SUBHARMONIC RESONANCES IN THE HYDROMECHANICAL TRANSMISSION OF THE WHEELED CHASSIS

СУБГАРМОНИЧЕСКИЕ РЕЗОНАНСЫ В ГИДРОМЕХАНИЧЕСКОЙ ТРАНСМИССИИ КОЛЕСНОГО ШАССИ

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Abstract: *In the article on the basis of the developed mathematical model the dynamics of the highly nonlinear system is investigated, consistent patterns of resonant mode occurrences are established, which is experimentally confirmed. It is established that one of the main reasons for high dynamic loading and HMT elements durability reduction is subharmonic resonances caused by close agreement of frequencies of free nonlinear system fluctuations with multiple value of frequencies of diesel engine disturbances. On the basis of the new consistent patterns, the area of stability and parameter move direction is defined for its provision. The offered damper design is synthesized for significantly reducing HMT dynamic loading.*

***KEYWORDS**: OSCILLATIONS, NATURAL FREQUENCY, SUBHARMONIC RESONANCE, DYNAMIC LOADING HYDROMECHANICAL TRANSMISSION*

1. Introduction

The Multiple Wheel Chassis are widely applied both as vehicles and as processing equipment in the Russia’s oil and gas sector in extreme operational conditions of Far North, the Polar Urals and West Siberia. The specified chassis is equipped with single-type hydromechanical transmissions which reliability is limited in many respects, in particular, in durability of matching reduction gearboxes [1].

Allison Company offers a project for modernization of MWC by installation of the engine and transmission assembly. However, the cost of the project makes 14 million RUB, which almost four times exceeds the cost of MWC complete overhaul. An increase in durability of matching reduction gearboxes, design development, their implementation, i.e. transmission modernization in the course of chassis maintenance is an effective way of increasing its reliability.

Application of a matching reduction gearbox with several tooth gears providing kinematic match of engine and hydraulic torque convertor characteristics is a peculiar feature of the transmission design under study. Strong nonlinearity of the system is caused by gear backlash opening in tooth gears and existing resonance elimination methods in this case are inefficient. In this regard, the proposed work devoted to analyzing the conditions of subharmonic resonant mode occurrences in the strongly nonlinear system and to substantiating the method of their detuning is topical.

The objective of this work is to define the ways of increasing durability of matching reduction gearboxes of MWC hydromechanical transmissions, to develop and implement design solutions providing detuning from subharmonic resonant oscillations on the basis of a new type torque vibration damper synthesis.

Scientific novelty of the work lies in studying consistent patterns of generation and provision of rationale for a method of eliminating subharmonic resonant modes in the system "Diesel Engine – Hydromechanical Transmission" on the basis of studying the dynamics of the strong nonlinear system with polyharmonic disturbances from the diesel engine.

2. The background of the problem and rationale for the key questions to be explored

Transmission dynamic loading is determined by influence of considerable sign-variable torques occurring in the steady-state regime under resonances, as well as under transient regime of engine start-up and dying out, acceleration and braking operation, gear shifting and torque convertor lockup.

The analysis of the failure rate shows that the greatest amount of element breakage of the mechanical system "Engine – Transmission - Transport Vehicle" deals with the part between the diesel (the activator of mechanical oscillations) and a pump wheel of the torque converter.

The solution of specific problems is hindered due to absence of an objective method for choosing a type of damper design and its parameter identification. It predetermines a large volume of experimental and test and evaluation work at the stage of developing the design when modification demands essential time, work and material expenditures.

The background for the research is scientific development of many domestic and foreign experts. Most fully the technique of tuning out resonant modes on the basis of oscillation damper synthesis is stated in the works of Grishkevich A. [1]. The results of researching the dynamic behavior of nonlinear systems leading to generation of such phenomena as clunk noise and shuffle are given in works of A. Crowther, R. Singh, Stahl, K., Pflaum, H., Meingassner, G. J., Lohmann, B. et al. [2,3,4,5]. The solution of this problem is stated in practical methods and reference books of Centa Company (Germany). The results of the company's research and its development are widely applied in mechanical engineering around the world, including Russia.

However, the existing works don't allow for considering high nonlinearity of the elastic characteristics peculiar to the system under study.

