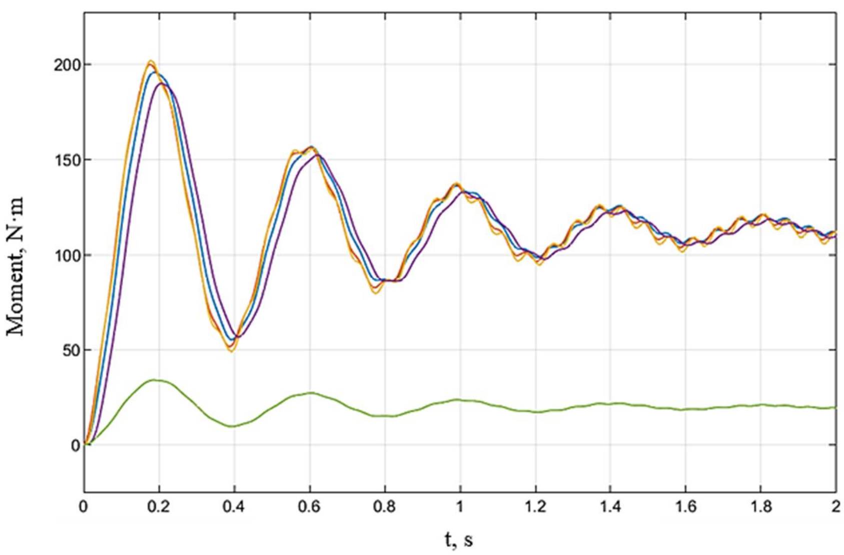


Fig. 10. Change in torque in drive links and DMF elements over time: f_{TR}=0; β_1=4 N·m·s/rad

the value of the torque in the drive links at the initial moment reaches 200 N·m and within 2 s decreases, approaching 120 N·m. The tension in the turns of the large DMM spring reaches 11.5×108 Pa, and in the turns of the small one – 8×108 Pa. After 1 s, the torque in the drive links and the tension in the DMF springs stabilize (Fig. 11).

When the coefficient of dry friction between the DMF links increases to f_{TR}=0.3 (Fig. 12), the magnitude of the torques in the drive links and the tension in the coils of the springs decrease insignificantly (the tension in the large spring is by 15 %, and in the small one by 4 %), but after 1 s the amplitude of torque fluctuations in the drive links increases and does not decay over time. This indicates an additional load on the elastic drive links.

Fig. 11. Change of tension in DMF elements with time: f_{TR}=0; β_1=4 N·m·s/rad; 1 – small spring; 2 – large spring

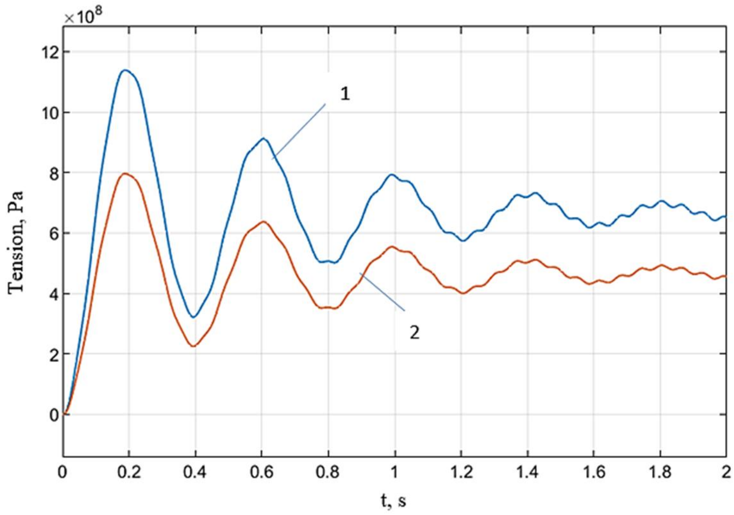


Fig. 11. Change of tension in DMF elements with time: f_{TR}=0; β_1=4 N·m·s/rad; 1 – small spring, 2 – large spring

