

DESIGN AND DYNAMICS OF ADAPTIVE TRANSFER BOX

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Abstract: *It is proved that the kinematic chain with two degrees of freedom containing the closed contour at presence of one input link is the mechanism. Such mechanism contains two external initial links with two generalized coordinates. The generalized coordinate of input initial link is set by the engine. The generalized coordinate of other initial link is determined by conditions of interaction of external and internal forces of the mechanism. The closed contour has equilibrium on the general equation of dynamics. The mechanism with the closed contour has property of force adaptation to variable technological loading.*

KEYWORDS: THEOREM, CLOSED CONTOUR, EQUILIBRIUM, ADAPTATION

1. Introduction

Smooth and adjustable process start from a place, stepless achievement of working speed, constructive simplicity and reliability of a demanded motion mode - the basic priorities of creation of transmissions of cars and the special hard loaded techniques.

The most widespread transmissions with operated stage transfer boxes provide the stage law of change of the twisting moment. Manual control in a stage transfer box is labor-consuming and imperfect. The automatic hydraulic control system of hydro dynamical transfer is more perfect and better carries out switching. However the law of change of the twisting moment remains stage. Besides following essential lacks are inherent in a hydrodynamic automatic transfer box: 1. Extreme complexity of a design. 2. The imperfection of a control system leading delay or an advancing of switching of transfers. 3. Rather low efficiency because of use of the hydro transformer.

Modern stepless transfers (a wedge belt variators) are capable to change smoothly the transfer ratio and the output twisting moment. However change of the transfer ratio occurs in narrow limits, and transfers have low reliability because of use of friction in the form of the functional factor.

Recently there were not switched gear boxes of transfers with constant gearing cogwheels.

Not switched continuous-leveling transmission is known [1]. This transmission is executed in the form of the hydrodynamic converter and the differential mechanism.

Lacks of this transmission: low efficiency because of use of the hydrodynamic converter, a small range of change of speed and low reliability at start-up that leads to necessity to use the additional mechanisms of a free course complicating a design and worsening quality of transfer work.

The device of transfer of energy with continuously variable transfer ratio [2] is known. This device is executed in the form of the differential mechanism with two degrees of freedom, contains a starting link and a brake.

Lacks of the device consist that it is not automatic, has low loading ability and reliability, low efficiency.

In the present work an object in view - theoretically to prove an opportunity of creation mechanical gear stepless transfer box providing reception of smooth performance data, simple on a design, reliable in work and highly effective. For achievement of an object in view it is offered to use the mechanism with two degrees of freedom at presence at it only one input.

The basic opportunity of adaptation of a kinematic chain to variable loading is based on introduction of an additional degree of freedom. For example, if on the lever the slider with the applied force of resistance is placed then the change of slider position on the lever will provide an opportunity of adaptation to variable force of resistance at constant motive force. However introduction of an additional degree of freedom demands additional control. According to terminology of theoretical mechanics « the least number of parameters which are necessary to get the system

possible position is called as number of its independent generalized coordinates» [3]. The additional generalized coordinate should provide an additional degree of freedom. In the theory of mechanisms and machines the concept «initial link» corresponds to concept «generalized coordinate» [4]. The kinematic chain has definability, if number of degrees of freedom is equal to number of initial links.

The kinematic chain with two degrees of freedom at presence of one input should contain one more initial part with the generalized coordinate.

However earlier [5, 6, 7] on the basis of a principle of possible works it has been shown that in that specific case a kinematic chain with two degrees of freedom with the mobile closed contour, having only one input, is in equilibrium and is definable. Hence, such chain having an input link can create one more initial link with the generalized coordinate.

The arbitrary kinematic chain (for example the chain containing structural group of two links instead of a contour) does not possess such ability. According to theoretical mechanics [3] « at absence of full definiteness of motion the principle of virtual works can not take a place». It is necessary to recognize: the additional generalized coordinate takes place if the kinematic chain has additional constraint between parameters.

The purpose of the scientific paper: to prove existence of unknown law in the theory of mechanisms and machines - a kinematic chain with two degrees of freedom containing the four-bar closed contour at presence of one input imposes additional constraint and provides definiteness of motion of all links.

In works [5, 6, 7] it is shown, that such chain realizes the force adaptation to variable external loading. Patents of Kazakhstan on an adaptive gear transfer [8, 9] are received.