On the basis of the analysis of the scientific works devoted to research and design of oscillation dampers, the conclusion is made that reduction in dynamic loading and damper synthesis, which consider real nonlinear properties, does not seem possible due to a complex interrelation of the system elements. The article substantiates the necessity of carrying out theoretical and pilot studies with application of up-to-date methods of nonlinear mechanics, of parametrical oscillation stability assessment, of simulation modeling and of processing of experimental data.

3. Theoretical research of dynamic loading

Hydromechanical transmission is a complex multiunit mechanical nonlinear system of a variable structure containing ring elements.

The detailed analysis of this model has allowed for drawing a conclusion that the dynamic torque on transmission shafts is formed by:

- a periodic component of the engine torque, both in steady-state and non-steady-state operating regimes;
- the dynamics of the mechanical system under transients of standing start, acceleration, gear shifting and torque converter lockup.

The dynamic loading of the matching reduction gearbox design elements (the kinematic scheme is given in Fig.1) is a defining factor limiting durability of engine and transmission assembly for the MWC family under study.

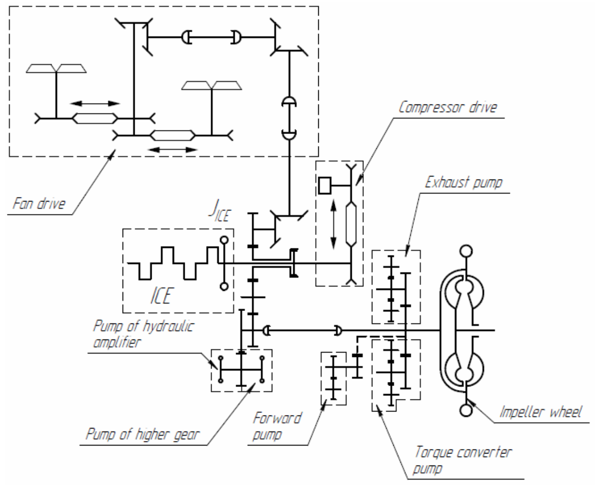


Fig. 1. *Kinematic scheme of pre-torque-converter zone*

The analysis has shown that the dynamic loading of the matching reduction gearbox design elements can be effectively defined on the basis of studying the dynamics of a two-mass system – the so-called pre-torque-converter zone [1]. This system (the computational scheme is given in Fig. 2) comprises an engine flywheel (the first mass), a torque converter impeller wheel (the second mass) with the inertial masses of a matching reduction gearbox (cooling system fans, compressors, etc.) connected to it.

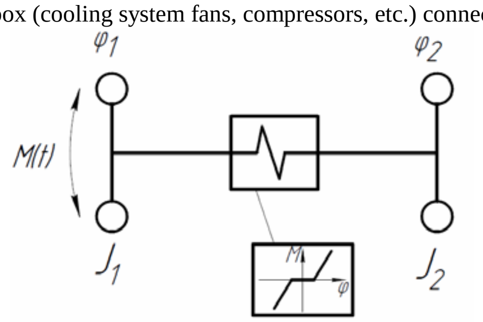


Fig. 2. *Design model of pre-torque-converter zone.*

The nature of interrelation between these two masses is built up by elastic and dissipative characteristics of the torsion damper (torsion, see Fig. 2) and tooth gears of a matching reduction gearbox, which backlash opening forms nonlinearity of an elastic interaction. The analysis of the dynamic process in the system under study and predetermination of the ways for reduction in dynamic loading are made on the basis of a mathematical model of the pre-torque-converter zone of the hydromechanical transmission as a nonlinear two-mass system (1).

$$\begin{cases} J_1\ddot{\varphi}_1 + \mathrm{M}(\varphi) = M(t) \\ J_2\ddot{\varphi}_2 - \mathrm{M}(\varphi) = 0 \\ \varphi = \varphi_1 - \varphi_2 \end{cases} \tag{1}$$

where J_1, J_2 are inertia moments of the engine and the impellewr respectively; $M(\varphi)$ is nonlinear elastic interaction function (Fig. 3); $M(t)$ - time function of engine and impeller wheel torque.

Nonlinear function - dependence of the moment on relative angular coordinate – is schematized (Fig. 3) and is accepted as symmetric with a dead zone without saturation [7], i.e.

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

the parameters of which δ and c are defined experimentally.

