OPTIMIZATION OF THE PARAMETERS OF THE MECHANICAL GEAR IN MEDICAL CENTRIFUGE (CHAIR OF BARANY)

Ts. Kachamachkov, D. Bakardzhiev, Hr. Bakardzhiev, V. Manoev

Technical University of Sofia, Bulgaria,

E-mails: eng_kachamamachkov@abv.bg; dim_bak@abv.bg; vmanoev@tu-sofia.bg

Abstract

We researched the influence of the mass, diameter and the structure of the mechanics on the moment of inertia, the mechanical time constant and the coefficient of efficiency for a class of the mechanisms. We offer, a specific solutions to optimize the mechanical units in medical centrifuge (chair of Barany) by these parameters: mechanical time constant, volume, weight, power, efficiency. Studies have been conducted for several mechanisms in order to find the optimal structure for the chair of Brany.

***Keywords**: mechanism, mechanical units, efficiency, mechanical time constant, volume, weight, power, chair of Barany.*

INTRODUCTION

The subject of the research in this work are the mechanical parts that make the mechanism of the rotating chair for studies of the vestibular system [1], [2]. Typical of it is that you have to provide a trapezoidal speed chart and sinusoidal oscillation, pendulous effect and stop impetus.

Representation

The chair is designed for responsible research in the vestibular system for civilian patients and pilots the electromechanical part is shown on Figure 1. It must provide the necessary conditions for the tests: The values of accelerations are strictly defined 0,06; 0,2; 0,8; 1,0; 3,0; 6,0; 9,0; 10,0; 15,0; 20,0; ${}^{o}/_{s^2}$, or 0,001; 0,0035; 0,014; 0,0174; 0,1047; 0,157; 0,174; 0,262; 0,35 ${}^{rad}/_{s^2}$.

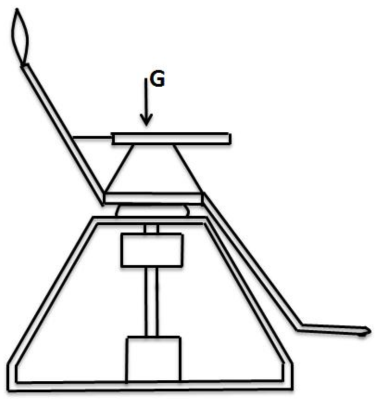


Fig. 1. Chair of Barany

The mechanical part comprises from:

1 - chair; 2 - electric motor; 3 - reducer;

For the sake of high efficiency we offer the following structural changes: standard, gearbox and engine are replaced by specifically designed for chair of Barany. Their production is justified by the wide-spread obtained by the chair of Barany over the past century. The mechanical part incorporates all interrelated moving masses or with other words engine, gearbox and actuator shown in Fig. 2. Motion is implemented by the engine Д, reducer КРП, perifleks coupling CM1, propels the mass m1 and patient with mass m2. Loading Actuator moves cargo m1 + m2.[3]

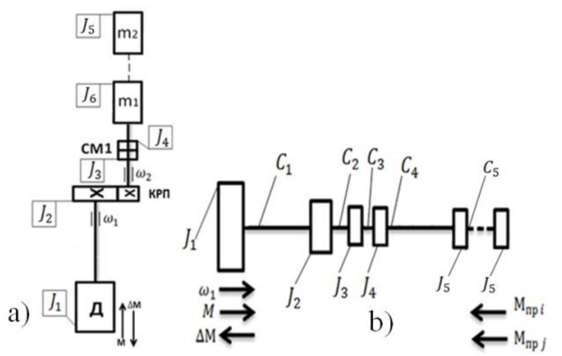


Fig. 2. Schematic diagram of the mechanical part a) and defined kinematic chain b) of medical centrifuge.

