DESIGN AND INVESTIGATION OF INTERMITTENT MOTION PLANETARY MECHANISMS WITH ELLIPTICAL GEARS

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Abstract

Machines with intermittent movement of working body are widely used in industry. Currently, mechanisms with unilateral constraints or variable structures are used as actuators of such machines. Output link stops in most mechanisms are provided by periodic rupture of the kinematic connection between the links. This disadvantage limits the use of these mechanisms in high speed machines, since impacts occur at the end or beginning of the motion phase. So, there is a relevant task to create, analyze and effectively put into practice intermittent motion mechanisms (IMMs) in which during operation the kinematic constraint between the links is not broken. The author proposes and analyzes planetary trains, which include modified elliptical wheels. The variable transfer function of a non-circular wheel pair and certain dimensions of the mechanism links make it possible to obtain required motion function. The kinematics of the proposed IMMs is analyzed, as a result of which the functions of the angle of rotation and the analogue of the output shaft angular velocity are found and constructed. Created mechanisms due to the use of gears perform reliability and compactness with the possibility of transferring great forces, and they can find application in metalworking machinery, automatic lines, robotics, transporters.

Keywords: Rotational motion, Intermittent motion, Elliptical gearwheels, Planetary mechanism, Kinematic analysis, Angular velocity

I. Introduction

In mechanical engineering, and first of all, in technological equipment of automatic and semi-automatic action, as a rule, there are used intermittent mechanisms [XXV], [III], [IV], [VIII], [XXXI], [II]. The most effective practical application they received as a drive of plotters, watches, indexable equipment, rotary tables, stepper conveyors, as well as to communicate movement with stops to the output links of the executive and auxiliary mechanisms.
In the IMM, the rotational movement of the drive (input link) is converted into motion with stops of the working body of the machine (output link). Currently, two types of mechanisms are used in industry:

- permanent structure mechanisms with unilateral constraints (anchor mechanisms, ratches);
- variable structure mechanisms (Geneva drives, starwheels, mutilated gears).

Today the Geneva drives are extensively used as a part of the drive systems in automatic machinery (the turrets of turret lathes, screw machines, turret drills; the tool changers; rotary tables and dividing heads) or conveyors. In [XXIX], it is proposed to combine the Maltese mechanism with linkages, gears and cams to reduce the shock load. Another method to improve kinematics and dynamics of the Maltese mechanism is based on adding curved slots to the mechanism diagram as reported in [IX], [XIII]. However, despite the synthesis and analysis of new constructions, in most mechanical units with intermittent motion is achieved by periodic rupture of the kinematic connection between the links. This leads to impacts at the beginning and end of the motion phase [XII], therefore, the described mechanisms cannot be used as part of high-speed vehicles. Thus, there is an urgent task of developing intermittent mechanisms devoid of the above disadvantages. Recently, a large amount of scientific papers have been published on the analysis and practical implementation of mechanisms with non-circular gears (NCGs) [XXXIV]. For example, there are proposed: chain drives with variable ratio for bicycles [X]; a mechanical device containing a Maltese cross mechanism and a gearing to achieve intermittent motion [VII]; the gear trains used for velocity variation and function generation [XVII]; constructions of NCGs as a part of rotor hydraulic machines [I] and high-performance pumps [XVIII]; a knee motion assist device for biomechatronic exoskeleton having grooved cams and non-circular wheels [XXX]; single planetary NCG with one internal and one external gear for bicycles with high efficiency [XIX]. NCGs are also used to decrease velocity and force oscillations in rotating shafts [VI], to compensate shaking torques in spatial mechanisms [XI]. The various use of non-circular gears is explained by the significant advances in their engineering and manufacturing [XXXIII]. So, one of the most perspective mechanisms that allow converting rotational motion of the input link into the required motion type of output link are non-circular planetary gear trains [XXXII], [XXIII],[XVIII].