Researches are executed for a kinematic chain with ideal constraints on the basis of use of equilibrium conditions of statics.

2. Statement of research problem

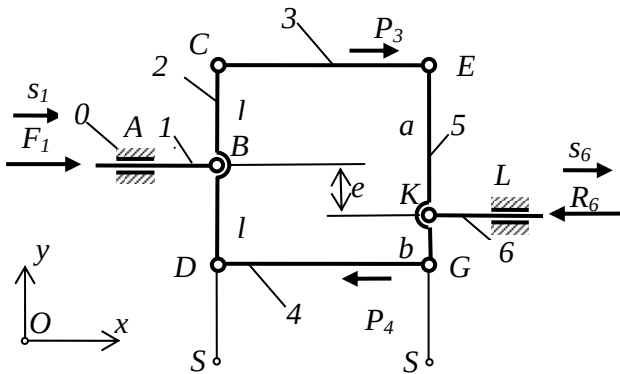
Object of research is the plane kinematic chain with two degrees of freedom having the closed four-bar contour with the external forces enclosed to chain. To prove that if for external forces of a chain the condition of equilibrium is carried out then for internal forces in a contour the conditions of equilibrium are carried out also.

Object of research in the form of a lever kinematic chain (Figure 1) represents a kinematic chain which contains two external links 1 and 6 connected with frame 0 and four-bar closed contour 2-3-4-5 which connects the external links among themselves. The four-bar closed contour is presented in the form of the structural group of the fourth class containing four mobile links and six kinematic pairs. External forces are enclosed to external links. The closed contour passes effort from external link 1 to external link 6.

Displacements of the kinematic chain points are presented on figure 2.

For a lever kinematic chain external links 1 and 6 are initial links and they determine the positions of all points of chain. We shall accept that external forces act only on initial links. If one of initial

Thus the considered kinematic chain with two degrees of freedom and with closed four-bar contour should have only one input link on conditions of equilibrium. Then conditions of equilibrium should provide determination of displacement of the second initial link (output link).



Further. Realization of kinetostatics equilibrium conditions for all chain will lead to equality of input motive force to output resistance force. But it is possible only when the closed contour represents rigid system at absence of internal mobility. Kinetostatics supposes an opportunity of realization of equilibrium for a moving system. The principle of virtual works represents a necessary equilibrium condition. The sum of works of external motive force and external resistance force on virtual displacements of their application points is equal to zero.

On basis of the description of object of research it is possible to formulate the statement of problem. - It is necessary to prove that the considered kinematic chain with the closed contour under action of forces is in equilibrium.

The proof of the theorem is brought to the proof of existence of a sufficient equilibrium condition if the necessary equilibrium condition takes place.

The plane kinematic chain with two degrees of freedom has a frame 0, one input link 1, the mobile closed contour with mobile links 2, 3, 4, 5 in the form of a parallelogram and one output link 6.

The sizes of links:

The kinematic chain has two initial links 1 both 6 and two generalized coordinates - displacements of links 1 and 6.

output link 6 . Forces and displacements are parallel to axis Ox .

B, C, D, L, M, K which belong to corresponding links i .

Constraints in kinematic pairs are ideal and stationary.

To prove the theorem we express internal forces of the contour in terms of external forces.

and R_6 by means of conditions of statics.

For the closed contour to be in equilibrium the equilibrium conditions are to be observed both for external forces and internal forces.

The internal forces on link 3 and link 4 independently will not match the conditions of statics at arbitrary external forces. However in a closed contour with its links being in motion there is an interacting work of forces.

There is the equilibrium conditions for nonadjacent links 2 and 5 below derived from a principle of virtual works.

For link 2

$$F_1 s_B = R_{32} s_C + R_{42} s_D. \quad (1)$$

For link 5

$$R_6 s_K = R_{35} s_E + R_{45} s_G. \quad (2)$$

We get the following after equations (1) and (2) are combined

$$F_1 s_B + R_6 s_K = R_{32} s_C + R_{42} s_D + R_{35} s_E + R_{45} s_G. \quad (3)$$

Here the internal forces acting in all points of a contour are expressed through active forces. Internal displacements can be defined through external displacements.

