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DYNAMIC DAMPING TORSIONAL VIBRATIONS IN THE TRANSMISSION OF REAR-WHEEL DRIVE AND ALL-WHEEL DRIVE VEHICLES

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ABSTRACT

A new system of dynamic absorption of vibrations in the vehicle's transmission is observed in this article. The proposed solution provides the absorption of torsional vibrations near the resonance frequency. The basis of the kinematic scheme is based on the principle of dynamic absorption of vibrations. The design of the damping device of the angular type is observed. A mathematical model of the transmission of the car with the damping device and the results of computer modeling in MathCad are considered. According to the results of the mathematical analysis of the results the parameters of the damping device are determined.

Keywords: dynamic, torsional vibrations, vehicles, movement, cooled air, heat and mass transfer, energy.

1. INTRODUCTION

In transmission, when driving, due to the work of ICE, as well as the influence of external disturbing factors, vibrational processes, which lead to the emergence of cyclic dynamic loads, occur. Additional dynamic loads reduce the reliability of components and assemblies of the vehicle [1-4] in general.

Like any mechanical system with elastic elements, the transmission has its own resonant frequency and the corresponding vehicle speed. Thus, for passenger cars of make GAS the speed is of 40-50 km/h, for other cars it is in the other range. There is increased vibration at the moving of vehicle with this speed. Damping devices in the form of rubber inserts, which are fixed on the transmission units or special systems with external control, are used for vibration damping [5-12].

2. DYNAMIC ABSORBER OF VIBRATIONS

For antihunting of vibrations near resonance frequencies of transmission of the car, the damping device, representing the dynamic absorber of vibrations of corner type can be used [1].It is a mass with moment of inertia смоментоминерции*Jn*, attached to the fly - weel with the help of spring with the stiffness *Cn* (Figure-1).

The magnitude of the moment of inertia Jп and the stiffness of the spring SP is chosen taking into account the resonance frequency of the transmission ω from the:

$$\omega = \sqrt{\frac{Cn}{Jn}}\ .$$

The resonant frequency depends on the size of the transmission elements and the stiffness of the elastic elements, determined by design features of the vehicle. The characteristics of the damping device should be selected according to the results of computer simulation of the operation of transmission.

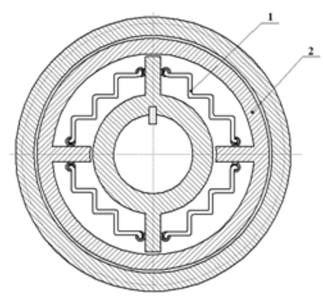

Figure-1. Dampingdevice: 1- springwithstiffness *Cn;* 2- fly -weel of damper with moment of inertia *Jn.*

3. MATHEMATICAL MODEL OF THE OPERATION OF TRANSMISSION OF THE VEHICLE

The mechanical transmission of the vehicle (Figure-2) can be regarded as a two-mass system connected between with an elastic connection. One of the masses J1 is connected with the rotating parts of the engine and the gearbox, and the second J2 - c rotating drive wheels and vehicle body. Elastic correlation is due to the stiffness of the propeller shaft and axle shafts.

Leading mass experiences a periodically varying loading $A \cdot Cos\omega \cdot t$, where*A*- the maximum value of changing moment; ω - rotary speed of ICE.

To maintain a constant speed of moving parts of the transmission, the flywheel is attached to dynamic damper in the form of a mass of elastic elements.

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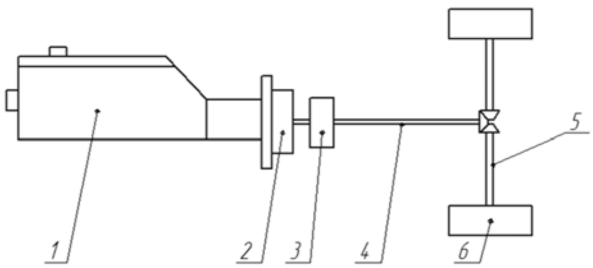

Figure-2. Vehicle transmission scheme.
1- engine device; 2- fly - weel; 3- gearbox; 4- карданныйвал; 5- axle shaft; 6- drive wheel

Thus, we have three-mass mechanical system with generalized coordinates, φ_1, φ_2, φ_n,
whereφ_1= generalized coordination of a given mass of ICE and gearbox;

φ_2=generalized coordination of a given mass of the body of the vehicle;

φ_n =generalized coordination of the given mass of the damper.

