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# DYNAMIC RESEARCH ON GEAR-AND-LEVER MOTION DRIVES

# BADANIA DYNAMICZNE NAPĘDÓW DŹWIGNIOWO-ZĘBATYCH

A b s t r a c t

Gear-and-lever is the most perspective for creating modern machines and devices. But the dynamic effect of these mechanisms on the frequency of rotation (over 1500 rotations per minute) in a modern warp knitting machine has not been investigated yet. The dynamic model must be understood as an ideal representation of the system in question for further usage in theoretical researches and engineering calculations. Any dynamic model appears to be limited and suitable under some conditions and with some questions taken into account.

*Keywords*: *gear-and-lever mechanisms, dynamic model*

S t r e s z c z e n i e

Mechanizmy dźwigniowo-zębate należą do grupy najbardziej perspektywicznych w projektowaniu maszyn i urządzeń. Jednak efekty dynamiczne związane z pracą tych mechanizmów (zwłaszcza przy prędkości obrotowej powyżej 1500 obr./min) nie zostały jeszcze wystarczająco zbadane. Model dynamiczny jest traktowany jako wyidealizowana reprezentacja układu, która jest wykorzystywana w badaniach teoretycznych i obliczeniach inżynierskich. Każdy model dynamiczny może pracować w danym zakresie wartości parametrów i przy uwzględnieniu określonych warunków pracy.

*Słowa kluczowe*: *mechanizm dźwigniowo-zębaty, model dynamiczny*

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One of the main tasks deciding of today's stage of development of mechanical engineering is the creation of high speed and balanced mechanisms of drive of working units, the design of which isn't too complicated. Gear-and-lever is the most perspective for creating modern machines and devices.

Gear-and lever is the type of mechanism which allows the reproduction of polyhedrons, because they began to be used for manufacturing the polyhedral profile. The other area where gear-and-lever mechanisms are used and have wide perspectives for further usage is the transformation of uniform velocity (rotation, as a rule) of entrance link into back-and-forth, oscillatory or rotary periodical movement of the initial link.

Khmelnitskiy National University, Ukraine, is carrying out such research (research supervisor Georgiy Paraska, Doctor of Sciences – Mechanical Engineering, Svitlana Smootko, Doctor of Philosophy).

The field of research is development of drive mechanisms of warp knitting machine tools and stitch-through machine tools.

The development includes a universal gear-and-lever motion drive which actuates all the tools (sinkers, ears, needles, sliders) of warp knitting machines and stitch-through ones. To change the displacement of tools in the mechanism it is sufficient to change 2 or 3 geometric parameters (eccentricity, initial angles), the diagram of the mechanism remaining unchanged.

As an example Fig. 1 shows the graphs (obtained by means of analytical and testing methods) of the displacements of tools which were implemented by the developed mechanism, with 2 or 3 geometric parameters changed.

