The Blagonravov Continuously Variable Transmission:
importance of internal force functions for torque
transforming

A V Yurkevich*, A V Tereshin and V A Soldatkin

Federal State Budgetary Scientific Institution ‘Institute of Engineering Science, Ural
Branch of the Russian Academy of Sciences’, the Russian Federation. 34,
Komsomolskaya St., Ekaterinburg, Russia

*urkeva@mail.ru

Abstract. It is shown that the use of properties of the potential force field generated by the
elastic forces to transform mechanical energy enables us to provide infinite kinematic and
significant power ranges of mechanical transmission. An example of a kinematic diagram of
the Blagonravov continuously variable transmission implementing this method is given. It is
shown that the use of elastic forces in a continuously variable transmission with kinematically
provided oscillatory motion of internal links in combination with mechanical rectifiers
provides in the stop mode of the transmission the energy return to the drive motor while
maintaining a significant moment on the transmission driven shaft. The mechanism of
formation and the role of power functions in the transformation of the moment are described.
The possibility of independent control of power functions and the rotational speed of the drive
shaft of the transmission is shown. This property is fundamental and practical.

1. Introduction

The Blagonravov continuously variable transmission [1, 2] is a new type of self-regulating mechanical
continuously variable transmission [3]. It is intended, like most such transmissions, for the use as a
transmission system of vehicles, as well as for drive systems of various technological equipment.

Among well-known concepts, one can distinguish the type of continuously variable transmissions,
in which the gear ratio changes forcibly. These include continuous friction variators [4] (the radii of
motion translation from the drive shaft to the driven shaft are regulated), pulse variators [5] (the
oscillation amplitude of internal links is regulated), hydraulic variators (the pump is cycled by
changing the angle of the washer).

Such transmissions require a rather complicated control system, and the gear ratio does not depend
on the transmitted moment. Transmissions typically have a limited power range.

A fundamentally different type of transmission has the ability to spontaneously, due to the physical
nature of the processes occurring in the technical system, change the gear ratio \( i (i = \omega_2 / \omega_1 \text{ where } \omega_2 \text{ is the angular speed of the driven shaft, and } \omega_1 \text{ is the drive shaft) depending on the magnitude of the external moment. In such transfers, the transformation of moment } M \text{ is carried out in accordance with the expression } M = dL / dt , \text{ where } L \text{ is the angular momentum of the system (kinetic moment) [6]. Implementation of this principle of action may correspond to transmissions of various designs.}
An example of such transmission is a torque converter. In it, the energy supplied from the engine to the pump is converted into the kinetic energy of the fluid flow in the circulation circle. Changing the moment of momentum on each wheel of the torque converter (pump, turbine, deflector) creates a moment equal to the external moment. The geometry of the hydrodynamic lattice of the circulation circle is such that when angular velocity $\omega_2$ of the turbine decreases, the difference $dL$ in the angular momentum of the flow at the turbine inlet and outlet increases. Therefore, when the external moment on the turbine shaft increases, its angular velocity automatically decreases, and the gear ratio $i$ decreases.

An example of a self-regulating mechanical system is an inertial moment transformer [7,8]. As in the torque converter, the moments on the link shafts are determined by the change in the angular momentum of the system. The oscillation amplitude of internal links of this transmission is spontaneously established with a dynamic balance of resistance forces and inertial forces. The automatic transmission is determined firstly by a decrease in the momentum (the momentum is the time integral over time) on the driven shaft with an increase in its angular velocity and, secondly, by an increase in cycle time $dt$ with an increase in the angular velocity of the driven shaft. External control of the transmission is carried out by changing the supplied kinetic energy by changing the speed of the drive shaft $\omega_1$.

There are known attempts [9,8] to use energy for transformation with ensuring internal automatism and kinetic and potential energy (elastic twisting of torsion shafts). In this case, the amplitude of the angular oscillations of the driven links also changes spontaneously, but with a dynamic equilibrium of changes in the kinetic and potential energy during the cycle. The level of kinetic energy in the cycle of such transmissions, as in the case of a torque converter and an inertial transformer, is proportional to the square of the drive shaft speed.

