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K.S. Ivanov[1], D.T. Tulekenova[2*], 2025.
[1,2]Almaty University of Power Engineering and Telecommunications named after G. Daukeev, Almaty, Kazakhstan;
[2]Al-Farabi Kazakh National University, Almaty, Kazakhstan.
E-mail: dana_tul@mail.ru

ENSURING THE DETERMINABILITY OF MOTION OF AN ADAPTIVE SPACECRAFT DRIVE BY INTRODUCING AN ADDITIONAL VELOCITY CONSTRAINT FORCE

Konstantin Samsonovich Ivanov – professor, doctor of technical sciences, Almaty University of Power Engineering and Telecommunications named after G. Daukeev, Almaty, Kazakhstan, E-mail: ivanovgreek@mail.ru, https://orcid.org/0000-0002-3750-8081;
Dana Toleubekovna Tulekenova – senior lector, master of natural sciences, Almaty University of Power Engineering and Telecommunications named after G. Daukeev, Almaty, Kazakhstan, E-mail: dana_tul@mail.ru, https://orcid.org/0000-0002-9509-4331.

Abstract: The adaptive drive of a docking mechanism is a critical component in automatic docking systems, widely used in spacecraft, marine vessels, and industrial robots. The primary goal of developing such a drive is to ensure precise and reliable connections between two objects under variable external conditions and within limited maneuvering space.

The reliable adaptation of a two-movable self-regulating mechanical drive consists in self-adaptability to an external load balancing friction clutch, which ensures the connection of the friction moment with the relative angular velocity. Recent studies of the interaction of force and kinematic parameters, reported at a Symposium dedicated to the 60th anniversary of the MMT journal, have opened up the possibility of creating an additional connection without losing the degree of freedom. Such a connection can be obtained by replacing the action of the above reaction with the action of the friction moment in the hinge of the intermediate link. The friction moment creates a force connection, which is taken into account in the equilibrium condition, and the friction joint retains relative mobility. The obtained equilibrium conditions ensure the definiteness of motion and the ability of self-regulation in the form of an inversely proportional dependence of the speed of the output link depending on the variable external load. The described method makes it possible to create a fundamentally new class of self-regulating mechanisms in all branches of technology.

The developed mathematical model of the existing central-type docking mechanism can be used for an adaptive drive docking mechanism by using the above simplifications of the changes. The paper briefly formulates the basics of the theory of power adaptation of a gear drive and develops a prototype of an adaptive drive for the docking mechanism of a spacecraft.

Key words: self-regulating transmission, planetary mechanism, force adaptation, mathematical model, design, simulation

К.С. Иванов[1], Д.Т. Тулекенова[2*], 2025.

[1,2]Ғ. Дәукеев атындағы Алматы энергетика және байланыс университеті, Алматы, Қазақстан;

[2]Әл-Фараби атындағы Қазақ ұлттық университеті, Алматы, Қазақстан.

E-mail: dana_tul@mail.ru

ЖЫЛДАМДЫҚ БАЙЛАНЫСЫНЫҢ ҚОСЫМША КҮШІН ЕНГІЗУ АРҚЫЛЫ ҒАРЫШ АППАРАТЫНЫҢ БЕЙІМДЕЛГЕН ЖЕТЕК ҚОЗҒАЛЫСЫНЫҢ АЙҚЫНДЫЛЫҒЫН ҚАМТАМАСЫЗ ЕТУ

Константин Самсонович Иванов – профессор, техника ғылымдарының докторы, Ғ. Дәукеев атындағы Алматы энергетика және байланыс университеті, Алматы, Қазақстан, E-mail: ivanovgreek@mail.ru, https://orcid.org/0000-0002-3750-8081;

Дана Толеубековна Тулекенова – аға оқытушы, жаратылыстану ғылымдарының магистрі, Ғ. Дәукеев атындағы Алматы энергетика және байланыс университеті, Алматы, Қазақстан, E-mail: dana_tul@mail.ru, https://orcid.org/0000-0002-9509-4331.

Аннотация: Қондыру механизмінің бейімделген жетегі ғарыш аппараттарында, теңіз кемелерінде және өнеркәсіптік роботтарда кеңінен қолданылатын автоматты қондыру жүйелерінің маңызды құрамдас бөлігі болып табылады. Мұндай жетекті дамытудың негізгі мақсаты – сыртқы жағдайларға бейімделетін және маневр жасау үшін шектеулі кеңістікте екі нысанның дәл және сенімді байланысын қамтамасыз ету.

