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Dynamics of Adaptive Gear Variator

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Abstract- The adaptive gear variator represents the wheelwork with constant engagement of the cogwheels independently adapting for variable force loading by change of motion speed. Adaptation is secured by the mobile closed contour in a kinematic chain with two degrees of freedom. Overcoming of the starting resistance moment is provided by creation of the dead center position which is wedging the closed contour. The exit from dead center position after starting is provided by the additional auxiliary transfer. Action of the adaptive variator is described by the simplest dynamics on basis of theorem on change of kinetic energy.

Keywords- *Gear Variator, Force Adaptation, Dynamics, Transitive Regime*

I. Introduction

The adaptive gear variator represents a wheelwork with constant engagement of the toothed wheels and having ability to be adapted for variable power loading at the expense of independent (without a control system) change of motion speed.

Attempts to create a gear variator (the adaptive gearing) were undertaken by many inventors [1, 2, 3, 4]. In the basis of Ivanov’s invention the two-mobile planetary kinematic chain [3, 4] was used. It has been proved [5, 6, 7] that if the kinematic chain contains the mobile closed contour at motion with two degree of freedom then the closed contour imposes additional constraint on motion of links and provides definability of motion. At the same time the kinematic chain gets property of force adaptation to a variable load. Such property takes place in motion operating condition at a relative uniform motion of all links.

However on the start (in the motion beginning) the output link is motionless, and the kinematic chain has one degree of freedom. Mechanism cannot transfer a force to output link in order to begin a motion. Definability of motion is absent. The using of brake on one of mobile links [2] demands a control and deprives autonomy transfer. Use of dynamic inertia parameters on the start [3] provides a small starting moment and is not reliable.

Reliable start with overcoming of the high starting moment can be provided by creation of the dead center position of the closed contour at chosen sizes of links [8]. Starting constraint is created by choosing of matching sizes of links and is eliminated after start. For elimination of starting constraint after the beginning of motion of all links the additional transfer which wedges out the closed contour is used [8]. This additional transfer takes place in parallel with the basic two-mobile planetary kinematic chain and does not obstacle to its motion. The wedging out of additional transfer can be created on the basis of coincidence of linear speeds of some links [9].

The developed kinematic analysis and force analysis of an adaptive variator in a uniform motion with two degree of freedom [10, 11, 12] install analytical regularity of interconnection of the kinematic and force parameters according to mechanics laws. Numerical instances prove the found regularity.

However autonomy of motion on start at transition from a condition with one degree of freedom into a two-mobile condition, and also in operating condition at loading change should be analyzed. It is necessary to describe and investigate dynamic transient for an estimation of reliability of autonomy and efficiency of gear variator. Work is devoted to description and research of dynamics of transients of adaptive gear variator.

II. Description of Adaptive Gear Variator

At the description of the device and work of the adaptive gear variator we will use following designations:

M_9, M_{10} – external moments on the input 9 and output 10 carriers,

F – input driving force,

R – output resistance force,

r_9, r_{10} – radiuses of input 9 and output 10 carriers,

u^{pl}_{9-5} – transfer ratio of the planetary kinematic chain from the input carrier 9 to the output satellite 5,

u^{ad}_{9-5} – transfer ratio of additional transfer from the input carrier 9 to the output satellite 5,

z_i, i=1, 2, 3, …, 8 – numbers of teeth of wheels,

ω_9, ω_{10} – angular velocities of the input 9 and output 10 carriers.

