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CREATION OF VIBRATION GEAR CONTINUOUSLY VARIABLE TRANSMISSION (CVT)

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Abstract. *Drives of machines which are used now do not possess ability to be adapted for extreme working conditions. Recently the technological direction of use of adaptive drive of machines is advanced. The adaptive drive mechanism engages the engine and the self-regulated transmission mechanism. The engine can transmit motion to the transmission mechanism even at full stop of working body. Conditions of start up of such mechanism admit substantial growth of a firing effort at the expense of small firing speed of a working body at constant engine power. It is offered to supply start of working body motion in the presence of a handicap for creation conditions of vibration effect on the working body which many times improving reliability start-up from a place. For this purpose it is offered to enter elastic links into the drive of mechanism. In the present work laws of creation of vibration effect in the mechanism continuously variable transmission are presented. The variable transmission leads to time-periodic angular speeds of the wheels and output carrier of the vibration gear.*

1 INTRODUCTION

Drives of machines which are using now do not possess ability to be accommodated for extreme working conditions. Such conditions are connected with possible difficulties of motion because of deviations from norms of maintenance (for example, because of long inactivity, a decline of conditions of greasing, insignificant faults, difference of temperatures, etc.). In the conditions of impossibility of elimination of small derangements on the move (for example, in an aeronautics) insignificant inaccuracy of maintenance of the service mechanism can serve as a cause of accident. The main deficiency of existing drives is a "rigid" constraint of the engine with the executive tool through the connecting gear with one degree of freedom. In this case the insignificant handicap on the move the tool or its wedging can call breakage of the drive.

Recently the scientific and technical direction of use of adaptive drives of machines develops [1-3]. The adaptive drive contains the engine and the self-controlled connecting gear. The adaptive gear connecting mechanism with two degree of freedom has ability to set in motion the executive tool with speed which is inverse - proportional to external loading at constant power of engine. It means that drive breakage will not occur even in the presence of a handicap on the tool motion. The engine can transfer motion to the connecting gear even at tool dead stop. Conditions of start-up of such mechanism admit substantial growth of a starting force at the expense of small starting speed of tool at constant power of engine. It is offered to use conditions of vibrating affecting on the tool to provide the beginning of motion of the tool in the presence of a handicap. For this purpose it is offered to include elastic links into the drive mechanism. In the present work the regularities of creation of vibrations in the mechanism of stepless adjustable transfer (CVT) are presented.

Work is executed on the basis of mechanics laws.

2 DESCRIPTION OF THE GEAR ADAPTIVE VIBRATING MECHANISM

Working out of an adaptive vibrating gear variator which provides a variable speed of motion of the target tool depending on loading on him and creates vibrating affecting on the tool is observed. Vibrating affecting in a combination with power adaptation will provide reliable overcoming of starting resistance and overloadings in operating conditions.

The adaptive vibrating drive represents the closed gear differential mechanism with two degree of freedom or the continuously variable transmission (Figure 1). The mechanism contains frame 0, input carrier H_1, input satellite 2, block of central toothed wheels with external teeth (solar wheels) 1 - 4, block of central toothed wheels with internal teeth (ring wheels) 3 - 6, output satellite 5 and output carrier H_2. Toothed wheels 4-1, 2, 3-6, 5 form the closed contour.

The external active moment M_{H1} is acting on the entrance carrier H_1. External moment of resistance M_{H2} is acting on the output carrier H_2.

The output carrier H_2 is moving quite definitely with a speed inversely proportional to output moment of resistance thanks to presence of two degree of freedom and differential constraint in the closed contour [1, 2]:

$$\omega_{H2} = M_{H1}\omega_{H1} / M_{H2} \tag{1}$$

Eq. (1) expresses effect of power adaptation of output link to a variable load.

When the shafts connecting wheels 1-4 and 3-6 are rigid angular speeds of wheels are defined under formulas [1-3]:

$$\omega_3 = \frac{\omega_{H2}(1-u_{46}^{(H2)}) - \omega_{H1}(1-u_{13}^{(H1)})}{u_{13}^{(H1)} - u_{46}^{(H2)}} \tag{2}$$

$$\omega_1 = u_{46}^{(H2)}(\omega_3 - \omega_{H2}) + \omega_{H2} \tag{3}$$

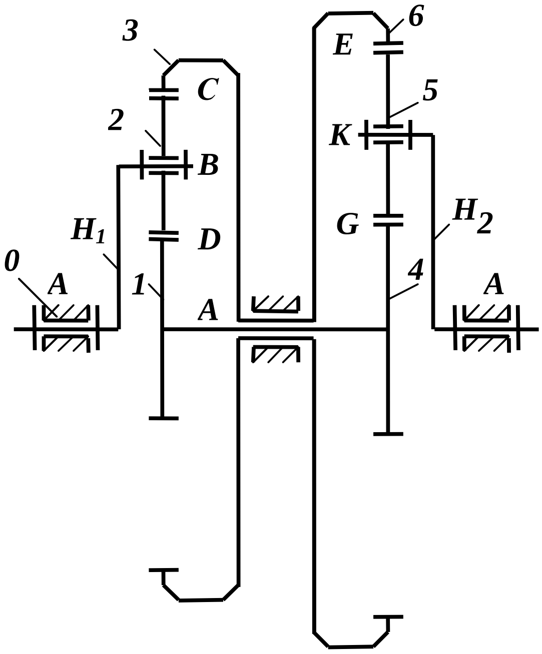

Figure 1. Gear adaptive vibrating mechanism.

