THE PARAMETRIC OSCILLATIONS OF STEEL FRICTION PLATES FOR A MULTIPLATE CLUTCHES

ПАРАМЕТРИЧЕСКИЕ КОЛЕБАНИЯ ФРИКЦИОННЫХ ДИСКОВ В ТРАНСМИССИЯХ ТРАНСПОРТНЫХ МАШИН

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Abstract: *The article substantiates the ways of tuning parametric oscillations providing for increasing durability of steel and friction plates of vehicle transmissions. The evaluation of dynamic stability is conducted on the basis of the analysis of Mathieu equation and Ince-Strutt diagram. It is offered to restrict the parameter of rigidity modulation depth by filtering low-frequency disturbances formed in the nonlinear system by the power unit (internal combustion engine (ICE) or electric motor (EM)), by hydrodynamic processes in the hydraulic transformer (torque converter or hydraulic coupling) and in the system of its oil supply. The performance evaluation of the developed actions on plate's durability is conducted.*

***KEYWORDS**: PLATE, FRICTION DESK, PARAMETRIC OSCILLATIONS, DYNAMIC STABILITY*

1. Introduction

The advanced multi-purpose tracked and wheeled vehicles operated in super severe conditions more and more perfect designs of hydromechanical, electromechanical and hybrid transmissions are to be developed [1]. Prototypes tests show the high dynamic loading which restricts component durability, in particular, the durability of plates of multiplate clutches and conjugated components [2, 3]. The article [4] considers the type of failure insufficiently investigated - plate disruption, which is observed in transmissions being developed and in designs of worldwide automotive leaders. It should be noted that the same type of failure is observed in the vehicles equipped with stepped ratio gear transmissions without torque converter and diesel engines.

The metallographic analysis of the disrupted plates shows that crack formation has a fatigue character, which is revealed within limited time of trial operation. It occurs due to generation of high-frequency oscillation. The article [4] shows that plate disruption take place because of resonant modes which are generated by hydrodynamic processes in the impeller-turbine interblade space and by working liquid pulsation in the oil supply circuit. However, because of the nonlinear elastic characteristics of plate and drum interaction at gap opening the parametric oscillations and resonances are excited which are more dangerous and are not eliminated with usual methods.

Thus, according to the aforesaid, there are three main points of the research subject. First, to study the processes of dynamic loading formation of wet clutch plates. Second, to prove the ways of durability increasing of wet clutch elements. Third, to improve the existing method of multiplate clutches calculation considering frequency characteristics and dynamic factors of loading.

Scientific novelty of the research involves to study conditions of resonance appearance. By the way, to develop the ways for eliminating resonant modes based on dynamics research of highly nonlinear system at polyharmonic disturbances. These disturbances are formed by a ICE or EM, by hydrodynamic processes in the impeller-turbine space of the torque converter and by pulsation of working liquid pressure in the oil supply circuit**.**

2. Analytical research of nonlinear system dynamics

The dynamic problem is solved on the basis of mathematical model of the system under study "Engine unit – Torque Converter – Planetary Transmission Elements (conjugated drum and lined plate)" (Fig. 1).

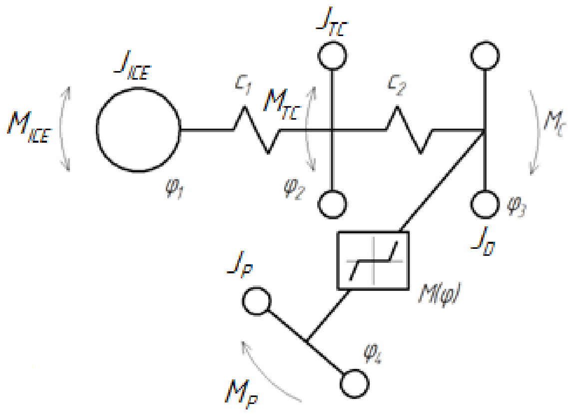

***Fig. 1.**Design model of the nonlinear system: the system structure.*

The scheme introduces the following notations: J_{ICE}, J_{TC}, J_D, J_P are moments of inertia of the engine, torque converter, drum and plate; c_1,c_2 are rigidities of the pre-torque-converter zone damper and transmission input shaft respectively; *M(φ)* is a nonlinear elastic characteristics of plate-to-drum interaction; φ_1; φ_2; φ_3; φ_4 are corresponding generalized coordinates of inertia masses. The movement of the elements is described by the following system of the differential equations (1):