However, quite often the working processes of machinery and equipment require various combinations of working moments and rotational speeds on the working bodies. However, at the same time, it is desirable that the drive power source (engine) operates in a fairly economical mode. Regulation, in which the operator can only change the rotation frequency of the motor shaft, does not provide such an opportunity.

Therefore, it was outlined in [9], and in [3] it was shown that the use of kinetic energy should be abandoned. There, Professor Blagonravov proposed to use the properties of the potential force field generated by elastic forces only in accordance with the expression $M = -dU/dx$ [6], where $U(x)$ is the force function, $x$ is the deformation of the elastic element to transform the moment. For the field of elasticity $U(x) = -\Pi$, where $\Pi$ is the potential energy of the mechanical system in its given position. It is equal to the work that the field forces will produce when the system moves from a given position to zero. Then, for mechanical transmission, the expression of potential energy for this force field can be found from the equation $\Pi = 0.5 \cdot cx^2$, where $c$ is the stiffness of the torsion shaft, $x$ is the angular deformation of the torsion. A property of the potential force field is that the work of the field forces will increase when the system moves from a given position to zero. Then, for mechanical transmission, the expression of potential energy for this force field can be found from the equation $\Pi = 0.5 \cdot cx^2$, where $c$ is the stiffness of the torsion shaft, $x$ is the angular deformation of the torsion. A property of the potential force field is that the work of the field forces is directed towards a decrease in the potential energy values. Internal automatism — a decrease in the moment $M$ with an increase in $i$ is achieved in this case not by changing the angular momentum (changing kinetic energy), but by tending to zero of the force function (potential spin energy of the torsion shafts). In this case, the control of the rotational speed of the drive shaft of the transmission (engine), of course, is preserved.

In the Blagonravov transmission, the above theoretical provisions were first implemented. In the scheme of this transmission, unlike the well-known Maltsev pulse variator [5], the free-wheeling mechanisms are supplemented with elastic shafts - torsions, as a result of which such a variator turns into a continuous transformer - a mechanical continuously variable transmission [3]. At the same time, the transmission acquires adjustable internal automaticity and continuity, which is maintained over a significant range of the gear ratio. In [10], at a qualitative level, the influence of power functions on the transmission properties and the features of its functioning is described in sufficient detail. The
quantitative assessment of power functions, the mechanism of formation and their role in the transformation of the moment is the task of this article.

2. Formation mechanism of power functions of the Blagonravov transmission

One of the possible options for the kinematic scheme of a mechanical continuously variable transmission with internal power functions is shown in Figure 1. The principle of operation is as follows: rotation of the drive shaft, consisting of the front 1 and the tail shaft 10 connected to it by 11 using the articulation linkage (common crank 3, connecting rods 4), is converted into angular vibrations of the leading elements of five mechanical rectifiers 5 – free-wheeling mechanisms (FWM). Rectifiers provide a kinematic condition by which the speed of their driven element can be greater, but cannot be less than the angular velocity of the rocker arm. Since there are five torsions working with a phase shift of 72 degrees, the transmission is multi-threaded. The leading elements of FWM - the rocker arms oscillate with a phase shift, and the driven elements are connected through torsion shafts 6 with gears 7 of the summing reducer. The central tooth gear 8 of this reducer is a driven transmission shaft 9. Torsions stretch momentum pulses in time, providing overlap and continuity of the moment on the output shaft. The transmission transmits torque to the driven shaft continuously and in multiple streams. It becomes impulse only when the internal gear ratio [3], which varies from 0 to 1, reaches a value greater than 0.9.