II. Structural Synthesis of Planetary IMM

Analysis and synthesis of machines traditionally suggests that mechanisms exist only in two types of spaces: the three-dimensional 3-mobility space ("planar" mechanisms) and three-dimensional 6-mobility space ("spatial" mechanisms). Such a limited choice of spaces leads to the fact that the structural formulas of Chebyshev, Somov-Malyshev and other authors often give incorrect results [XXVI]. In [XXVI, XXVII], it was shown that mechanisms can exist in spaces of different dimension and mobility, and based on this, there was developed following equation for determining the mobility of simple mechanisms:

\[ \text{Mobility} = \text{Dimension} \times \text{Mobility} \]
where $W$ is the mechanism mobility (DOF); $p_i$ is the amount of kinematic pairs of the $i$-th mobility (DOF); $k$ is the amount of independent closed circuits; $\Pi$ is the mobility of the space in which there is a mechanism.

The amount of independent closed circuits is determined by the equation [XXVI]:

$$k = p - n$$

where $p = \sum_{i=1}^{n-1} p_i$ is the total amount of mechanism kinematic pairs, $n$ is the total amount of movable links.

To construct a structural mathematical model of mechanisms and machines, the notion of a $t$-vertex link was introduced in [XXVI]. The amount of vertices in the mechanism link is determined not by the amount of kinematic pairs (the link does not have them), but by the amount of elements of the kinematic pairs, by which it is connected to other links and the rack. If the link contains two elements of kinematic pairs ($t = 2$), it is two-vertex; three ($t = 3$) – a three-vertex link; four ($t = 4$) – four-vertex, etc. In addition, [XXVI] also introduced the notion of the number $S$ of elements of kinematic pairs, by means of which the kinematic chains of the mechanism are attached to racks.

Taking as the base link in the analyzed mechanism the vertex $T$ with the largest amount of elements of kinematic pairs, the total amount of kinematic pairs $p$ (the amount of elements of kinematic pairs equals twice the amount of kinematic pairs) in the mechanism is determined by the equation:

$$p = \frac{1}{2} \left( \sum_{i=t-j}^{2} tn_i + S \right)$$

where $T$ is the amount of vertices of the base link; $t$ is the amount of vertices of the links; $n_i$ is the amount of movable links with $t$ vertices, $j$ is an integer index.

Independent closed loops in mechanisms are formed only if its kinematic chains contain two or more vertex links ($t \geq 2$). Then the basic link of the mechanism, which has $k$ independent closed loops, can have $T$ vertices:

$$T \leq k + 1$$

Combining the obtained equations (1)-(4), we have a structural mathematical model of mechanisms with closed kinematic chains [XXVI]:

$$W = \sum_{i=1}^{n-1} p_i - k\Pi$$
Let’s carry out the synthesis of IMMs structural schemes under the following initial conditions [XXVIII]. The mechanism exists in a three-motion space \((\Pi = 3)\), has two closed contours \((k = 2)\), a three-vertex base link \((T = 3)\), one \(- (p_1)\) and two \(- (p_2)\) kinematic pairs. The structural mathematical model (5) after substitution of the initial synthesis conditions into it will take the form:

\[
\begin{align*}
 p &= \frac{1}{2} \left( \sum_{i=T-j}^{2} n_i + S \right); \\
 n &= \sum_{i=T-j}^{2} n_i; \\
 W &= \sum_{i=1}^{[i-1]} ip_i - k\Pi; \\
 k &= p - n; \\
 p &= \sum_{i=1}^{[i-1]} p_i; \\
 T &\leq k + 1.
\end{align*}
\]