$$\begin{cases} J_{ICE}\ddot{\varphi}_1 + c_1(\varphi_1 - \varphi_2) = H_0 + H_i sin(\omega_i t + \alpha_i) \\ J_{TC}\ddot{\varphi}_2 - c_1(\varphi_1 - \varphi_2) + c_2(\varphi_2 - \varphi_3) = H_m sin(\omega_m t + \alpha_m) \\ J_D\ddot{\varphi}_3 + b(\dot{\varphi}_3 - \dot{\varphi}_4) - c_2(\varphi_2 - \varphi_3) + M(\varphi) = M_C \\ J_P\ddot{\varphi}_4 - b(\dot{\varphi}_3 - \dot{\varphi}_4) - M(\varphi) = H_D \end{cases} \quad (1)$$

In this system H_i, ω_i, α_i are amplitudes, frequencies and initial stages of engines harmonics respectively; H_j,ω_j,α_j are amplitudes, frequencies and epoch angle of harmonics respectively, being formed in the impeller-turbine space of the torque converter; M_C is a reduced torque of resistance to motion; H_D is drag torque losses.

In the design model plate and drum connection is carried out on the clearance fit centering according to the diameter of the reference circle of teeth which providing axial movement of plate relatively to the drum when engage or disengage. The design model of plate and drum gear connection is shown in Fig. 2.

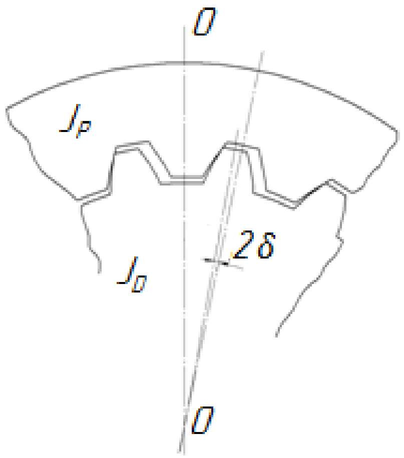

Fig. 2.*Design model of the nonlinear system: drum and plate connection.*

A nonlinear function - nonlinear elastic characteristics on angular coordinate, is schematized (Fig. 3) and is accepted symmetric with a gap, i.e.:

$$M(\varphi) = \begin{cases} 0 \ at \ |\varphi| \le \delta \\ c \cdot |\varphi| \cdot sign(\varphi) \ at \ |\varphi| > \delta. \end{cases}$$

The parameter δ was determined by engineering drawings (*δ=0,3 degrees)*; *c* is rigidity of the couple "plate tooth - drum tooth" on the basis of modeling a stress-strain behavior of tooth gearing of a plate with a drum (*$c=6.845 \cdot 10^5$ N·m/rad)*. For determination of plate-to-drum rigidity it is necessary to define the uniformity factor of loading between teeth. Mutual position of drum relatively to plate has a periodic character during functioning. For a quantitative evaluation of the disturbances formed by the nonlinear system under the study, it is accepted that contact rigidity is defined by one, two, three, etc. couples of teeth, and during the parallel work of elements the rigidity is summarized.

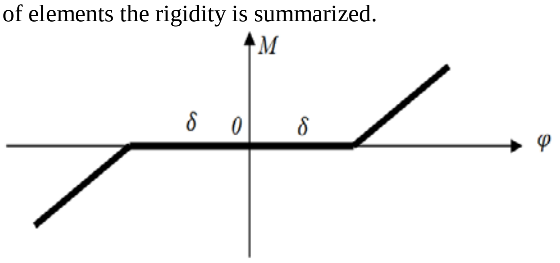

Fig. 3.*Idealization of the nonlinear function, dependence of the moment on an angular coordinate (nonlinearity with a clearance).*

The results of modeling the systems dynamics are shown in Fig. 4.