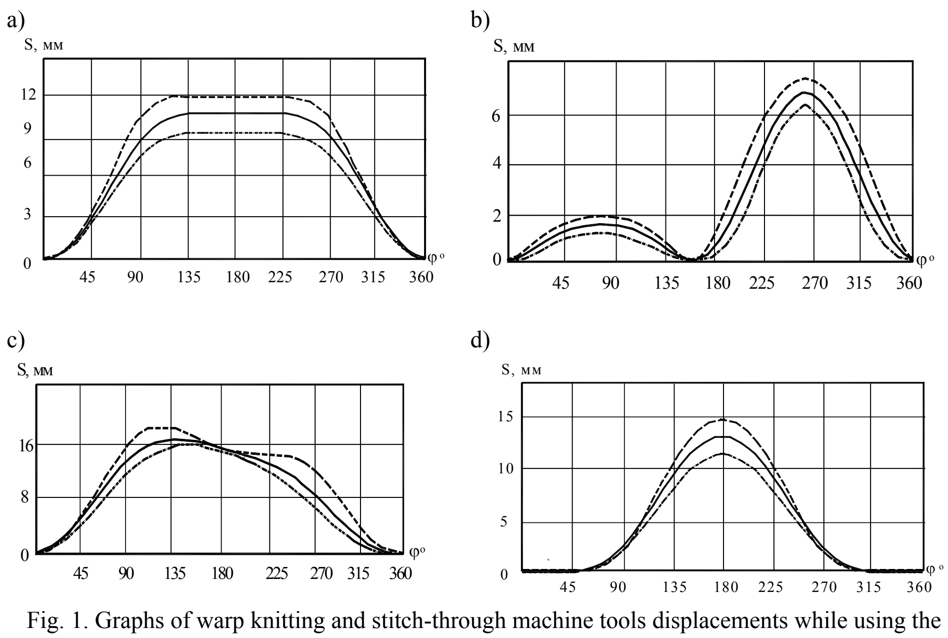


Fig. 1. Graphs of warp knitting and stitch-through machine tools displacements while using the developed mechanism

Rys. 1. Wykresy przemieszczeń elementów maszyny dziewiarskiej po zastosowaniu zaprojektowanego mechanizmu

CAD of the developed gear-and-lever motion drive confirms the possibility of obtaining practically any desirable low of warp knitting and stitch-through machines tools displacement. Using the developed universal drive in warp-knitting machines and stitch-through ones essentially reduces the cost of the whole machine. But the dynamic effect of these mechanisms on the frequency of circulation (over 1500 rotations per minute) in modern warp knitting machine has not been investigated yet. The occurrence of unbalanced masses on such circulation frequencies leads to substantial loading on the shaft and mechanism holder.

According to I.I. Vulfson [2] the dynamic model must be understood as an ideal reflection of the system in question for further usage in theoretical research and engineering calculations. Any dynamic model appears to be limited and suitable under some conditions, with certain questions taken into account.

The method proposed by I.I. Vulfson was used for the creation of the dynamic gear-and-lever planetary mechanisms. He distinguished the method of elements which are necessary for modelling any dynamic structure:

– electric motor,
– transmission mechanism Π with one of several degrees of motion, the function of transmission defined by the law of motion of working units,
– dynamic links with centred operation factors.

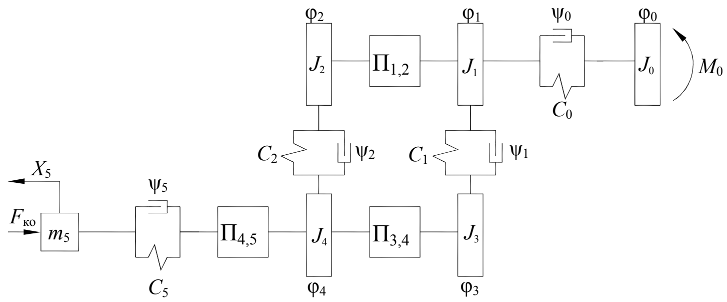


Fig. 2. Dynamic model of simple gear-and-lever of planetary mechanism, where: $\Pi_{1,2}$, $\Pi_{3,4}$, $\Pi_{4,5}$ – transmission function, $C_0$, $C_1$, $C_2$, $C_5$ – stiffness of a link, $\varphi_0$, $\varphi_1$, $\varphi_2$, $\varphi_3$, $\varphi_4$ – angle coordinates of masses; $\psi_0$, $\psi_1$, $\psi_2$, $\psi_3$, – coefficients of scattering which took into consideration the properties of links in the system, $J_0$, $J_1$, $J_2$, $J_3$, $J_4$ – the moments of links inertia, $m_5$ – the mass of initial link, $x_5$ – translational axis of initial link, $F_{ко}$ – sustaining effective power