The integer roots of model (6) are the following values:

\[
\begin{align*}
 p_1 &= 1, p_2 = 3, p = 4, n_1 = 1, n_2 = 1, n = 2, S = 3; \\
 p_1 &= 1, p_2 = 3, p = 4, n_3 = 0, n_2 = 2, n = 2, S = 2; \\
 p_1 &= 3, p_2 = 2, p = 5, n_2 = 2, n_1 = 1, n = 3, S = 3; \\
 p_1 &= 3, p_2 = 2, p = 5, n_1 = 1, n_2 = 2, n = 3, S = 2; \\
 p_1 &= 3, p_2 = 2, p = 5, n_2 = 0, n_3 = 3, n = 3, S = 1; \\
 p_1 &= 5, p_2 = 1, p = 6, n_2 = 2, n_3 = 1, n = 4, S = 3; \\
 p_1 &= 5, p_2 = 1, p = 6, n_3 = 1, n_2 = 3, n = 4, S = 2; \\
 p_1 &= 5, p_2 = 1, p = 6, n_1 = 1, n = 4, S = 2. \\
\end{align*}
\]

After analyzing the decisions (7) – (14) it is proposed to synthesize the mechanism schemes according to the solution (9), which corresponds to a double-row planetary gear with external gearing (Fig. 1) [XXVIII]. The mechanism scheme contains three 1-DOF kinematic pairs \((A, C, E)\) and two 2-DOF kinematic pairs \((B, D)\); one three-vertices link (link 2) and two links having two vertices (links 1, 3);
three connections to the rack 0 ($S = 3$). The kinematic analysis showed the possibility of obtaining a reciprocating-rotational movement of the output shaft. This mechanism has been effectively applied as a stirred tank actuator [XXIX], [XXI], [XXII], [XXIII]. Subsequently, a dynamic analysis was carried out, which made it possible to obtain and analyze the equations of motion of the actuator links [XXI].

Fig. 1. Planetary mechanism with elliptical gears.

The proposed mechanism [XXVIII] under certain size of spur gears (Fig. 2) performs the movement with stops [XXX]. There was performed a mechanism kinematics [XXIV], as a result we showed that mechanism allows to realize the intermittent motion of the output link during rotational motion with a constant angular velocity of the input link. This is achieved by the fact that the point $D$ in velocity plan will not cross the zero line.

Fig. 2. Intermittent motion planetary mechanism.
However, the mechanism shown in Fig. 2 is difficult for balancing, as elliptical gears centers of masses are not on the axis of rotation. To correct this disadvantage it is proposed to use a pair of modified elliptical gears [XXXII] with the axis of rotation at the center of symmetry (Fig 3).

The modified elliptical gear centrode is represented by the equation:

\[ \rho(\varphi) = \frac{b}{\sqrt{1 - e^2 \cos^2 \varphi}} \]  

(15)

where \( b \) is the semi-minor axis of the elliptical gears, \( e \) is the eccentricity, \( \varphi \) is rotation angle of the drive gear.

Mechanism having the pair of modified elliptical gears is shown on Fig. 4. The radii of the central (\( R_1 \)) and peripheral (\( R_2 \)) spur gears (Fig. 4a) are determined as:

\[ R_1 = a; \]  

(16)

\[ R_2 = b, \]  

(15)

where \( a \) and \( b \) are the semiaxes of the elliptical gears.
Fig. 4. Intermittent motion planetary mechanism with modified elliptical gears: a – kinematic scheme; b – construction.

Planetary mechanism (Fig. 4 b) consists of an input shaft 1, a carrier 2, an output shaft 3, a sun spur stationary wheel 4, an elliptical gear 5, a satellite, which consists of a spur planet gear 6, an elliptical planet gear 7 and a shaft 8. The mechanism has three connections to the rack 0: through two rotational kinematic pairs and one gear kinematic pair.

The input shaft 1 performs rotational motion, which is transmitted to the carrier 2, thereby the spur gear 6 rotates around a sun wheel 4. The rotational movement of the spur gear 6 transmitted to the satellite shaft 8 and the elliptical gear 7 which drives the elliptical gear 5 and the output shaft 3 respectively. At moment when the gear ratio of elliptical gears pair is equal to the gear ratio of cylindrical gears pair, the output shaft 3 is stopped. Further, the output shaft angular velocity increases to a maximum value and then decreases again to zero. This provides the intermittent movement of the output link.

