

Gear of Hydraulic Pumps for Ship Equipment

Gear of hidrauličke pumpe za brodsku opremu

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Summary

The aim of this article is to analyse the involute gear of a ship's gear pump. The gear applied in a direct gear is usually designed to comply with requirements for sustainability, reliability and operability under unfavourable working conditions. These requirements can be achieved by adjusting the axis distance, angular correction, as well as by rearranging the pressure angle. These modifications are directly interconnected.

KEY WORDS

hydraulic gear pump, direct front cogging, gear adjustment, angle adjustment, pulsation

Sažetak

Cilj ovog članka je analizirati spiralni mehanizam brodske pumpe. Mehanizam se primjenjuje kod direktne opreme koja je obično dizajnirana da bi udovoljila zahtjevima održivosti, pouzdanosti i operabilnosti u nepovoljnim radnim uvjetima. Ovi zahtjevi se mogu postići poravnanjem udaljenosti osovine, kutnim ispravkom i jednako tako preuređenjem kuta pritiska. Ove modifikacije su izravno međusobno povezane.

KLJUČNE RIJEČI

hidrauličke pumpe opreme, direktni prednji zubac zupčanika, podešavanje opreme, podešavanje kuta, pulsacija

1. INTRODUCTION

Modern ship equipment contains hydraulic pumps that offer considerable advantages over mechanical energy transfer. For consistencies sake, basic pumps and standard sizes have been introduced. Producers are mainly interested in the price, technical parameters, noise level, reliability, and sustainability. The design and production of modern hydraulic pumps requires large amounts of capital investment in advanced technologies. To determine whether a possible capital investment is worthwhile it is appropriate to assess the financial situation of the company, for example by using financial ratios or different benchmarking methods [1]. In order to make returns on this investment modern hydraulic pumps are therefore produced in long series. During the design process close attention is paid to the results of basic and applied research, which have been proven in practical experiments. In practice, gear pumps are much more affordable in price than axis pumps, although the parameters are more difficult to regulate. The parameters are influenced by the engine revs. Gear pumps are the most commonly used rotary hydrostatic pumps. The first mention of a gear pump dates back to 1600 and is ascribed to J. Keppeler [2]. The next reference dates to 1604 and is ascribed to a Prague clockmaker, J. Buergi. However, the gear pump first started to be effectively used by Pappenheim in 1636. Gear pumps are popular for their simple construction, sustainability and reliability. They do not carry sensitive distribution elements or even inflow.

Of importance is their reversibility (the pump may function in the pump and/or engine mode). Salem [3] dealt with the influence of the geometrical parameters of cogging and their output. As a result, an adjusted gear mechanism can cope with

a smaller pulsation inflow than a standard non-adjusted one. An alternative method for the analysis of cogging with inclined cogs was introduced by Kapelevich and Kleiss [4]. The layout included non-symmetrical cogs and a gear mechanism whereby the number of cogs amounted to . Manring and Kasaragadda examined the influence of a different number of cogs and came to the conclusion that the ratio could be smaller when , and the pulsation inflow of hydraulic oil is reduced by half. The ratios of strength change on the cogging shaft bearings. The number of cogs, angle meshing, and modulus of the maximum hydraulic liquid inflow were optimized by Huang and Chen. Also of importance is the pulsation pressure of the output. The issue was investigated by Klarecil, Rabsztyn and Hermańczyk [5]. They carried out an experiment to measure the dependency of the pulsation pressure on the shaft revs on the basis of frequency analysis. The parameters were verified in a mathematical-physical model created by G. Dalpiaz [6]. The conclusion was drawn that the model functioned as a cog pump noise-level and vibration indicator.