On the diagram in Fig. 2 with arrows are shown the applied to the individual mases in the system, adduced moments of the active in the system external forces $M_{\text{пр}\ i}$ and $\mathrm{M}_{\text{пр}\ j}$. To the rotor of the engine J_1 is applied the electromagnetic moment of the engine M and the moment of the mechanical es ΔM, to properly calculate we assumed to be positive sign the direction of the angular velocity ω_1. In simplifying of the scheme it is necessary to calculate all externally applied forces on the masses which are connected by means of solid bodies. The study of the dynamics of the electric drive shows that the direct calculation of mechanical scheme in most cases gives the same output as the detailed calculation of the individual forming mechanism. Therefore we identify the main masses and stiffness and bring it down to a two mass system shown on fig. 3, where the system is broth down to the unit with the lowest stiffness and result inertial momentum.[4]

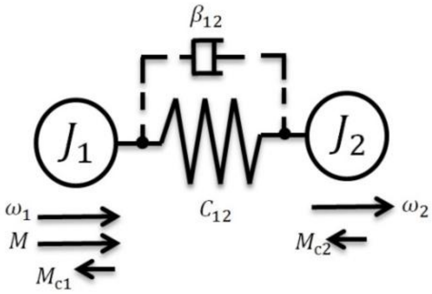


Fig. 3. Summarized two - mass scheme for medical centrifuge

Parameters of two – mass flexible mechanical system (fig. 3), are reduced to the total moment of inertia moments $J_1\ and\ J_2$, directed to the mechanical connection between them. We accept the moment of elasticity which is brought to the stiffness of a mechanical link J_1 and $J_2 - C_{12}$. The first mass represents the rotor of the engine himself and the mechanical components which have direct contact with him. To this we add the applied electromagnetic moment of the engine M and the moment of static load $\mathrm{M_{c1}}$. To the intermediate masses in the mechanism J_2 is applied the resistance moment M_c. In calculating the above-quoted static moment M_C, when all active forces and momentums in the mechanism are defined. In most cases, losses from friction in the mechanism are unknown and are calculated with the use of the efficiency coefficient of the mechanism. [4]

$$\eta_{\text{мех}} = \eta_1\eta_2\eta_3 \dots, \qquad (1),[6]$$

Where $\eta_1, \eta_2, \eta_3 -$ are efficiency coefficients of the units in the kinematic chain.

If the moment $\mathrm{M_{MEX}}$ of the load in the mechanism is positive, the moment of the static load is detriment from the equations. [4],[5]

$$\mathrm{M_C}\omega_1 = \mathrm{M_{мех}}\omega_{\text{мех}}/\eta_{\text{мех}} + \Delta\mathrm{M}\omega_1 \qquad (2)$$

Therefore

$$\mathrm{M_C} = \mathrm{M_{мех}}/i_0\eta_{\text{мех}} + \Delta M, \qquad (3)$$

Where $\Delta M -$ is the moment of mechanical losses in the engine.

$i_0 = \frac{\omega_1}{\omega_{\text{MEX}}} = i_1 i_2 i_3 \dots -$ overall gear ratio from the motor to the actuating device.

The equation of the power capacity can be written in the following way, thanks to the efficiency - in the system.

$$\mathrm{M_C}\omega_1 = M_{\text{мех}}\omega_{\text{мех}}\eta_{\text{мех}} - \Delta\mathrm{M}\omega_1 \qquad (4)$$

In this case:

$$\mathrm{M_C} = (\mathrm{M_{мех}}/i_0)\eta_{\text{мех}} - \Delta M. \qquad (5)$$

The moment caused by mechanical losses in the engine isn't bigger then 1-5 % of the rated torque of the engine. In this case we assume ΔM≈0, and $\mathrm{M_{мех}}$ becomes:

$$\mathrm{M_C} = \mathrm{M_{мех}}/i_0\eta_{\text{мех}}; \qquad (6)$$

For motor rotating in the opposite direction:

$$\mathrm{M_C} = (\mathrm{M_{мех}}/i_0)\eta_{\text{мех}} \qquad (7)$$

When $\Delta\mathrm{M} = 0$, the equation can be written:

$$M_C\omega_1 = F_{\text{мех}}v_{\text{мех}}/\eta_{\text{мех}} \qquad (8)$$

From where:

$$\mathrm{M_C} = (F_{\text{мех}}/\eta_{\text{мех}})\mathrm{p} \qquad (9)$$

So for motor rotating in the opposite direction:

$$\mathrm{M_C} = F_{\text{мех}}\mathrm{p}\eta_{\text{мех}} \qquad (10)$$

From the kinematic chain in fig. 4 we can write the current example the three most - significant masses are given the rotor, the engine with inertia moment $J_{\text{дв}}$ and load J_c.

$$1/C_{e\text{кв}} = 1/C_1 + 1/C_2 + 1/C_3 + \dots \ (11)[7]$$

The behavior of the system depends on the variable parameter T_M to get the required quality of the transition process. [1] The algorithm taking into account the variable mechanical time constant. Is calculated and explained in [1].