III. Kinematics of Planetary Gear Train

Let’s study the kinematics of the mechanism (Fig. 4) using the construction and analysis of the plan of linear velocities of its links (Fig. 5) [XIV].

The vector CC′ from the point C lying at the same level as the point C in the mechanism diagram shows the velocity of the point C of a carrier. Connecting the point C′ and the point A that corresponds to the stationary point A on the carrier axis gives the line C′A showing the linear velocity distribution of the carrier CA. For a satellite, there are known the velocities of two points: that of the point C, which is common for a carrier and a satellite, and the point B, the velocity of which is equal to zero according to the condition of rolling of the initial circle of a gear 6 on the initial circle of a gear 4. Point B corresponds to point B in the velocity diagram. Connecting point B with point C′ gives the linear velocity distribution line of a satellite. Point D′ lying on this line is the end of vector DD′, which shows the velocity of point D. This
point is common for the pair of elliptical gears 5 and 7. Hence, connecting point \( D' \) with point \( A \) gives the linear velocity distribution line of an elliptical gear 5 and, thus, that of a takeoff shaft 3.

According to Figure 5, the angular velocity analogue of the output shaft 3 is determined by:

\[
\varphi_3 = \frac{\omega_3}{\omega_1} = \frac{v_D \cdot AC}{v_C \cdot DE} = \frac{DD' \cdot AC}{CC' \cdot DE} = \frac{BD \cdot AC}{BC \cdot DE}
\]  

(18)

where \( \omega_3 \) is the output shaft 3 angular velocity; \( \omega_1 \) is the input shaft 1 angular velocity; \( BD, AC, BC, DE \) are distances from Figure 5.

The distances \( AC \) and \( BC \) in the equation (18) are defined as:

\[
AC = R_1 + R_2 = a + b
\]

(19)

\[
BC = R_2 = b
\]

(20)

As can be seen from Fig. 5, point \( D \) changes its position relative to point \( B \), at the same time, the velocity vector \( DD' \) changes its value. The length of the vector \( DD' \) decreases and when the points \( B \) and \( D \) coincide, the mechanism output shaft is stopped. To determine the segments \( BD \) and \( DE \), it is necessary to find the length of the segment \( CD \). We use the equation (15) of elliptical gear centrode:

\[
CD = \rho_7 = \frac{b}{\sqrt{1 - e^2 \cos^2 \varphi_7}}
\]

(21)

where \( \varphi_7 = \frac{R_1}{R_2} \cdot \varphi_1 + \frac{\pi}{2} \) is the elliptical gear 7 rotation angle.
Then, according to Fig. 5 and the equation (21), we define the segments $BD$ and $DE$:

\[ BD = BC - CD = b - \rho \gamma \]  
\[ DE = AC - CD = a + b - \rho \gamma \]  

Substituting (19) – (23) to (18), we obtain the equation for the output shaft $3$ velocity analogue:

\[ \dot{\varphi}_3 = \frac{(b - \rho \gamma) \cdot (a + b)}{b \cdot (a + b - \rho \gamma)} \]  

For further dynamic analysis of machines with a planetary IMM, in addition to the found angular velocity analogue of the output shaft $\dot{\varphi}_3$, it is necessary to determine the kinematic characteristics of the satellite:

- angular velocity analogue of the satellite:
  \[ \dot{\varphi}'_s = \frac{\omega_s}{\omega_1} \]  

- velocity analogue of the satellite center of mass – point $C$:
  \[ S'_C = \frac{\nu_C}{\omega_1} \]  

The velocity of point $C$ is determined as:

\[ \nu_C = \omega_1 (R_1 + R_2) = \omega_1 (a + b) \]  

Point $B$ is the instantaneous velocity center of the satellite (Fig. 5), then, using equation (27), we get:

\[ \omega_s = \frac{\nu_C}{BC} = \frac{\omega_1 (a + b)}{b} \]  

Substituting (28), (27) in (25), (26), we obtain:

\[ \dot{\varphi}'_s = \frac{a + b}{b} \]  
\[ S'_C = a + b \]  

Equations (24), (29), (30) allow us to study the kinematics of the mechanism, since the use of differentiation and integration operations will allow us to obtain functions of the acceleration analogue and the rotation angle, respectively.

