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SIMULATION OF THE INSTANT SUPPLY IN GEAR HYDRAULIC MACHINES

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Abstract: The article proposes a new technique for estimating the instantaneous supply in gear hydraulic machines. It takes into account the peculiarity of the movement of the radius connecting the centers of rotation of the gears with the point of engagement. The dependence of the angular velocity of rotation and the length of the radius on the position of the engagement point due to slipping of the working profiles of the gear teeth are determined. Estimation of the degree of non-uniformity of the supply, in accordance with the obtained model, differs significantly from the traditional ideas about the pulsation of the instantaneous supply in gear hydraulic machines.

KEYWORDS: gear hydraulic machines, degree of unevenness of supply, supply pulsation, instantaneous supply

1 Introduction

Raising the technical level of gear pumps is impossible without further theoretical and experimental studies. This work is devoted to one of the insufficiently studied theoretical aspects of the functioning of gear pumps, concerning the laws of instantaneous supply of fluid from the angle of rotation of the gears.

Several works are devoted to the problem of uneven supply and pressure, in which the results of both theoretical and experimental studies are presented [1 – 17].

The magnitude of the pulsation is measured by the degree of feed unevenness, which is determined in accordance with the relationship [1]:

$$\sigma = (Q_{max} - Q_{min})/Q_{max} \tag{1}$$

or [1]:

$$\sigma = (Q_{max} - Q_{min})/0.5 \cdot (Q_{max} + Q_{min}) \tag{2}$$

where, Q_{max} and Q_{min} respectively, the maximum and minimum pump supply.

In [1-13], various theoretical approaches were used to study this problem. However, the obtained dependencies do not give confidence in the truth of the results, since they show an extremely wide range of variation in the degree of feed unevenness - from a minimum of 1,25% [2] to an average value of 15 ... 23% [1] and a maximum of 63 ... 73% [10, 11].

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We should note that a very small number of works have been devoted to a theoretical explanation of a gear pump's supply and pressure pulsation. Italian researchers [4, 7, 8] dealt with these issues in the most detail, in which the error explained the phenomenon of pulsation in the manufacture of gear teeth.

The results of experimental studies showed a spread in the values of the supply and pressure pulsations [1-17] in an even wider range: from 1.25 ... 6.0% [4, 5, 7] and 2.0 ... 4.0% in the work [16] up to 55…100% [15, 17]. These results can be explained by the rather high complexity of the problem. Unfortunately, in all the mentioned works, there is no or insufficiently detailed methodology for conducting experimental studies in order to have an idea of the degree of validity and reliability of the results presented.

Thus, conflicting information about the pulsation of instantaneous feed and pressure indicates that the developed physical and mathematical models of the instantaneous feed process are not able to fully explain this phenomenon. Until now, there is no comprehensive understanding of the causes of supply and pressure pulsations from a physical point of view.

All this is a limiting factor for further improvement of the gear pump design and suggests the need for new in-depth theoretical and experimental studies of the gear pump feeding process.

The purpose of this study is to improve the existing ideas about the process of instantaneous supply in a gear pump.

Let us consider the main reasons for the discrepancy between the results of experimental and theoretical studies.

In published works, researchers do not see the difference between the geometric and hydrodynamic feed. The first is connected purely with the movement of gear elements. Hydrodynamic supply, based on the laws of geometric flow, considers the peculiarities of the movement of the working fluid under the influence of the elements of the pump gears.

This article will consider a mathematical model of geometric instantaneous feed.

Researchers do not fully take into account the features of the mutual movement of the involute profiles of gear teeth. They consist in the fact that between the working profiles of the gear teeth there is a mutual rotation with sliding along the entire length of the engagement, except the engagement pole, where the sliding turns into rolling [18]. In the article [20], theoretical dependencies of the movement of gear elements were obtained, in which these features of the movement of involute profiles of gear teeth were taken into account. We will use these patterns to develop a mathematical model for the instantaneous delivery of a gear.

2 Methodology

The research methodology is based on the theory of gearing, the geometry of involute gearing, the laws of motion of a material point along the gearing line.

The principle of operation of the gear pump has some features. One of them is that the discharge chamber is separated from the suction chamber in the center space by a pair of teeth that are in engagement. In this case, the boundary of the discharge and suction chambers is movable. This movement is directed away from the pumping elements of the pump, which somewhat reduces the rate of displacement of the working fluid into the injection chamber and the working volume of the gear pump (Fig. 1).

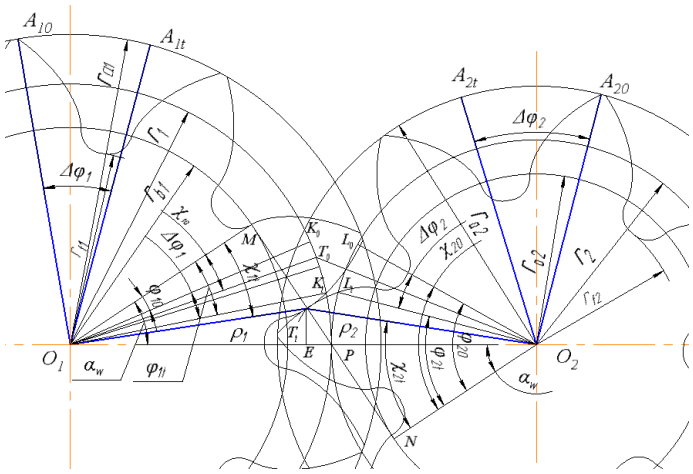


Fig. 1 Scheme of the movement of gear elements from the initial position – 0 to the position of the gear elements at the current time – *t*

At the first stage of research, to simplify the geometric model, we will replace the body of the gear teeth with plates (Fig. 1) [19].