![Figure 1. Kinematic scheme of a continuously variable transmission with internal power functions.](image-url)

The oscillation amplitude of rocker arms is determined by the distance from the axis of general crank 3 to the axis of drive shaft 1. The amplitude is changed by lead 2 by turning it relative to the input shaft by a control mechanism, which consists of two identical planetary gears 14 and 15 united by a pinion carrier. The sun gear of the planetary gear 14 is rigidly connected to drive shaft 1. The epicyclic of this gear set is connected to the worm wheel of the gearbox 16 through its outer gear ring and intermediate gears. When controlling the amplitude of oscillations, rotation of this wheel provides a proportional rotation of the sun gear of planetary gear 15 relative to the sun gear of row 14 by an angle \( \Delta \phi = -K\phi_3 \), where \( K \) is a characteristic of planetary gears 14 and 15, and \( \phi_3 \) is the angle of rotation of the controlled epicyclic gear of planetary gear 14. The sun gear 15 of the row through lead 2 unfolds common crank 3 relative to rotating shaft 1, which leads to a change in the oscillation amplitude of the rocker arms of rectifiers 5. The epicyclic gear of row 15 is rigidly connected to the transmission body. With the stationary epicyclic gears of planetary gears 14 and 15, the sun gears of these gears rotate with the same speed coinciding with the drive shaft, and the common pinion carrier rotates with a frequency proportional to the characteristic \( K \) of the planetary gears. The transmission lubrication system includes an external gear oil pump 12 with a pressure reducing valve. The pump is driven from drive shaft 9 through gear 13 and roller 11.

The nature of the transmission within one revolution of the drive shaft (one cycle) at different loading modes of the torsions is different. If the driven transmission shaft is stationary (stop mode \( i = 0 \)), then the rear ends of the torsions are also stationary \( \omega_2 = 0 \), and the front ends connected to the driven elements of the rectifiers turn towards the working operation of the rocker arm. At the same time, rectifiers are switched on during the entire rotation of the drive shaft. As a result, the torsions are
twisted, their elastic deformation occurs. The energy spent on this elastic deformation forms force function \( U \). The force function creates the potential energy \( \Pi = -U \) equal in absolute value to it — work that can be done with a decrease in the elastic deformation of the torsions.

In a twisted torsion, the moments acting on its opposite ends are equal in magnitude but opposite in direction. The first half of oscillation period \( \omega t < \pi \) the input end of the torsion is rotated against the direction of the moment, while the potential energy increases from zero to maximum. The second half of oscillation period \( \omega t > \pi \) the direction of movement of the rocker arm is reversed. Potential energy of the swirling torsion will be consumed only from the input end and through the oscillation transformer will return to the drive shaft. Thus, during the second half of the oscillation period, all the potential energy of the torsion accumulated during the first half of the cycle is returned to the engine. Therefore, an average engine torque during one cycle required to create the potential energy of the torsions remains the same as is needed only to overcome friction losses in a loaded converter. At the same time, a torque is created on the driven transmission shaft equal to the sum of five torsion average times averaged over the period, taking into account the gear ratio of the summing reducer. As a result, the maximum value of the transformation coefficient — the ratio of the moments at the output and the input — can turn out to be very large.

If the driven shaft rotates \( i > 0 \), then the rear end of the torsion also rotates \( \omega_2 > 0 \). The ON phase of the rectifiers is reduced. The direction of rotation of the rear end of the torsion and the direction of the moment acting from its twist coincide. Moreover, during one cycle, work is performed equal to the product of this moment by the angle of rotation of the shaft. Therefore, the potential spin energy is consumed and transmitted to the driven shaft.

The total level of potential energy of the twist of the torsion shafts is reduced. The average value of the transmitted moment \( M_2 \) to the driven shaft also decreases. The intensity of the expenditure of this energy depends on the speed of rotation of the torsion shaft end. By changing the amplitude of the oscillations of leading links \( \varphi_0 \) and thereby the possible maximum torsion angles of twist control, it is possible to obtain controlled internal automatism. Since only a part of the potential energy \( \Delta \Pi \) returns to the drive shaft in the second half of the oscillation period, the average moment of the engine \( M_2 = M_1 \) during one cycle required to create the potential energy of the torsions increases.

When \( i = 1 \), potential energy is not generated, the moments on the drive shaft and the driven shaft are zero.