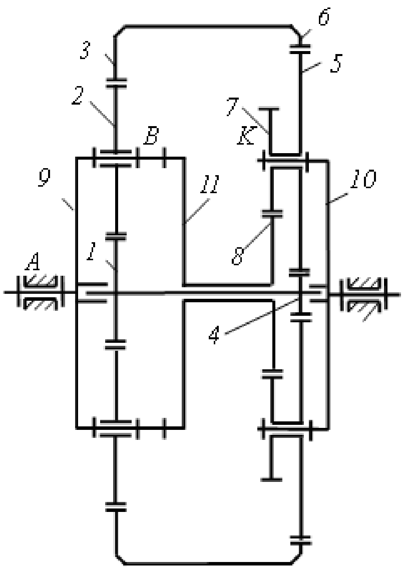

Figure 1. Adaptive gear variator

The adaptive gear variator (Fig. 1) contains the following parts: input carrier 9, input satellite 2, block of solar wheels 1-4 fixed on intermediate shaft, block of ring wheels 3-6 leaning against satellites, output satellite 5 and output carrier 10.

Input 9 and output 10 carriers are executed with equal sizes (radiuses) $r_9 = r_{10}$, that matches to the formula connecting numbers of teeth of wheels 1, 2 and 4, 5: $z_1 + z_2 = z_4 + z_5$. It leads to transfer wedging on start.

Additional transfer is executed in the form of the gearing from the input carrier 9 to the output satellite 5 containing the toothed wheel 8 connected to the input carrier 9 by means of disk 11 and wheel 7 rigidly connected to the output satellite 8. Additional transfer provides the wedging out of the kinematic chain after the motion beginning. Additional transfer in the form of wheels 8 and 7 has transfer ratio u^{ad}_{9-5} and doubles a planetary kinematic chain from input carrier 9 to output satellite 5 with transfer ratio u^{pl}_{9-5}. The equality $u^{ad}_{9-5} = u^{pl}_{9-5}$ takes place, where

$$u^{ad}_{9-5} = -z_8 / z_7, \tag{1}$$

$$u^{pl}_{9-5} = \frac{u^{(9)}_{13} - u^{\{10)}_{46}}{u^{(10)}_{56}(u^{(9)}_{13} - 1)}. \tag{2}$$

Here $u^{(9)}_{13} = -z_3 / z_1$ – the transfer ratio of wheels 1 and 3 at the motionless carrier 9, $u^{(10)}_{46} = -z_6 / z_4$ - the transfer ratio of wheels 4 and 6 at the motionless carrier 10, $u^{(10)}_{56} = z_6 / z_5$ – the transfer ratio of wheels 5 and 6 at the motionless carrier 10.

After substitution of these values in (2) we will gain

$$u^{pl}_{9-5} = \frac{z_3 z_4 z_5 - z_1 z_5 z_6}{z_3 z_4 z_6 + z_1 z_4 z_5}.$$

From a condition of equality of the transfer ratios expressed by (1) and (2), we will gain a condition of interconnection of numbers of teeth of wheels of the mechanism allowing synthesizing the mechanism

$$\frac{-z_8}{z_7} = \frac{z_3 z_4 z_5 - z_1 z_5 z_6}{z_3 z_4 z_6 + z_1 z_4 z_5}. \tag{3}$$

The adaptive gearing works as follows.

In the beginning of motion (on start) the output carrier 10 is motionless, transfer has one degree of freedom and can be in free move at relative mobility of wheels 1-4, 2, 3-6, 5 of closed contour. Force interacting of links of transfer is presented on a side view (Fig. 2). The load 11 with weight *G* and mass *m*, attached by a flexible filament to shaft *A* of output carrier 10, creates a tractive resistance. Relative motion of links on the start is possible generally when carriers 9 and 10 have different radiuses, and transfer has eccentricity $e = r_9 - r_{10}$ allowing the creating moment $M = Fe$ turn the satellite 5 around motionless point *K* of the output carrier 10.

Thus, closed contour of toothed wheels gets internal relative mobility. However at equal radiuses of carriers 9 and 10 the kinematic chain of transfer appears wedged because the line of acting of driving force *F* from the part of input carrier 9 in point *B* passes through point *K* of the output carrier 10 and force *F* is directed oppositely to resistance force *R*. Eccentricity is equal $e = 0$, and the driving moment which rotates the output satellite 5 and all closed contour in relative motion is absent. As a result of wedging the kinematic chain loses one degree of freedom and can begin motion only in the condition of the overcoming force of resistance *R* and output starting moment of resistance on the carrier 10. Start from a place becomes absolutely reliable (as in the usual mechanism with one degree of freedom).