Rys. 2. Model dynamiczny prostego dźwigniowo-zębatego mechanizmu planetarnego, gdzie: $\Pi_{1,2}$, $\Pi_{3,4}$, $\Pi_{4,5}$ – przełożenia, $C_0$, $C_1$, $C_2$, $C_5$ – sztywność połączenia, $\varphi_0$, $\varphi_1$, $\varphi_2$, $\varphi_3$, $\varphi_4$ – współrzędne kątowe mas, $\psi_0$, $\psi_1$, $\psi_2$, $\psi_3$ – współczynniki rozpraszania energii uwzględniające parametry połączeń, $J_0$, $J_1$, $J_2$, $J_3$, $J_4$ – momenty bezwładności, $m_5$ – masa pierwszego ogniwa, $x_5$ – oś ruchu posuwistego pierwszego ogniwa, $F_{ко}$ – moc efektywna

These schemata are useful for determining the real loading, depending on the aims of investigation and structural peculiarities of mechanism They contain the same mass or system of masses (2, 3 sometimes 4), which are linked by springs [3]. These masses can be either constant or variable. Commonly the stiffness of link and outer forces (motive and resistant forces) are variable, which depends on the position of the system and speed of leading element.

All masses lead to the point of mass being fixed, which moves at some speed $v_0$, which can be written [3]

$$\frac{m_n v_0^2}{2} = \frac{m_1 v_1^2}{2} + \frac{m_2 v_2^2}{2} + \frac{m_3 v_3^2}{2} + \dots, \tag{1}$$

consequently we have

$$m_n = m_1 \frac{v_1^2}{v_0^2} + m_2 \frac{v_2^2}{v_0^2} + m_3 \frac{v_3^2}{v_0^2} + \dots, \tag{2}$$

where:

$m_n$ – the value of mechanism equivalent mass,
$m_1$, $m_2$, $m_3$ – masses of mechanism elements,
$v_1$, $v_2$, $v_3$ – speed of their motion.

Taking into consideration that

$$\frac{v_1}{v_0} = i_1,\ \frac{v_2}{v_0} = i_2,\ \frac{v_3}{v_0} = i_3,\ \dots, \tag{3}$$

we receive

$$m_n = m_1 i_1^2 + m_2 i_2^2 + m_3 i_3^2 + \dots, \tag{4}$$

Thus, the mass is equal to the amount of mean products squared. If the scheme of mechanisms contains rotary masses, setting inertia moment of some masses will be done similarly as above

$$J_n = J_1 J_1^2 + J_2 J_2^2 + J_3 J_3^2 + \dots, \tag{5}$$

where:

$J_n$ – value of equivalent inertia moment of masses of all mechanism elements,
$J_1$, $J_2$, $J_3$ – inertia moments of masses of elements of real scheme mechanism,
$i_1$, $i_2$, $i_3$ – corresponding transmission relations.

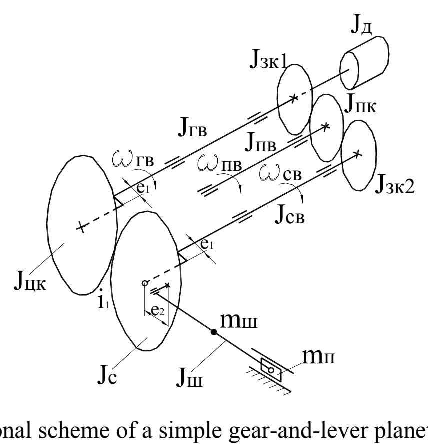


Fig. 3. Conditional scheme of a simple gear-and-lever planetary mechanism

Rys. 3. Diagram warunków pracy prostego dźwigniowo-zębatego mechanizmu planetarnego

