Mathematical modeling of torsional vibrations of the wheel-motor unit of mains diesel locomotive UZTE16M

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Abstract. The article considers methods for studying torsional vibrations of the wheel-motor block of a diesel locomotive UZTE16M. The diesel locomotives in question were modernized at the structural enterprise "O'ztemiryo'lmashta'mir" (Machine building and repair in Uzbekistan Railways company) under the joint-stock company "O'zbekiston temir yo'llari" (Uzbekistan railways). The authors of the article developed a dynamic model for the analysis of torsional vibrational movements that occur in the traction transmissions of the UZTE16M diesel locomotive, taking into account the geometric dimensions of the electric traction motors of the main locomotives and the degree of their wear.
Based on the model of the wheel-motor block of mainline locomotives, differential equations of the torsional-vibrational motion of traction gears are obtained in the form of Lagrange equations of the 2nd kind. The results of numerical studies of the amplitudes of torsional vibrations of the wheel-motor block of the UzTE16M locomotive at different frequencies are presented. All calculations were performed in the MATHCAD 15 environment.

1 Introduction

JSC "O'zbekiston temir yo'llari" carries out tasks not only for the regular and uninterrupted transportation of general-purpose goods but also to increase the economy of using both locomotives and diesel locomotives. The central unit of any rolling stock, in particular, a diesel locomotive of the UzTE16M type, is a wheel-motor unit, which, after the first repair, sharply reduces the traction force of the locomotive and reduces the coefficient of adhesion of the wheel to the rails. The repair process requires radical improvement, which is impossible without the creation of a mathematical model and dynamic models.

The interaction of the wheel-motor block of the locomotive with the rails is the most important factor influencing the mechanical part of the rolling stock. The wheel-motor unit includes a wheel pair, axlebox assembly, gearbox, elastic coupling, and electric traction motor.

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Through the efforts of these scientists, the theoretical foundations for modeling mechanical vibrations of wheel-motor units of locomotives and their technical diagnostics, including methods for modeling shaft vibrations of electric traction motors, have been developed and developed, original designs and circuit solutions of these systems and their technical means. At the same time, not enough attention has been paid to improving the methods for modeling mechanical oscillations of the wheel-motor units of locomotives, taking into account the complex effect of external vibrations and developing an improved method for diagnosing them to extend the useful life [1-6].

The research aims to increase the efficiency of using mainline diesel locomotives by modeling mechanical vibrations of the wheel-motor blocks of mainline diesel locomotives, improving the calculation methodology.

2 Methods

To calculate the torsional vibrations of the TED armature shaft, it is necessary to consider the entire system of the wheel-motor unit (WMU) as a whole.

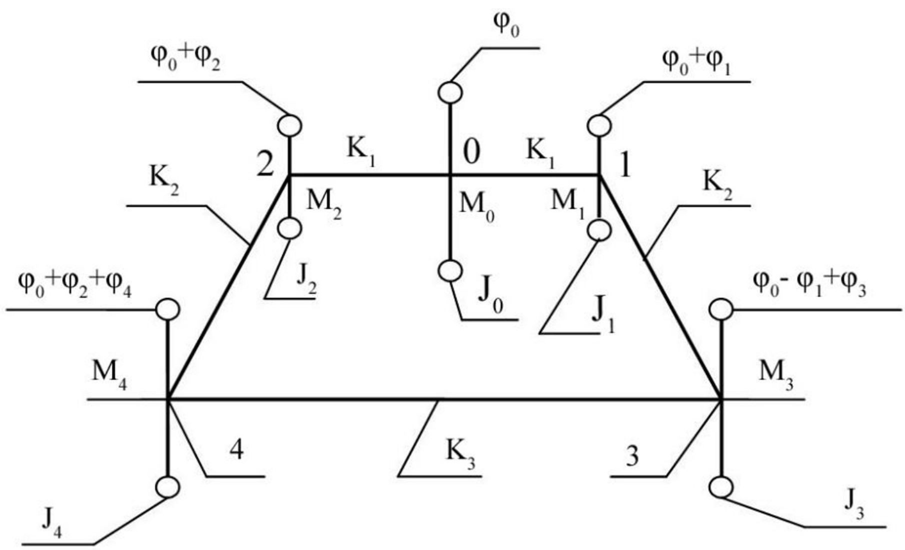

Fig. 1. Calculation scheme of torsional vibrations of wheel-motor unit (WMU) of UzTE16M diesel locomotive