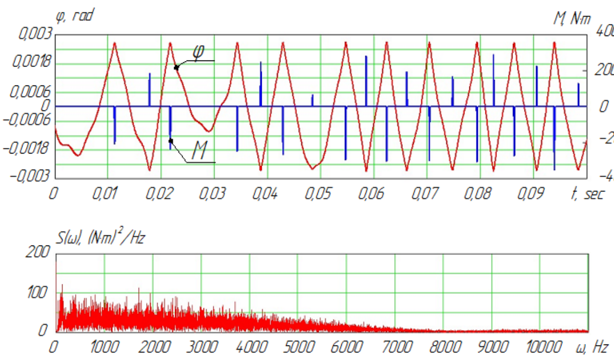

Fig. 4.*Results of modeling the system dynamics.*

The results of the computational solution of the system (1) show that nonlinear high-frequency interaction plate-to-drum generates a wide disturbing forced spectrum (from 0 to 6,000 Hz) with high power. Thus, the spectrum of forced frequencies crosses the spectrum of the system natural frequencies. By the way, this spectrum crosses forms of plate oscillations (if system is considered as system with the distributed parameters). This fact can provide generation of resonant oscillations of plates. The analysis for eliminating of resonances appearance was performed. This problem is solved on the basis of studying two-mass mathematical model of the nonlinear system “Drum – Plate”. This model accounts on equivalent moment of inertia of TC turbine, elements of planetary gear sets, pack of *I* plates, gap between plate and drum. The model was modified to a nonlinear differential equation (2) after addition a coordinate of the relative angular movement $\varphi = \varphi_1 - \varphi_2$:

$$J_{dr}\ddot{\varphi} + b\dot{\varphi} + M(\varphi) = M(t) \quad (2)$$

In this equation J_{dr} is a driven inertia moment $\left(J_{dr} = \frac{J_D \cdot J_P \cdot i}{J_D + J_P \cdot i}\right)$; *b* is a dissipation factor depending on a frequency, oscillation amplitude and the hysteresis loop area (which occures due to imperfectly elastic collision of the plate with the drum, *b=4.5 N·m·sec*); *M(t)*is temporary function of the disturbing torque formed by the engine, by the hydrodynamic processes in the impeller-turbine space of the torque converter, by the pulsation of working liquid pressure in the oil supply circuit; J_P is a moment of plate inertia; *i* is a number of plates. The moment of inertia J_{dr} calculated with number of assumption. In this regard, the value J_{dr} is determined by experimental data in compliance with the frequency of its natural oscillations in the dynamic system.

3. *Analysis of stability of nonlinear system periodic solutions*

It is known that the solution of the nonlinear equation (2) is ambiguous and several stationary modes with various amplitudes $\varphi_i = \varphi_i(t)$ may be existed as well as unstable.

The analysis of stability is carried out with application of parametrical oscillation approach. For this purpose the mathematical model (2) is modified to Mathieu equation [5,6]:

$$\ddot{\varphi} + 2\varepsilon\dot{\varphi} + \omega_0^2\left[1 - \frac{q_{dyn}cos(pt)}{q_{st}}\right]\varphi = 0 \qquad (3)$$

The equation does not take into account the presence of dissipative forces. The moment of friction in off-mode, when plates free rotate, is considered when determining a modulation parameter in the formula $\frac{q_{д}}{q_{ст}}$ in formula (3) and determines the value of the parameter h in the Ince-Strutt diagram. Evaluation of the numerical values of dissipation coefficient b in the system of equations (1), arising as a result of inelastic interactions of the plate with the drum, has no significant effect on the value of the natural frequency of the dynamic system and unsignificantly reduces the instability area in the Ince-Strutt diagram (Fig. 5, blue line) .

In this equation $= \frac{b}{2J_{dr}}$; ω_0^2 is a square of natural oscillations frequency of the nonlinear system corresponding to the value of harmonic linearization factor q_{dyn}; *p* is external forcing frequency.

The rigidity of nonlinear interaction «Drum-to-Plate» was taken as modulation depth $2\mu = \frac{q_{dyn(A)}}{q_{st(A)}}$. The numerator was determined by four-mass-model solution. The dominator was determined according to the drag torque.

For the assumed values of the amplitudes of angular oscillation values $q_{dyn(A)}$ stiffness *(A)* and $q_{st(A)}$ (A) determined by the method of harmonic linearization (4) for the above non-linearity (Fig. 3):

$$q(A) = c_0 - \frac{2c_0}{\pi}\left(arcsin\frac{\delta}{A} + \frac{\delta}{A}\sqrt{1 - \frac{\delta^2}{A^2}}\right), \quad (4)$$

where *A* is an angular oscillation amplitude; δ is a gap (play). It is important to note that an assumed value in computation $q_{dyn(A)}$ gives the value ω_0 within the range between 600 and 700 Hz, which is close enough to the lowest natural frequency (the second form of oscillations of the multiplate clutch lined plate) of the transmission under study.