Екі дәрежелі өзін-өзі реттейтін механикалық жетектің сенімді бейімделуі үйкеліс моментінің салыстырмалы бұрыштық жылдамдықпен байланысын қамтамасыз ететін жүктемені теңестіретін сыртқы үйкеліс муфтасына өзін-өзі реттеу болып табылады. ММТ журналының 60 жылдығына арналған Симпозиумда баяндалған күш пен кинематикалық параметрлердің өзара әрекеттесуі туралы соңғы зерттеулер еркіндік дәрежесін жоғалтпай қосымша байланыс орнатуға мүмкіндік берді. Мұндай байланысты жоғарыда аталған реакцияның әрекетін аралық буынның топсасындағы үйкеліс моментінің әсерімен алмастыру арқылы алуға болады. Үйкеліс моменті тепе-теңдік жағдайында ескерілетін күш байланысын тудырады, ал үйкеліс буыны салыстырмалы қозғалғыштығын сақтайды. Алынған тепе-теңдік шарттары

қозғалыстың анықтығын және ауыспалы сыртқы жүктемеге байланысты шығыс байланысының жылдамдығының кері пропорционалды тәуелділігі түрінде өзін-өзі реттеу қабілетін қамтамасыз етеді. Сипатталған әдіс технологияның барлық салаларында өзін-өзі реттейтін механизмдердің түбегейлі жаңа класын құруға мүмкіндік береді.

Қолданыстағы орталық типті қондыру механизмінің математикалық моделін жоғарыда келтірілген өзгертулерді жеңілдету арқылы бейімделген жетекті қондыру механизмі үшін пайдалануға болады. Мақалада беріліс вариаторының күштік бейімделу теориясының негіздері қысқаша сипатталған және ғарыш аппаратының қондыру механизмі үшін бейімделген жетектің прототипі жасалған.

Түйін сөздер: бейімделген тісті жетек, күштік бейімделу, планетарлық тісті механизм, математикалық моделі, жобалау, моделдеу.

К.С. Иванов[1], Д.Т. Тулекенова[2*], 2025.
[1,2]Алматинский университет энергетики и связи имени Г. Даукеева, Алматы, Казахстан;
[2]Казахский национальный университет имени аль-Фараби, Алматы, Казахстан.
E-mail: dana_tul@mail.ru

ОБЕСПЕЧЕНИЕ ОПРЕДЕЛИМОСТИ ДВИЖЕНИЯ АДАПТИВНОГО ПРИВОДА КОСМИЧЕСКОГО АППАРАТА С ПОМОЩЬЮ ВВЕДЕНИЯ ДОПОЛНИТЕЛЬНОЙ СИЛЫ СКОРОСТНОЙ СВЯЗИ

Константин Самсонович Иванов – профессор, доктор технических наук, Алматинский университет энергетики и связи, Алматы, Казахстан, E-mail: ivanovgreek@mail.ru, https://orcid.org/0000-0002-3750-8081;
Дана Толеубековна Тулекенова – старший преподаватель, магистр естественных наук, Алматинский университет энергетики и связи, Алматы, Казахстан, E-mail: dana_tul@mail.ru, https://orcid.org/0000-0002-9509-4331.

Аннотация. Адаптивный привод стыковочного механизма является важнейшим компонентом систем автоматической стыковки, широко используемых в космических аппаратах, морских судах и промышленных роботах. Основная цель разработки такого привода – обеспечить точное и надежное соединение двух объектов в изменяющихся внешних условиях и в ограниченном пространстве для маневрирования.

Надежная адаптация двухподвижного саморегулирующегося механического привода заключается в самоприспособляемости к внешней уравновешивающей нагрузку фрикционной муфте, обеспечивающей связь фрикционного момента с относительной угловой скоростью. Последние исследования взаимодействия силовых и кинематических параметров, представленные на Симпозиуме, посвященном 60-летию журнала MMT, открыли возможность создать дополнительную связь без потери степени

свободы. Такая связь может быть получена путем замены действия вышеупомянутой реакции действием фрикционного момента в шарнире промежуточного звена. Фрикционный момент создает силовую связь, учитываемую в условии равновесия, а фрикционный шарнир сохраняет относительную подвижность. Полученные условия равновесия обеспечивают определимость движения и способность саморегулирования в виде обратной пропорциональной зависимости скорости выходного звена в зависимости от переменной внешней нагрузки. Описанный способ позволяет создать принципиально новый класс саморегулирующихся механизмов во всех отраслях техники.