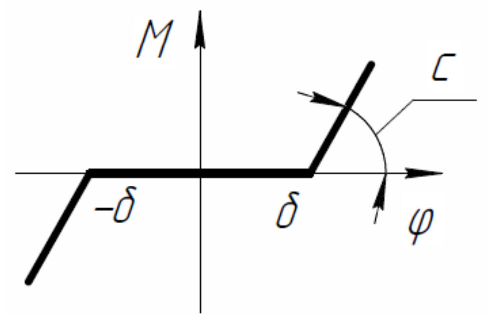


Fig. 3.*Schematization of nonlinear function with dependence of the moment on the angular coordinate (nonlinearity with a dead band without saturation).*

The time function of the engine torque *M(t)* is accepted in Fourier's series form, the parameters of which are defined according to the data of the manufacturing factory.

$$M(t) = M_0 + \sum_{m=1}^{\infty} M_m \cos(m\omega t + \beta_m)$$

The harmonious analysis of the engine torque shows that the most dangerous harmonics of the engines YaMZ-8401, YaMZ-240, D-12 (according to the manufacturer, JSC Avtodiesel) is the sixth, i.e. in the engine rpm range, the exciting frequency makes from 70 to 85 Hz that considerably (3 … 4 times) exceeds the frequencies of natural oscillations of the system under study (12 …17 Hz).

The results of the numerical solution (1) at values of the parameters corresponding to the object of the pilot study are given in Fig. 4. As comes from this Fig. 4 oscillations of the torque occur regarding the established value with amplitude 3 … 4 times exceeding the established value.

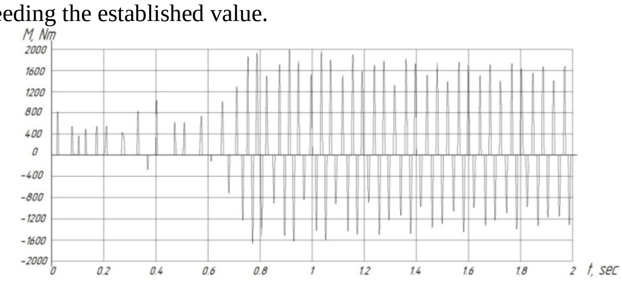
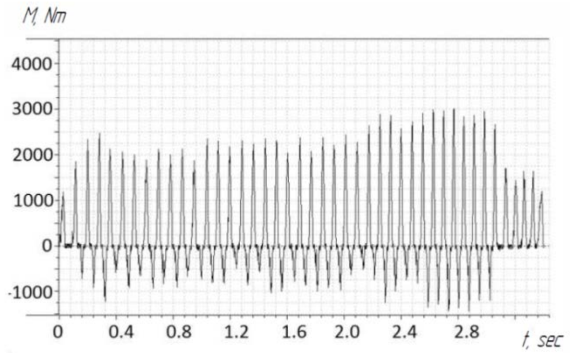


Fig. 4. *The results of the numerical solution (1) at values of the parameters corresponding to the object of the pilot study (n_{ICE}=740 rev/min)*

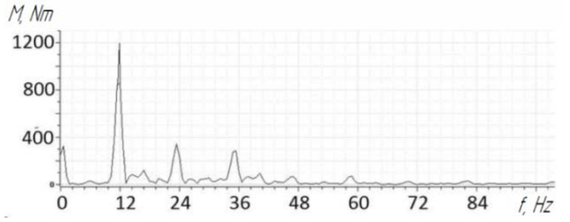
The characteristics of the periodic process under study (moment amplitude, natural oscillations) significantly depend on the initial conditions which is characteristic for nonlinear systems. In fig. 4, two various types of movement characterized by the size of backlash opening are shown. It leads both to an amplitude change and frequency of periodic oscillations of the system under study.

The results of numerical modeling are confirmed experimentally at running trials of the wheel chassis KZKT-7428, equipped with YaMZ-8401.10. Similar results were obtained at trials of wheeled chassis KZKT-74286 with engine YaMZ-240 and MWC MAZ-537 with the engine D-12 (the trial results of KZKT-74286 and MAZ-537 are not given).