In Figure 2.a. the design diagram of loading the torsion shaft of one stream is shown. The following designations are adopted: \( OABC \) - four-link oscillation transducer; \( OA = r \) is radius of the general crank; \( AB = l_2 \) - connecting rod; \( BC = l_3 \) - the rocker arm of the rectifier connected to the input end of the torsion shaft through a mechanical rectifier (FWM) and oscillating with the amplitude \( \varphi_0 = \varphi_0 = \arcsin(r/l_3) \). The frequency of rotation \( \omega_0 \) of the common crank is equal to the frequency of rotation of the drive shaft of the transmission, and the moment \( M_1 = M_0 \). The rotational speed \( \omega_2 \) of the output end of the torsion shaft corresponds to the rotational speed of the driven transmission shaft, taking into account the gear ratio of the summing reducer. The moment \( M_2 \) at the output end of the torsion corresponds to the resistance moment given to it on the driven transmission shaft.
Figure 2. The design scheme of loading the torsion shaft. a) – design scheme, b) – formation of the torsion shaft twisting angle (power function); \(\omega_1 t\) – angle of rotation of the drive shaft, when the rectifier is turned on, \(\omega_2 t\) – angle of rotation of the drive shaft when the rectifier is turned off.

In accordance with [3], the assumptions are made: - the output end of the torsion shaft rotates with \(\omega_2 = \text{const}\); - the rocker arm of the rectifier \(CB\) oscillates according to a harmonic law; - energy losses in kinematic pairs are neglected; - turning on and off the rectifiers occurs without slipping the slave and leading elements of the rectifiers. Then the kinematic relations for the ends of the torsion shaft will take the following form

\[
\begin{align*}
\varphi_x &= -\varphi_0 \cos(\omega_1 t), \\
\omega_x &= \varphi_0 \omega_1 \sin(\omega_1 t), \\
\varphi_2 &= i \omega_1 t, \\
\omega_2 &= i \omega_1, 
\end{align*}
\]

where \(t\) – time; \(\varphi_x\), \(\omega_x\) – angle of rotation and angular velocity of the rocker arm; \(\varphi_2\) – angle of rotation of the output end of the torsion; \(i\) – overall gear ratio.

Gear ratio \(i\) is determined by

\[
\begin{align*}
i &= i_{HF} i_P, 
\end{align*}
\]

where \(i_{HF}\) – maximum gear ratio of the vibration transducer [3] (for harmonic law \(i_{HF} = \varphi_0\)); \(i_T\) – torsion internal gear ratio, varying from 0 to 1 [3]; \(i_P\) – gear ratio of the summing gear reducer for the circuit under consideration \(i_P = 1\).

Figure 2. \(\delta\) shows the results of the calculation according to formulas (1–5) for \(\varphi_0 = 0.2\) and \(i = 0.25\). The rectifier turns on at the phase angle of rotation of the drive shaft \(\omega_1 t_1\), and turns off at the phase angle \(\omega_1 t_3\) at which the angle \(\varphi_{ST}\) becomes equal to zero. According to [3] \(\omega_1 t_1 = \arcsin(i_P)\), and \(\omega_1 t_3\) is determined by solving the transcendental equation \(\cos(\omega_1 t_1) - \cos(\omega_2 t) - i_T \omega_1 (t_3 - t_1) = 0\). The torsion angle of rotation is determined as \(\varphi_{ST} = \varphi_x - \varphi_2\), or taking into account (1, 3 and 5) it will be determined

\[
\varphi_{ST} = \varphi_0 \left[ \cos(\omega_1 t_1) - \cos(\omega_2 t) - i_T (t - t_1) \right] 
\]

Using the equation of the relation between the angular deformations and the stiffness of the torsion \(c_T\), the value of the moment twisting the torsion is

\[
M_T = c_T \varphi_{ST} 
\]

The value of the moment loading the engine is determined

\[
M_\delta = M_1 = i M_2 
\]

Integration (6) from \(\omega_1 t_1\) to \(\omega_1 t_3\) enables you to determine the average value of the torsion shaft twist angle in one cycle
\[
\phi_{st}^p = \phi_0 \left[ \cos(\omega_1 t_1)(\omega_3 t_3 - \omega_1 t_1) - \sin(\omega_1 t_1) + \sin(\omega_3 t_3 - 0.5 \omega_1 (\omega_3 t_3 - \omega_1 t_1)^2 \right] (2\pi)^{-1}
\]