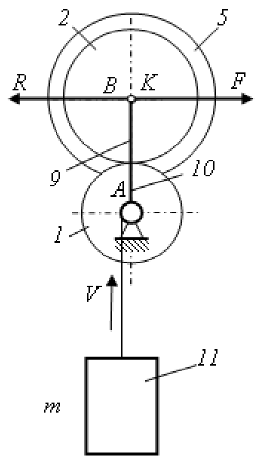

Figure 2. An adaptive gear variator on a side view

After start from a place additional (parallel) transfer (Fig. 1) through wheels 8 and 7 provides transfer of the driving moment directly from the input carrier 9 on the output satellite 5, keeping its relative motion in the closed contour, and eliminates wedging. The mechanism passes into a condition with two degree of freedom with relative mobility of links of the contour. In this condition the equilibrium of the mechanism is carried out by a principle of possible works with adaptation to a variable output moment of resistance by formula which is resulted in [6]

$$\omega_{10} = \frac{M_9 \omega_9}{M_{10}}. \tag{4}$$

Thus, the offered design provides automatic overcoming of high starting resistance and definability of motion in two mobile conditions.

III. Dynamics of Transient of Gear Variator in Stage of Running Start

For the dynamic analysis we will use the theorem on change of kinetic energy: change of the kinetic energy of the mechanism within some interval of time is equal to work of external forces.

$$A_M - A_R = T - T_0. \tag{5}$$

T, T_0 – kinetic energy in the end and in the beginning of motion interval,

A_M, A_R –- work of driving force and work of resistance force.

It is convenient to use powers of the driving and resistance forces P_M, P_R instead of their works A_M, A_R. For this purpose we divide the equation (5) into an interval of time t and obtain

$$P_M - P_R = (T - T_0)/t. \tag{6}$$

On initial transient two kinds of transitive motion take place: 1) start – overcoming of the starting moment of resistance during time of start t_S; 2) acceleration – before transition into an operational mode of uniform motion during time t_a. Total time of initial transient $T_b = t_s + t_a$.

Let us determine parameters of the start. Kinetic energy of the mechanism can be determined as $T = 0.5mV^2$, where m – the reduced mass of all links, V – speed of a point of reduction. However, for simplification it is possible to neglect masses of links of the mechanism for they are small in comparison with mass of a moving load. Then $T_0 = 0$, $T = 0.5mV_s^2$, where m – mass of the load (fig. 2), $V_s = \omega_{10s} r_{10}$ – the given starting initial speed of motion of the load, ω_{10s} – starting initial angular speed, r_{10} – output shaft radius.

Power of resistance on start $P_{Rs} = GV_s$, where G – the load weight ($G = mg$).

From (6) we can determine the required power of the engine for start providing.

Let us determine parameters of the acceleration after the start. Interconnection of parameters of acceleration in matching with (6) has following format

$$P_M - P_{Rs} = (T_a - T_s)/t_a, \tag{7}$$

where t_a – time of acceleration.

From (7)

$$\begin{aligned} P_M &= P_{Rs} + 0.5m(V_a^2 - V_s^2)/t_a = \\ &= P_{Rs} + 0.5m(V_a - V_s)(V_a + V_s)/t_a = \\ &= P_{Rs} + 0.5ma_a(V_a + V_s)/t_a \end{aligned}$$

or

$$P_M = P_{Rs} + 0.5ma_a(V_a + V_s), \tag{8}$$

where $a_a = (V_a - V_s)/t_a$ – given (allowable) initial acceleration.

