Development of Energy Efficient Mechatronic Module for Alternative Energy

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Abstract. The article describes the creation of mechatronic module for follow-up systems. In the mechanical part of the module the authors applied a reducer, which is based on cam coaxial radial flat transmission with intermediate rolling bodies. Transmission ratio is determined and equation of tooth profile is derived for the transmission. Transmission kinematics with different number of rolling bodies was also analyzed. Mechatronic module was designed to limit overall dimensions. The analysis of stress-strain state of the most loaded elements of the transmission is conducted. Structural diagram of mechatronic module is developed. Structure of optimal control of BLDC motor is presented as well as a function of minimizing the dependence on characteristics of a transition process of motor speed is selected. An adaptive PID controller is synthesized using genetic algorithm.

1. Introduction
For mechatronic systems, the requirements of technical characteristics of modern mechanisms are applied. Gearmotors are one of the main driving devices of mechatronic systems. They are subject of high requirements in terms of mass-size-power ratio. The reducing mechanism is mainly responsible for providing this ratio. This ratio is best provided by rotation mechanisms made on the basis of kinematic wave transmissions [1, 2]. These transmissions receive a load of about half the teeth of the cycloid wheel, other words the teeth of the gear at an angle of approximately 180° are in the working area [3, 4]. This makes it possible to transmit large rotating moments relative to the toothed gears, with the same mass-dimensional characteristics. Modern drives use a compact motor with a hollow shaft like 3DBM (Figure 1). And in order to develop a mechatronic module based on this motor and cycloidal transmission with intermediate rolling bodies (IRB), it was a task to fit the reducer into the cavity of the motor shaft. This will minimize the overall dimensions of the mechatronic module and the entire mechanism; as well as develop an electric drive control system that allows to protect the drive from overheating and controlling the rotation speed of the motor shaft.
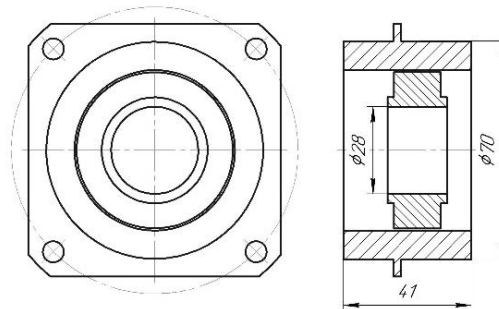
Figure 1. General view of the electric motor 3 DBM

Mechatronic module provides the following technical data:
- Reducer is designed for electric motor 3 DBM 70;
- Transmission ratio is 8;
- Rated torque, not less than 1 Nm;
- Maximum torque, not less than 1.5 Nm;
- Efficiency, not less than 85%;
- Weight of the reducer, not more than 0.25 kg.

There is a three-phase brushless direct current (BLDC) motor with the supply voltage of 27 volts, positioning accuracy on a turning angle 10 minutes, rotating speed of 3000 revolutions per minute that is considered as main characteristics of an electronic part of the module.

Currently, there are four generations of mechatronic modules. The most modern one is synergy of a mechanical part (the motor and reducer), an electronic part (drivers and microcontrollers) and control algorithms. The feature of fourth generation is application of optimal or adaptive control algorithms using methods of artificial intelligence.

2. Design of Compact Cycloidal Reducer
In order to minimize the dimensions of the reducer, we use a cam coaxial radial flat transmission with IRB (Figure 2), which is an analogue of the kinematic wave transmission.
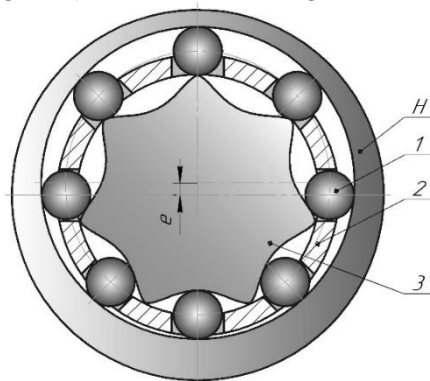

Figure 2. The cross section of cam coaxial radial flat transmission with intermediate roller bodies: H – generator (planet carrier); 1 – intermediate rolling body; 2 – separator; 3 – central gear; e – eccentricity of the transmission.

The transmission works following way: the planet carrier H rotate and made intermediate rolling bodies to move radial along grooves of separator 2. The separator 2 is fixed rigidly in housing of the transmission. The rollers contact with cycloid profile of central gear 3. By the radial movement rollers made to the central gear 3 rotated.