The left part of the equation (3) represents the sum of works of external forces of the contour. When external forces are in equilibrium according to the principle of virtual works for the contour (or for all chain)

$$F_1 s_B + R_6 s_K = 0. \quad (4)$$

For the equilibrium condition to be met, equation (4) is to contain one unknown parameter. If we accept active forces F_1 , R_6 and generalized coordinate s_B as set parameters, then it is possible to determine external displacement s_K from equation (4).

Displacement s_K is output generalized coordinate. As the generalized coordinates are known then displacements of all points of a contour are known (are presented in a picture of displacements – fig. 1) and they can be expressed through the generalized coordinates.

The right part of equation (3) represents the sum of works of internal forces of the contour. In the right part of the equation internal forces are determined through known external forces, internal displacements are determined through external displacements. Then the observance of condition (4) will also lead to the observance of internal forces works zero equality on virtual internal displacements

$$R_{32} s_C + R_{42} s_D + R_{35} s_E + R_{45} s_G = 0. \quad (5)$$

It means that equilibrium condition (4) for external parameters of the contour ensures also the execution of equilibrium condition (5) for internal parameters of the contour.

As equilibrium of external and internal forces acting on the on the contour takes place, kinematic chain with two degrees of freedom containing two links with the applied forces and placed between them closed four-bar contour at presence of one input link is in equilibrium, quod erat demonstrandum.

4. Consequences

Consequence 1. Equation (4) represents an additional condition of equilibrium to equations of statics. Equation (4) or corresponding equation (5) is the equation of differential constraint of a closed contour. This equation superimposes constraint on motion of the contour. The kinematic chain with two degrees of freedom and one input containing the closed contour gets additional constraint and turns to the mechanism with one degree of freedom.

Consequence 2. The kinematic chain with two degrees of freedom, one input and a closed contour shows the effect of force adaptation. From formula (4) and in view of the sign of works we get the following $s_K = F_1 s_B / R_6$.

Or when time is considered

$$v_K = F_1 v_B / R_6. \quad (6)$$

Here v_B , v_K - speeds of points B , K .

Formula (6) characterizes the force adaptation effect shown by a kinematic chain with a closed contour: output speed v_K is in inverse proportional dependence on variable force of resistance R_6 at constant parameters of input power F_1 , v_B .

Consequence 3. Outside and inside of the closed contour the equilibrium of works takes place according to equations (4), (5) but not that of forces.

5 Example (numerical confirmation of the theorem)

For kinematic chain presenting on figure 1

It is given: $F_1 = 12 \text{ N}$, $R_6 = 16 \text{ N}$,

$$s_B = s_1 = 6 \text{ m}, l = 2 \text{ m}, a = 3 \text{ m}, b = 1 \text{ m}, e = 1 \text{ m}$$

It is necessary to prove the existence of equilibrium.

We determine responses R_{32} , R_{42} through external force F_1 for link 2

$$R_{32} = R_{42} = F_1 / 2 = 12 / 2 = 6 \text{ N}.$$

We determine responses R_{35} , R_{45} through external force R_6 for link 5

$$R_{35} = R_6 b / 2l = 16 \cdot 1 / 2 \cdot 2 = 4 \text{ N},$$

$$R_{45} = R_6 a / 2l = 16 \cdot 3 / 2 \cdot 2 = 12 \text{ N}.$$

Equilibrium of external forces takes place under formula (4) if output displacement s_K is equal to

$$s_K = s_6 = F_1 s_B / R_6 = 12 \cdot 6 / 16 = 4.5 \text{ m}.$$

Then the equilibrium of internal forces of the contour is to be observed under formula (5). In this equation we accept that $s_C = s_E = s_3$, $s_D = s_G = s_4$. In view of the signs of works we get the following

$$-R_{32} s_3 - R_{42} s_4 + R_{35} s_3 + R_{45} s_4 = 0.$$

In view of

$$R_{23} = -R_{32}, R_{24} = -R_{42}, -R_{53} = R_{35}, -R_{54} = R_{45} \quad \text{we}$$

get the following

$$(R_{23} - R_{53}) s_3 + (R_{24} - R_{54}) s_4 = 0$$

$$\text{or } (R_{23} - R_{53}) s_3 = (R_{54} - R_{24}) s_4.$$

We take the following designations

$$R_{23} - R_{53} = P_3, R_{54} - R_{24} = P_4 \quad \text{which are forces acting on}$$

links 3 and 4. Then the equation of equilibrium for internal forces of the contour looks as follows

$$P_3 s_3 = P_4 s_4. \quad (7)$$

We execute the numerical verification of equation (7).