The UzTE16M diesel locomotive block has a support-axial suspension of the traction electric motor ED118B(A) and a one-way gear train. The power of a locomotive diesel engine is realized through a gear wheel, in the form of traction forces, during the interaction of wheelsets with rails. The electric traction motor (TEM) ED118B(A) on one side is rigidly supported on the axle of the wheel pair through motor-axial bearings, and on the other side, it is connected through a spring suspension to the bogie frame. With such a suspension, almost half of the TEM mass is rigidly connected with the unsprung masses of the wheelset. It amounts to about 4250 kg on one WMU, which causes the occurrence of significant mechanical vibrations on the shafts in the system "railway - wheelset - traction engine", both torsional and bending and longitudinal [7-13].

The calculation scheme of the WMU is shown in Figure 1, on which the points *0, 1, 2, 3, 4* of the connections are highlighted and:

- moments of inertia of the armature TEM – J_0 and sections of the armature shaft, and gear J_1, J_2, of gears and axle sections of wheel sets J_3, J_4;

- torsional stiffness between fixing points 0 and 1 – K_1, 0 and 2 – K_1, *1* and *3* – K_2, *2* and *4* – K_2, *4* and *3* – K_3;
- $\varphi_0(t)$ – the angle of rotation of the TEM armature relative to the stator, taking into account the rotational (kinematic) motion functions of all masses J;
- angles of elastic oscillations of mass moments of inertia J_1 and J_2 relative to $J_0 \rightarrow \varphi_1(t)$ and $\varphi_2(t)$, J_3 relative to $J_1 \rightarrow \varphi_3(t)$, J_4 relative to $J_2 \rightarrow \varphi_4(t)$, $J_3 \rightarrow \varphi_1(t) + \varphi_3(t)$ and $J_4 \rightarrow \varphi_2(t) + \varphi_4(t)$;
- the motion torque M_0 at point *0,* and the rotation resistance torque M_3 and M_4 transmitted through the elastic bond sections *0, 1, 3,* and *0, 2, 4*.

The fluctuations of the moments of inertia of the system are characterized by a set of angles of torsional oscillations:

$$J_0 \rightarrow \varphi_0(t);\ J_1 \rightarrow (\varphi_0(t) + \varphi_1(t));\ J_2 \rightarrow (\varphi_0(t) + \varphi_2(t));$$
$$J_3 \rightarrow (\varphi_0(t) + \varphi_1(t) + \varphi_3(t));\ J_4 \rightarrow (\varphi_0(t) + \varphi_2(t) + \varphi_4(t)).$$

The Lagrange method and functions were used to derive the equations of oscillation of moments of inertia:
Kinetic energy:

$$T = \frac{1}{2} \cdot \begin{bmatrix} J_0 \cdot \dot{\varphi}_0^2 + J_1 \cdot (\dot{\varphi}_0 + \dot{\varphi}_1)^2 + J_2 \cdot (\dot{\varphi}_0 + \dot{\varphi}_2)^2 + \\ +J_3 \cdot (\dot{\varphi}_0 + \dot{\varphi}_1 + \dot{\varphi}_3)^2 + J_4 \cdot (\dot{\varphi}_0 + \dot{\varphi}_2 + \dot{\varphi}_4)^2 \end{bmatrix} \tag{1}$$

Potential energy:

$$\Pi = \frac{1}{2} \cdot \begin{bmatrix} K_1 \cdot \varphi_1^2 + K_1 \cdot \varphi_2^2 + K_2 \cdot \varphi_3^2 + K_2 \cdot \varphi_4^2 + \\ +K_3 \cdot (\varphi_1 + \varphi_3 - \varphi_2 - \varphi_4)^2 \end{bmatrix} \tag{2}$$

Operation of external forces (torques):

$$\partial A = M_0 \cdot \delta\varphi_0 - M_3 \cdot (\delta\varphi_0 + \delta\varphi_1 + \delta\varphi_3) - M_4 \cdot (\delta\varphi_0 + \delta\varphi_2 + \delta\varphi_4) \tag{3}$$

The Lagrange equation for each coordinate φ_0, φ_1, φ_2, φ_3, φ_4 in the form of:

$$\frac{\partial}{\partial t} \cdot \left[\frac{\partial}{\partial \dot{\varphi}_i}\right] + \frac{\partial T}{\partial \varphi_i} = \frac{\delta A}{\delta \varphi_i} \tag{4}$$