There is no detailed description in the literature on the design of such gears, so it is necessary to determine the transmission ratio and infer the tooth profile equation for the gear.

In the transmission, rolling bodies have planetary rotation, i.e. this one can be attributed to epicyclic gear train [6]. Indeed, the wave generator can be called a planet carrier because it support the rolling

bodies having movable axes: in the slots of the separator bodies reciprocate and rotate around their own axis. Thus, the Willis method is applicable to the present transmission [7]. This method based on the motion conversion (method of planet carrier stopping in the mind): all links of the mechanism are mentally given opposite rotation with angular speed equal to the angular speed of the planet carrier. Let us take wave generator with eccentric as planet carrier, rolling bodies with separator – link 2, profile gear – link 3. Since the separator is fixed, its angular velocity in the non-rotated mechanism will be equal to zero. We will draw up a table of links angular velocity for reverse and non-reverse transmission.

Table 1. Angular velocity of the transmission links

Mechanism	H	2	3
non-reverse	ω_H	0	ω_3
reverse	0	$-\omega_H$	$\omega_3-\omega_H$

Let us determine the transmission ratio from the third link to the second link in the reverse mechanism:

$$i_{32}^{(H)} = \frac{\omega_3-\omega_H}{-\omega_H}, \tag{1}$$

By simplifying the expression (1), we get:

$$i_{32}^{(H)} = \frac{Z_2}{Z_3}. \tag{2}$$

Transmission ratio from carrier H to link 3:

$$i_{H3} = \frac{1}{1-i_{32}^{(H)}}. \tag{3}$$

Substituting (2) in (3) and converting, we get:

$$i_{H3} = \frac{Z_3}{Z_3-Z_2}. \tag{4}$$

Equation (4) defines transmission ratio of cam coaxial radial flat transmission with free rolling bodies. It can be seen from the equation that the smaller the difference between rolling bodies number and the teeth number of the gear, the higher the gear ratio. The maximum transmission ratio can be obtained with the difference Z_2 and Z_3 per unity.

Maximum transmission ratio can be obtained in two cases:

$$Z_3 > Z_2 \text{ or } Z_3 < Z_2, \tag{5}$$

Then (4) will take the form:

$$i_{H3} = Z_3 \text{ or } i_{H3} = -Z_3. \tag{6}$$

It can be seen from the expressions (6) that the rotation can be organized unidirectional or different-directional for input and output shafts in the transmission.

Let us get the equation of the gear tooth profile. For this purpose we will replace of the mechanism with higher kinematic pairs by the mechanism with lower pairs [8]. Then the task of the transmission kinematic investigation can be reduced to the corresponding equivalent axial slider-crank mechanism investigation [7] (Fig.3).
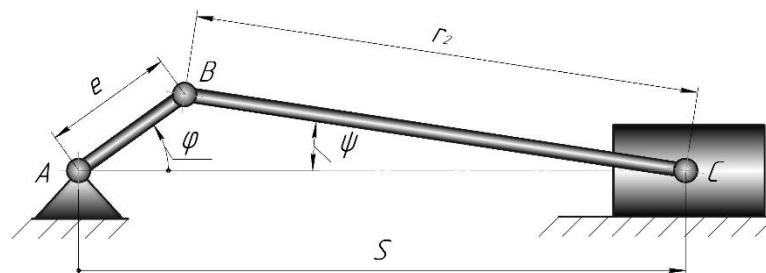

Figure 3. Replacement axial slider-crank mechanism

The input eccentric generator acts as the driving crank (Fig.3). Driving crank length is equal to generator eccentricity e. Connecting rod presents like line is connecting centers of rolling body and eccentric. Let us name the length of the connecting rod as the radius of the rolling body centers. Distance S (Figure 3) can be written in vector form:

$$\vec{S} = \vec{e} + \vec{r_2} \tag{7}$$

Let us define the angle using the sine theorem

$$\frac{r_2}{\sin\varphi} = \frac{e}{\sin\psi}. \tag{8}$$

or

$$\psi = \arcsin\left(\frac{e}{r_2}\sin\varphi\right). \tag{9}$$

The displacement $\vec{S}$ formula in scalar form is written as:

$$S = r_2\cos\psi + e\cos\varphi. \tag{10}$$

where e – eccentricity of the transmission, value of which is recommended to be determined depending on radius of rolling body [9].