Next, the Mathieu equation was modified to the parameters of the Ince-Strutt diagram *a* and *h* (Fig. 5). Entering the parameters of rigidity modulation depth μ and the frequency of its alternation p, Mathieu equation without taking into consideration dissipation is modified to (5):

$$\ddot{\varphi} + [a - 2h\cos(2\tau)]\varphi = 0. \quad (5)$$

In this equation a and h are parameters (abscissa and ordinate) of Ince-Strutt diagram (Fig. 5): $a = \left(\frac{2\omega}{p}\right)^2$; $h = a\mu$; $2\tau = pt$. Such representation allows tor analyze stability of parametrical oscillations according to the specified diagram and according to the positions of points with given coordinates a and h.

On the diagram the instability regions are shaded, stability regions are not shaded. In the Figure 5 a straight line 1 characterize the function $h = a\mu$ of the dynamic system "Drum - Plate" under study, (p=1,200 and 680 Hz, $\mu = 1.275$). As it comes from the diagram practically in all the range of functioning Line 1 is in instability region. This means that parametric oscillations take place.

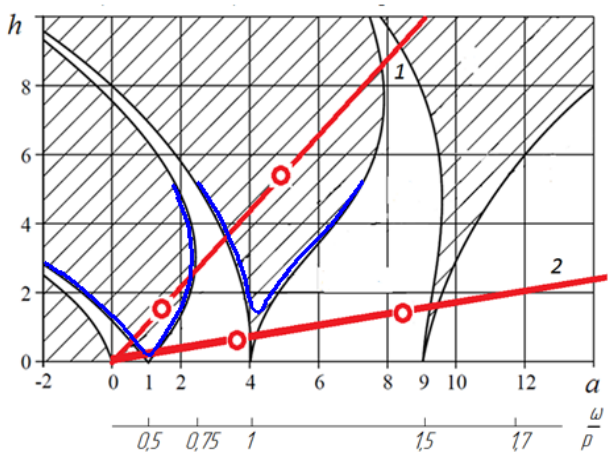

Fig. 5.*Ince-Strutt diagram for the system under study.*

For a simplification of data analysis in the diagram the scale of natural frequency to forcing frequency ratio is changed in addition ω/p. For analysis simplification the diagram was redrawn in the dependence of depth rigidity on frequencies relation (see Fig. 6). Fig. 6 shows zones of the main and the second parametrical resonances (the shaded areas) with damping logarithmic decrement of attenuation Δ=0.3.

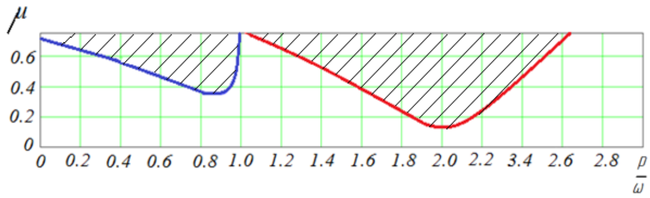

Fig. 6 - Zones of stability in relation to forcing frequency p for the main and the second parametrical resonance.

As follows from the diagram, it is possible to prevent parametric oscillations in two ways. First, to increase dissipation in the system. On the diagram it is reflected as lifting of the tips above abscissa axis. And second, to reduce rigidity depth to less than 0,1. The first way is structurally pointless. The second one is possible.

Structurally it may be realized by installation of a filter of low-frequency oscillations which will not let high-frequency disturbances generated to be transferred to the drum. This analytical conclusion received on the basis of the Ince-Strutt diagram, analysis is verified when performing computational modeling according to the model (1).

Besides, the required values of the modulation depth and natural frequency of the system providing expansion of the stability zone are established. For the system under study decrease in modulation depth parameter μ up to the value *0.1* (Line 2 in Fig. 5) allows to narrow essentially the range of parametrical resonant oscillation. The variation of modulation depth parameter μ is implemented by introduction of the low-frequency oscillation filter, which carries out filtration of the high-frequency disturbances generated in the system and vibroprotection of the drum and multiplate clutch elements. As a result a low-frequency filter is synthesized. The results of the synthesis are implemented, for example, in the form of development of the torsion damper, which can be installed between a turbine wheel of the torque converter and input shaft of the planetary gearbox.