Разработанная математическая модель существующего стыковочного механизма центрального типа может быть использована для стыковочного механизма с адаптивным приводом благодаря вышеописанным упрощениям и модификациям. В статье кратко излагаются основы теории силовой адаптации зубчатого привода и разрабатывается прототип адаптивного привода для стыковочного механизма космического аппарата.

Ключевые слова: саморегулирующаяся трансмиссия, планетарный механизм, силовая адаптация, математическая модель, проектирование, моделирование.

Introduction. The existing drive has a limited range of action due to the lack of sufficient conditions for determining the movement of the required power factor of the existing design is not enough to achieve the required control range. The existing total moment of friction in kinematic pairs turned out to be insufficient to achieve a practical control range (Syromyatnikov, et al., 1984).

The drive mechanism of a spacecraft docking system is a critical component that ensures the precise and reliable connection between two spacecraft. This mechanism must operate effectively in the unique and challenging environment of space, which includes factors such as weightlessness, high relative velocities, and potential external disturbances. One innovative approach to improving the control and determinacy of the docking mechanism drive is through the introduction of additional velocity coupling forces (Fehse, et al., 2003).

The drive mechanism of a spacecraft docking system plays a vital role in ensuring successful and reliable docking operations. By incorporating additional velocity coupling forces into the control system, the docking mechanism can achieve higher levels of precision, stability, and adaptability. This approach addresses the challenges posed by the space environment and enhances the overall performance and reliability of the docking process (Boesso, et al., 2013).

Docking mechanisms have been employed in space missions for more than 50 years. With the start of the Space Stations era, the new concept of standardization was born and the contribution of different participants from various countries led to the definition of common design requirements (Yaskevich, 2019).

Materials and methods of research.

1) The proposed solution for the drive of the docking mechanism.

Fig. 1 shows one of the options for an illustrative example of a spacecraft docking mechanism drive. In the drive of the tightening device, two duplicate electric motors 1 relate to the output shaft 5 through locking clutches 2, planetary mechanism (differential) 3 and two planetary gears 4.

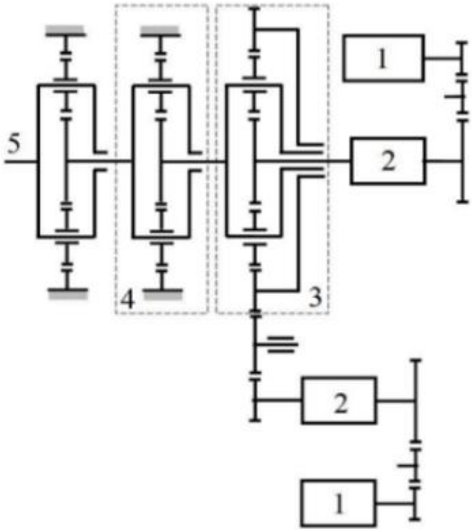

Figure 1. An example of drive of docking mechanism as a kinematic design: planetary unit-4,5; planetary gear-3; motors-1,2 (Golubev, 2019).

The locking clutches prevent transmission of motion from the main wheels of the differential to the electric motors.

The drives of electromechanical docking devices use DC collector motors. Permanent magnet motors in four-pole design, with a duplicated brush assembly and armature winding, as practice shows, are highly reliable. These motors have a high ratio of starting torque to nominal torque for the applied power range (Mp/Mn = 5...10) and therefore have a good overload capacity, which favorably affects the reliability of the drives, but requires the use of safety couplings. Despite the presence of a brush collector, the engines have good performance in vacuum. To further increase the life of engines and other elements in outer space, it is possible to completely seal housings, use AC motors, brushless DC motors. Disadvantages of motors can be indicated in: need for on-board DC-to-AC converters; large mass and dimensions of the engines themselves; lower efficiency (for low-power engines); 4) worse overload capacity (lower starting torque multiplicity Mp/Mn = 1.5 ... 2).

Disadvantages of the existing drive of the docking mechanism: 1. The complexity of the design; 2. Large dimensions and weight; 3. Large energy losses; 4. Relatively low reliability.

2) Design of an adaptive drive.