It’s reasonable to reduce a gear-and-lever planetary mechanism to a double-mass model, leading masses and inertia moments of all links to the main shaft. Let us look at a common case, when gear wheels are not balanced. Taking into consideration the conditions of conservation of kinetic energy for this mechanism we have

$$\frac{J_{пр1}\cdot\omega_{гв}^2}{2}=\frac{J_{зк1}\cdot\omega_{св}^2}{2}+\frac{\left(J_{св}/2\right)\cdot\omega_{св}^2}{2}+\frac{J_{дв}\cdot\omega_{гв}^2}{2}+\frac{J_{пк}\cdot\omega_{пв}^2}{2}+\frac{J_{пв}\cdot\omega_{пв}^2}{2}+ \\ +\frac{\left(J_{св}/2\right)\cdot\omega_{св}^2}{2}+\frac{J_{зк2}\cdot\omega_{св}^2}{2}+ \tag{6}$$

$$\frac{J_{пр2}\cdot\omega_{гв}^2}{2}=\left(\frac{J_{цк}\cdot\omega_{гв}^2}{2}+\frac{m_{цк}\cdot v_{цк}^2}{2}\right)+\frac{\left(J_{гв}/2\right)\cdot\omega_{гв}^2}{2}+\left(\frac{J_{c}\cdot\omega_{гв}^2}{2}+\frac{m_{c}\cdot v_{c}^2}{2}\right)+ \\ +\frac{\left(J_{св}/2\right)\cdot\omega_{св}^2}{2}+\left(\frac{J_{ш}\cdot\omega_{ш}^2}{2}+\frac{m_{ш}\cdot v_{ш}^2}{2}\right)+\frac{m_{ш}\cdot v_{2}^2}{2} \tag{7}$$

Let us use equations (6) and (7) to define $J_{пр1}$ and $J_{пр2}$; making canonical transformation we receive, taking into consideration that $\omega_{гв}=\omega_{св1}=\omega_{св2}$, and $v_{цк}=\omega_{гв}\cdot e_1$ i $v_c=\omega_{св}\cdot e_1$

$$J_{пр1}=J_{цк}+m_{цк}\cdot e_1^2+\frac{J_{гв}}{2}+J_c+m_c\cdot e_1^2+\frac{J_{св}}{2}+J_{ш}\frac{\omega_{ш}^2}{\omega_1^2}+m_{ш}\frac{v_{ш}^2}{\omega_1^2}+m_{п}\frac{v_{п}^2}{\omega_1^2} \tag{8}$$

$$J_{пр1}=J_{цк}+m_{цк}\cdot e_1^2+\frac{J_{гв}}{2}+J_c+m_c\cdot e_1^2+\frac{J_{св}}{2}+J_{ш}\frac{\omega_{ш}^2}{\omega_1^2}+m_{ш}\frac{v_{ш}^2}{\omega_1^2}+m_{п}\frac{v_{п}^2}{\omega_1^2} \tag{9}$$

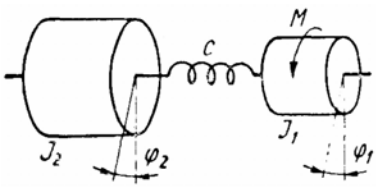


Fig. 4. Double-mass system

Rys. 4. Układ dwumasowy

Let’s designate $J_{пр1}=J_1$, $J_{пр2}=J_2$ for simplification. Thus, we receive a double-mass high frequency system (Fig. 4), whose differential equation of motion under loading during operation is [3]

$$\left.\begin{aligned} J_1 \frac{d^2\varphi_1}{dt^2} + \left(\varphi_1 - \varphi_2\right)c = M; \\ J_1 \frac{d^2\varphi_1}{dt^2} - \left(\varphi_1 - \varphi_2\right)c = -M_c \end{aligned}\right\}$$

where:

| | | |
|---|---|---|
| $J_1$, $J_2$ | – | equivalent inertia moment of leading and known mass systems, |
| $c$ | – | equivalent stiffness of the main shaft, |
| $M$ | – | moment, which is created by the engine, |
| $\varphi_1$, $\varphi_2$ | – | angles of masses coordinates, |
| $M_c$ | – | outer loading, which depends on the system position, time or speed of the leading mass. |

Thus, by setting the parameters of a dynamic system, we can find the angles of mass rotations and, consequently, rotation of the main shaft under different working conditions, which is the aim of our further research.