$$P_3 = R_{23} - R_{53} = 6 - 4 = 2 \text{ N},$$

$$P_4 = R_{54} - R_{24} = 12 - 6 = 6 \text{ N}.$$

Displacements of link points according to the picture of displacements are interrelated (linked) by formulas

$$(s_3 - s_1) / l = (s_1 - s_6) / e, (s_6 - s_4) / b = (s_1 - s_6) / e.$$

From here we get the following

$$s_3 = s_1 + (s_1 - s_6) l / e = 6 + (6 - 4.5) 2 / 1 = 9 \text{ m},$$

$$s_4 = s_6 - (s_1 - s_6) b / e = 4.5 - (6 - 4.5) 1 / 1 = 3 \text{ m}.$$

After substitution to equation (7) is done, we get $2 \cdot 9 = 6 \cdot 3$. It means that equation (7) is executed.

As equilibrium conditions (4) and (7) are executed for external and internal forces of the contour, takes place the equilibrium of a kinematic chain with two degrees of freedom and a single input. Such chain is already a mechanism.

On the basis of the proved theorem it is possible to create gear adaptive transfer with constant engagement of toothed wheels. For this purpose it is necessary to get laws of interrelation of parameters of the gear mechanism with the closed contour on the basis of the found laws of a lever kinematic chain.

6. Interrelation of parameters of gear mechanism with the closed contour

The gear mechanism with the closed contour (figure 3) looks like the differential gear mechanism and contains input carrier H_1 , the input satellite 2, block of the central toothed wheels 1 - 4 with external teeth, block of the central ring toothed wheels 3 - 6 with internal teeth, the output satellite 5 and output carrier H_2 . Toothed wheels 4-1, 2, 3-6, 5 form the closed contour. The external links (carriers H_1 and H_2) are acting on closed contour. Carrier H_1 passes on the closed contour the input motive force F_{H1} . Carrier H_2 passes on the closed contour the output resistance force R_{H2} .

The interrelation of kinematic and force parameters of the gear mechanism is determined by a principle of virtual works (or powers) under formulas which correspond to formulas (1) - (5). Parameters correspond to designations which are accepted on figure 1 and figure 2.

For gear mechanism in formulas (1) - (5) acting forces can be expressed through the moments which are applied to links of the gear mechanism, and linear displacements can be expressed through angular displacements of links:

$$\begin{aligned} F_1 &= F_{H1} = M_{H1} / r_{H1}, R_6 = R_{H2} = M_{H2} / r_{H2}, \\ R_{32} &= -R_{23} = -M_{23} / r_3, R_{42} = -R_{24} = -M_{24} / r_4, \\ R_{35} &= -R_{53} = -M_{53} / r_6, R_{42} = -R_{24} = -M_{24} / r_4, \\ S_B &= \varphi_{H1} r_{H1}, S_K = \varphi_{H2} r_{H2}, S_C = \varphi_3 r_3, \\ S_D &= \varphi_6 r_6, S_E = \varphi_1 r_1, S_G = \varphi_4, \end{aligned}$$

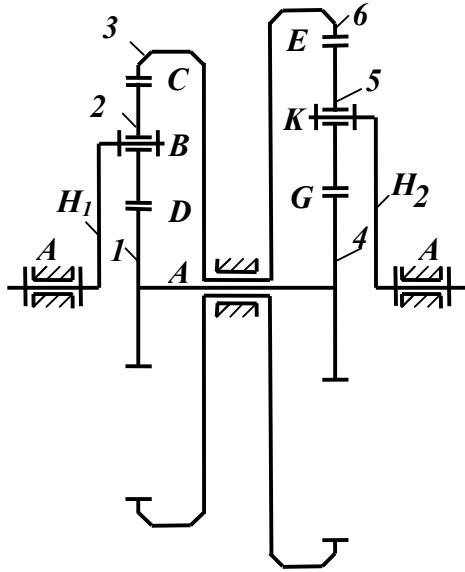


Fig. 3. Gear adaptive mechanism

Where symbol M means external moment, symbol φ means angle of turn of link, symbol r means radius of link, and the subscript at these symbols means the link (for example, M_{H1} - moment on link H_1 , φ_{H1} - angle of turn of link H_1 , r_{H1} - radius of link H_1).