On driving power it is possible to select the engine (propeller) with matching angular velocity ω_M and determine the driving moment

$$M_M = \frac{P_M}{\omega_M}. \tag{9}$$

The driving moment must be checked on a condition of serviceability of transfer [8] using radiuses of toothed wheels 1, 4, 9, 10 and the given moment of resistance $M_{Rs} = P_{Rs}/\omega_{10s}$

$$M_9 \geq M_{Rs} \frac{R_4 R_9}{R_1 R_{10}}. \tag{10}$$

After that it is possible to determine the start time under the formula received from (6)

$$t_s = \frac{mV_s^2}{2(P_M - P_{Rs})}. \tag{11}$$

After the start the acceleration begins – the sped up motion with transition into motion operating condition. Equation (6) becomes

$$P_M - P_{Rs} = 0.5m(V^2 - V_s^2)/t_a, \tag{12}$$

where t_a – acceleration time, V – the given speed of motion of a load in the end of initial transient. The acceleration proceeds before achievement of equality of powers of driving and resistance forces. We will determine acceleration time

$$t_a = \frac{0.5m(V^2 - V_s^2)}{P_M - P_{Rs}}. \tag{13}$$

Further the motion operating condition begins. At equality of powers of the driving and resistance forces ($P_R = P_M$) the uniform motion without change of kinetic energy occurs. At instant change of the resistance moment the transitive regime with change of kinetic energy before achievement of equality of powers of the driving and resistance forces at the expense of change of output angular speed according to (6) occurs. In this case driving power P_M remains without change, power of resistance matches to the changed moment resistance M_R at former angular velocity ω_{R0}, that is $P_R = M_R\omega_{R0}$, and the kinetic energy in the end of transient will match to new angular speed $\omega_R = P_M / M_R$. Then from (6) it will be possible to determine time t of transient to the changed moment of resistance in motion operating condition. The further motion will be uniform.

IV. Numerical Instance of Dynamic Calculation

Let us execute dynamic calculation of the mechanism (Figs. 1, 2).

Initial data:

Load weight and mass $G = 10000\ Nm, m = 1000\ Ns^2 / m$.

The kinematic parameters of the beginning of motion

$V_s = 0.1\,m/s, V_a = 1\,m/s, a_s = 7.27 m/s^2$

Geometrics

$R_1 = 0.044 m,\ R_4 = 0.012 m,$

$R_9 = R_{10} = 0.056 m,\ r_{10} = 0.010 m$.

We will determine: the time of start t_s, acceleration time t_a, total time of the beginning of motion T_b for transition into motion operating condition with $V = V_a = 1 m/s$ and with operating condition motion parameters M_R, ω_R. Numerical results we will show on the diagram of a tractive characteristic of a gear variator (Fig. 3).

The solution

1. Output angular velocity on start $\omega_{10s} = V_s / r_{10} = 0.1/0.01 = 10 s^{-1}$.

2. Output moment of resistance on start $M_{Rs} = Gr_{10} = 10000 \cdot 0.01 = 100 Nm$.

3. Power of resistance on start $P_{Rs} = M_{Rs}\omega_{10s} = 100 \cdot 10 = 1000 Nm/s$. – Points *A* and *B* on Fig.3.

4. The driving power consumed for overcoming of starting resistance and acceleration on the equation (8)

$$P_M = P_{Rs} + 0.5 m a_s (V_a + V_s) = \\ = 1000 + 0.5 \cdot 1000 \cdot 7.27 \cdot (1 + 0.1) = 5000\ Nm/s.$$

We select the electric motor: power of *5 kW*, a rotational speed of *1500 rpm* ($\omega_M = \omega_9 = 150 s^{-1}$).

5. Driving moment $M_M = M_9 = \dfrac{P_M}{\omega_M} = \dfrac{5000}{150} = 33.3 Nm$.

Check of the driving moment on possibility of start

$$M_9 \geq M_{Rs}\frac{R_4 R_9}{R_1 R_{10}} = 100\frac{0.012 \cdot 0.056}{0.044 \cdot 0.056} = 27.2 Nm \cdot$$

6. The start time (getaway) by (11)

$$t_s = \frac{mV_s^2}{2(P_M - P_{Rs})} = \frac{1000 \cdot 0.1^2}{2(5000 - 1000)} = 0.00125\ s.$$

Start is presented by section *AB*.