The expression (10) determines the position of the center of the rolling body relative to the transmission axis in the fixed coordinate system. It is the main geometric expression from which all another geometric equations are derived [9]. The equations of the centers of the rolling bodies, relative to the center of the gear, using the transition matrix [10] will take the form:

$$\begin{cases} X_2 = S \cdot \sin\beta; \\ Y_2 = S \cdot \cos\beta. \end{cases} \tag{11}$$

where β – rotation angle of circle of centers of rolling bodies relative to generator.

This angle is related to the generator rotation angle by the following constraint:

$$\beta = \frac{\varphi}{i_{H3}}. \tag{12}$$

Let us define the theoretical profile of the gear that point K of the rolling body surface will draw (Figure 4).

Figure 4. To define normal at contact point of rolling body and tooth profile

The vector $\vec{n}$ is normal of tangent to the gear profile at the point of contact with the rolling body. In addition, it is just as normal to the relative velocity vector $\vec{V}^{12}$. In order to determine the normal vector $\vec{n}$, we will decompose the velocity vector into components: radial $\vec{V}^1$, and tangential $\vec{V}^2$. The vector $\vec{V}^1$ of radial velocity can be determined by derivative (10) over time. The radial component of the relative velocity can then be determined as:

$$V^1 = \frac{\partial S}{\partial t} = -r_2 \sin\psi \frac{\partial\psi}{\partial t} - e \sin\varphi \frac{\partial\varphi}{\partial t}. \tag{13}$$

Differentiating by time (8) and setting the result to (13), we get:

$$V^1 = \frac{\left(-r_2 \sin \psi \frac{e}{r_2} \cos \varphi\right)\omega}{\cos \psi} - e \sin \varphi \cdot \omega. \tag{14}$$

The expression (14) is simplified, let us present it as an analogue (i.e., change by angular velocity), we obtain:

$$V^1 = -e \operatorname{tg} \psi \cos \varphi - e \sin \varphi. \tag{15}$$

Tangential component of relative speed $\vec{V}^2$ is determined through distance S (Figure 4) and angular velocity of input link. So as an analogue:

$$V^2 = \frac{S}{i_{H3}}. \tag{16}$$

Let us determine pressure angle γ [8] through relative velocity components:

$$\gamma = \operatorname{arctg} \frac{V^1}{V^2}. \tag{17}$$

The coordinates of the point K are defined through the radius-vector R, which can be found from the expression:

$$R = Sj + r = Sj + (r_{tk} \sin \gamma)i - (r_{tk} \cos \gamma)j, \tag{18}$$

where j – unit vector of axis Y;

i – unit vector of axis X.

By simplifying the expression (18) and bringing it into matrix form, we get:

$$R = \begin{pmatrix} r_{tk} \sin \gamma \\ S - r_{tk} \cos \gamma \\ 1 \end{pmatrix}. \tag{19}$$

The expression (19) is the coordinates of the point K in the fixed coordinate system. With the help of transfer matrix [10] we pass to coordinate system rigidly connected to rotating profile gear. Then the gear profile equation will take the form:

$$\begin{cases} X_3 = r_{tk} \sin(\gamma - \beta) + S \sin \beta \\ Y_3 = -r_{tk} \cos(\gamma - \beta) + S \cos \beta \end{cases}. \tag{20}$$

Equations system (20) defines parametric equation of gear profile in fixed coordinate system.

Equations of gear profiles at rotation opposite direction of output link (6) taking into account that sine of angle is even function, and cosine is odd function [11], will take the form:

$$\begin{cases} X_3 = -(r_{tk} \sin(\gamma - \beta) + S \sin \beta) \\ Y_3 = -r_{tk} \cos(\gamma - \beta) + S \cos \beta \end{cases}. \tag{21}$$
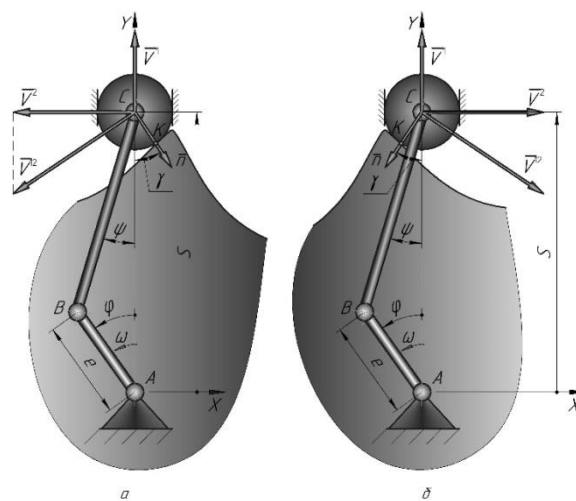

Figure 5. Direction of the output transmission link rotation: a - differently directed rotation; b - unidirectional rotation.