Let's determine the dependencies of the instantaneous feed on the angle of rotation of the drive gear – φ_1. In [20], the dependence of the length of the radii connecting the axes of rotation of the gears with the point of engagement of the teeth $\rho_1(\varphi_1)$ and $\rho_2(\varphi_2)$ on the angle of rotation of the drive gear φ_1 were determined:

$$\rho_1(\phi_1) = r_{b1}\sqrt{(inv\alpha_w + \phi_1)^2 + 1} \tag{3}$$

$$\rho_2(\phi_1) = r_{b2}\sqrt{(tg\alpha_w + (\alpha_w - \phi_1)/i)^2 + 1} \tag{4}$$

And also, the dependencies of the angles of rotation of the radii $\rho_1(\varphi_1)$ and $\rho_2(\varphi_2)$ on the angle of rotation of the drive gear φ_1 were determined respectively $\chi_1(\varphi_1)$ and $\chi_1(\varphi_1)$.

$$\chi_1(\phi_1) = arctg(\phi_1 + inv\alpha_w) \tag{5}$$

$$\chi_2(\phi_1) = arctg[(\alpha_w - \phi_1)/i + tg\alpha_w] \tag{6}$$

where φ – argument – the angle of rotation of the drive gear;

r_{b1}, r_{b2} – the radii of the main circles, respectively, of the driving and driven gears;

a_w – engagement angle;

Between the increment of the angle of rotation of the driving gear $\Delta\varphi_1$ and the increment of the angle of rotation of the driven gear $\Delta\varphi_2$ there is a relationship:

$$\Delta\phi_2 = \Delta\phi_1/i \tag{7}$$

where i – gear ratio,

$$i = z_2/z_1 = r_2/r_1 = \omega_2/\omega_1 = n_2/n_1 \tag{8}$$

where z_1, z_2 – the number of teeth of the driving and driven gears;

r_1, r_{b1} – the radii of the initial circles, respectively, of the driving and driven gears;

ω_1, ω_2 – angular speeds of rotation, respectively, of the driving and driven gears;

n_1, n_2 – frequency of rotation, respectively, of the driving and driven gears;

From fig. 1, we see that during the rotation of the gears, the plates O_1A_1 and O_2A_2 move towards each other, displacing the working fluid located between them into the injection chamber. And the O_1T and O_2T plates move away from the discharge chamber, thereby reducing

the rate of displacement of the working fluid and reducing the volume of the pump working chamber.

The injection of the working fluid occurs because the total length of the O_1T and O_2T plates is less than the total length of the O_1A_1 and O_2A_2 plates. Also, the rotation speed of the O_1T and O_2T plates is less than the rotation speed of the O_1A_1 and O_2A_2 plates [20]. This feature of the mutual movement of the elements of the pump gears has not been considered in detail by most researchers.

Let at the initial moment the drive gear turn relative to the radius O_1M by an angle φ_1 (Fig. 1), then the driven gear will turn relative to the radius O_2N by an angle $a_w + (a_w - \varphi_1)/i$ [20].

Let's give the drive gear an increment – $\Delta\varphi_1$, then:
- the angle of rotation of the radii $O_1A_1 = r_{a1}$ will be $\Delta\varphi_1$ from the initial position;
- the angle of rotation of the radii $O_2A_2 = r_{a2}$, will be $\Delta\varphi_1/i$ from the initial position.

The area of the circular sector described by the radius $O_1A_1 = r_{a1}$ will be

$$\Delta S_{r_{a1}}(\Delta\phi_1) = 0{,}5 \cdot \left(r_{a1}{}^2 - r_{f1}{}^2\right) \cdot \Delta\phi_1 \tag{9}$$

The area of the sector described by the radius $O_2A_2 = r_{a2}$ will be

$$\Delta S_{r_{a2}}(\Delta\phi_1) = 0{,}5 \cdot \left(r_{a2}{}^2 - r_{f2}{}^2\right) \cdot \Delta\phi_1/i \tag{10}$$

where r_{a1} and r_{a2} are the radius of the circle of the tops of the heads of the teeth of the driving and driven gears, respectively; r_{f1} and r_{f2} are the radius of the circle of the recesses of the driving and driven gears, respectively.

Note that the radii of the gears $O_1A_1 = r_{a1}$ and $O_2A_2 = r_{a2}$ are constant. The length of the radii $\rho_1(\varphi_1)$ and $\rho_2(\varphi_1)$ is determined by the angles of their rotation $\chi_1(\varphi_1)$ and $\chi_2(\varphi_1)$. And therefore, to find the increment of the radius $\rho_1(\varphi_1)$ when the argument is incremented by $\Delta\varphi_1$, they find a difference in the form

$$\Delta\rho_1(\phi_1) = \rho_1(\phi_1 + \Delta\phi_1) - \rho_1(\phi_1) \tag{11}$$

where φ_1 is the initial position of the radius r_{a1} of the drive gear.