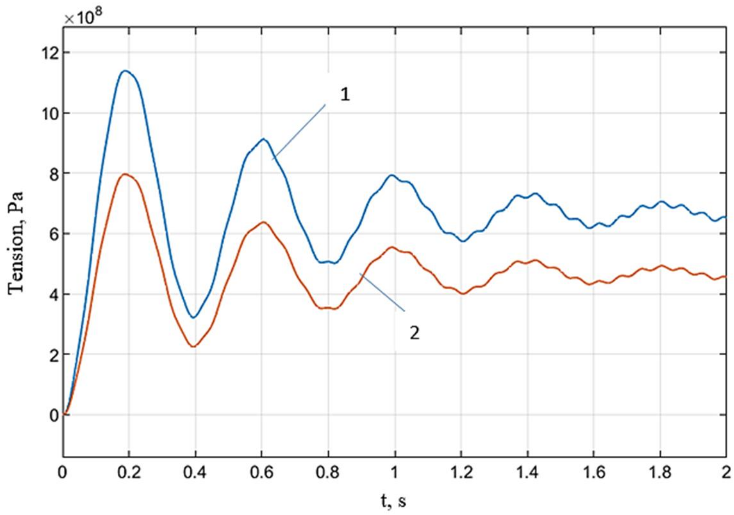
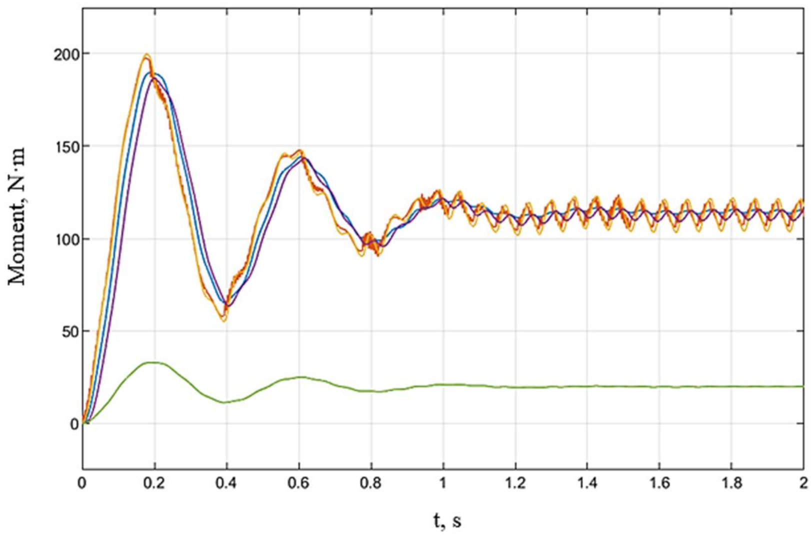


Fig. 12. Change of torque in drive links and DMF elements from time: f_{TR}=0.3; β_1=4 N·m·s/rad



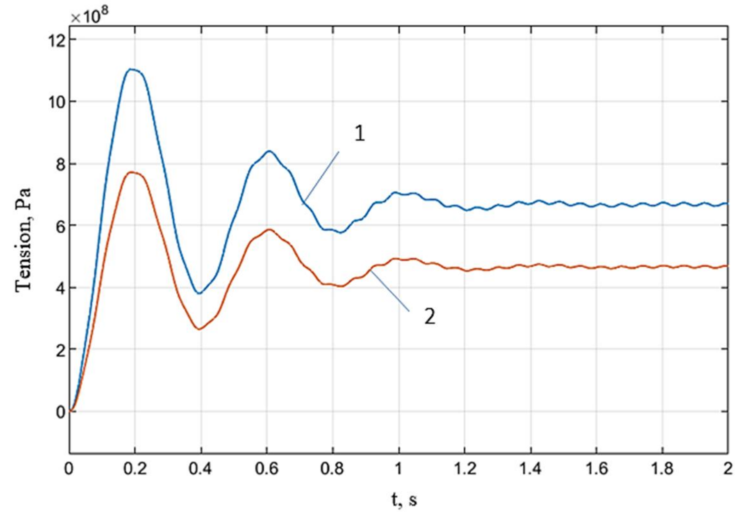


Fig. 13. Change of tension in DMF elements from time:
f_{TR}=0.3; β_1=4 N·m·s/rad;
1 – small spring; 2 – large spring