Electromechanical time constant is:

$$T_M = \frac{J_{\Sigma.} r_e}{(с\Phi)^2}; = с\Phi = \frac{М_Н}{I_Н}; \quad (12)$$

$$J_{\Sigma.} = J_{дв} + J_P + J_{CM1} + \frac{J_C}{\eta_{мех}\, i_p^2}; \quad (13)$$

$$J_C = \frac{GD^2}{4g}; J_P = 0{,}2J_{дв}; J_{\Sigma.} = 1{,}2J_{дв} + J_{CM1} + \frac{GD^2}{4g\eta_{мех}\, i_p^2}; G = G_C + G_П; \quad (14)$$

Where:

$J_{дв}$ – motor inertial moment,

J_{CM1} – inertial moment, perifleks coupler.

J_P – inertial moment of the profiled designed gear,

J_c – inertial moment of the load,

i_p – ratio of the gearbox,

η_{MEX} – efficiency,

In this current example it is shown the parameters of the gear of medical centrifuge used in the of Barany.

Reducer used at the moment.

- transmission ratio i = 63
- rotational speed of the input n_1=1512 min^{-1}
- rotational speed of the output shaft $n_n = 24\, min^{-1}$
- transmitted rated power (output) P = 400 W
- the number of teeth of the worm wheel Z_2 = 63
- input torque $T_1 = 3{,}7\, N.m$
- output torque $T_2 = 119{,}4\, N.m$
- the axle spread a = 52 mm
- the functional length of the worm wheel $b_1 = 62\, mm$
- superlative diameter $d_2 = 82\, mm$
- width of the worm wheel $b_2 = 25{,}2\, mm$
- the distance between the bearings and the worm shaft l1 = 157 mm
- the distance between the bearings l_2 = 108 mm

Reducer designed for chair of Barany

- transmission ratio i = 88
- rotational speed of the input n_1= 3000 min^{-1}
- rotational speed of the output shaft $n_n = 34\, min^{-1}$
- transmitted rated power (output) P= 270 W
- the number of teeth of the worm wheel Z_2 = 88 бр.
- input torque $T_1 = 1{,}3\, N.m$
- output torque $T_2 = 76\, N.m$
- the axle spread a = 10 mm
- the functional length of the worm wheel $b_1 = 65\, mm$
- superlative diameter $d_2 = 36\, mm$
- width of the worm wheel $b_2 = 8{,}5\, mm$
- the distance between the bearings and the worm shaft l_1 = 40 mm
- the distance between the bearings l_2 = 64 mm

Determine the required power of the electric motor

$$P_{ел} = \frac{P_{изх}}{\eta_{мех}} \quad (15)[7]$$

Where:

$P_{изх}$ – Output power

$\eta_{мех}$ – Efficiency of the system

$P_{ел}$ – rated power of the electric motor.

$$d = m.q \quad (16)$$

Where:

m – Modules of engagement of the worm gearing.

q – Coefficient of the diameter of the worm.

$$d_2 = \sqrt[3]{\frac{T_2}{0{,}2.\tau_{доп}}} \quad (17)$$

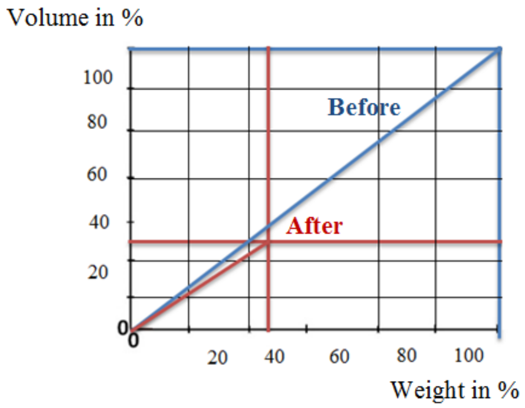
Where:

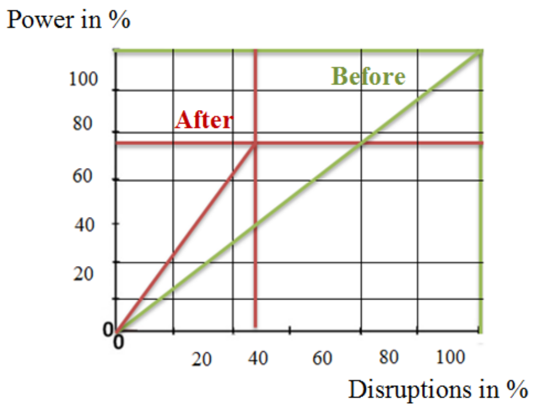
$\tau_{доп.ус}$ – is allowable tension twist.

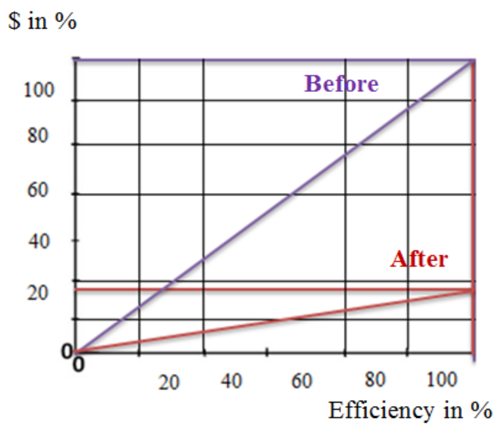
T_2 – output torque.

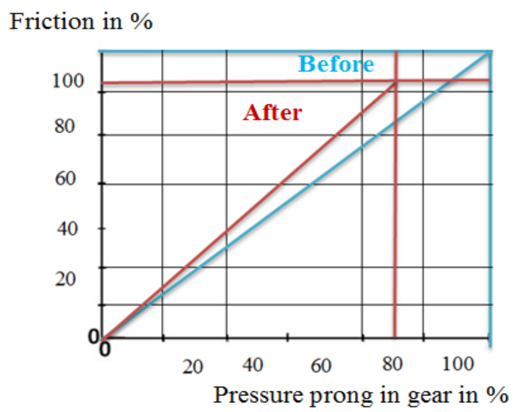
d – the diameter of the outlet end of the worm wheel shaft.

Comparative analysis on parameters in % of change from previous to current.









CONCLUSION

In applying the designed gearbox, friction in the system is reduced the volume of the used gear is reduced, weight decreased, the power required to drive also falls, the money required for production drops.

References

[1] Bakardjiev D., B. C. Bakardjiev, D. Angov M. Zhilevski, management kupolometar to study the vestibular system, Proceedings of the Technical University of Rousse, Scientific and technical conference with international participation, 2005.

[2] Bakardjiev D., Zhilevski M. Bakardjiev Hr., Zhilevska M. Kochevska V. Zhilevski M., Management swivel chair for Medical Research Generation IV, Science University of Rousse Volume 48 3.1 series, 2009.

[3] Theory of electric VI key, Moscow Energoatomisdat 2001 BBK 31,191, K 52, 62-83 UDC (075.8)

[4] Теория электропривода В. И. Ключев, Москва энергоатомиздат 2001 г. ББК 31.191, К 52, УДК 62-83(075.8)

[5] The total electric rate, M. G. Chilikin, A. S. Sandler, Moscow Energoizdat 1981.

[6] Machine Parts, G. Angelov, fourth edition, CI,, TECHNOLOGY "Sofia - 1964

[7] Tasks and course design theory of mechanisms and machines, Eng. Tcanko K. Nedev, Eng. Alexander G. Vrigazov, Eng. Mihail I. Milkov State Publishing House,, Technique "Sofia 1963.