2. METHODOLOGY

The basic inflow parameters for a gear pump are dependent on the revs and geometrical parameters. For a standard gear this can be expressed as follows [7]:

$$Q = f(z, m, h_a^*, c^*, \alpha, n, \dots) \quad (1)$$

where

Q (l/min)	flow rate of the pump
z (-)	number of teeth
m (mm)	module

- h_a^* (-) addendum module
- c^* (-) bottom depth coefficient
- α (°) pressure angle on the reference circle
- n (rpm) rotation speed of the pump

The inflow parameters for an adjusted non-standard gear can be expressed as follows [8]:

$$Q = f(z, m, h_a^*, c^*, \alpha, x, \Delta y, n, \dots) \quad (2)$$

where

- x (-) correction factor
- Δy (-) arrangement coefficient

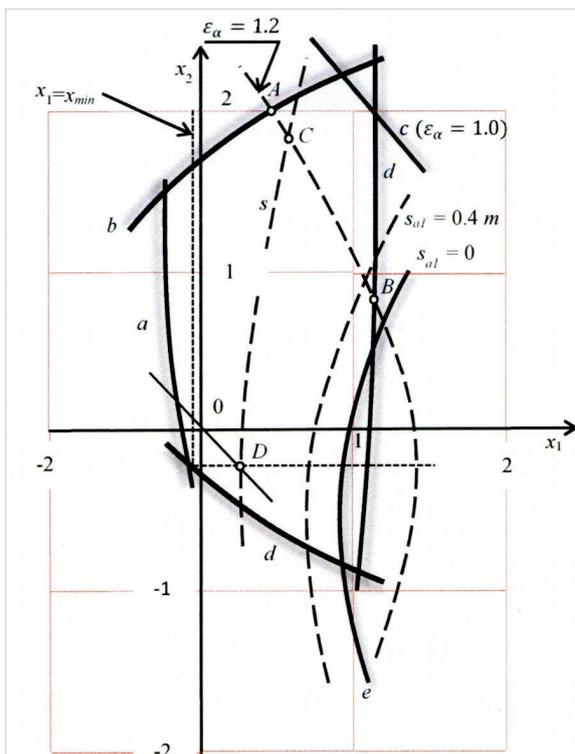
Two types of corrections can be made in practice to improve the parameters. Firstly, angle correction. Secondly, rearrangement. By rearranging an adjusted gear mechanism to a positive or negative direction the following could be achieved [2,5]:

- exact axis distance;
- prevention of cog wear - cog wear is often the result of a small number of cogs, whereby the coefficient reduces contact ratio and reduces cog sustainability;
- prevention of cog angle being too sharp;
- prevention of production and assembly interference with the cog;
- contact ratio enhancement;
- specific slide enhancement;
- noise-level and cogging vibration reduction;
- increase in mechanical efficiency;
- enhancement of cogging touch, flexion, seizure, wear and sustainability.

Cog wear may be prevented by the enlargement of the pressure angle. The minimum number of cogs may be reduced as part of the angle adjustment. As a result, the cogs should

not wear. As a matter of fact, in order for the angle of the cogs to not be too sharp it is limited to 32°. On the other hand, the disadvantage of the angle adjustment is the necessity for a special tool with an abnormal meshing angle. Such an adjustment can be used on cogwheels for which the production is on a large-scale [7].