Similarly for the angle of rotation of the radius $\rho_1(\varphi_1)$ - $\chi_1(\varphi_1)$

$$\Delta\chi_1(\phi_1) = arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w) - arctg(\phi_1 + inv\alpha_w) \tag{12}$$

Let at a given moment of time the radius $\rho_1(\varphi_1)$ be in the position determined by the angle of rotation $\chi_1(\varphi_1)$. Let's give the argument φ_1 an increment $\Delta\varphi_1$. Then the area of the right-angled triangle O_1MT_t (Fig. 1) is determined in accordance with the dependence

$$S_{O_1MT_t} = 0{,}5 \cdot r_{b1} \cdot r_{b1} \cdot tg(\chi_1(\phi_1 + \Delta\phi_1)) \tag{13}$$

Substituting the value from expression (5) into dependence (13), we obtain

$$S_{O_1MT_t} = 0{,}5\, r_{b1}{}^2\, (\phi_1 + \Delta\phi_1 + inv\alpha_w) \tag{14}$$

Since the working fluid is transferred only in the cavities between the teeth (Fig. 1), then from the result obtained (14) it is necessary to subtract the sector described by the radius of the gear cavities – r_{f1} and its angle of rotation – $\chi_1(\Delta\varphi_1)$:

$$S_{rft} = 0{,}5 \cdot r_{f1}^2 \cdot arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w) \tag{15}$$

we get

$$S_{O_1MT_t} - S_{rft} = 0{,}5\left[{r_{b1}}^2 \cdot (\phi_1 + \Delta\phi_1 + inv\alpha_w) - r_{f1}^2 \cdot arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w)\right] \tag{16}$$

The area of a right-angled triangle O_1MT_0 (Fig. 1) is determined in accordance with the dependence

$$S_{O_1MT_0} = 0{,}5 \cdot r_{b1} \cdot r_{b1} \cdot tg(\chi_1(\phi_1) \tag{17}$$

Substituting the values $\chi_1(\varphi_1)$ from expression (5) into dependence (17), we obtain

$$S_{O_1MT_0} = 0{,}5\ {r_{b1}}^2\ (\phi_1 + inv\alpha_w) \tag{18}$$

Taking into account the area of the circular sector of radius r_{f1} and the angle of its rotation $\chi_1(\varphi_1)$:

$$S_{rf0} = 0{,}5 \cdot r_f^2 \cdot arctg(\phi_1 + inv\alpha_w) \tag{19}$$

$$S_{O_1MT_0} - S_{rf0} = 0{,}5 \cdot {r_{b1}}^2 \cdot (\phi_1 + inv\alpha_w) - 0{,}5 \cdot r_{f1}^2 \cdot arctg(\phi_1 + inv\alpha_w) \tag{20}$$

Then, we finally get an increment of the area described by the radius $\rho_1(\varphi_1)$

$$\Delta S_{\rho 1} = \left(S_{O_1MT_t} - S_{rft}\right) - \left(S_{O_1MT_0} - S_{rf0}\right) \tag{21}$$

Substituting into (22) the values from (17) and (21), after transformations, we obtain

$$\begin{aligned}\Delta S_{\rho 1} = 0{,}5 \cdot {r_{b1}}^2 \cdot \Delta\phi_1 - 0{,}5 \cdot r_{f1}^2 \\ \cdot [arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w) - arctg(\phi_1 + inv\alpha_w)]\end{aligned} \tag{22}$$

Same for the driven gear. Let at a given moment of time the radius $\rho_2(\varphi_1)$ be in the position determined by the angle of rotation $\chi_2(\varphi_1)$. Let us give the argument φ_1 an increment $\Delta\varphi_1$. Then, after the corresponding mathematical operations, we obtain

$$\begin{aligned}\Delta S_{\rho 2} = 0{,}5 \cdot {r_{b2}}^2 \cdot (\Delta\phi_1)/i - -0{,}5 \cdot r_{f2}^2 \\ \cdot \{arctg[(\alpha_w - \phi_1)/i + tg\alpha_w] \\ - arctg[(\alpha_w - \phi_1 - \Delta\phi_1)/i + tg\alpha_w]\}\end{aligned} \tag{23}$$

The algebraic sum of the areas described by the radii $O_1A_1 = r_{a1}$, $O_2A_2 = r_{a2}$, $\rho_1(\varphi_1)$ and $\rho_2(\varphi_1)$ will be

$$\Delta S(\Delta\phi_1) = \Delta S_{r_{a1}}(\Delta\phi_1) + \Delta S_{r_{a2}}(\Delta\phi_1) - \Delta S_{\rho 1}(\Delta\phi_1) - \Delta S_{\rho 2}(\Delta\phi_1) \tag{24}$$

Substituting in (24) the expressions of its components from (9), (10) and (22), and multiplying the resulting expression by the width of the gear – b, after the transformation, we proceed to change the volume from a change in the argument φ_1

$$\begin{aligned}&\Delta V(\Delta\phi_1) = \\ &= 0{,}5 \cdot b \cdot \left\{\begin{matrix}\left({r_{a1}}^2 - {r_{f1}}^2\right) \cdot \Delta\phi_1 + \left({r_{a2}}^2 - {r_{f2}}^2\right) \cdot \Delta\phi_1/i - {r_{b1}}^2 \cdot \Delta\phi_1 - {r_{b2}}^2 \cdot (\Delta\phi_1)/i \\ +r_{f1}^2 \cdot [arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w) - arctg(\phi_1 + inv\alpha_w)] + \\ +r_{f2}^2 \cdot \{arctg[(\alpha_w - \phi_1)/i + tg\alpha_w] - arctg[(\alpha_w - \phi_1 - \Delta\phi_1)/i + tg\alpha_w]\}\end{matrix}\right\}\end{aligned} \tag{25}$$