The obtained expressions allow us to find the velocities of all the links and their centers of mass, which will allow to determine the reduced moment of inertia of the mechanism and then construct and investigate a single-mass dynamic model.
IV. Results and Discussion

Synthesized intermittent mechanism scheme has two gear pairs (Fig. 4 b): spur gears 4 and 6 with radii $R_1$ and $R_2$, and equal elliptical gears 5 and 7 with semi-axes $a$ and $b$. By exchanging gears 4 and 6, as well as the gear pairs 4, 6 and 5, 7, we get four kinematic schemes of IMMs (Fig. 6).

The study of kinematics will be carried out with the following parameters. In each planetary mechanism, the elliptical gearwheels have dimensions of semi-axes $a = 20$ mm and $b = 25$ mm, and the radii of spur gears $R_1$ and $R_2$ are calculated according to equations (16), (17).

![Image of kinematic schemes](image)

Fig. 6. Kinematic schemes of intermittent motion mechanisms: a, b Schemes with spur sun gear; c, d. Schemes with elliptical sun gear.

The equation (24) allows us to find the velocity analogue of output shaft for scheme 6a. By integrating (24) according to the generalized coordinate, we obtain a function of the output shaft rotation angle, which graph is shown in Fig. 7a.

Applying the proposed methods to different kinematic schemes (Fig. 6b – 6d), we also can determine segments $BD$, $AC$, $BC$, $DE$ in equation (18) and find...
angular velocities and rotation angle functions for these mechanisms. Graphs of functions \( \varphi_3(\varphi_1) \) for schemes 6b – 6d are shown in Fig. 7b – 7d, respectively.

As we can see from Fig. 7a, in a single turn of the input shaft 1 the output shaft 3 of mechanism 6a makes two stops. The amount of stops \( N \) per revolution is determined by the gear ratio of the first pair of gears:

\[
N = \frac{R_1}{R_2} \cdot 2
\]  

(31)

The mechanisms 6c and 6d have two stops for one turn of the input shaft because elliptical wheels are identical. The graphs \( \varphi_3(\varphi_1) \) in Fig. 7 also show that the input and output shafts in the mechanism 6b and 6c rotate in one direction and in the mechanisms 6a and 6d – in the opposite direction.

![Graphs of functions \( \varphi_3(\varphi_1) \)](https://via.placeholder.com/150)

Fig. 7. Graphs of functions \( \varphi_3(\varphi_1) \)
V. Conclusions

Using the structural mathematical model of mechanisms and machines, the author proposes new kinematic schemes of the 1-DOF planetary mechanisms with elliptical gears. To prove the provision of the output shaft intermittent motion, to conduct further dynamic studies and design, kinematic analysis was performed, as a result of which the velocity analogues of all links and their centers of mass were determined.

Since developed mechanisms are more compact, reliable and easy for balancing, in comparison with known converters of motion, they can be recommended for wide practical application. Furthermore, the intermittent motion is provided without rupture of the kinematic chain, which will increase the speed of automatic machines.

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References

I. AnI. K., “Synthesis, geometric and strength calculations of rotor hydromachines planetary mechanisms with non-circular gears”, Doctor thesis, Tomsk Polytechnic University, Tomsk, 2001

II. Artobolevskii I., “Mechanisms in modern engineering design”, MIR Press, Moscow, USSR, 1986.

III. Bickford J. H., “Mechanisms for intermittent motion”, Industrial Press Inc., New York, USA, 1972.

IV. Chang Z., Xu C., Pan T., Wang L., Zhang X., “A general framework for geometry design of indexing cam mechanism”, Mechanism and Machine Theory, Volume: 44, Pages: 2079 – 2084, 2009. https://doi.org/10.1016/j.mechmachtheory.2009.05.010