The fragment of the oscillogram of the torque changing in the steady-state mode, at the engine rpm range of 700 … 850 rpm, is given in Fig. 5a. From the provided data, it follows that the change of the torque has an oscillatory character with an amplitude up to 2,500 Newton meters, and the process frequency makes from 10 to 19 Hz with a disturbing frequency of the engine from 70 to 85 Hz (the 6th main engine harmonics of the engine YaMZ-8401). This alternating character of the torque within the time frame (Fig. 4, Fig. 5), corresponds to subharmonic resonant oscillations in the nonlinear mechanical system with a dead zone.



a)



b)

Figure 5. *The fragment of the oscillogram of the torque changing at the torsion shaft (a) and its spectrum (b) in an idle mode (NICE=740 rev/min.)*

The spectral analysis of the obtained numerical solution verifies the abovementioned conclusion about generation of oscillations at subharmonic frequencies multiple times smaller than the main harmonics (the 6^{th}) of the YaMZ-8401 engine. In this case, the subharmonic resonance oscillations in nonlinear systems coexist with principal forced oscillations.

Thus, the solution of the system of nonlinear differential equations is not unambiguous. In addition, existence of several steady-state modes with various amplitudes is possible $\varphi_i = \varphi_i(t)$, including an unstable mode. Research of oscillation process stability and provision of a rationale for the direction of resonant modes elimination is carried out on the basis of the device of parametrical oscillations [8]. To implement this approach, an equation system of (1) by introduction of a coordinate of relative angular movement $\varphi = \varphi_1 - \varphi_2$ is reduced to the form of nonlinear differential equation (2)

$$J_{reduced}\ddot{\varphi} + M(\varphi) = M(t), \qquad (2)$$

where $J_{reduced}$ – reduced moment of inertia. Other functions introduced into the equation (2) are given above. The complexity of the analytical determination of the reduced inertia moment is that it is necessary along with the moment of flywheel inertia to consider response time of the torque converter impeller filled with hydraulic fluid. Besides, in the design of MWC transmission, it is from the pump wheel that many mechanical devices are actuated (fans, pumps, compressors, etc.). In this regard this size is determined experimentally [6] ($J_{ICE} = 8{,}5\ kgm^2$).

To analyze stability of the system, the mathematical (2) model is modified to Mathieu equation [8]:

$$\ddot{\varphi} + 2\varepsilon\dot{\varphi} + \omega^2\left[1 - \frac{q_{dyn}\cos(\omega_6 t)}{q_{st}}\right]\varphi = 0 \qquad (3)$$

In this equation $\varepsilon = {}^{b}/_{2J_{reduced}}$ is a dissipation parameter; ω^2 is a square of natural frequencies of the nonlinear system, determined considering the constant component of the engine moment. The value of this moment is determined by losses in the matching reduction gearbox, by losses on overcoming resistance in the compressor drive, the drive of fans, etc. The frequency of natural oscillations is defined by the following response characteristics:

$$\omega = \sqrt{\frac{c - \frac{c}{\pi}\left(arcsin\frac{\delta - x}{\varphi} + arcsin\frac{\delta + x}{\varphi} + \frac{\delta - x}{\varphi}\sqrt{1 - \frac{(\delta - x)^2}{\varphi^2}} + \frac{\delta + x}{\varphi}\sqrt{1 - \frac{(\delta + x)^2}{\varphi^2}}\right)}{J_{reduced}}}, \qquad 4)$$

where φ is angular amplitude at the neutral point in the transmission and at the shaft speed matching to idle speed; x is the shift of the zero line determined by a constant component of loading. Introducing the parameters of depth of disturbance μ and its alternation frequency p, Mathieu equation without considering dissipation is modified to

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

where $a = \left(\frac{2\omega}{p}\right)^2$; $h = a\mu$; $2\tau = pt$. Ince-Strutt diagram with the coordinates *h(a)* is given in Fig. 6 (stability area is shaded). The parameters of the system under study are represented by point A, being in the unstable zone. The coordinates of the point are determined as follows. The abscissa of the point A is determined according to the formula $a = \left(\frac{2\omega}{p}\right)^2$, where ω is the frequency of natural oscillations of the nonlinear system, determined according to the formula (4) and corresponding to the amplitude of the moment of the main motor harmonics with a frequency p (p=6, M=1,200 Nm). To determine the parameter *h*, it is necessary to evaluate modulation depth μ, characterizing the amplitude of a variable parameter of the system (relation ${}^{q_{dyn}}/_{q_{st}}$ in formula (3)) being determined for this system according to the formula (6).