The subscript at the reactive moment (internal moment) M means numbers of links in direction of transfer of the moment (for example, M_{23} - the moment which passed from link 2 on link 3).

After substitution of these values in formulas (4), (5) we shall receive

$$M_{H1} \varphi_{H1} + M_{H2} \varphi_{H2} = 0. \quad (8)$$

$$M_{23} \varphi_3 + M_{24} \varphi_1 + M_{53} \varphi_6 + M_{54} \varphi_4 = 0. \quad (9)$$

The formula (8) or the matching formula (9) determines additional constraint to conditions of statics which the closed contour imposes on motion of links. This constraint provides the transformation of a kinematic chain with two degrees of freedom and with one input in the mechanism.

In the equation (9) for blocks of wheels 1-4 and 3-6 it is

considered $\varphi_4 = \varphi_1$, $\varphi_6 = \varphi_3$. We shall receive in view of signs of works

$$(M_{23} - M_{53}) \varphi_3 + (M_{24} - M_{54}) \varphi_1 = 0.$$

$$\text{Or } (M_{23} - M_{53}) \varphi_3 = (M_{54} - M_{24}) \varphi_1.$$

Let's mean $M_{23} - M_{53} = M_3$, $M_{54} - M_{24} = M_4$ - the total (summarized) internal moments which are acting on blocks of wheels 3-6 and 1-4. Then the equation of equilibrium of the internal moments of a contour will become

$$M_3 \varphi_3 = M_4 \varphi_1. \quad (10)$$

Hence, in the closed contour the equilibrium of works of the internal moments takes place instead of the equilibrium of moments.

From the formula (8) in view of signs of works and with account of time follows

$$\omega_{H2} = M_{H1} \omega_{H1} / M_{H2}. \quad (11)$$

Here ω_{H1} , ω_{H2} - angular speeds of links H_1 , H_2 .

The formula (11) characterizes effect of force adaptation of the gear mechanism with the closed contour: at constant parameters of input power M_{H1} , ω_{H1} the output angular speed ω_{H2} is in inverse proportional dependence on the variable moment of resistance M_{H2} .

7. Kinematic and force analysis of gear adaptive mechanism

The problem statement: The external moments M_{H1} , M_{H2} and input angular speed ω_{H1} are the set parameters. To determine the output angular output speed, the intermediate angular speeds ω_{H2} , ω_1 , ω_3 of links H_2 , 1, 3 and to determine the internal forces (reactions).

We shall determine the output angular speed under the formula (11) using constraint (8).

The interrelation of kinematic and geometrical parameters (geometrical constraint) in view of $\omega_4 = \omega_1$, $\omega_6 = \omega_3$ is determined by formulas:

$$\frac{\omega_1 - \omega_{H1}}{\omega_3 - \omega_{H1}} = u_{13}^{(H1)}, \quad (12)$$

$$\frac{\omega_1 - \omega_{H2}}{\omega_3 - \omega_{H2}} = u_{46}^{(H2)}, \quad (13)$$

Where $u_{13}^{(H1)} = -z_3 / z_1$, $u_{46}^{(H2)} = -z_6 / z_4$ - the mechanism transfer ratios expressed through numbers of teeth of wheels 3, 1 and 6, 4.

Solving the system of equations (12), (13) we shall determine angular speeds ω_3 , ω_1 of blocks of wheels 3-6 and 1-4

$$\omega_3 = \frac{(u_{13}^{(H1)} - 1) \omega_{H1} - (u_{46}^{(H2)} - 1) \omega_{H2}}{u_{13}^{(H1)} - u_{46}^{(H2)}}, \quad (14)$$

$$\omega_1 = u_{13}^{(H1)} (\omega_3 - \omega_{H1}) + \omega_{H1}. \quad (15)$$

Reactions in kinematic pairs are determined on the set moments of external forces using conditions of a statics.

Start up of working body occurs at motionless position of an output shaft of a gear mechanism (carrier H_2) in a status with one degree of freedom.

On start the satellite 5 which is placed on the motionless output carrier H_1 is the output link. The output satellite 5 should have the moment of resistance to pass the effort from input carrier H_1 on motionless carrier H_2 . Then the input moment on carrier H_1 will be equal to the output moment on carrier H_2 on conditions of statics for all mechanism. This moment is the starting moment.

The satellite 5 should have the demanded moment of inertia and the accounted dynamic moment arising because of accelerated motion of the satellite 5 to execute start up and to begin motion.