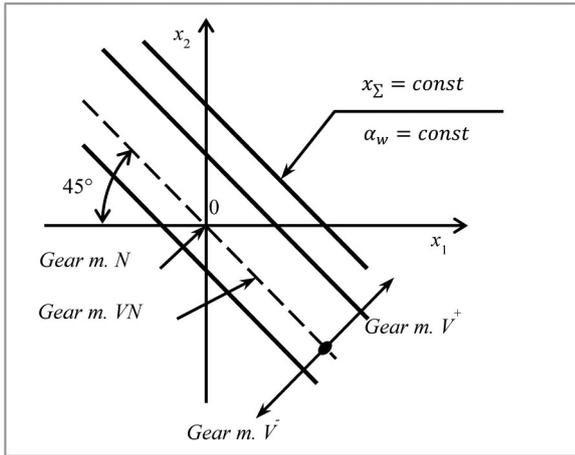
- *Interference* means a limited condition where the profiles of the cog meshing simultaneously engage a certain part of the space – overlap and collide with each other.
- *Interference with a transition curve* comes when the head of the cog collides with the transition curve of another cog (this curve represents a side of a cog without a cut).
- *Head Interference* means a cog head collision of the cogwheels.
- *Operation Interference* happens when a cogwheel does not delineate its trajectory; however it follows the transition curve of the cooperating cogwheel. The interfering cogwheels carry a different transmission ratio. Hence, an additional dynamic strain occurs.
- *Production Interference* means a head collision of the cogwheel with the transition curve of the tool. In fact, the production tools are designed in order to rule out any possible production interference. On the other hand, production interference may occur as long as the size of the cogging heads has been significantly changed. On the whole, the interference collision does not seriously endanger the functioning of the gear mechanism; it only in effect reduces the head size.
- As a matter of fact, there is no possibility to insert one cogwheel into another during *assembly interference*. This sort of interference may be carried out as long as it is possible to install the sprocket axially. However, in case of *assembly interference*, it is advisable to remove the adjustment.



Source: author

Figure 1 Limit Adjustment Graph

- curve a – gear mechanism with the maximum acceptable cut of the sprocket cog
- curve b – gear mechanism endangered by the interference on the transition curve of the cogs
- curve c – gear mechanism with limited value of the meshing coefficient $\epsilon_\alpha = 1$
- curve d – gear mechanism on the brink of interference with the transition curve of the sprocket cogs
- curve e – gear mechanism of which the sprocket borders with sharpness
- curve s – gear mechanism with equal values of the specific slides within the marginal points of the meshing line segment
- point A – suggested gear mechanism requires increased contact strength of the cog sides (resistance to pitting)
- point B – suggested gear mechanism requires increased flexibility strength of the cogs
- point C – suggested gear mechanism requires increased resistance to the wear of the cog sides; point C is situated on the curve of the equal specific slides. The equal specific slide may be achieved by the usage of a VN gear mechanism
- point D – cogwheels carry equal major adjustments and hold equal specific slides; VN gear mechanism.



Source: author

Figure 2.1. Adjustment Graph

The gear mechanism carrying [9,10] device arrangement $x_1 = x_2 = 0$ does not bear adjustment – non-adjusted gear mechanism N . The axial distance of the shafts is $a_w = a = \frac{m(z_1 + z_2)}{2}$, the meshing angle is $\alpha_w = \alpha$, and the simultaneous distance sprocket circles are equal $d_{w1} = d_1$, $d_{w2} = d_2$.

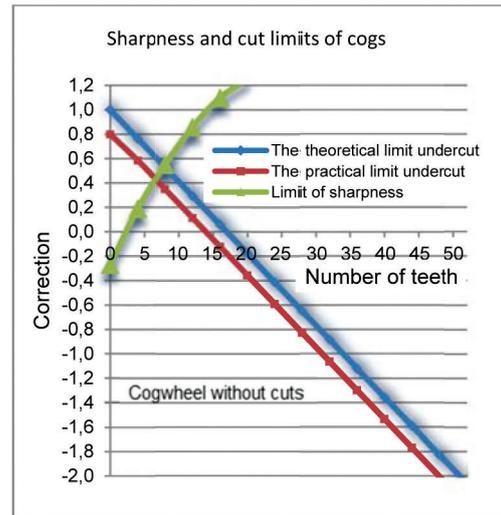
The gear mechanisms $x_\Sigma = x_1 + x_2 \geq 0$ are V gear mechanisms. These gear mechanisms bear the coefficient x_Σ and can be expressed in the following equations [8,11]:

$$x_\Sigma > 0, \text{ then } \alpha_w > \alpha, \alpha_w > \alpha,$$

$$x_\Sigma < 0, \text{ then } \alpha_w < \alpha, \alpha_w < \alpha,$$

$x_\Sigma = 0$, then $\alpha_w = \alpha$, $\alpha_w = \alpha$. where $x_\Sigma = 0$, $x_1 = -x_2 \neq 0$, the gear mechanism is VN . For such gear mechanisms, the rolling circle is identical to the distance circles.