V. Coxeter H.S.M., “Introduction to Geometry”, Wiley Publishing House. New York, USA, 1969.

VI. Dooner D. B., Palermo A., Mundo D., “An intermittent motion mechanism incorporating a Geneva wheel and a gear train”, Transactions of the Canadian Society for Mechanical Engineering, Volume: 38(3), Pages: 359 – 372, 2014. https://doi.org/10.1139/tcsme-2014-0026
VII. Dooner D. B., “Use of noncircular gears to reduce torque and speed fluctuations in rotating shafts”, Journal of Mechanical Design, Volume: 119(2), Pages: 299 – 306, 1997. https://doi.org/10.1115/1.2826251

VIII. Figliolini G., Angeles J., “Synthesis of conjugate Geneva mechanisms with curved slots”, Mechanism and Machine Theory, Volume: 37, Pages: 1043 – 1061, 2002. https://doi.org/10.1016/S0094-114X(02)00062-9

IX. Figliolini G., Rea P., Angeles J., “The pure-rolling cam-equivalent of the Geneva mechanism”, Mechanism and Machine Theory, Volume: 41, Pages: 1320 – 1335, 2006. https://doi.org/10.1016/j.mechmachtheory.2006.01.002

X. Freudenstein F., Chen C. K., “Variable-ratio chain drives with noncircular sprockets and minimum slack—theory and application”, Journal of Mechanical Design, Volume: 113(3), Pages: 253 – 262, 1991. https://doi.org/10.1115/1.2912777

XI. Han J. Y., “Complete balancing of the shaking moment in spatial linkages by adding planar noncircular gears”, Archive of Applied Mechanics. Volume: 67(1-2), Pages: 44 – 49, 1997. https://doi.org/10.1007/BF00787138

XII. Kozhevnikov S. N., Esipenko Y. I., Raskin Y. M., “Mechanisms”, Mechanical Engineering Press, Moscow, USSR, 1976.

XIII. Lee J. J., Jan B. H., “Design of Geneva mechanisms with curved slots for non-undercutting manufacturing”, Mechanism and Machine Theory, Volume: 44, Pages: 1192 – 1200, 2009. https://doi.org/10.1016/j.mechmachtheory.2008.09.003

XIV. Levitskiy N. I., “Theory of mechanisms and machines”, Nauka Press, Moscow, USSR, 1979

XV. Lin C., Xia X., Li P., “Geometric design and kinematics analysis of coplanar double internal meshing non-circular planetary gear train”, Advances in Mechanical Engineering. Volume: 10(12), Pages 27 – 35, 2018. https://doi.org/10.1177/1687814018818910

XVI. Litvin F. L., Fuentes A., “Gear geometry and applied theory”, Cambridge University Press: Cambridge, UK, 2004.

XVII. Litvin F. L., Gonzalez-Perez I., Fuentes A., Hayasaka K., “Design and investigation of gear drives with non-circular gears applied for speed variation and generation of functions”, Computer Methods in Applied Mechanics and Engineering. Volume: 197, Pages: 3783 – 3802, 2008. https://doi.org/10.1016/j.cma.2008.03.001

XVIII. Liu D., Ba Y., Ren T., “Flow fluctuation abatement of high-order elliptical gear pump by external noncircular gear drive”, Mechanism and Machine Theory. Volume: 134, Pages: 338 – 348, 2019. https://doi.org/10.1016/j.mechmachtheory.2019.01.011
XIX. MundoD., “Geometric design of a planetary gear train with non-circular gears,” Mechanism and Machine Theory, Volume: 41, Pages: 456 – 472. 2006. https://doi.org/10.1016/j.mechmachtheory.2005.06.003