By coordinate φ_0:

$$\ddot{\varphi}_0 \cdot (J_0 + J_1 + J_2 + J_3 + J_4) + \ddot{\varphi}_1 \cdot (J_1 + J_3) + \ddot{\varphi}_2 \cdot (J_2 + J_4) + \ddot{\varphi}_3 \cdot J_3 +$$
$$+\ddot{\varphi}_4 \cdot J_4 = M_0 - M_3 - M_4 \tag{5}$$

By coordinate φ_1:

$$\ddot{\varphi}_0 \cdot (J_1 + J_3) + \ddot{\varphi}_1 \cdot (J_1 + J_3) + K_1 \cdot \varphi_1 + K_3 \cdot \varphi_1 + \ddot{\varphi}_3 \cdot J_3 = -M_3 \tag{6}$$

By coordinate φ_2 :

$$\ddot{\varphi}_0 \cdot (J_2 + J_4) + \ddot{\varphi}_2 \cdot (J_2 + J_4) + K_1 \cdot \varphi_2 - K_3 \cdot \varphi_2 + \ddot{\varphi}_4 \cdot J_4 = -M_4 \tag{7}$$

By coordinate φ_3:

$$\ddot{\varphi}_0 \cdot J_3 + \ddot{\varphi}_3 \cdot J_3 + K_2 \cdot \varphi_3 + K_3 \cdot \varphi_3 = -M_3 \quad (8)$$

By coordinate φ_4:

$$\ddot{\varphi}_0 \cdot J_4 + \ddot{\varphi}_4 \cdot J_4 + K_2 \cdot \varphi_4 - K_3 \cdot \varphi_4 = -M_4 \quad (9)$$

Solutions of the resulting system of equations take into account components from solutions:
- The system of similar equations, when $M_i = 0$ and $\ddot{\varphi}_0(t) = 0$;
- of a system with variable rotation, when $\ddot{\varphi}_0(t) \neq 0$;
- when variable loads are acting on the system $M_i \neq 0$.

The first version of the solution of the system (5) ÷ (9) was performed for the conditions of non-uniform rotation of the masses of the wheel-engine model:

$$\ddot{\varphi}_0(t) = \bar{\varphi}_0 \cos(\omega \cdot t) \quad (10)$$

where: $\bar{\varphi}_0$- acceleration amplitude of circular frequency ω.
We solved the system in the form of functions:

$$\varphi_1(t) = \varphi_1 \cos(\omega \cdot t);\ \varphi_2(t) = \varphi_2 \cos(\omega \cdot t);$$
$$\varphi_3(t) = \varphi_3 \cos(\omega \cdot t);\ \varphi_4(t) = \varphi_4 \cos(\omega \cdot t) \quad (11)$$

where φ_i – amplitudes of oscillations of the model masses according to the figure 1.
After substituting the time derivatives of (11) into (5) ÷ (9), we obtain a system of algebraic equations to determine the amplitudes φ_i:

$$-\varphi_1 \cdot \omega^2 \cdot (J_1 + J_3) - \varphi_2 \cdot \omega^2 \cdot (J_2 + J_4) - \varphi_3 \cdot \omega^2 \cdot J_3 - \varphi_4 \cdot \omega^2 \cdot J_4 = \varphi_0 \cdot \omega^2 \cdot (J_0 + J_1 + J_2 + J_3 + J_4) = B_1 \quad (12)$$

$$\varphi_1 \cdot [K_1 + K_3 - \omega^2 \cdot (J_1 + J_3)] - \varphi_3 \cdot \omega^2 \cdot J_3 = \varphi_0 \cdot \omega^2 \cdot (J_1 + J_3) = B_2 \quad (13)$$

$$\varphi_2 \cdot [K_1 - K_3 - \omega^2 \cdot (J_2 + J_4)] - \varphi_4 \cdot \omega^2 \cdot J_4 = \varphi_0 \cdot \omega^2 \cdot (J_2 + J_4) = B_3 \quad (14)$$

$$\varphi_3 \cdot [K_1 + K_3 - \omega^2 \cdot J_3] = \varphi_0 \cdot \omega^2 \cdot J_3 = B_4 \quad (15)$$

$$\varphi_4 \cdot [K_2 - K_3 - \omega^2 \cdot J_4] = \varphi_0 \cdot \omega^2 \cdot J_4 = B_5 \quad (16)$$