$$\mu = \frac{c - \frac{c}{\pi}\left(arcsin\frac{\delta - x}{\varphi_d} + arcsin\frac{\delta + x}{\varphi_d} + \frac{\delta - x}{\varphi_d}\sqrt{1 - \frac{(\delta - x)^2}{{\varphi_d}^2}} + \frac{\delta + x}{\varphi_d}\sqrt{1 - \frac{(\delta + x)^2}{{\varphi_d}^2}}\right)}{c - \frac{2c}{\pi}\left(arcsin\frac{\delta}{\varphi_s} + \frac{\delta}{\varphi_s}\sqrt{1 - \frac{(\delta)^2}{{\varphi_s}^2}}\right)} - 1, \qquad (6)$$

where c is rigidity of an elastic element (torsion), $c = 72{,}000\ Nm/rad$; φ_d is the amplitude of the sixth engine harmonics; φ_s is the amplitude corresponding to the engine load moment (loss torque); δ is a dead zone defined by backlashes in the gear of the matching reduction bocgear (δ=0.026 rad); *x* is a shift of the zero line (*x*=0.029).

For the given parameters at rpm engine speed of 850 rev/min, Ince-Strutt diagram parameters makes $a = 0.07$ $h = 1.033$.

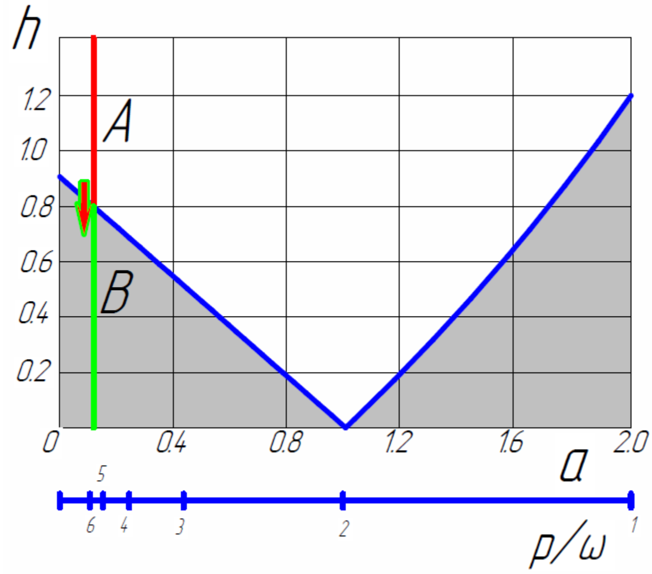


Fig. 6. *Ince-Strutt Diagram.*

Analyzing the results of numerous calculations, corresponding to the diagram it appears that at the given parameters of the dynamic system in the range of parameter values from 0 to 0.111 (frequency range of the disturbance we are interested which is determined by the detuning factor $\frac{p}{\omega} = 6$ at the minimum allowable speed ICE – 600 rev/min) the parameter *h* exceeds the value of 0.9. Therefore, all estimated values of set points in h-a coordinates reside in an instability zone ***A*** (shaded) to the left from the red line. Such a situation remains unchanged up to 900 rev/min when periodic oscillations of the dynamical system have an unstable (resonance) character.

Solution of the inverse problem of choosing optimal parameters of a dynamical system, being unable to control the modulation depth μ and to change the characteristics of the nonlinear elastic relationship (dead zone value), is possible only if to change the setting of detuning the system ${}^{p}/_{\omega}$ (relation ${}^{p}/_{\omega}$ is given in Fig. 6 as an additional axis of abscissa).Thus, the stability is ensured by choosing torsion vibration damper compliance considering the condition that an operating point with coordinates *h – a* will get into the shaded zone ***B***, located in the diagram to the left of the green line.

For example, taking the minimum allowable point of rpm engine speed at 600 rev/min an gradually reducing the rigidity of the elastic element, we determine the Ince-Strutt diagram parameters. The calculation is performed before moving the operating point with coordinates *h – a* from unstable zone to the stable one, i.e. from zone ***A*** to zone ***B***. With respect to the dynamical system, the desired effect is achieved at the rigidity of the elastic element of 45,000 … 50,000 Nm/rad. Furthermore, the Ince-Strutt diagram parameters made $a=0.089$ $h=0.684$ *at* $c=45000$ *Nm/rad and* $a=0.098$ $h=0.854$ *at* $c=50000$ *Nm/rad.*

When the reliability of the results obtained by comparison with experimental data, they can become the basis for the method of limiting torque oscillation amplitudes and, respectively, of dynamic loading.