Having found a relation of the form $\Delta V(\Delta\varphi_1)/\Delta\varphi_1$, we will determine the average rate of volume change that occurs during the rotation of the pump gears. In engineering, this value is called the average pump flow.

$$\frac{\Delta Q(\Delta\phi_1)}{\Delta\phi_1} = \\ = 0{,}5 \cdot b \cdot \left\{ \begin{matrix} \left(r_{a1}{}^2 - r_{f1}{}^2\right) + \left(r_{a2}{}^2 - r_{f2}{}^2\right)/i - r_{b1}{}^2 - r_{b2}{}^2 /i \\ + \dfrac{r_{f1}^2 \cdot [arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w) - arctg(\phi_1 + inv\alpha_w)]}{\Delta\phi_1} + \\ + \dfrac{r_{f2}^2 \cdot \{arctg[(\alpha_w - \phi_1)/i + tg\alpha_w] - arctg[(\alpha_w - \phi_1 - \Delta\phi_1)/i + tg\alpha_w]\}}{\Delta\phi_1} \end{matrix} \right\} \quad (26)$$

Going to the limit, we carry out the transition from average to instantaneous feed, at a point of rotation of the gears:

$$Q_{ins}(\Delta\phi_1) = 0{,}5 \cdot b \cdot \lim_{\Delta\phi_1 \to 0} \frac{\Delta V(\Delta\phi_1)}{\Delta\phi_1} \quad (27)$$

Given that

$$\lim_{\Delta\phi_1 \to 0} \frac{arctg(\phi_1 + \Delta\phi_1 + inv\alpha_w) - arctg(\phi_1 + inv\alpha_w)}{\Delta\phi_1} = \frac{1}{[(\phi_1 + inv\alpha_w)]^2 + 1} \quad (28)$$

We obtain an expression for the instantaneous feed in the form

$$Q_{ins}(\phi_1) = 0{,}5 \cdot b \cdot \left\{ \begin{matrix} \left(r_{a1}{}^2 - r_{f1}{}^2 + \dfrac{r_{a2}{}^2 - r_{f2}{}^2}{i}\right) - \left[\dfrac{i \cdot r_{b1}{}^2 + r_{b2}{}^2}{i} - \right. \\ \left. - \dfrac{r_{f1}^2}{(\phi_1 + inv\alpha_w)^2} - \dfrac{r_{f2}^2 \cdot i}{(\alpha_w - \phi_1 + i \cdot tg\alpha_w)^2 + i^2}\right] \end{matrix} \right\} \quad (29)$$

For further analysis of the dependence (29), we present it as components corresponding to the geometric parameters of the pump elements, which have a direct impact on the fluid supply.

$$Q_{ins}(\phi_1) = Q_{ins\ ra1} + Q_{ins\ ra2} + Q_{ins\ \rho1}(\phi_1) + Q_{ins\ \rho2}(\phi_1) \quad (30)$$

where $Q_{ins\ ra1}$, $Q_{ins\ ra2}$ are the constant components of the instantaneous feed provided by the gear teeth, respectively, the driving and driven gears:

$$Q_{ins\ ra1} = 0{,}5 \cdot b \cdot \left(r_{a1}{}^2 - r_{f1}{}^2\right) \quad (31)$$

$$Q_{ins\ ra2} = 0{,}5 \cdot b \cdot \left(\frac{r_{a2}{}^2 - r_{f2}{}^2}{i}\right) \quad (32)$$

$Q_{ins\ ra1}(\varphi_1)$ and $Q_{ins\ ra2}(\varphi_1)$ variable components of instantaneous feed, dependent on the angle of rotation of the gears, provided by radius $\rho_1(\varphi)$ and $\rho_2(\varphi)$

$$Q_{ins\ \rho1}(\phi_1) = -0{,}5 \cdot b \cdot \left[r_{b1}{}^2 - \frac{r_{f1}^2}{(\phi_1 + inv\alpha_w)^2 + 1}\right] \quad (33)$$

$$Q_{ins\ \rho2}(\phi_1) = -0{,}5 \cdot b \cdot \left[\frac{r_{b2}{}^2}{i} - \frac{r_{f2}^2 \cdot i}{(\alpha_w - \phi_1 + i \cdot tg\alpha_w)^2 + i^2}\right] \quad (34)$$

For gearing with the same number of teeth, expression (30) is converted to the form

$$Q_{ins}(\phi_1) = b \cdot \left(r_a{}^2 - r_f{}^2\right) - 0{,}5 \\ \cdot b \left\{2\ r_{b1}{}^2 - r_{f1}^2 \cdot \left[\frac{1}{(\phi_1 + inv\alpha_w)^2 + 1} + \frac{1}{(\alpha_w - \phi_1 + tg\alpha_w)^2 + 1}\right]\right\} \quad (35)$$