The proposed design is based on previous works for new design of a gear drive. The adjustable drive proposed in this paper contains an electric motor and a gear variator. Adaptive gear variator contains input driver , closed loop of gears 1–2–3–6–5–4 a closed loop of gears and an output driver . The closed loop contains the input satellite 2, solar wheel unit 1-4, ring wheel unit 3-6 and output satellite 5 (Ivanov, 2019).

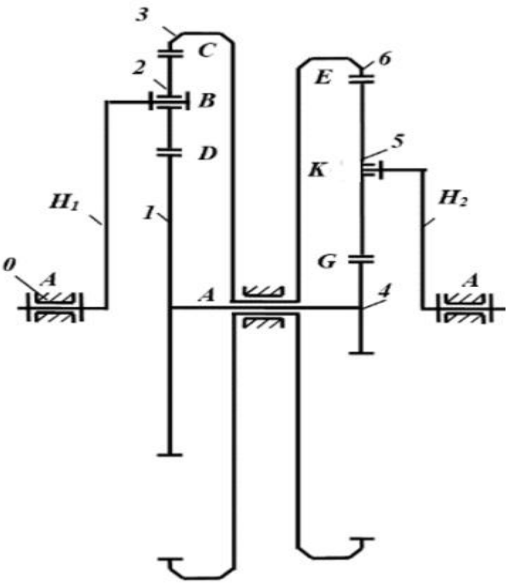

Figure 2. Kinematic scheme of the proposed adaptive drive

The drive in Fig. 2 is considered to replace the existing drive Fig. 1 of the spacecraft docking mechanism.

The formulation of the problem of force analysis of a mechanism with two degrees of freedom (Fig. 2) and with two inputs is as follows: according to the given external forces, reactions in kinematic pairs and generalized external forces F_{H1} and F_{H2} are determined (or moments $M_{H1} = F_{H1}r_{H1}$ and $M_{H2} = F_{H2}r_{H2}$) on the two input leads H_1 and H_2. The force analysis should begin by considering the structural group 1-2-3-6-5-4 in the form of a four-link closed circuit consisting of gears. The structural group contains a block of solar wheels 1-4, satellite 2, a block of epicyclic wheels 3-6 and satellite 5.

These conditions can be represented as equilibrium conditions based on the principle of possible displacements.

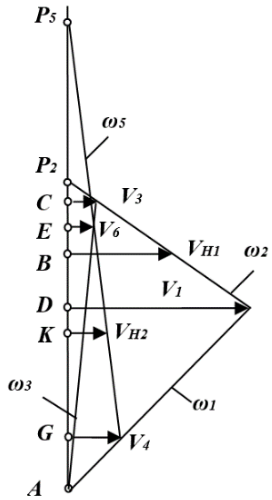

Figure 3. Linear velocity plan of the adaptive drive

Linear velocities are determined using known angular velocities and wheel radii using the formula $V_i = \omega_i r_i$.

A closed loop allows you to create static equations. Let's make up the equilibrium conditions for the links of the contour 2 and 5.

$$R_{12} + R_{32} = F_{H1}, \quad (1)$$

$$R_{45} + R_{65} = F_{H2}. \quad (2)$$

For satellite 2, we obtain from the equations of moments

$$R_{12} = 0.5F_{H1}, \quad (3)$$

$$R_{32} = 0.5F_{H1} \quad (4)$$

Multiply equation (3) by V_1 (the velocity of the point *D* of the satellite 2 or the circumferential velocity of the wheel 1). Multiply equation (4) by (the velocity of the satellite point 2 or the circumferential velocity of the gear 3).

$$R_{12}V_1 = 0.5F_{H1}V_1 \quad (5)$$

$$R_{32}V_3 = 0.5F_{H1}V_3 \quad (6)$$

Add up equations (5) and (6).

$$R_{12}V_1 + R_{32}V_3 = 0.5F_{H1}\,V_3 \quad (7)$$

According to the linear velocity plan of the mechanism (Fig. 3) $0.5\ V_3 = V_H$ where V_{H1} is the velocity of point *B* the satellite point 2 or the circumferential velocity of the carrier H_1.