7. Acceleration time after start

$$t_a = \frac{m(V_a^2 - V_s^2)}{2(P_M - P_{Rs})} = \frac{1000 \cdot (1^2 - 0.1^2)}{2(5000 - 1000)} = 0.123 s.$$

This is presented by section *BC*.

8. The time of the beginning of motion transition from start into motion operating condition

$$T_b = t_s + t_a = 0.00125 + 0.123 = 0.12425\ s$$

V. Dynamics of Transient of Gear Variator in Stage of Steady Motion

After the beginning of motion in operating condition with a constant resistance moment the uniform motion with constant angular velocity takes place. Power of resistance is equal in the beginning of operating condition to driving power $P_R = P_M = 5000 Nm$. Parameters of power of resistance are equal $M_R = 100 Nm,\ \omega_{10} = 50\ s^{-1}$ – the point *C* on Fig.3.

In motion operating condition at a process with decrease of the moment of resistance there is an increase of angular velocity and kinetic energy increasing according to (6). For example, with decrease of the moment of resistance to value $M_R = 50 Nm$ we obtain

$$\omega_{10} = \frac{P_M}{M_R} = \frac{5000}{50} = 100\ s^{-1} \text{– point } D.$$

In the case of a transitive regime on (13) it is necessary to use following parameters.

At point *C* initial speed is equal $V_s = \omega_{10} r_{10} = 50 \cdot 0.01 = 0.5\ m/s$, at point *D* it is matching to new value of moment of resistance $M_R = 50 Nm$, so the terminal speed is matching to angular velocity $\omega_{10} = 100\ s^{-1}$ and equal $V = \omega_{10} r_{10} = 100 \cdot 0.01 = 1\ m/s$. Driving power P_M is remaining invariable. Power of resistance is equal $P_R = M_R\omega_{10} = 50 \cdot 50 = 2500\ Nm/s$.

Then by (13) we will gain time of acceleration and of transition into a new regime of motion with changed moment of resistance M_R and power P_R

$$t_a = \frac{0.5m(V^2 - V_s^2)}{P_M - P_R} = \frac{0.5 \cdot 1000 \cdot (1 - 0.5^2)}{5000 - 2500} = 0.15\ s\ .$$

In the end of this transitive regime (point *D*) input and output powers will be made equal again, and there will be further a regime of a uniform motion with parameters of point *D*.

Generally transitive process in operating condition occurs at change (decrease or increase) of the moment of resistance and power of resistance that leads to respective alteration of speeds of motion of links and kinetic energy of the mechanism.

When the minimum moment of resistance which equals to driving moment $M_R = 33.3\,Nm$ takes place we will gain $\omega_R = \frac{P_M}{M_R} = \frac{5000}{33.3} = 150\ s^{-1}$ – point *E*.

VI. Dynamics of Transient of Gear Variator in Stage of Steady Motion

Initial data match to the previous instance.

Power of resistance is equal in the beginning of operating conditions to driving power $P_R = P_M = 5000Nm$. Parameters of power of resistance are equal in the beginning of a stage of installed motion $M_{Rb} = 100Nm, \omega_{10b} = 50\ s^{-1}$ – point *C*. New value of the resistance moment $M_{Rn} = 50Nm$.

We will determine the output angular velocity ω_{10e} in the stage of the installed motion operating condition at the end of motion interval and the transition time in a new regime of motion *t*.

Solution

1. According to (6) final angular velocity $\omega_{10e} = \frac{P_M}{M_{Rn}} = \frac{5000}{50} = 100\ s^{-1}$ – at point *D*. An initial winding speed of load in point *C* $V_b = \omega_{10b} r_{10} = 50 \cdot 0.01 = 0.5\ m/s$. Speed in the end of the interval $V_e = \omega_{10e} r_{10} = 100 \cdot 0.01 = 1\ m/s$.