(9)

For multithreaded transmission (for example, for 5 equivalent flows), the average values of the moments on the driven and driving transmission shafts will be determined taking into account (9)

\[
M_2^{op} = 5c_T \phi_0^p \cdot (i_p)^{-1}
\]

(10)

\[
M_1^{op} = 5\phi_0^p c_T \phi_{st}^p
\]

(11)

Figure 3.a shows the results of the calculation according to (8 and 7) in relative units \((M_z / c_T \phi_0)\) for various values of the realized gear ratio \(i_T\).

\[\text{Figure 3. Change of moments on transmission shafts.}\]

a) \(1 - M_1, 2 - M_2, \phi_0 = 0.1, \) solid line - \(i_T = 0.05,\) dotted line \(i_T = 0.5;\)

b) fragment of the loading transmission oscillogram \(i_T \approx 0.\)

In Figure 3.b, a fragment of the oscillogram of the loading process of an experimental sample of such a transmission [11] with only one torsion installed in the body is shown. As can be seen from the figure, the nature of the change of \(M_1\) and \(M_2\) is identical to the calculated one. The average value per cycle in stop mode \(M_1^{op} \approx 0\) and \(M_2^{op} \neq 0\), which fully corresponds to the theoretical concepts of the functioning of such a transmission.

### 3. Power functions and moments on transmission shafts

When the drive shaft of the gearbox rotates, the engine spends work on the elastic twisting of the torsion and on completing useful work to overcome the external moment of resistance on the driven transmission shaft, which equals \(M_2\). The magnitude of the transmitted moment is completely determined by the magnitude of the torsion angle of rotation. That is, the movement is carried out in a potential force field of torsion elasticity. The magnitude of the power function is determined [6]

\[
U = -c_T \left( \phi_{st}^p \right)^2 / 2 + \text{const}
\]

(12)

This force function is associated with the potential energy of the twisted torsion shafts \(\Pi\) by the ratio \(U = -\Pi + \text{const}\). Such a function has an important property - its value depends only on the torsion angles of rotation and does not depend on speeds.

We compose a differential equation of energy balance for one stream (torsion). At any moment of time, during one cycle, the elementary work of the engine \(dA_1\) to rotate the drive shaft up to an efficiency is equal to the elementary work performed by the rocker arm of rectifiers \(dA_k\) connected through the rectifier to the input end of the torsion.

\[
dA_k = M_2 d\phi_k
\]

(13)

where \(d\phi_k\) is the elementary angle of rotation of the rocker arm at a constant moment \(M_2\), determined by (7).

An elementary angle \(d\phi_k\) is defined as a differential \(\phi_k\). Substituting \(d\phi_k\) in (13) we obtain

\[
dA_k = c_T \phi_{st}^p \phi_0 \omega_1 \sin(\omega_1 t) dt
\]

(14)
The twist angle of the torsion $\varphi_{T}^{n}$ is determined by (6). The work performed by the engine (rocker) at time $t$ will be determined as

$$A_t = A_c = c_T \varphi_0 \omega_1 \int_{t}^{t_1} \varphi_{T} \sin(\omega_1 t) dt$$

where $t_1$ - point in time at which the rectifier is jammed.

We further determine the work that the output end of the torsion will perform. Since the moment at the input and output ends of the torsion is the same, the elementary work performed by the output end is defined as

$$dA_2 = M_2 d\varphi_2$$

where $d\varphi_2$ is the elementary rotation angle of the output end of the torsion, which is determined by differentiating expression (3) taking into account that $i_H = \varphi_0$ and $i_P = 1$. Substituting $d\varphi_2$ into expression (16) we obtain

$$dA_2 = c_T \varphi_{T} \phi_{i} \omega_1 dt$$

The work performed by the output end of the torsion shaft at time $t$ will be determined by the integral

$$A_2 = c_T \varphi_{i} \omega_1 \int_{t}^{t_1} \varphi_{T} dt$$

The equation of work (energy) balance in a differential form $dA_1 = dA_2 + d\Pi$ by substituting (15 and 16) and subsequent integration enables us to obtain an expression for potential energy

$$\Pi = c_T \varphi_0 \omega_1 \left( \int_{t}^{t_1} \varphi_{T} \sin(\omega_1 t) dt - i_T \int_{t}^{t_1} \varphi_{T} dt \right)$$