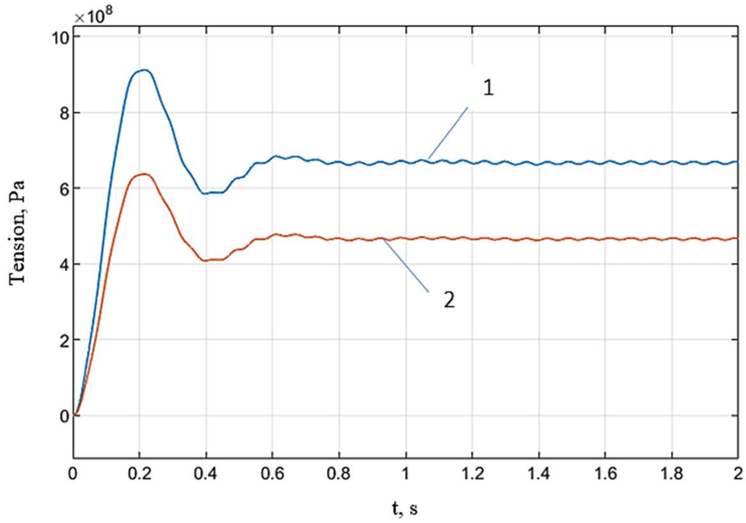


Fig. 14. Change of tension in DMF elements from time:
f_{TR}=0; β_1=12 N·m·s/rad;
1 – small spring, 2 – large spring

Otherwise, the system responds to changes in viscous friction between DMF elements. In Fig. 14 shows graphs of tension changes in DMF spring coils at f_{TR}=0; β_1=12 N·m·s/rad.

It can be seen from the graphs that an increase in the energy dissipation coefficient in DMF β_1 to 12 N·m·s/rad leads to a significant decrease in the tensions in the springs – to 6.4×108 Pa for a large spring and to 9.1×108 Pa – for a small spring, which is 19 and 18 %, respectively. The duration of the damping of stress fluctuations is also reduced to 0.6 s in contrast to the previous case (1.2 s).

7. CONCLUSIONS

1. The effect of dry and viscous friction between the elements of a two-mass flywheel on the damping of oscillations in the links of its elastic-damping system and the drive links of the car was studied.

2. It is shown that an increase in the coefficient of dry friction between DMF elements from 0 to 0.3 does not provide a noticeable damping of oscillations in the drive links and tension in the DMF springs. The coefficient of viscous friction between the links of the DMF has a significant influence on the amount of stress in the springs of the DMF.

3. To increase the resource of the DMF, it is advisable to install separators made of polymer material between the elastic links with a small coefficient of friction between it and the steel body of the DMF.

References

1. Magdolen, Ľ., Danko, J., Milesich, T., Nemec, T., Sloboda, K., & Bucha, J. (2021). Dual Mass Flywheel Parameter Identification. *Journal of Mechanical Engineering. 71*(2), 167–178. doi: 10.2478/scjme-2021-0027 (in English).

2. Litvin, R. (2016). Analiz efektyvnosti vykorystannia dvomasovoho makhovyka na avtomobili [Analysis of the efficiency of using a dual-mass flywheel on a car]. *Visnyk Natsionalnoho universytetu "Lvivska politekhnika". Seriia: Dynamika, mitsnist ta proektuvannia mashyn i pryladiv: zbirnyk naukovykh prats. [Bulletin of Lviv Polytechnic National University. Series: Dynamics, Strength and Design of Machines and Devices: Collection of Scientific Works], 838,* 180–185. (in Ukrainian).

3. Website "Valeo.prowly.com". Retrieved from: https://valeo.prowly.com/50687-a-completely-new-look-at-dual-mass-flywheels-valeo-vblade (in English).

4. Kindratskyy, B., & Litvin, R. (2018). Klasyfikatsiia nespravnostei dvomasovykh makhovykiv u pryvodakh avtomobiliv ta prychyny yikh vynyknennia [The Classification of Malfunction of Dual Mass Flywheel in Automotive Vehicles Drives and Causes of Their Appearance]. *Naukovo-tekhnichnyi zbirnyk "Visnyk Natsionalnoho transportnoho universytetu" seriia "Tekhnichni nauky". [The National Transport University Bulletin: A Scientific and Technical Journal, Series "Technical Sciences"], 3*(42), 46–53. (in Ukrainian).

5. Xie, M., Wang, S., Yan, Z., Wang, L., & Tan, G. (2022). Optimization of Dual-Mass Flywheel Parameters Based on a Multicondition Simulation Test. *Shock and Vibration. Volume 2022*, 1–15. doi: 10.1155/2022/2954825 (in English).