2. Power of resistance in the beginning of the considered interval of motion – at point *C* $P_{Rn} = M_{Rn}\omega_{10b} = 50 \cdot 50 = 2500Nm/s$.

3. Transition time in the new regime of motion

$$t = \frac{m(V_e^2 - V_b^2)}{2(P_M - P_{Rn})} = \frac{1000 \cdot (1^2 - 0.5^2)}{2(5000 - 2500)} = 0.15s\ .$$

4. At the new minimum moment of resistance equal to the driving moment $M_{Rn} = 33.3\,Nm$, we obtain $\omega_{10n} = \frac{P_M}{M_{Rn}} = \frac{5000}{33.3} = 150\ s^{-1}$ – point *E*. The speed of load $V_e = \omega_{10n} r_{10} = 150 \cdot 0.01 = 1.5\ m/s$.

5. Transition time in a regime of motion from position in point *C* with the minimum resistance – point *E*.

$$t = \frac{m(V_e^2 - V_b^2)}{2(P_M - P_{Rn})} = \frac{1000 \cdot (1.5^2 - 0.5^2)}{2(5000 - 2500)} = 0.4s\ .$$

Parameters of the tractive characteristic:

Point *A* –- start with parameters $\omega_{10} = 0, M_R = 100\ Nm$. *AB* – transitive motion regime on start with overcoming of starting resistance (parameters of point *B*: $\omega_{10} = 10\ s^{-1}, M_R = 100\ Nm$). *BC* – a section of the regime of acceleration and increase of kinetic energy. Point *C* – the beginning of operating condition of motion with two degree of freedom (parameters $\omega_{10} = 50\ s^{-1}, M_R = 100\ Nm$). *CD* – transitive regime of operational motion in a condition with the reduced moment of resistance (angular velocity and kinetic energy are increasing). Point *D* – an intermediate state of operating condition of motion (parameters $\omega_{10} = 100\ s^{-1}, M_R = 50\ Nm$). The return motion (for example, from point *D* to point *E*) takes place when moment of resistance is increasing with decreasing of angular velocity and kinetic energy.

VII. Conclusion

The gear variator is created on the basis of the kinematic chain with two degrees of freedom that determines its basic difference from existing transfer mechanisms. Research of dynamics of the adaptive gear variator allows presenting a full picture of its action in all regimes of motion. The elementary method based on the theorem on change of kinetic energy is used for dynamic research.

The start transitive regime of motion provides the start from place and acceleration of mechanism. The start takes place when the motion power is exceeding power of force of resistance. Reliability of the beginning of motion provides the starting wedging of the kinematic chain. The gear variator overcomes the maximum starting resistance in the accelerated motion with increase in kinetic energy.

Operational motion regime (after breaking) occurs at equality of motion power and power of resistance force. This motion is uniform. At change of powers balance the corresponding transitive regime changing parameters of motion takes place. Here the closed contour of the mechanism gets the compelled internal mobility and provides adaptation to variable loading.

The made dynamic analysis confirms the efficiency and reliability of work of the two-mobile mechanism containing necessary additional constraints in all regimes of motion.

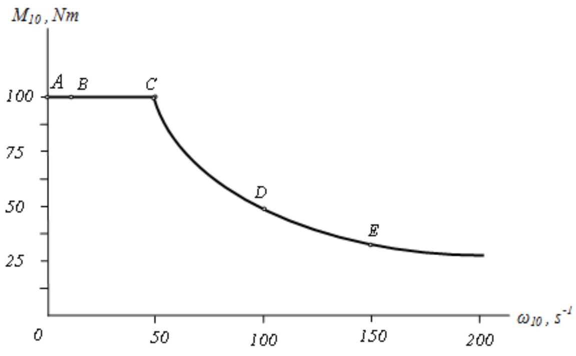

Figure 3. Tractive characteristic of gear adaptive variator

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