While developing mathematical models of the transmission of the vehicle with the damping device the following assumptions are made.

a) The system has a stationary two-way communications.

b) The energy dissipation in the nodes of the device of antihunting is absent.

c) The antihunting device has a constant mass.

d) Gaps and clearances are missing.

e) The elastic properties of the clutch spring and device, propeller shaft and axle shafts are taken into account. The remaining elements are taken as rigid.

Figure-3 shows the calculated transmission scheme of the vehicle. Propeller shaft is accepted as the reduction link.

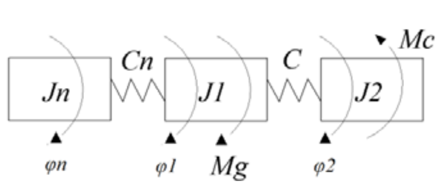

Figure-3. The analytical model of vehicle transmission.

The kinetic energy of a mechanical system is equal to:

$$T = \frac{J1}{2}\cdot\dot{\varphi}_1^2 + \frac{J2}{2}\cdot\dot{\varphi}_2^2 + \frac{Jn}{2}\cdot\dot{\varphi}_n^2 \;;$$

and potential energy:

$$\Pi = \frac{(\varphi_1-\varphi_2)^2}{2}\cdot C + \frac{(\varphi_1-\varphi_n)^2}{2}\cdot 2C_n\,.$$

Equivalent moment of inertia of the vehicle will be:

$$J_2 = J_\kappa \cdot n + \frac{Q\cdot D^2}{4\cdot g\cdot i^2}\;;$$

Where
J_κ= the moment of the inertia of wheel;
n = the number of driving wheels;
Q =the weight of the vehicle;
D = the diameter of the drive wheel;
i= ratio of number of teeth of the final drive.
Unit stiffness will be equal to:

$$C = \frac{2C_1\cdot C_2}{2C_1\cdot C_2 + 2C_2 + i^2\cdot C_2}\;;$$

Where
C_1 =the rigidity of propeller shaft andиclutch springs
C_2 =the rigidity of half axles.

Taking into account the Lagrange equation of the second kind, we will receive:

$$\left.\begin{aligned} &J_1\cdot\ddot{\varphi}_1 + C(\varphi_1-\varphi_2) + C(\varphi_1-\varphi_n) = M_g + M_B\cos\omega\cdot t\\ &J_2\cdot\ddot{\varphi}_2 + C(\varphi_1-\varphi_2) = -M_c\\ &J_n\cdot\ddot{\varphi}_n + C_n(\varphi_1-\varphi_n) = 0 \end{aligned}\right\} \quad (1)$$

where M_g =moment of engine device;

M_c =the moment of resistance to motion of the vehicle;

M_B =the perturbation component of the moment of engine device;

ω =the frequency of distributing influence corresponding to the resonance frequency of the system:

$$\omega = \sqrt{\frac{J1}{C}}\,.$$

Using a system of differential equations (1) it is possible to shape various processes in the transmission under various driving modes of the vehicle and to

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determine the load in the elastic links. Modeling of processes in mechanical systems allows optimizing the number of their characteristics.

The system of differential equations of motion of an equivalent mass of a transmission of the vehicle (1) can be solved by a system of mathematical calculations MathCad, using various functions [13].For solving the problem of modeling of different modes of motion of an reduced mass of vehicle transmission the function *rkfixed* is used, which gives a solution of a system of ordinary differential equations. This function returns a matrix solution by the Runge-Kutta method with the initial conditions in the vector, the right parts of which are written in a character vector at a certain interval with a fixed step.

As the system of differential equations (1) is of second order, then each equation in the calculation should be transformed into a system of two equations of the first order.

At the solution the function y is defined, the calculation interval φ_1 and φ_2, the number of points and matrix functions G.

Figure-4 shows a program for calculating the differential equations system (1) for the GAZ at established rotational rate of engine shaft $\omega=196c^{-1}$, corresponding to the resonant frequency without taking into account the viscous fiction in the fiction nodes.