During the research an experimental test was carried out. This test proved the idea about exclusion of a high-frequency disturbance by means of low-frequency filter. The object of the research was the hydromechanical transmission, in which the plates of one of multiplate clutches in this transmission were in resonant mode at a frequency of 456 Hz due to nonlinear disturbance from electric motor and resulting torsional vibrations (Fig. 7). Two torque measurement sensors were installed in the transmission. The first one was connected with the shaft which is separated from the multiplate clutch by torsion damper (sensor A). The second sensor measured torsion torque of the shaft, which had rigid kinematical connection with the drum of the multiplate clutch under study. The electric motor was a source of vibrations in the transmission. The results of the test show that sensor A did not register oscillations at the resonant frequency of 456 Hz, but sensor B did. This proves efficiency of the suggested method of exclusion of resonant oscillations (as superharmonic) in non-linear system due to installation of torsion damper (which operates in the capacity of a low-frequency-filter).

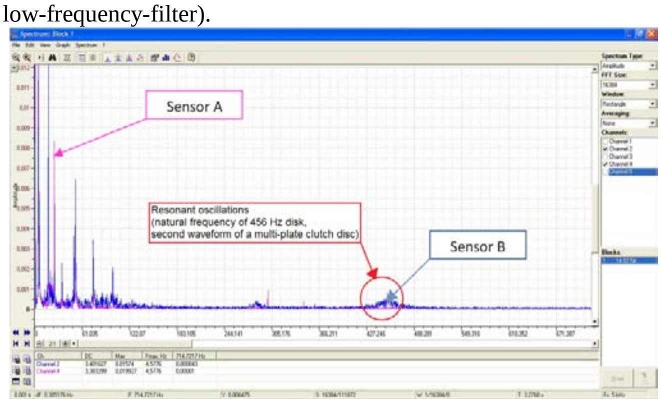

Fig. 7. *Screenshot of the spectrum analyzer to confirm the idea of exclusions high-frequency perturbations due to the use of low-pass filter*

Figure 8 shows the results of "Engine – Torque Converter – Drum – Plate" nonlinear system dynamics modeling after performing actions connected with tuning out parametrical resonant oscillations according to suggested way. Comparing the results of modeling (Fig. 4 and Fig. 8) it is evident that the power of disturbing spectral density generated by the system is reduced in comparison with the initial one over 2,5 times, and the amplitude of the moment decreases over 3 times. The assessment of fatigue durability of the plate design shows that if tuning parametrical resonances by the suggested way, the plate resource raises enormously.

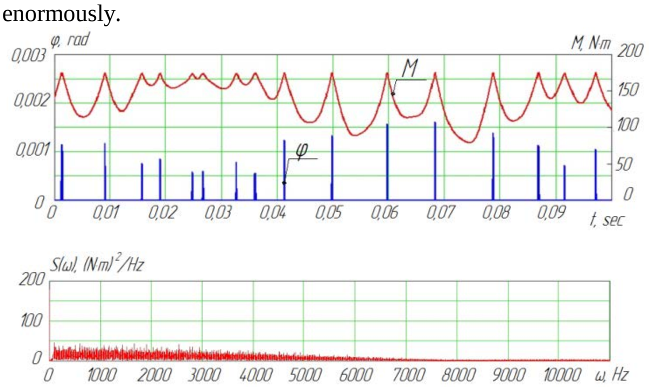

Fig. 8.*The results of "Engine – Torque Converter – Drum – Lined Plate" nonlinear system dynamics modeling, after performing actions connected with tuning out parametrical resonant oscillations by suggested way.*

Summary

The new results about formation of dynamic loading of multiplate clutches plates are the basis of development of the described design calculation method and is distinguished from existing one by taking into account probability of generating parametrical resonant oscillations.

According to the research results it is established that the resonant mode in the multiplate clutch plates can be generated at various frequencies and by various disturbing sources. Resonance disturbance in the nonlinear system can be generated by oscillations of the ICE and EM torque, as well as by hydrodynamic processes in the impeller-turbine passage circulation of the torque converter, by oscillations of working liquid pressure in the system of its oil supply. Tuning out resonances at one of its frequencies can excite it on the adjacent ones. In this regard the most effective way of eliminating resonant modes is to filter oscillations generating a resonance.

Filtration of oscillations in pre- and post-converter zones can be carried out by synthesis of a high-frequency oscillation damper eliminating resonance in these zones, practically without receiving [5, 6] disturbances with frequencies significantly exceeding its natural frequency. Thus, to filter high-frequency oscillations generated in the transmission, as well as formed by a nonlinear characteristic of elastic interaction of tooth gearing of a drum and a plate, requires introduction of an additional torsion damper.

The executed assessment of the performance of the developed actions shows that elimination of resonant modes significantly increases durability of multiplate clutch plates.

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