It can be seen from the equations (20) and (21), when the direction rotation of the output link is changed, the coordinate Y of the gear profile remains unchanged and the coordinate X changes its sign to the opposite. At the same time rolling bodies will transmit load either right part of the profile or left part (Figure 5). Using the equations (20) and (21) it is possible to manufacture profile wheels by various methods [12, 13].

3. Development of the mechatronic module electronic part

As in the set BLDC motor there are no built-in current sensors, for control of speed of motor shaft the AS5147P sensor is used. It has 4096 values on one turn of a shaft that gives definition accuracy less than 6 minutes. This sensor is powered from 3.3 V and issues data on a shaft turning angle on the serial peripheral interface (SPI). For definition of a turning angle of a reducer shaft, the sensor of absolute situation is installed. It transfers data by industry standard RS422 to signal of universal asynchronous receiver-transmitter (UART) is transformed for processing by the microcontroller. For motor management the driver of STM L6235Q was selected, management of which requires 7 exits (3 PWM and 4 digital) and maintaining voltage to 52 V and current of 5.6 A. The industry standard is used RS485 for the set of the necessary rotational speed of the motor and a turning angle of the reducer shaft. The standard is transformed to the UART interface for interaction with the controller. The controller of STM STM32F407GT was selected as a result of the choice of these devices, as the managing microcontroller. The described system provides:

- data processing from the user and from the sensor of absolute position (two hardware UART interfaces);
- motor management (through 4 digital and 3 hardware PWM exit);
- data processing from the motor position sensor (the hardware SPI);
- operation frequency of the microcontroller – 64MHz;
- compact sizes.

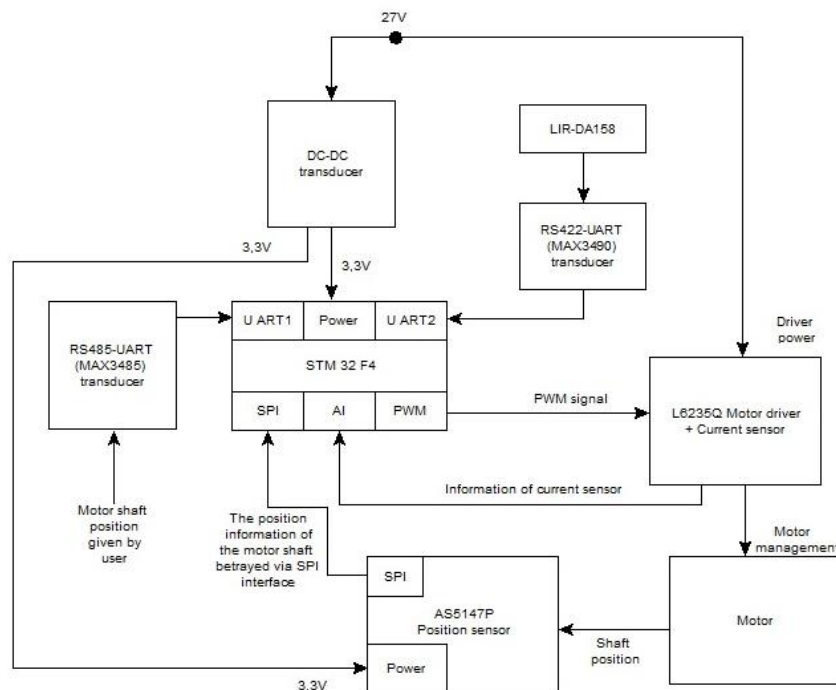The block diagram of the mechatronic module (figure 6) is a result of development.

Figure 6. Structural diagram of the drive control system

To adjust the drive, a mathematical model BLDC motor was taken, it presented in [14]. During initial adjustment of proportional integral derivative (PID) controller the following factors were obtained: P = 0.33678, I = 37.5306, D = 0.0007489.

Since it is very difficult to establish a clear functional relationship between the indicators of the quality of the transition process and the coefficients of the PID controller, the authors preferred the use of artificial intelligence methods, which showed their validity both in the configuration of the PID controller parameters [15] and in the management of complex objects [16].

4. Example

On the basis of the data from the technical assignment, transmission parameters with intermediate rolling bodies were selected, the dimensions of which are located in the hollow part of the rotor (Fig.1). Transmission parameters are given in Table 2, the diagram of reduction gear box assembly is shown in Figure 7.