$m := 0.03 \quad A := 50 \quad v := 1 \quad p := 220 \quad c := 1160 \quad k := 200$

$w := 196 \quad r := 0.005 \quad q := 15.5$

$$y := \begin{pmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{pmatrix}$$

$$G(t,y) := \begin{pmatrix} y_1 \\ \frac{-c}{m}\cdot y_0 + \frac{A\cdot\cos(w\cdot t)}{m} + \frac{c}{m}\cdot y_2 - \frac{k}{m}\cdot y_0 + \frac{k}{m}\cdot y_4 \\ y_3 \\ \frac{c}{q}\cdot y_0 - \frac{c}{q}\cdot y_2 \\ y_5 \\ \frac{k}{r}\cdot y_0 - \frac{k}{r}\cdot y_4 \end{pmatrix}$$

$U := \mathrm{rkfixed}(y, 0, 1, 200, G) \quad n := 0..200$

$T := U^{\langle 0\rangle} \quad M := c\cdot\left(U^{\langle 1\rangle} - U^{\langle 3\rangle}\right) + p \quad D := k\cdot\left(U^{\langle 1\rangle} - U^{\langle 5\rangle}\right) + p$

$V1 := \left(U^{\langle 2\rangle} + w\right) \quad V2 := \left(U^{\langle 4\rangle}\right) + w \quad V3 := \left(U^{\langle 6\rangle}\right) + w$

Figure-4. Program for calculation of systems of differential equations of motion of a reduced mass of vehicle transmission.

At the free-running speed of motion the moment M_g is equal to the moment of the resistance motion M_c, and the first mass is effected by the distributing component of the moment of engine device.

Figure-5a shows a graph of load changes in the elastic link transmission without damping, as Figure-5b shows the change of the angular displacement of the first mass. These changes are cyclical in nature with increasing amplitude.

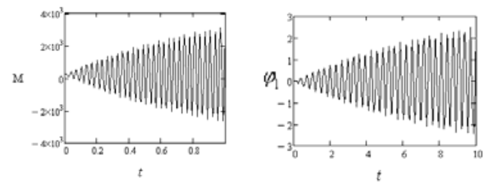

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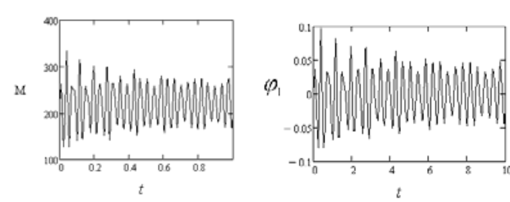
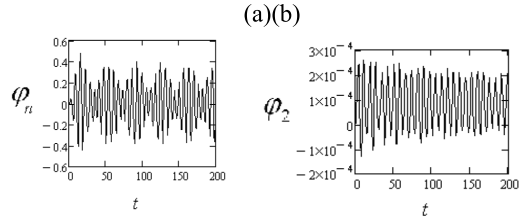
Figure-5. The solution of the equations of motion without damping: M - turning torque in the elastic element; *t* - время; φ-the amplitude of the angular displacement.

Figure-6(a) shows the change of the load in the elastic link transmission with the dynamic absorber of vibrations. Here takes place symmetric loop with the dynamic factor K_u =1,05. Figure-6(b) shows the change of the angular displacement of the first mass with the amplitude φ_1=0,1 rad., and the Figure-6(c) - angular movements of the mass absorber of vibrations with amplitude φ_n=0,5 rad. The change of the angular displacements of the second mass (Figure-6d) is insignificant and areφ_2=2,5 rad.

Optimization calculation was carried out on the minimum value of the amplitude fluctuations of turning engine torque. The optimization characteristic is the spring stiffness and the moment of inertia of the absorber, which for this task is: $C_n = 200$Н·м, a $J_n = 5\cdot 10^{-2}$ Н·мc^2.



(a)(b)



(c)(d)

Figure-6. The solution of the equations of motion with dynamic absorber of vibrations.

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Thus, modeling in Mathcad system confirms the effectiveness of using dynamic damping in the transmission of the vehicle and determines the optimal characteristics of the absorber for a specific vehicle model.

5. CONCLUSIONS

a) Damping device of angular type, providing absorption of torque vibrations in the drive train of the vehicle in the area of resonant frequencies is proposed.

b) The mathematical model of the transmission of the vehicle that allows to model its performance in transient conditions and to analyze the obtained results is elaborated.

c) It is possible to choose optimal characteristics of the damping device based on the minimization of dynamic loads in the elements of transmission.

d) The considered technical solution allows to reduce the dynamic loads at the nodes of the transmission in the zone of resonance frequencies and to increase the reliability of the vehicle in general.

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