6. Kindratskyy, B., Litvin, R., & Osmak, O. (2022). Influence of Vehicle Acceleration Intensity on Dual-Mass Flywheel Elements and Transmission Load. *Transport Technologies. 3*(1), 65–76. doi: 10.23939/tt2022. 01.065 (in English).

7. Kindratskyy, B., & Litvin, R. (2019). Oscillation Processes in a Transmission with a Dual-Mass Flywheel while Moving a Car from Rest. *Ukrainian Journal of Mechanical Engineering and Materials Science. 5*(1), 93–104. doi: 10.23939/ujmems2019.01.093 (in English).

8. Gaojie, H., Yongchen, X., Yawei, Z., & Xianxian, C. (2022). Optimized Design of Dual-mass Flywheel to Improve Booming Vibration and Shift-shock. *Noise and Vibration Control. Volume 42*(2), 253–258. doi: 10.3969/j.issn.1006-1355.2022.02.043 (in Chinese).

9. Berbyuk, V. (2022). Weight-Vibration Pareto Optimization of a Dual Mass Flywheel. *Journal of Mathematical Sciences. 263*(1), 1–14. doi: 10.1007/s10958-022-05917-6 (in English).

10. Zeng, L., Liu, J., Wan, Z., Song, L., & Yuan, X. (2022). Modeling and vibration characteristics analysis of a DMF rotor system. *Journal of Mechanical Science and Technology*. *36*(6), 2799–2810. doi: 10.1007/s12206-022-0512-0 (in English).

11. Zeng, L., Xu, Y., Huang, J., & Song, L. (2021). Dynamic Characteristics Analysis of a Circumferential Short Spring Dual Mass Flywheel with the Influence of Centrifugal Force and Friction. *Symmetry. 13*(8), 1501, 1–21. doi: 10.3390/sym13081501 (in English).

12. Wu, G., & Zhao, G. (2022). Parameter Influence Law Analysis and Optimal Design of a Dual Mass Flywheel. *International Journal of Mechanical System Dynamics*. *Volume 2*(2), 165–177. doi: 10.1002/msd2.12046 (in English).

Received 05.10.2022; Accepted in revised form 18.10.2022.

ВПЛИВ ТЕРТЯ МІЖ ЕЛЕМЕНТАМИ ДВОМАСОВОГО МАХОВИКА НА КОЛИВАЛЬНІ ЯВИЩА У ТРАНСМІСІЇ АВТОМОБІЛЯ

Анотація. *Автовиробники при проєктуванні нових автомобілів, все частіше стикаються з необхідністю зменшення ваги складових компонентів, щоб досягти потрібного рівня споживання палива та екологічних норм. Як наслідок проєктуються і виготовляються двигуни внутрішнього згорання (ДВЗ) з невеликою кількістю циліндрів, що дозволяє досягти збільшення вихідної потужності за рахунок підвищення тиску в циліндрі та більш ефективного згорання палива. В результаті цього на колінчастому валу виникають крутильні коливання, які передаються і негативно впливають на трансмісію, передчасно виводячи її з ладу. Демпфувальні властивості двомасових маховиків (ДММ) напряму залежать від їхньої будови та конструктивних параметрів. Усі сучасні ДММ містять у собі певну кількість густого мастила, що так чи інакше покращує його характеристики. Але, крім деталей які постійно працюють в середовищі з мастилом, маховики містять елементи, між якими* *відбувається сухе тертя. Тому можна припустити, що його наявність може впливати на пружно-демпфувальні властивості ДММ. Метою праці є розроблення симуляційних моделей і дослідження впливу тертя між елементами ДММ на коливальні процеси у трансмісії автомобіля та розроблення рекомендацій щодо зменшення навантаження на елементи ДММ і ланки трансмісії. Досліджено вплив сухого і в'язкого тертя між елементами ДММ на затухання коливань у його пружно-демпфувальній системі. Показано, що збільшення коефіцієнта сухого тертя між елементами ДММ з 0 до 0,3 не дає відчутного затухання коливань у ланках приводу і ДММ. Істотніший вплив на затухання коливань має в'язке тертя між ланками ДММ. Для збільшення ресурсу ДММ доцільно між його пружними ланками встановлювати сепаратори з полімерного матеріалу з невеликим коефіцієнтом тертя між ним і стальним корпусом ДММ.*

Ключові слова: *трансмісія, двомасовий маховик, симуляційна модель, коливальні явища, напруження, крутний момент.*