Then from equation (7) we obtain the equilibrium equation of satellite 2 according to the principle of possible displacements using capacities instead of work

$$R_{12}V_1 + R_{32}V_3 = F_{H1}V_{H1} \tag{8}$$

In a similar way, we obtain the equilibrium condition of the satellite 5

$$R_{45}V_4 + R_{65}V_6 = F_{H2}V_{H2} \tag{9}$$

Where V_4, V_6, V_{H2} are the velocities of the points *E,G,K* of the satellite 5 or the circumferential velocities of the wheels 4, 6 and the carrier H_2.

Using equations (8), (9) it is possible to obtain an equilibrium equation based on the principle of possible displacements for the entire mechanism. Add up the equations (8), (9)

$$R_{12}V_1 + R_{32}V_3 + R_{45}V_4 + R_{65}V_6 = F_{H2}V_{H2} + F_{H2}V_{H2} \tag{10}$$

It is convenient in equation (10) to convert the linear parameters of the satellites into the angular parameters of the central wheels, as well as the linear parameters of the drivers into angular parameters.

To do this, we will use a replacement according to the formula. $V = \omega r$ with the appropriate indices for speeds, and for forces, we will replace reactions on satellites with reactions applied to the central wheels according to the $R_{12} = -R_{21}$ principle, etc.

$$-R_{12}\omega_1 r_1 - R_{23}\omega_3 r_3 - R_{54}\omega_4 r_4 - R_{56}\omega_6 r_6 = F_{H1}\omega_{H1}r_{H1} + F_{H2}\omega_{H2}r_{H2} \tag{11}$$

The product of the force by the radius determines the moment using the appropriate indices. Equation (11) will take the form

$$-M_{21}\omega_1 - M_{23}\omega_3 - M_{54}\omega_4 - M_{56}\omega_6 = M_{H1}\omega_{H1} + M_{H2}\omega_{H1} \tag{12}$$

Equation (12) contains the parameters of all the links of the mechanism and represents the equilibrium equation of the entire mechanism according to the principle of possible displacements. Note that such an equation can be composed only if there is a closed loop.

Let's transform equation (12) considering the equality of the angular velocities of the wheels in the wheel blocks $\omega_4 = \omega_1, \omega_6 = \omega_3$

$$-M_{21}\omega_1 - M_{23}\omega_3 - M_{54}\omega_1 - M_{56}\omega_3 = M_{H1}\omega_{H1} + M_{H2}\omega_{H1} \tag{13}$$

According to equation (13), the sum of the powers of the moments of internal forces on the blocks of the central wheels 1-4 and 3-6 is equal to the sum of the powers of the moments of external forces on the input drivers.

On the left side of equation (13), there is a sum of capacities (corresponding to the sum of work) of the internal forces of the contour. The connections in the kinematic pairs of the contour are ideal and stationary. The work of external forces cannot turn into the work of internal forces. Therefore, the work (power) of internal forces on possible displacements is zero

$$M_{21}\omega_1 + M_{23}\omega_3 + M_{54}\omega_1 + M_{56}\omega_3 = 0 \qquad (14)$$

The right side of equation (13) is the sum of the capacities (corresponding to the sum of the work) of the external forces of the contour. When condition (14) is fulfilled, we obtain from equation (13) the equilibrium condition for external forces according to the principle of possible displacements.

$$M_{H1}\omega_{H1} + M_{H2}\omega_{H2} = 0 \qquad (15)$$

Equation (15) analytically represents an additional connection to the static conditions between the parameters of the kinematic chain. Consequently, a closed loop in a conventional kinematic circuit with two degrees of freedom and two input links also imposes an additional connection on the movement of the links.

Equation (15) allows us to determine the output angular velocity.

The combination of two degrees of freedom with additional coupling ensures the dependence of the output angular velocity on the external load. This property follows from the formula (15)

$$\omega_{H2} = M_{H1}\,\omega_{H1}/M_{H2} \qquad (16)$$

Here M_{H1} is the input driving torque, and M_{H2} is the output torque of the resistance (external load).

Equation (16) expresses the main theoretical result – the effect of force adaptation in mechanics. The effect of force adaptation has the following essence: for given constant input power parameters and a given output moment of resistance, the output angular velocity is inversely proportional to the variable output moment of resistance.

Results and discussion.

The CAD design of adaptive drive in Fig. 2. is a self-regulating mechanism.

An experiment was conducted with an adaptive drive. At the beginning, it is considered when the adaptive drive is made of metal (Fig. 5). The second case is when the adaptive drive is made of plastic (Fig. 8).