The operation of the engine $A_1$ increases only up to the point in time $t_x = \pi/\omega_1$ - at which the angle of rotation of the drive shaft is equal $\omega_1 t_x = \pi$, because wherein $\sin(\omega_1 t_x) \geq 0$. Upon further rotation of the drive shaft, only part of the stored torsion potential energy is returned to the engine, since wherein $\sin(\omega_1 t_x) \leq 0$. However, another part of the potential energy continues to do useful work on the driven shaft. Using the well-known expression $\Pi = 0.5 c_T (\varphi_{T})^2$ does not allow to ‘divide’ potential energy into parts - to determine how much is returned to the engine and how much is given to the driven shaft.

The engine performs positive work only up to the phase angle of rotation $\omega_1 t_x = \pi$, and the total work performed during the cycle at the output end of the torsion is completed at the phase angle of rotation $\omega_3 t_3$. Therefore, the part of potential energy returned to the drive shaft can be determined by the formula

$$\Delta \Pi = A_1^* - A_2^*$$

where $A_1^*$ - work performed by the engine, determined by (15) for $t = t_x$; $A_2^*$ - work performed by the output end of the torsion shaft, determined by (18) for $t = t_3$. In relative units, expression (20) will have the following form

$$\Delta \Pi / \Pi_{t=0} = \left( A_1^* - A_2^* \right) / \Pi_{t=0}$$

where $\Pi_{t=0}$ is the maximum possible value of the potential twist torsion energy in the ‘stop’ mode, equal to $c_T (2\varphi_0)^2 / 2 = 2c_T \varphi_0^2$.

Expression (21) no longer depends on the amplitude of oscillations of the rocker arms of rectifiers $\varphi_0$, but depends only on the internal gear ratio of the torsion and characterizes the change in the fraction of potential energy returned to the drive shaft when the load on the driven transmission shaft changes (property of self-regulation of the transmission).
The relations obtained above make it possible to quantify the strength functions and their role in the transformation of moments on the transmission shafts. As an illustration, the influence of the internal gear ratio \( i_r \) characterizing the property of self-regulation of the transmission when the external load on the driven transmission shaft changes is shown in Figure 4.a.

![Figure 4](image)

Figure 4. Change in work \( A \) and potential energy \( P \) (power function) in one cycle. 1 – when \( \phi_0 = 0.1, i_r = 0.05 \); 2 – when \( \phi_0 = 0.1, i_r = 0.5 \); 3 – when \( \phi_0 = 0.3, i_r = 0.5 \); a) influence of internal gear ratio (self-regulation); b) influence of the set oscillation amplitude \( \phi_0 \).

Results are presented in terms of being relative to \( c_y \phi_0 \). The dashed line shows the nature of the change in work performed by the engine (rocker arm) \( A_1 \), the solid line shows the work performed by the output end of the torsion \( A_2 \).

The difference of these works forms the internal force function (potential energy \( \Pi \)). \( \Delta \Pi \) value - the part of potential energy returned to the engine (drive shaft). The nature of the change in \( \Pi \) during the cycle is presented in the lower graph. The maximum value of \( \Pi \) takes place at the maximum twist angle of the torsion shaft \( \phi_{1T} \). If provide control over the amplitude of oscillations \( \phi_0 \) (Figure 4.b.), then we obtain an adjustable control of potential energy and, therefore, an adjustable automatic transmission. Essentially, the transmission is an adjustable mechanical battery of potential energy.

The fraction of potential energy returned to the drive shaft of the engine \( \Delta \Pi / \Pi_{i=0} \) is shown in Figure 5.a and depends only on the internal gear ratio \( i_r \) of the torsion.