The coefficient of the meshing duration of two united wheels defines the meshing continuity and is defined as the ratio of the length of the meshing segment line to the distance of the basic



Source: author

Figure 2.2. Sharpness and cut limits of cogs

circle for the adjusted gear mechanism V [2]:

$$\varepsilon_{\alpha(V)} = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \sqrt{d_{a2}^2 - d_{b2}^2} - 2a_w \sin \alpha_w}{2\pi m \cos \alpha} \quad (-) \quad (3)$$

The coefficient of the meshing duration of two united wheels defines the meshing continuity and is defined as the ratio of the length of the meshing segment line to the distance of the basic circle for the adjusted gear mechanism V [7]:

$$\varepsilon_{\alpha(N)} = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \sqrt{d_{a2}^2 - d_{b2}^2} - 2a \sin \alpha}{2\pi m \cos \alpha} \quad (-) \quad (4)$$

The non-standard g.m. consists of the maximum meshing coefficient [6]:

$$\varepsilon_{\alpha \max} = \frac{4}{\pi \sin 2\alpha}; \quad \varepsilon_{\alpha \max} \quad (-) \quad (5)$$

For the theoretical values of the meshing coefficient, the radii of the cooperating wheels would have to be interminably large. Table 3 shows the results for two strained racks for the angles given.

Table 1 Dimensional calculation of a front external gear mechanism with direct cogs; type N with zero arrangement (non-adjusted g. m.) [2,7]

Title	Sprocket	Wheel
Number of teeth	Z_1	Z_2
Normal module	m	
Basic profile parameters		
Pressure angle	α	
Addendum coefficient	h_a^*	
Coefficient of dedendum	h_f^*	
Bottom clearance coefficient	c^*	
Radius coefficient of circle roundness	ρ_f^*	
Correction ratio	x_1	x_2
Centre distance	$a = \frac{1}{2}m(z_1 + z_2)$	
Diameter of reference circle	$d_1 = mz_1$	$d_2 = mz_2$
Diameter of base circle	$d_{b1} = d_1 \cos \alpha$	$d_{b2} = d_2 \cos \alpha$
Diameter of addendum circle	$d_{a1} = d_1 + 2mh_a^*$	$d_{a2} = d_2 + 2mh_a^*$
Diameter of dedendum circle	$d_{f1} = d_1 - 2m(h_a^* + c^*)$	$d_{f2} = d_2 - 2m(h_a^* + c^*)$
Addendum	$h_{a1} = mh_a^*$	$h_{a2} = mh_a^*$
Dedendum	$h_{f1} = m(h_a^* + c^*)$	$h_{f2} = m(h_a^* + c^*)$
Tooth depth	$h_1 = h_{a1} + h_{f1}$	$h_2 = h_{a2} + h_{f2}$
Circular pitch	$p = \pi m$	

Table 2 Dimensional calculation of a front external gear mechanism with direct cogs; type V with non-zero arrangement (adjusted g.m.) [2,7]