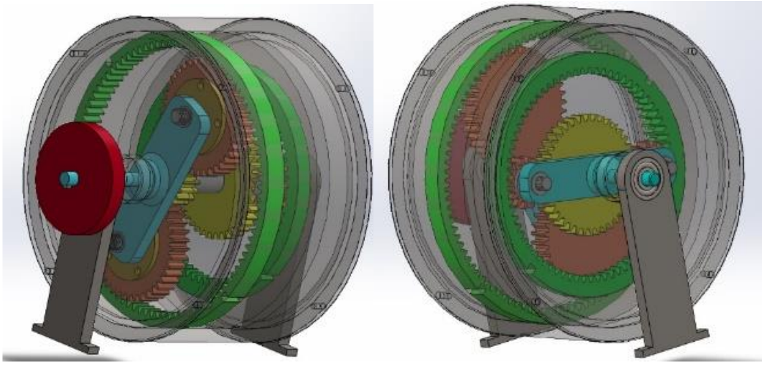

Fig. 4.1. Fig. 4.2.

Figure 4. Structural description of a three-dimensional adaptive drive model left view (4.1) and right view (4.2)

The module is equal to two, the number of teeth is shown in Table 1.

Table 1. Number of teeth and radius for gears in Fig.4

Gear	Number of teeth	Diameter (mm)
Input sun gear, Z1	40	80
Input satellite, Z2	16	32
Input ring gear, Z3	72	144
Output sun gear, Z4	16	32
Output satellite, Z5	40	80
Output ring gear, Z6	96	192

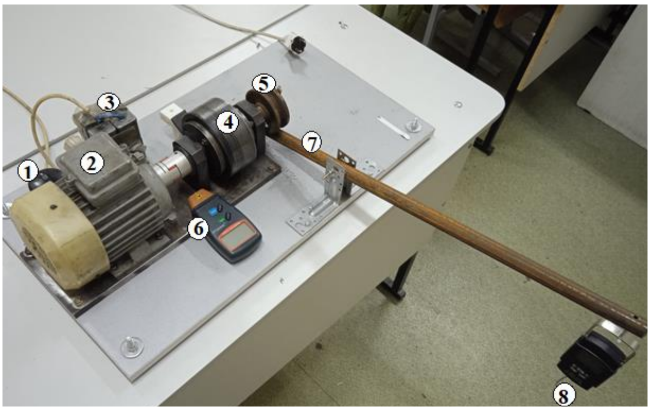

Figure 5. The prototype design of the metal Adaptive drive (Ivanov, 2019)

Fig. 5 shows the prototype of metal adaptive drive and consists of the following parts: 1 – the switch, 2 – motor, 3 – battery, 4 – adaptive drive, 5 – pulley, 6 – device for measuring angular velocity, 7 – lever, 8 – load.

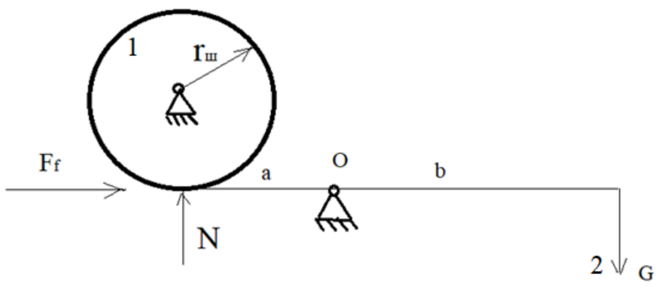

Figure 6. Torque sensor using a dynamometer

Fig. 6 shows the prototype of metal adaptive drive and consists of the following parts: 1 – pulley; 2 – load; a and b – lever arms.

The experimental moment is found by this formula.

$$M_f = Nfr_{ш} \qquad (17)$$

Friction power $$P_f = M_f\omega_f \qquad (18)$$

where $M_f\omega_f$ – an additional non-holonomic connection or connection that introduces force restrictions, but does not change the number of degrees of freedom equal to two.

where $F_f = Nf, \quad Na = Gb, N = Gb/a, \; G{=}mg \qquad (19)$

here N – normal force, f – coefficient of friction, $f = 0.6$, G – gravity; $r_{ш}$ – pulley radius; a, b – lever arms. The results of the experiment of a metal adaptive drive at