Table 2. Angular velocity of the transmission links

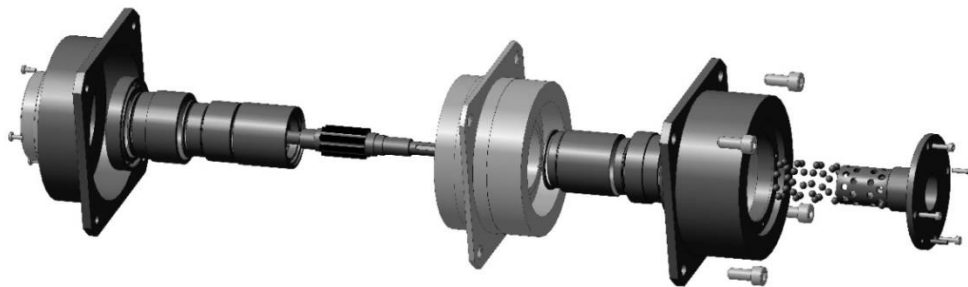| Parameter name	Designation	Value
Number of rolling bodies	Z_2	8
Rows number of rolling bodies	–	4
Radius of the rolling body centres, mm	r_2	7.925
Rolling body radius, mm	r_{tk}	1,6
Eccentricity, mm	e	0,8
Internal radius of generator, mm	–	9.525
External radius of generator, mm	–	14
One row width of rolling bodies, mm	–	4,3

Figure 7. Assembly diagram of mechatronic module

Results of stress-strain state mathematical simulation are shown in Figure 8 for separator and the gear. For the most loaded parts of the reducer simulation was carried out in the SolidWorks system with maximum torque at 1.5 Nm.
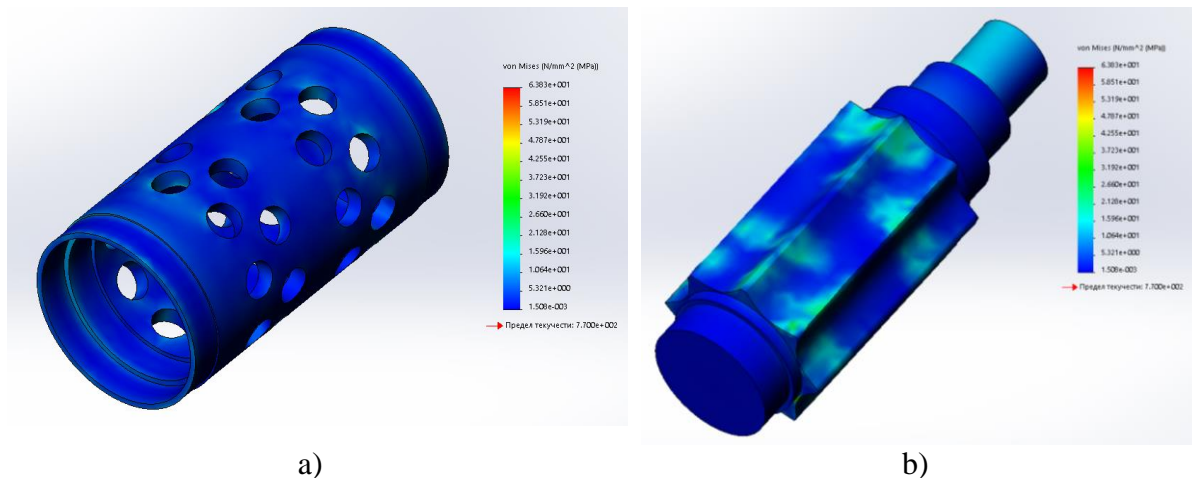
a)   b)

Figure 8. Stress distribution in parts of reducer: a - separator; b - gear profile

5. Conclusions

On the basis of the electric motor of 3 DBM 70-0.16-3-3 with dimensions of 70x81x81 mm and weighing about 900 grams, the mechatronic module is developed.

It was possible to reduce overall dimensions of mechatronic module due to installation of coaxial reducer with intermediate rolling bodies into cavity of electric motor rotor.

Transmission ratio of cam coaxial radial flat transmission with IRB is determined and equation of profile is derived for the gear.

Intelligent control system of electric drive is proposed which to protect the module from overheating and optimize rotation speed of motor shaft by means of using PID controller with self-tuning function based on genetic algorithms.

Acknowledgments

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