Introduced the coefficient designations for φ_i in (12) ÷ (16):

$$\begin{aligned} A_{11} \cdot \varphi_1 + A_{12} \cdot \varphi_2 + A_{13} \cdot \varphi_3 + A_{14} \cdot \varphi_4 = B_2 \\ A_{21} \cdot \varphi_1 + A_{22} \cdot \varphi_2 + A_{23} \cdot \varphi_3 + A_{24} \cdot \varphi_4 = B_3 \\ A_{31} \cdot \varphi_1 + A_{32} \cdot \varphi_2 + A_{33} \cdot \varphi_3 + A_{34} \cdot \varphi_4 = B_4 \\ A_{41} \cdot \varphi_1 + A_{42} \cdot \varphi_2 + A_{43} \cdot \varphi_3 + A_{44} \cdot \varphi_4 = B_4 \end{aligned} \quad (17)$$

For this model, the conditions for excitation of oscillations are:
$K_1 - \omega^2 \cdot (J_1 + J_3) = 0$, whence the formula for determining the frequency of natural oscillations is obtained:

$$\omega_1 = \sqrt{\frac{K_1}{J_1 + J_3}} \quad (18)$$

$K_1 - \omega^2 \cdot (J_2 + J_4) = 0$, Whence:

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$$\omega_2 = \sqrt{\frac{K_1}{J_2+J_4}} \tag{19}$$

$K_2 - \omega^2 \cdot J_3 = 0$, Whence:

$$\omega_3 = \sqrt{\frac{K_2}{J_3}} \tag{20}$$

$- K_2 - \omega^2 \cdot J_4 = 0$, Whence:

$$\omega_4 = \sqrt{\frac{K_2}{J_4}} \tag{21}$$

3 Results and discussion

Numerical studies were carried out in the programming environment MATHCAD 15 for the model of torsional vibrations in the wheel-motor unit of UzTE16M locomotive. The calculation program is presented in Appendix 1 (Program 1. "Methodology for calculating parameters of the model of torsional oscillations in the wheel-motor unit of UzTE16M locomotive") [14-19].

For numerical studies, 2 main modes of loading by armature speed are adopted:

in continuous (continuous) mode, the armature speed of TEM ED-118A n = 476 rpm;

at maximum load mode, the speed of ED-118A armature is n = 2290 rpm.

Consider the starting torque at traction motor speed n = 476 rpm and t = 0. The angular acceleration at the given parameters in the program is $\bar{\varphi}_0 = 1{,}193$ 1/sec^2. The initial values of the amplitudes are as follows:

$$\varphi_1 = -3.362 \cdot 10^{-5}, \varphi_2 = -1.406 \cdot 10^{-5}, \varphi_3 = -1.406 \cdot 10^{-5}, \varphi_4 = -0.871 \cdot 10^{-5}$$

The results of the calculations are shown in the form of torsional vibration graphs for the wheel motor unit (Fig. 2-7).

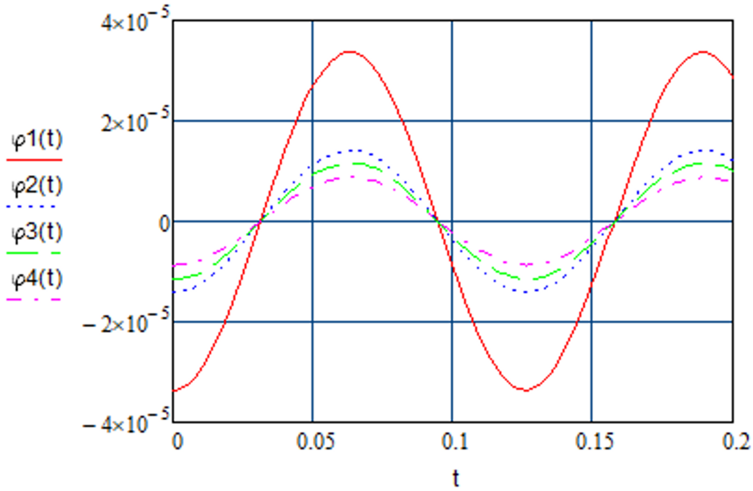

Fig. 2. Amplitudes of torsional oscillations by mass of WMU of UZTE16M locomotive at n=476 rpm.