The efficiency of the gear adaptive mechanism corresponds to efficiency of the closed differential mechanism.

The adaptive gear mechanism has the following essentially new functional properties:

1. A wide range of change of the transfer ratio: from 1 up to 10. Higher transfer ratio leads to excessive increase in angular speeds of intermediate links.
2. The smooth stepless automatic regulation of the transfer ratio without any control.
3. Absolute conformity of the variable transfer ratio to external force resistance.
4. High efficiency because of absence of power losses on transformation of the transfer ratio.
5. High reliability in connection with use of tooth wheels which are in constant engagement.
6. Extreme simplicity of a design.

8. Practical realization of gear adaptive mechanism

The developed assembly drawing of the continuously variable transmission (fig. 4).

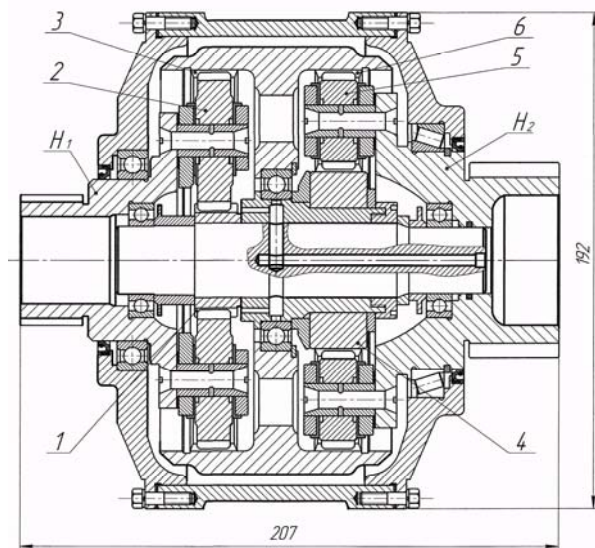


Fig. 4. Gear continuously variable transmission. Assembly drawing

On figure 5 the animation model of the working adaptive gear mechanism is presented.

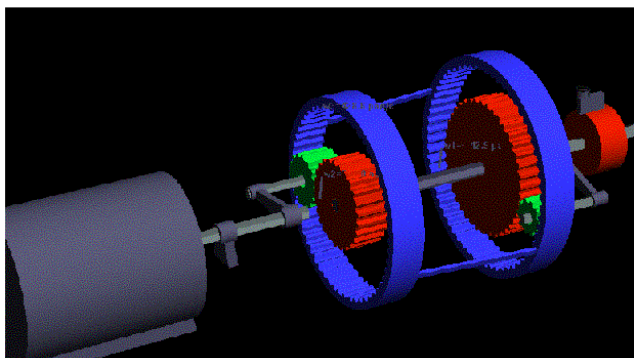


Fig. 5. Animation model of working adaptive gear mechanism

This animation model contains the engine (at the left), gear adaptive mechanism matching scheme on figure 2 and the brake which creates the external loading (at the right). Animation model of working adaptive gear mechanism is presented on web-site <http://www.madbass.narod.ru>. The changing of the external loading by keys W, Q leads to the changing of output angle speed.

This web-site allows to see work of mechanism at constant input power and variable torque, and also to download the model.

9. Conclusion

Existence of unknown law in the theory of mechanisms and machines is proved - a kinematic chain with two degrees of freedom containing two links with the applied forces and placed between them a closed four-bar contour at presence of one input link is in equilibrium. The mobile four-bar closed contour which is under action of arbitrary external forces superimposes an additional constraint condition on motion of links because of connection of forces with displacements. Such kinematic chain has essentially new property of force adaptation. - Speed of motion of an output link of the mechanism is inverse to resistance force on constant parameters of input power. Force adaptation has huge practical value. It allows creating adaptive drives of machines with the variable transfer ratio depending on technological resistance.

The found structural laws of an adaptive lever kinematic chain are used to create the gear adaptive mechanism with constant engagement of tooth wheels.

Analytical expressions found for the adaptive lever mechanism are used to describe interrelation of kinematic and force parameters of the gear adaptive mechanism.

The gear mechanism with two degrees of freedom at presence only one input realizes effect of force adaptation which allows using it as a gear automatic adaptive variator with constant engagement of tooth wheels. Gear adaptive transfer possesses essentially new functional properties defining its high efficiency.

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