And the components of dependence (35) will have the form

$$Q_{ins\ ra1} = Q_{ins\ ra2} = 0{,}5 \cdot b \cdot \left({r_a}^2 - {r_f}^2\right) \tag{36}$$

$$Q_{ins\ \rho1}(\phi_1) = -0{,}5 \cdot b \cdot \left[{r_b}^2 - \frac{r_f^2}{(\phi_1 + inv\alpha_w)^2 + 1}\right] \tag{37}$$

$$Q_{ins\ \rho2}(\phi_1) = -0{,}5 \cdot b \cdot \left[{r_b}^2 - \frac{r_f^2}{(\alpha_w - \phi_1 + tg\alpha_w)^2 + 1}\right] \tag{38}$$

The average feed can be determined in accordance with the dependence

$$Q_{avr}(\phi) = \frac{1}{2 \cdot \pi \cdot \frac{\varepsilon}{z_1}} \cdot \int_{\alpha - \pi \cdot \frac{\varepsilon}{z_1}}^{\alpha + \pi \cdot \frac{\varepsilon}{z_1}} Q_{мгн}(\phi_1) \cdot d\phi \tag{39}$$

Turning to the analysis of the obtained expressions, it is worth noting that the graphs of the dependences under study are constructed for gears with different parameters calculated for pumps of the NSh-32 UK type. The choice of this pump is explained by the fact that its gearing is a typical representative of gear pumps. The parameters of the studied gearing are presented in Table. 1 [20].

Dependence graphs of instantaneous feed (30) and its constant components (32) and (33) and variable components (34) and (35) are presented in Fig. 2 for the case z_1=z_2=8, in Fig. 3 for the case z_1=9 z_2=7 and in Fig. 4 for the case z_1=7 z_2=9, respectively.

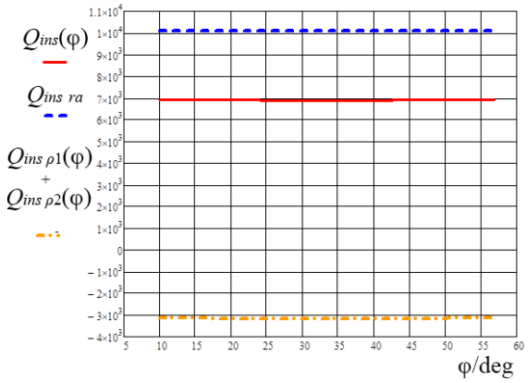


Fig. 2 Graph of instantaneous flow and its constant and variable components for a gear pump with gears with an equal number of teeth z_1=z_2=8

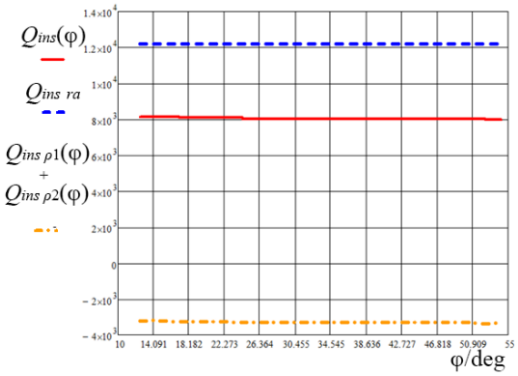


Fig. 3 Graph of instantaneous flow and its constant and variable components for a gear pump with gears with a number of teeth z_1= 9 z_2=7

The analysis of the given graphic dependences shows the almost complete absence of pulsation of the instantaneous supply of the gear pump, which contradicts the traditional ideas about this process. In addition, comparing the graphs in Fig. 2 – Fig. 4, it can be seen that the instantaneous feed increases under the condition $z_1 \geq z_2$ and vice versa, it decreases if $z_1 \leq z_2$.

Figures 5-7 show the plots of instantaneous feed (30) and its average value (40) on an enlarged scale. On Fig. 5 for the case z_1=z_2=8, in Fig. 6 for the case z_1=9 z_2=7 and in Fig. 7 for the case z_1=7 z_2=9, respectively.

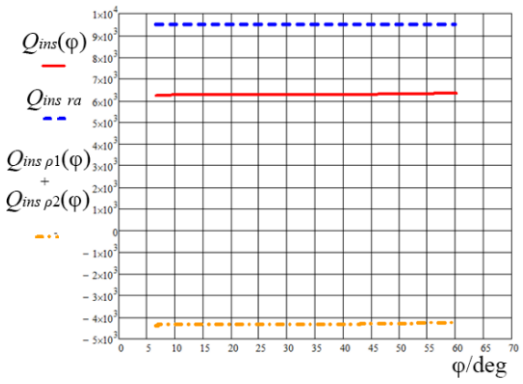


Fig. 4 Graph of instantaneous flow and its constant and variable components for a gear pump with gears with a number of teeth z_1= 7 z_2=9

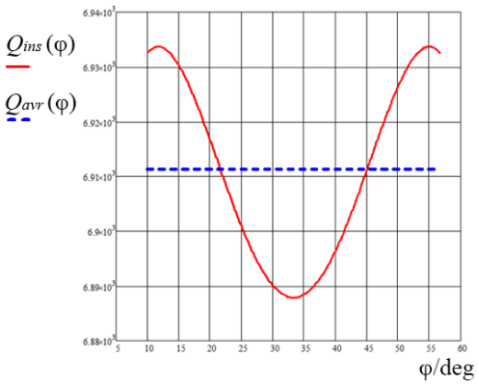


Fig. 5 Graph of the dependence of the instantaneous flow and the average flow of a gear pump with gears with an equal number of teeth z_1=z_2=8

Table 1. The degree of unevenness of the instantaneous supply of a gear pump, depending on the ratio of the number of teeth of its gears.