Let's make these calculations for different rotation frequencies of torque electric motors.

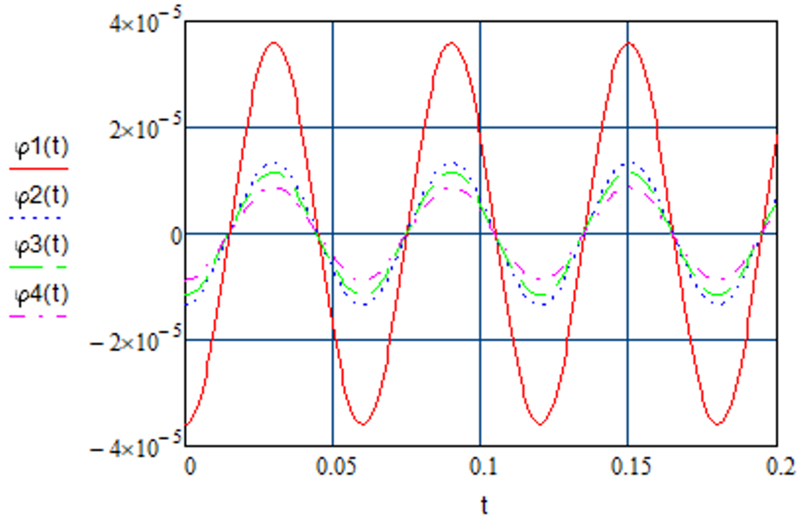

Fig. 3. Amplitudes of torsional oscillations by mass of WMU of UZTE16M locomotive at n=1000 rpm

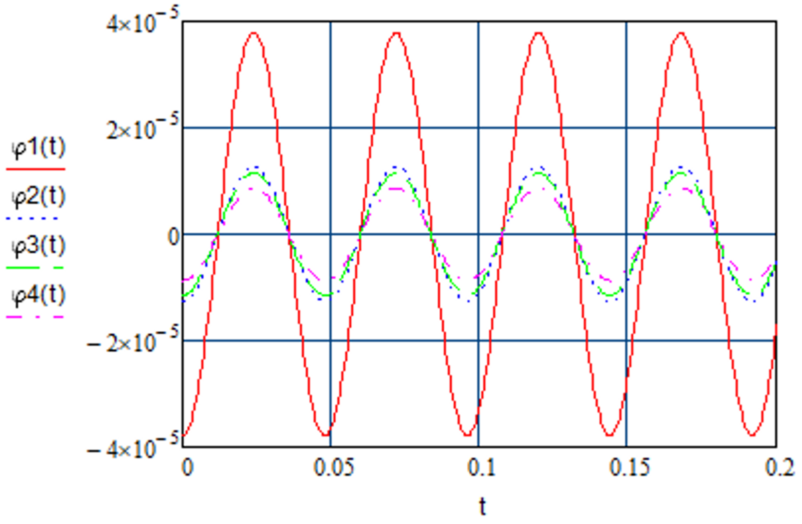

Fig. 4. Amplitudes of torsional oscillations by mass of WMU of UZTE16M locomotive at n=1250 rpm.

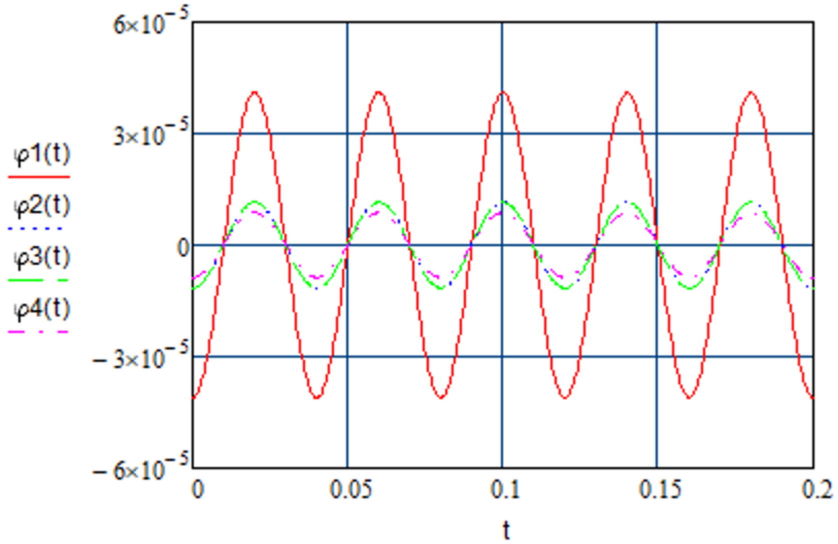

Fig. 5. Amplitudes of torsional oscillations by mass of WMU of UZTE16M locomotive at n=1500 rpm