4. Analysis of the research results

Qualitative and quantitative comparison of the research results was carried out on the base of analysis for amplitudes and frequencies of high frequency oscillations in the pre-torque-converter zone (see. Fig 3 and Fig 4). Some differences of amplitude and frequency might be explained by deviation of initial conditions when modeling.

On the basis of the research results the method of elimination of resonant modes in the nonlinear system is developed. According to nonlinear elastic characteristic parameters on the basis of the developed mathematical model, the zone of stability and probability of subharmonic resonant mode generation is defined. For their eliminating, it is necessary to vary parameters of Ince-Strutt diagram (h and a) from a condition of getting into the stability zone. The most effective and easy way is tuning out the natural frequency of the system by changing angular rigidity of matching reduction gearbox. In particular, for reduction in dynamic loading of the HMT under study by elimination of a resonant mode, it is necessary that torsion damper angular rigidity was not higher than 50,000 Newton meters per radian.

Accordingly, a design of a matching reduction gearbox (Fig. 7) with a new type of torsion vibration damper with the rigidity of 21,000 Nm/rad. This silicone elastic element comes from the line of the Centa Company dampers and fit well into the existing design of the matching gearbox

The proposed design of the matching gearbox (Fig. 7) is produced as a three-shaft transmission with overdrive skew gears. In the case 1 on the bearing supports, there is a drive shaft-mounted gear 2, intermediate 3 and driven 4 shafts. The case 1 is fastened with the bolts to the housing 5 of the engine flywheel 6. The support 9 is connected with the engine flywheel 6. The disk 10 is connected to the support 9 with female splines. The elastic element, made as a silicone ring 11, with the help of the splines on the outside surface is connected with the disk 10, and the splines on the internal surface with multiple piece hub group 12, carried by bearings 15, installed in the support 9. The transmission shutdown mechanism made in the form of a two-stage gear coupling located in the cavity formed by the front part of the case. The gear clutch consists of a driven half coupling 8, connected by internal splines with the left end of the extended drive-shaft of the gear 2. The half-coupling 8 is connected by internal splines with the sleeve 7. Thus, the sleeve 7 provides for connection of the driven half coupling 8 with multiple piece hub group 12 on the outside surface where there are splines. At that, the sleeve 7 is fixed at two positions with ball locking mechanism 16. The driven shaft 4 through cardan shaft rotates a pump wheel of the torque convertor of hydromechanical transmission of the transport vehicle (fig. 7 does not show it).

The proposed design of the matching reduction boxgear works in the following way. With steady rotation of the shaft, the flywheel 6, the support 9, the disk 10, the elastic ring 11, the multiple hub group 12 due to the sleeve connection 7 in an operating condition, the drive shaft-mounted gear 2 rotate as one unit. As the tooth rim of the drive shaft-mounted gear 2 by means of the gear 13 of the intermediate shaft 3 is connected with the gear 14 of the driven shaft 4, the latter rotates taking into account the gear ratio transmission with an angular velocity greater than the drive shaft speed.

With unsteady rotation of the shaft, the flywheel 6, the support 9, the disk 10, can oscillate in regard to the multiple hub group 12, and respectively, to the drive shaft-mounted gear 2 within the angular allowable deformation of the elastic ring 11.

As it is shown above, the parameters of the elastic ring 11 are chosen to provide the dynamic system stability, decrease the frequency of natural oscillations of the mechanical system «Engine-Damper-Impeller Wheel» take resonant modes out of the limits of an engine rpm speed operating range. This provides the desired resource of the matching reduction gearbox.

The efficiency of research results is defined on the basis of experimental determination of torque convertor loading at trial runs.

Dynamic loading of the hydromechanical transmission with an improved design of a matching reduction gearbox is evaluated at road test simulation. Efficiency of the proposed solutions was evaluated by a value of the dynamic torque on the drive shaft according to the amplitude and reverse characteristic (see Fig. 8).