The ratio of the number of teeth of gears	z_1=z_2=8	z_1=9 z_2=7	z_1=7 z_2=9
Degree of non-uniformity of giving, %	0,667	1,728	1,742

Analysis of the graphs presented in Fig. 5 – Fig. 7 showed that the geometric instantaneous feed has a pulsating character. However, simple calculations in accordance with dependence (2) gave the following results.

From Table. 1 we see that the actual degree of instantaneous feed unevenness is insignificant and is in the range of 0,667% ... 1,742%, which is at least an order of magnitude less in comparison with traditional ideas about this process [1 – 17]. But the results obtained agree very well with the results of experimental studies presented in the works of Italian authors Casoli P., Vacca A., Berta G. [4, 5, 7] and Stroganova N.V. [16] – in accordance with which, the degree of uneven supply is 1,25 ... 6,0%.

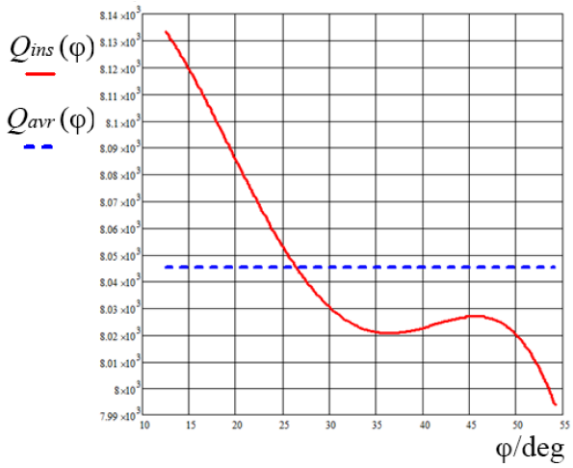


Fig. 6 Graph of instantaneous flow and average flow of a gear pump with gears with an equal number of teeth z_1=z_2=8

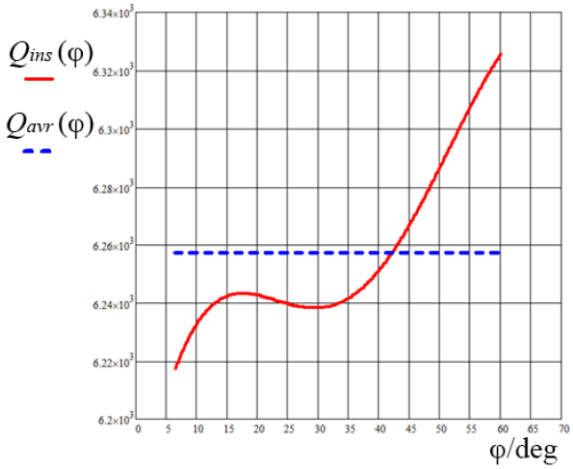


Fig. 7 Graph of the dependence of the instantaneous flow and the average flow of a gear pump with gears with an equal number of teeth z_1=7 z_2=9

The pulsation of the instantaneous delivery of the gear pump, which does not exceed 1,742%, can be explained by the non-linear behavior of the variable components of the instantaneous delivery of the pump (34) and (35). On Fig. 8 – Fig. 10 shows graphs of the variable components of the instantaneous supply of the gear pump (34) and (35) for the case $z_1=z_2=8$ in Fig. 8, for the case $z_1=9$ $z_2=7$ in Fig. 9 and for the case $z_1=7$ $z_2=9$ in Fig. 10 respectively.

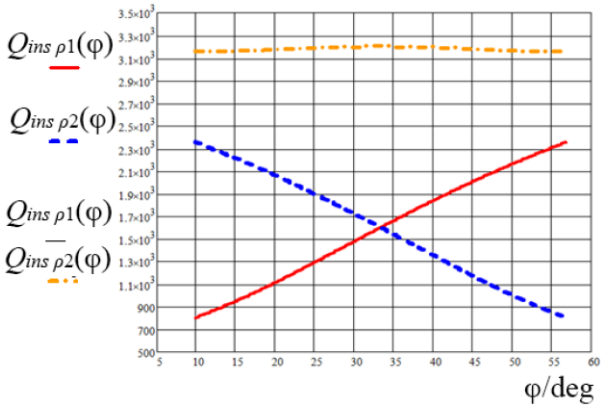


Fig. 8 Graphs of variable components of instantaneous delivery for a gear pump with gears with an equal number of teeth $z_1=9$ $z_2=7$

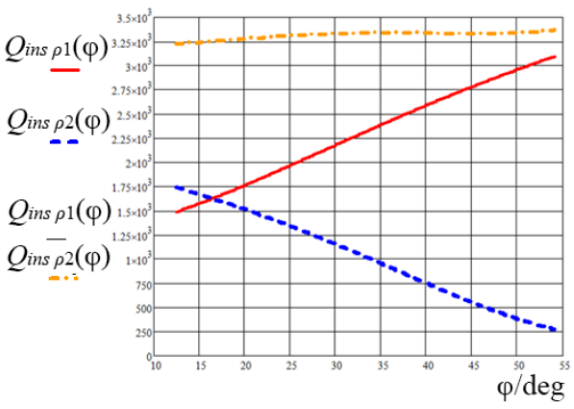


Fig. 9 Graphs of variable components of instantaneous delivery for a gear pump with gears with an equal number of teeth $z_1=9$ $z_2=7$