XX. Prikhod'koA.A., SmelyaginA.I., “Dynamic analysis of rotationally reciprocating stirred tank with multiple impellers”, Proceedings of 2015 International Conference on Mechanical Engineering, Automation and Control Systems, Pages: 1 – 5. 2015. https://doi.org/10.1109/MEACS.2015.7414910

XXI. Prikhod'koA.A., SmelyaginA.I., “Dynamic analysis of rotationally reciprocating stirred tank with planetary actuator”, Journal of Physics:Conference Series. Volume: 858, 012026. 2017.https://doi.org/10.1088/1742-6596/858/1/012026

XXII. Prikhod’koA.A., SmelyaginA.I., “Investigation of power consumption in a mixing device with swinging movement of the actuating element”, Chemical and Petroleum Engineering. Volume: 54, Pages: 150 – 155. 2018.https://doi.org/10.1007/s10556-018-0454-7

XXIII. Prikhod'koA.A., SmelyaginA.I., “Kinematic analysis of mechanism for converting rotational motion into reciprocating rotational motion”, Procedia Engineering. Volume: 129, Pages: 87–92. 2015. https://doi.org/10.1016/j.proeng.2015.12.013

XXIV. Prikhod'koA.A., SmelyaginA.I., TsybinA.D., “Kinematics of planetary mechanisms with intermittent motion”, Procedia Engineering, Volume: 206, Pages: 380–385. https://doi.org/10.1016/j.proeng.2017.10.489

XXV. SclaterN., ChironisN. P., “Mechanisms and mechanical devices sourcebook”, McGraw-Hill Press. New York, USA, 2001.

XXVI. SmelyaginA.I., “Structure of mechanisms and machines”, Novosibirsk State Technical University Press. Novosibirsk, Russia, 2001.

XXVII. SmelyaginA.I., ChoosovitynN. A., Konstantinoval.V., Sannikova J. V., “The structural and parametrical synthesis of machines and mechanisms”, Proceedings of The Fifth Russian-Korean International Symposium on Science and Technology, Pages: 257 – 261. 2001. https://doi.org/10.1109/KORUS.2001.975117.

XXVIII. SmelyaginA.I., Prikhod’koA.A., “Structure and kinematics of a planetary converter of the rotational motion into the reciprocating rotary motion”, Journal of Machinery Manufacture and Reliability. Volume: 45(6), Pages: 500 – 505. 2016. https://doi.org/10.3103/S1052618816060108

XXIX. SujanV. A., MeggiolaroM.A., “Dynamic optimization of Geneva mechanisms”, Proceedings of the International Conference on Gearing, Transmissions and Mechanical Systems, Pages: 687 – 696, 2000.
XXX. Terada H., Zhu Y., Suzuki M., Cheng C., Takahashi R., “Developments of a knee motion assist mechanism for wearable robot with a non-circular gear and grooved cams”, Mechanisms and Machine Science. Volume: 3, Pages: 69 – 76. 2012, https://doi.org/10.1007/978-94-007-2727-4_6

XXXI. Waldron K. J., Kinzel G. L., “Kinematics, Dynamics, and Design of Machinery”, John Wiley & Sons Press Inc, New York, USA, 1999.

XXXII. Zheng F., Hua L., Han X., Chen D., “Generation of non-circular bevel gears with free-form tooth profile and curvilinear tooth lengthwise”, Journal of Mechanical Design. Volume: 138(6), 064501. 2016. https://doi.org/10.1115/1.4033396

XXXIII. Zheng F., Hua L., Han X., Li B., Chen D., “Linkage model and manufacturing process of shaping non-circular gears”, Mechanism and Machine Theory. Volume: 96, Pages: 192 – 212. 2016. https://doi.org/10.1016/j.mechmachtheory.2015.09.010

XXXIV. Zheng F., Hua L., Han X., Li B., Chen D., “Synthesis of indexing mechanisms with non-circular gears”, Mechanism and Machine Theory, Volume: 105, Pages: 108 – 128, 2016, https://doi.org/10.1016/j.mechmachtheory.2016.06.019