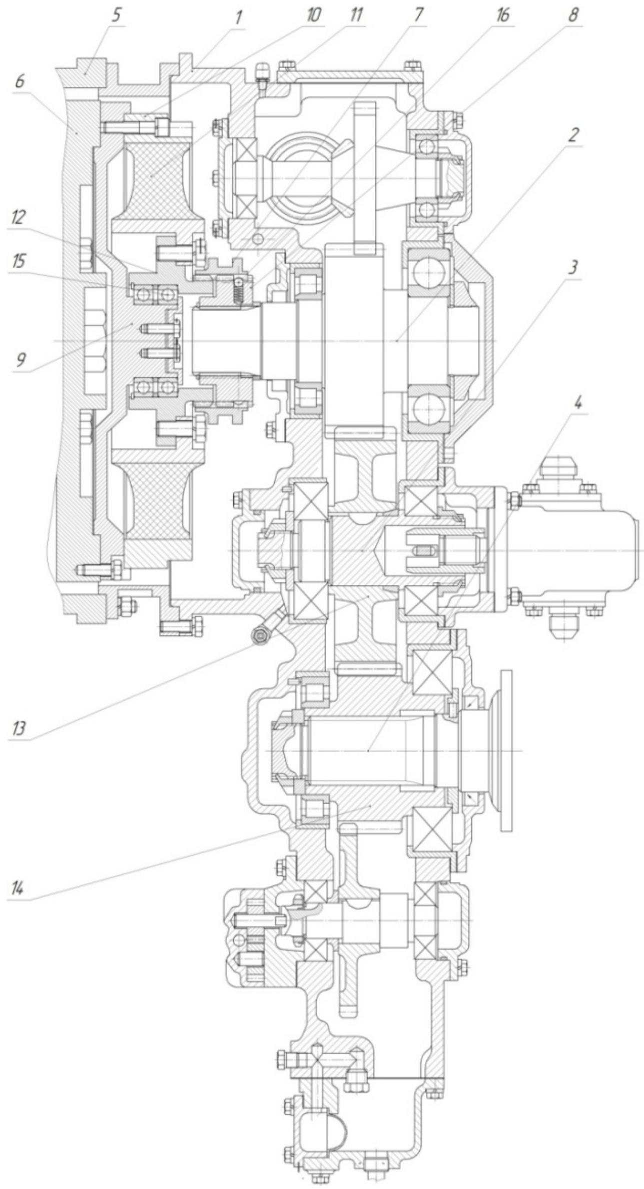


Fig. 7. *Modernized design of matching reduction gearbox.*

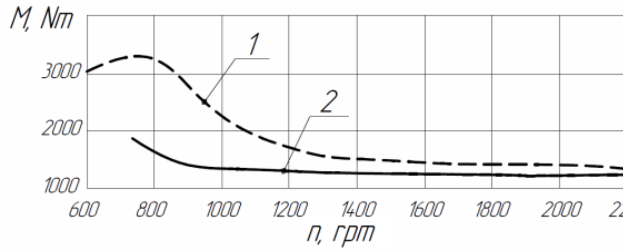


Fig. 8. *Amplitude and reverse characteristics of the dynamic torque in the pre-torque-converter zone of the HMT: 1 – original (serial produced) version of a matching reduction gearbox; 2 – the developed sample of a matching reduction gearbox.*

Comparison of the amplitude and reverse characteristics (lines 1, 2 in Fig. 5) shows that the proposed design of a matching reduction gearbox allows for tuning resonance out of engine rpm working range and reduce the dynamic torque in 5... 6 times. Thus, durability of matching reduction gearbox elements increased significantly.

Conclusions

1. The mathematical model applied in the research and a package of computer programs give a possibility to investigate dynamics of highly nonlinear system, to determine consistent patterns of generating resonant modes, verified experimentally

2. It is substantiated that one of the main reasons for high dynamic loading and HMT element durability restriction is subharmonic resonance caused by close coincidence of natural frequencies of the nonlinear system with multiple value of diesel engine disturbing frequencies. On the basis of the determined consistent patterns the area of stability and the direction of parameter variation for its maintenance is defined.

3. The conducted pilot researches of dynamic loading of the hydromechanical transmission of three MWC models, statistical data processing confirms validity of the developed mathematical model reflecting physical processes in resonant modes and of correctness of the basic assumptions.

4. On the basis of the obtained results, the engineering solutions, allowing for reducing dynamic loading of the transmission are proposed and designs of new type dampers are developed for matching reduction gearboxes of the HMT of various models of MWC. It allows for tuning resonant modes out of engine rpm working range and for reducing dynamic loading of the transmission up to 5 … 6 times, thus predetermining an increase of matching reduction gearbox elements durability.

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