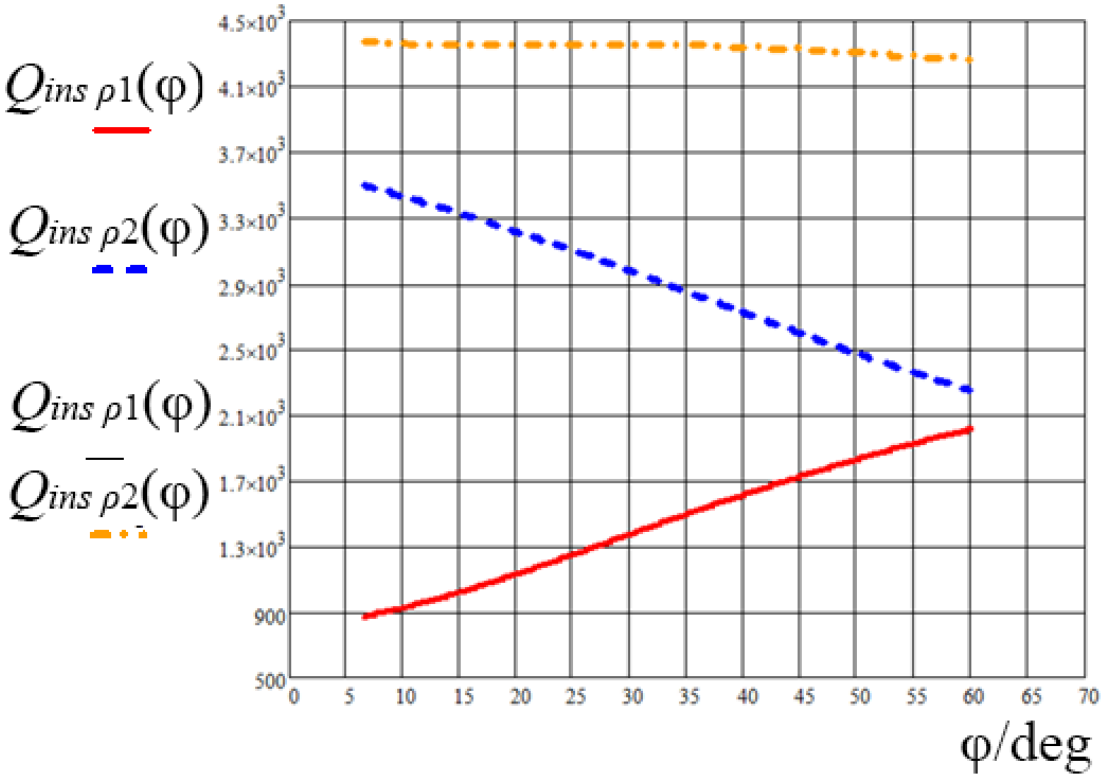


Fig. 10 Graph of the dependence of the instantaneous flow and the average flow of a gear pump with gears with an equal number of teeth $z_1=7$ $z_2=9$

CONCLUSION

1. The analysis of sources of scientific and technical information showed the inconsistency of the results of theoretical and experimental studies of the pulsation of the instantaneous supply of a gear pump, which was the reason for the search for new approaches to solve this problem.

2. A new approach has been proposed in the development of a mathematical model of instantaneous supply, the essence of which is that the proposed model takes into account the features of the process of pumping a working fluid consisting in the interaction of two oppositely directed processes of reducing and increasing the volume of the pressure chamber of a gear pump.

3. The obtained dependences of the instantaneous feed on the angle of rotation of the drive gear consist of constant and variable components. The constant component of the instantaneous feed is provided by the radii of the driving r_{a1} and driven r_{a1} gears and the angular speed of their rotation and contributes to the growth of feed and pressure. The variable component of the instantaneous feed is provided by the radii ρ_1 of the driving and driven ρ_2 gears and the angular speed of their rotation r_{a1} and r_{a2}, which is less than the speed of rotation of the radii and due to the presence of sliding of the involute profiles of the gear teeth. At the same time, the movement of these radii ρ_1 ρ_2 is directed in the opposite direction from the discharge chamber, which reduces the pump flow. The variable component is the cause of a slight, not exceeding 2%, instantaneous feed pulsation.

4. The proposed model explains the principle of operation of a gear pump when pumping a working fluid, which allows a new approach to improving the gear pump in the direction of increasing its supply. At the same time, the results obtained refute the traditional ideas about the pulsating nature of the instantaneous feed; in any case, theoretical evidence has been obtained that the pulsation of the instantaneous feed is an order of magnitude smaller than is commonly believed.

REFERENCES

[1] Yudin, E.M. “Gear pumps. Basic parameters and their calculation / Yudin E. M. – [2nd ed. revised and add.]”, – M.: Mashinostroenie, 236 p., **1964**.

[2] Kuleshkov, Yu.V. “Improvement of the mathematical model of the instantaneous supply of a gear pump”, / Kuleshkov Yu.V., Osin R.A., Rudenko T.V., Matvienko O.O. //Collection of science practices of the Kirovohrad National Technical University "Technology in the Agricultural Production, Galuzev machine–building, Mutomation" – Kirovohrad, KNTU, – No. 21., pp. 253 – 262, **2008**.