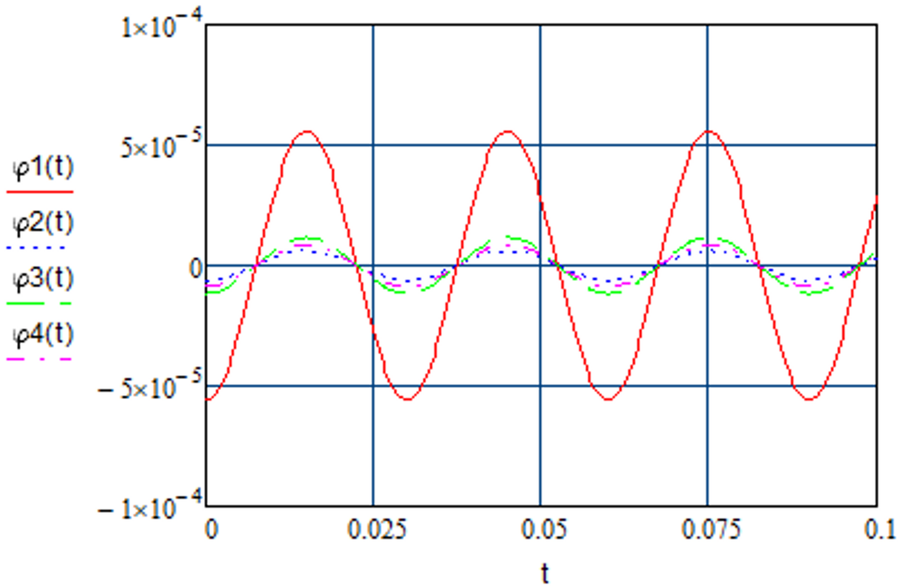

Fig. 6. Amplitudes of torsional oscillations by mass of WMU of UZTE16M locomotive at n=2000 rpm

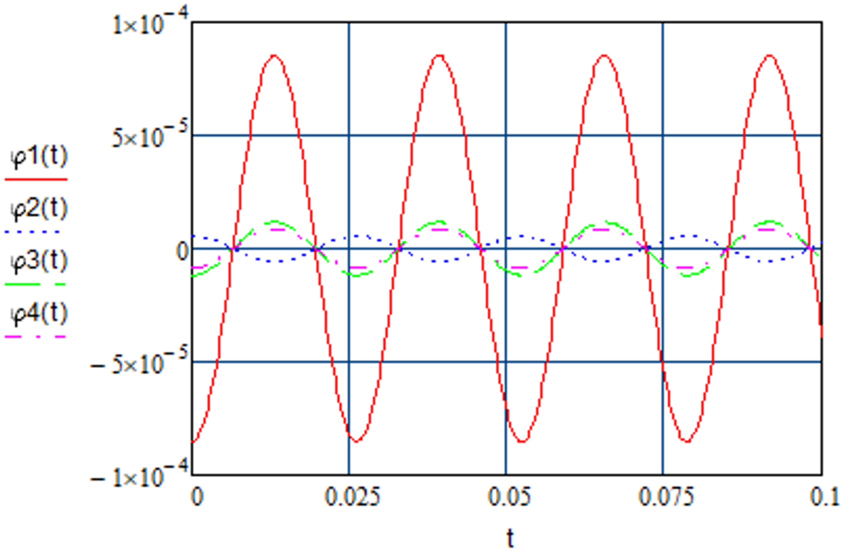

Fig. 7. Amplitudes of torsional oscillations by mass of WMU of UZTE16M locomotive at n=2290 rpm

4 Conclusion

As a result of the research, a new method of calculating torsional oscillations in the wheel-motor unit of UzTE16M locomotive has been developed. It was found that by increasing the TEM armature rotation frequency of the UzTE16M locomotive, the amplitude of torsional oscillations WMU will increase, and the period of oscillations WMU will decrease.

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