Here transfer ratios are next: $u_{13}^{(H1)} = -z_3 / z_1$, $u_{46}^{(H2)} = -z_6 / z_4$.

Presence of the elastic shafts connecting wheels 1, 4 and 3, 6 will lead to emersion of dynamic pulses at transfer of motion on the output carrier.

3 DYNAMIC ANALYSIS OF MOTION OF MECHANICAL SYSTEM WITH ELASTIC SHAFTS

Let's consider the possibility to receive of definability of system with two degree of freedom when elastic shafts connect wheels 1 and 4 and wheels 3 and 6.

Reactions $R_{23} = F_{H1}/2, R_{21} = F_{H1}/2$ are transferred on intermediate links 3 and 1 from entrance satellite 2. Reactions $R_{54} = R_{H2}/2, R_{56} = R_{H2}/2$ are transferred on intermediate links 4 and 6 from output satellite 5.

Here $F_{H1} = \dfrac{M_{H1}}{r_{H1}}$, $R_{H2} = \dfrac{M_{H2}}{r_{H2}}$,

M_{H1}, M_{H2} - moments on input and output carriers,

r_{H1}, r_{H2} - radiuses of input and output carriers,

r_i $(i = 1, 2...6)$ - radiuses of wheels.

Moments are transferred to wheels 4 and 6:

$$M_4 = 0.5 M_{H2} \frac{r_4}{r_{H2}}, \; M_6 = 0.5 M_{H2} \frac{r_6}{r_{H2}} \tag{4}$$

When elastic constraint between wheels 1-4 and 3-6 differential equations of motion of wheels 4 and 6 look like

$$J_4 \ddot{\varphi}_4 = M_4 - c_4(\varphi_4 - \varphi_1) \tag{5}$$

$$J_6 \ddot{\varphi}_6 = M_6 - c_6(\varphi_6 - \varphi_3) \tag{6}$$

Here J_4, J_6 - moments of inertia of wheels 4 and 6,

c_4, c_6 - torsion rigidity of the shafts connecting wheels 1-4 and 3-6.

According to [4] on a method torsion rigidities at enough high power of the engine it is possible to consider angular speed ω_1 as a constant, then $\varphi_1 = \omega_1 t$ .The angle φ_4 differs from φ_1 a little. Therefore it is convenient to take for generalized co-ordinate a difference $\varphi = \varphi_4 - \varphi_1$ instead φ_4. Then the differential Eq. (5) becomes:

$$J_4 \ddot{\varphi} + c_4 \varphi = M_4 \tag{7}$$

The solution of Eq. (7) at zero entry conditions looks like:

$$\varphi = \frac{M_4}{c_4}[1 - \cos(k_4 t)] \tag{8}$$

Here $k_4 = \sqrt{\frac{c_4}{J_4}}$ - oscillation frequency of wheel 4 concerning wheel 1 owing to elastics of the shaft connecting wheels 1-4.

Motion of a wheel 4 can be considered as the motion consisting of the basic motion with constant angular speed ω_1 and additional motion with speed $\dot{\varphi}$ having oscillatory character:

$$\dot{\varphi} = A_{\omega 4} \sin(k_4 t) \tag{9}$$

Here amplitude of angular speed $A_{\omega 4} = \frac{M_4}{c_4} k_4 = \frac{M_4}{\sqrt{c_4 J_4}}$.

Thus angular speed of wheel 4 in the presence of elastic constraint is

$$\omega_4 = \omega_1 + \frac{M_4}{\sqrt{c_4 J_4}} \sin(k_4 t) \tag{10}$$

Angular speed of a wheel 6 is defined analogously:

$$\omega_6 = \omega_3 + \frac{M_6}{\sqrt{c_6 J_6}} \sin(k_6 t) \tag{11}$$

Here $k_6 = \sqrt{\frac{c_6}{J_6}}$ - oscillation frequency of wheel 6 concerning wheel 3 owing to elastics of the shaft connecting wheels 3 and 6.