![Figure 5](image)

Figure 5. Modelling results. a) – fraction of potential energy returned to the drive shaft of the engine; b) – dimensionless transmission characteristic for rocker oscillation amplitude, 1- \( \phi_0 = 0.1 \) rad, 2 - \( \phi_0 = 0.2 \) rad, 3 - \( \phi_0 = 0.3 \) rad.
In stop mode $i_T = 0$, all potential energy is returned to the drive shaft. Therefore, on the dimensionless transmission characteristic (Figure 5.b.), $M_2^{sp}/c_T\Phi_{omax} = 0$ although $M_2^{sp}/c_T\Phi_{omax}$ assumes the maximum value (Solid lines - $M_2$ calculation according to (10) for one stream, dashed – calculation $M_1$ according to (11)). At $i_T = 0.725$, the return of potential energy does not occur $\Delta\Pi/\Pi_{i=0} = 0$. This property is fundamental and practical.

For example, when starting off at a vehicle equipped with such a transmission, the energy of the internal combustion engine is first spent on twisting the torsion bars (creating a power function) and returns. But at the same time, significant torque is applied to the wheels. This property completely eliminates the possibility of the engine stalling with increasing resistance to the movement of the vehicle, including when the vehicle completely stops.

4. Conclusion
In the Blagonravov transmission, power functions provide a continuously variable transformation of mechanical energy. In this case, the moment transmitted to the driven shaft is created only by changing the force function per cycle — the total potential strain energy of all the elastic transmission links. It depends on the gear ratio, but does not depend on the rotation speed of the drive shaft of the transmission and can be adjusted by moving the control, the position of which determines the amplitude of the oscillations of the internal links. The transmission has an adjustable internal automaticity and continuity, which is maintained over a significant range of gear ratios. Transmission provides infinite kinematic and significant power ranges.

References
[1] Patent No. 2211971 of the Russian Federation. Mechanical variable transmission, Blagonravov A.A. - Publ. 2003, Bull. Number 25.
[2] Patent No. 2211971 of the Russian Federation. Mechanical variable transmission, Blagonravov A.A. - Publ. 2003, Bull. Number 25.
[3] Blagonravov A.A. 2011 Calculation of the external characteristics of a mechanical transformer with oscillatory movement of internal links (Vestnik mashinostroenija Publ.) No. 10. - pp. 8-13.
[4] Li C, Li H, Li Q, Zhang S, & Yao J 2019 Modeling, kinematics and traction performance of no-spin mechanism based on roller-disk type of traction drive continuously variable transmission (Mechanism and Machine Theory) V.133, pp. 278–294. doi:10.1016/j.mechmachtheory.2018.11.017
[5] Maltsev V.F. 1978 Mechanical impulse transmissions (M.: Mashinostroenie Publ.) 367 p.
[6] Leach J.W. Classical mechanics (Transl. from English. Ya. I. Szekerz-Zenkowicz. M.: Publishing House of Foreign Literature) 172 p.
[7] Leonov A.I. 1978 Inertial automatic transformers of a torque (M.: Mashinostroenie Publ.) 224 p.
[8] Leonov A.I., Dubrovskiy A.F. 1984 Mechanical continuously variable transmissions (Moscow: Mashinostroenie Publ.) 191 p.
[9] Blagonravov A.A. 2005 Mechanical continuously variable transmissions (Yekaterinburg: Ural Branch of the Russian Academy of Sciences) 202 p.
[10] Blagonravov A.A., Yurkevich A.V., Tereshin A.V. Adjustable internal automaticity of a stepless mechanical transformer (Vestnik Mashinostroeniya Publ.) 2014. No. 2. – pp. 3-7.
[11] An experimental study of the loading of the main elements of a mechanical stepless transformer of the moment, the rationale for choosing the range of variation of the internal power function: R&D report (conc.): A.A. Blagonravov, A.V. Yurkevich et al. - Kurgan, 2016.-- 156 p. - No. PH 01201368139. - ICRBS AAAA-B16-216032150071-4