$\omega_{H1} = 155{,}5$ s^{-1}, $M_{H1} = 1{,}6$ Nm

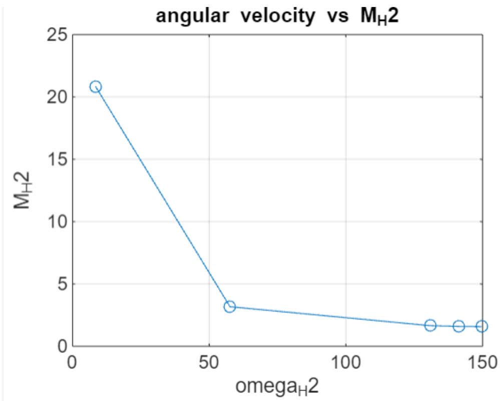

Figure 7. Computed output moment of resistance from the output angular velocity

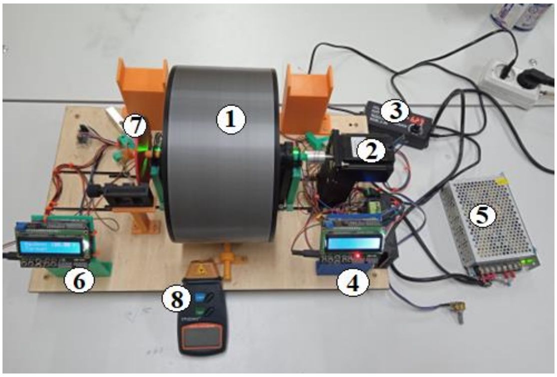

Figure 8. The prototype design of the plastic Adaptive drive (Tulekenova, et al, 2021)

Fig. 8 shows the prototype of adaptive drive and consists of the following parts: Drive: 1 – electric motor, 2 – adaptive gear variator; Test bench: 3 – AC to DC converter, 4 – voltage converter, 5 – input RPM meter, 6 – output RPM meter, 7 – output disc, 8 – device for measuring angular velocity.

The results of the experiment of plastic adaptive drive at ω_{H1} = 11,94 s^{-1}, M_{H1} = 1,8 *Nm*

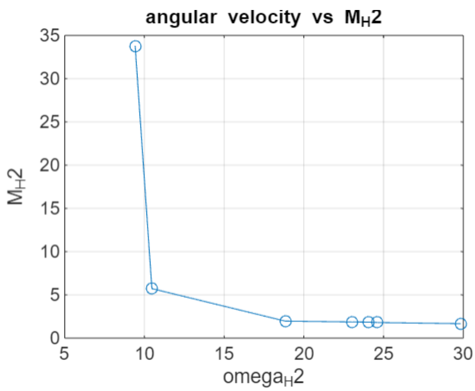

Figure 9. Computed output moment of resistance from the output angular velocity

As shown in (Fig.7) and (Fig.9) the output angular velocity decreases and the output torque increases accordingly. These graphs show the dependence of the output moment of resistance on the output angular velocity from numerical calculation, from the CAD model and the experimental sample. This explains the operating mode of the power adaptation of the drive.

The readings of the speed counters on the test bench correspond to the theoretical values obtained. Fig. 6 shows the traction characteristic of the prototype of a toothed adaptive variator in the form of a graph of the change in the traction torque on the output shaft in Nm depending on its rotation speed in rad/sec. The theoretical results are consistent with the test results on the stand. A closed loop as part of a kinematic chain with two degrees of freedom, in the presence of ideal connections, provides confidence in movement both in a state with two degrees of freedom and in a state with one degree of freedom.

Conclusion. The experiment confirms the presence of definability of movement and the effect of force adaptation. The adaptive drive with two degrees of freedom was developed considering aspects of mechanical design and kinematic modeling. The design of the adaptive drive is shown in the kinematic diagram. The mathematical model with an adaptive drive reduces loads over the entire range of initial docking conditions and allows you to implement specified external design constraints and the possibility of docking with nodes with different shapes of the receiving cone due to a slight narrowing of the set of permissible combinations of parameters from this range: 1) A scheme for actuating a coupling mechanism with an adaptive drive has been developed, based on the design requirements for the drive. The drive is greatly simplified. The brake motor, an additional degree of freedom, a differential mechanism, planetary mechanisms (2) and a control system that changes the gear ratio to the output shaft have been replaced. 2) The main design parameters of the adaptive drive (engagement module, number of teeth, gear ratio adjustment range, etc.) are calculated. 3) A dynamic analysis of the driving modes is carried out. 4) The analysis confirmed the advantages of the developed drive design.

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