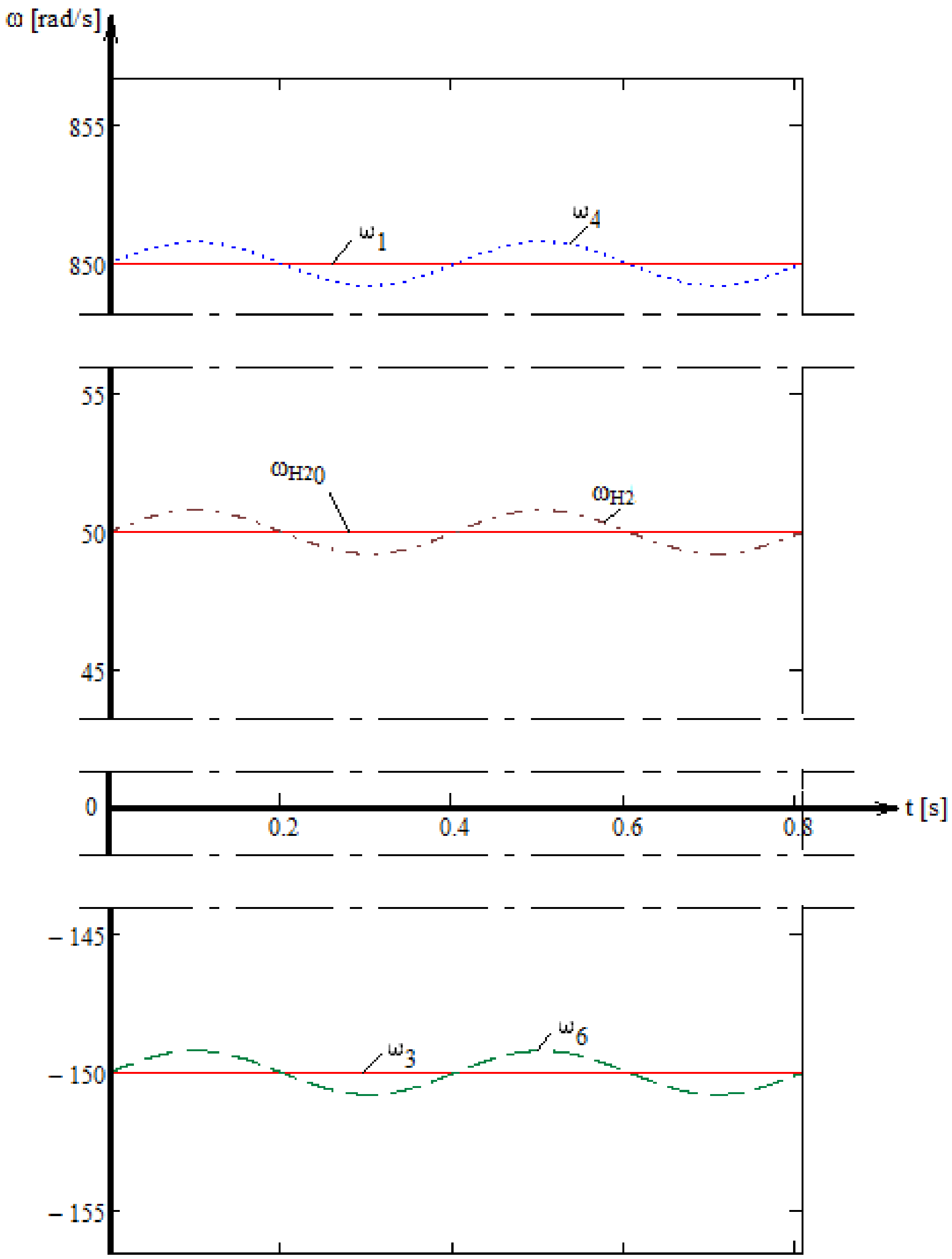

Figure 2. Schedule of change of angular speeds.

Thus for transfer of oscillations on the satellite 5 and the target carrier H_2 without distortions (with frequency and amplitude of oscillation conservation) it is necessary to satisfy condition $k_4 = k_6$. From this condition which is condition for the selection of rigid and inertia parameters follows:

$$\frac{c_6}{c_4} = \frac{J_6}{J_4} \tag{12}$$

The calculations have shown that at performance of Eq. (12) the equality of amplitudes of oscillation of wheels 4 and 6 takes place also.

Amplitudes and periods of oscillations are selected by the set of shafts torsion rigidities and moments of inertia.

On the found angular speeds of wheels 4 and 6 the angular speed of target carrier H_2 is defined:

$$\omega_{H2} = \frac{\omega_4 - u_{46}^{H2}\omega_6}{1 - u_{46}^{H2}} \tag{13}$$

As is shown in Figure 2 the angular speed ω_{H2} of the target carrier H_2 varies under the harmonious law about a mean ω_{H20} defined by the Eq. (1).

4 CONCLUSIONS

The oscillation frequency in an elastic contour is high and has vibrating character. Vibrating affecting in combination with effect of power adaptation predetermines high reliability start and overcoming of emergency overloadings.

The executed scientific researches allow creating simple and reliable adaptive vibrating drive of the service mechanism for the techniques working in extreme conditions (in aircraft, in space, etc).

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