[3] “The theoretical flow ripple of an external gear pump” a Thesis presented lo the Faculty of the Graduate School University of Missouri–Columbia In Partial Fulfillment Of the Requirements for the Degree Master of Science By Suresh Babu Kasaragadda dr. Noah d. Manring. Thesis Supervisor december **2003**.

[4] Casoli, P., Vacca, A., Berta, G. L. “Optimization of relevant design parameters of external gear pumps”, Department of Industrial Engineering, University of Parma Viale G. P. Usberti 181/a, 43100 Parma, Italy. Electronic resource http://www.jfps.jp/proceedings/toyama2008/pdf/p1–05.pdf.

[5] Andrea Vacca Projects GearUnits / A. Vacca – Purdue University, **2006**. access mode: https://engineering.purdue.edu/maha_avteam/research%20projects/projectsgearunitssept16.pdf.

[6] Tkach, P., Reviakina, O., Kryvosheia, A., Melnyk, V., Ustynenko, O., Protasov, R. “Meshing characteristics of profile shifted cylindrical quasi-involute arc-tooth-trace gears. Part 1. Theoretical base”, Strojnícky časopis – Journal of Mechanical Engineering 72 (2), pp. 201 – 210, **2022**. DOI: 10.2478/scjme-2022-0029

[7] “A numerical model for the simulation of external gear pumps”, Paolo Casoli, Andrea Vacca and Germano Franzoni. Dipartimento di Ingegneria Industriale Università degli Studi di Parma Parco Area delle Scienze 181/a, 43100 Parma, Italy.

[8] Casoli, P., Vacca, A., Berta, G. L. “A numerical model for the simulation of flow in hydraulic external gear machines [Tener]: Power Transmission and Motion Control”, pp. 147 – 165, **2006**.

[9] Hujo, Ľ., Kaszkowiak, J., Nosian, J., Michalides, M. “Test of oil filters on laboratory test device”, Strojnícky časopis – Journal of Mechanical Engineering 72 (3), pp. 27 – 34, **2022**. DOI: 10.2478/scjme-2022-0037

[10] Kosenok, B. B. “Mathematical vector modeling of fluid supply pulsations by a gear pumping unit”, / B.B. Kosenok, A.N. Kryuchkov, L.V. Rodionov, E.V. Schakhmatov. // Proceedings of the samara scientific center of the russian academy of sciences 11 (3), pp. 28 – 37, **2009**.

[11] Rodionov, L. V. “Development of methods for calculating and improving the dynamic characteristics of gear pumps”, diss. cand. tech. sciences / Rodionov Leonid Valerievich. – Samara, 154 p., **2009**.

[12] Andrea, V. “Modeling and experimental validation of external spur gear machines for fluid power applications” / Andrea V., Guidetti M. – MAHA Fluid Power Research Center, Purdue University, West Lafayette, in, USA, CASAPPA SPA, Cavalli di Collecchio, Parma, Italy – **2011**. access mode: http://www.sciencedirect.com/science/article/pii/s1569190x11001018

[13] Kuleshkov Yu., V. “Analysis of theoretical studies of the pulsation of the instantaneous supply of a gear pump”, / Kuleshkov Yu.V., Rudenko T.V., Krasota M.V., Kuleshkova K.Yu. // Foreign state international scientific and technical collection "Construction, design and operation of agricultural machines". – Kirovograd, KDTU – No. 43, part 1 – p. 83 – 96, **2013**.

[14] Zhao X, Vacca A. “Theoretical Investigation into the Ripple Source of External Gear Pumps”, Energies 12 (3): 535, **2019**

[15] Gasparov, M. S. “Hydrodynamics and vibroacoustics of aviation combined pump units” diss. cand. tech. sciences / Gasparov Markar Sergeevich. – Samara, 166 p., **2006**.

[16] Stroganova, N. V. “Automated modeling system for gear hydraulic pumps and their diagnostic complexes. diss. cand. tech. sciences / Stroganova Nadezhda Vasilievna. – Omsk, 121 p., **2009**.

[17] Kuleshkov Yu., V. “Analysis of experimental studies of instantaneous feed pulsation of a gear pump”, / Kuleshkov Yu.V., Rudenko T.V., Krasota M.V., Kuleshkova K.Yu. // Foreign state international scientific and technical collection "Construction, design and operation of agricultural machines". – Kirovograd, KDTU – no. 43, part ii – p. 134 – 148, **2013**.

[18] Artobolevsky, I. I. “Theory of mechanisms and machines” / Artobolevsky I. I. - [4th ed. revised and add.] – M.: Science. Ch. ed. Phys.-Math. lit., 640 p., **1988**.

[19] Bashta, T. M. “Volumetric pumps and hydraulic engines of hydraulic systems”, / Bashta T.M. – M.: Mashinostroenie, 606 p., **1974**.

[20] Kuleshkov, Y. V., Rudenko, T. V., Krasota, M. V., Bošanský, M., Tóth, F. “Performance features of tooth gearing in gear hydraulic machines”, Acta technologica agriculturae 2. Nitra, Slovak University Agriculture in Nitra, pp. 84 – 91, **2021**.