Title	Sprocket	Wheel
Number of teeth	Z_1	Z_2
Normal module	m	
Basic profile parameters		
Pressure angle	α	
Addendum coefficient	h_a^*	
Coefficient of dedendum	h_f^*	
Bottom clearance coefficient	c^*	
Radius coefficient of circle roundness	ρ_f^*	
Correction ratio	x_1	x_2
Involute operating pressure angle	$inv \alpha_w = 2tg\alpha \frac{x_1 + x_2}{Z_1 + Z_2} + inv \alpha$	
Centre distance	$a = \frac{1}{2}m(Z_1 + Z_2)$	
Operational axis distance	$a_w = a + m(x_1 + x_2 - \Delta y)$	
Diameter of reference circle	$d_1 = mz_1$	$d_2 = mz_2$
Diameter of base circle	$d_{b1} = d_1 \cos \alpha$	$d_{b2} = d_2 \cos \alpha$
Diameter of addendum circle	$d_{a1} = d_1 + 2m(h_a^* + x_1 - \Delta y)$	$d_{a2} = d_2 + 2m(h_a^* + x_2 - \Delta y)$
Diameter of dedendum circle	$d_{f1} = d_1 - 2m(h_a^* + c^* - x_1)$	$d_{f2} = d_2 - 2m(h_a^* + c^* - x_2)$
Addendum	$h_{a1} = m(h_a^* + x_1 - \Delta y)$	$h_{a2} = m(h_a^* + x_2 - \Delta y)$
Dedendum	$h_{f1} = m(h_a^* + c^* - x_1)$	$h_{f2} = m(h_a^* + c^* - x_2)$
Tooth depth	$h_1 = h_{a1} + h_{f1}$	$h_2 = h_{a2} + h_{f2}$
Circular pitch	$p = \pi m$	
Diameter of pitch circle	$d_{w1} = mz_1 \frac{\cos \alpha}{\cos \alpha_w}$	$d_{w2} = mz_2 \frac{\cos \alpha}{\cos \alpha_w}$

Table 3 Maximum values of the meshing coefficient for a non-adjusted g.m. N

Meshing angle(°)	Meshing duration length
15.0	2.5465
17.5	2.2198
20.0	1.9808
22.5	1.8006
25.0	1.6621
27.5	1.5543
30.0	1.4702
32.5	1.4049

Source: author

The minimum number of cogs necessary for the production of an external cogged tool in the shape of a cogwheel (Fellow) is given by [2,5]:

$$z_{min} = \sqrt{z_0^2 + \frac{4h_a^*}{\sin^2 \alpha} (z_0 + h_a^*)} - z_0 \quad (-) \quad (6)$$

where z_0 signifies the number of the cogs of the device (Fellow).

The specific slide of the sprocket is given by [3,7]:

$$\vartheta_1 = 1 - \frac{d_{b2}}{d_{b1}} \cdot \frac{1}{i} \quad (-) \quad (7)$$

where $i = \frac{Z_2}{Z_1}$ signifies the transition ratio.

The specific slide of the wheel is given by [8]:

$$\vartheta_2 = 1 - \frac{d_{b1}}{d_{b2}} \cdot i \quad (-) \quad (8)$$

The specific slide of the sprocket base is given by [8,10]:

$$\vartheta_{A1} = 1 - \frac{\sqrt{r_{a2}^2 - r_{b2}^2}}{a_w \sin^2 \alpha - \sqrt{r_{a2}^2 - r_{b2}^2}} \cdot \frac{1}{i} \quad (-) \quad (9)$$

The specific slide of the wheel head is given by:

$$\vartheta_{A2} = 1 - i \frac{a_w \sin^2 \alpha - \sqrt{r_{a2}^2 - r_{b2}^2}}{\sqrt{r_{a2}^2 - r_{b2}^2}} \quad (-) \quad (10)$$

The specific slide of the sprocket head is given by [2,3]:

$$\vartheta_{B1} = 1 - \frac{a_w \sin^2 \alpha - \sqrt{r_{a1}^2 - r_{b1}^2}}{\sqrt{r_{a1}^2 - r_{b1}^2}} \cdot \frac{1}{i} \quad (-) \quad (11)$$

Specific slide of the wheel base is given by [12]:

$$\vartheta_{B2} = 1 - i \frac{\sqrt{r_{a1}^2 - r_{b1}^2}}{a_w \sin^2 \alpha - \sqrt{r_{a1}^2 - r_{b1}^2}} \quad (-) \quad (12)$$

The total of the specific slides is the total of the single absolute values of the specific slides as given by:

$$\sum |\vartheta| = |\vartheta_{A1}| + |\vartheta_{A2}| + |\vartheta_{B1}| + |\vartheta_{B2}| \quad (-) \quad (13)$$

The theoretical inflow of a hydraulic pump for a standard cogging – non-adjusted cogging N with an equal number of cogs within the wheel and sprocket is given by:

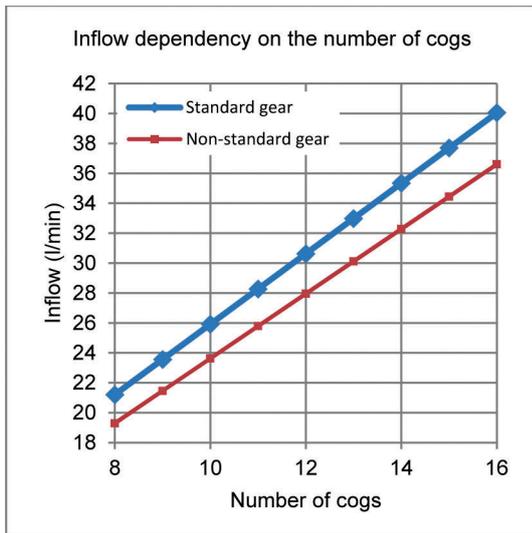
$$Q_N = 2\pi b m^2 \left[z + \left(1 - \frac{\pi^2 \cos^2 \alpha}{12} \right) \right] \cdot 10^{-6} \quad (l/min) \quad (14)$$

The theoretical inflow of a hydraulic pump for a non-standard cogging – adjusted cogging N with an equal number of cogs within the wheel and sprocket is given by [3,13]:

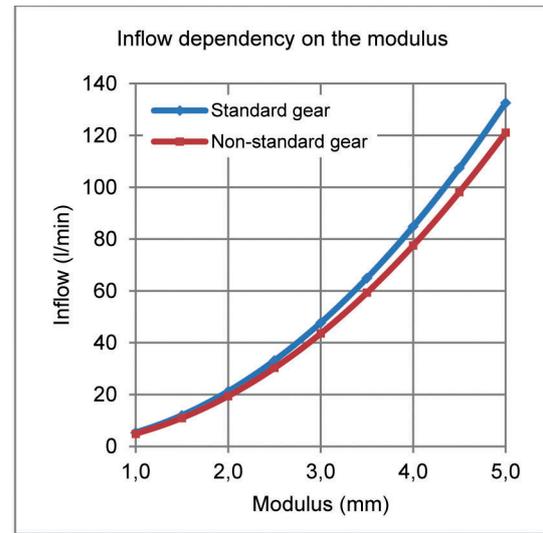
$$Q_V = \pi b n \left[\left(m z + x z + m - m x \frac{z}{2} - \frac{m z^2}{2} \right)^2 - \left(m \frac{(z+2xy)}{2} \right)^2 - \frac{m^2 \pi^2 \cos^2 \alpha}{6} \right] \cdot 10^{-6} \quad (l/min) \quad (15)$$

The pulsation inflow factor for a non-adjusted cogging N with an equal number of cogs within the wheel and sprocket is given by:

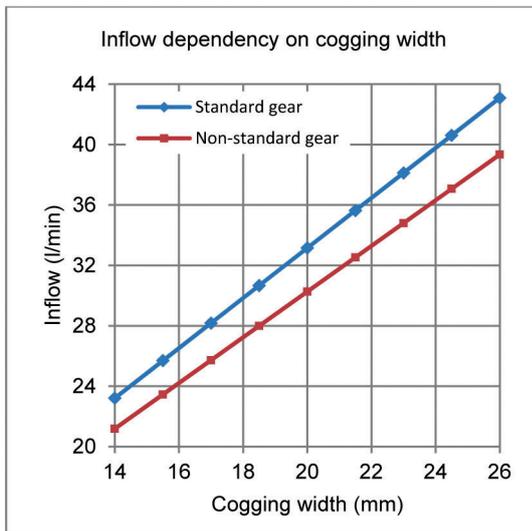
$$\delta_N = \frac{\pi^2 \cos^2 \alpha}{4 \left(z + 1 - \frac{\pi^2 \cos^2 \alpha}{12} \right)} \quad (-) \quad (16)$$



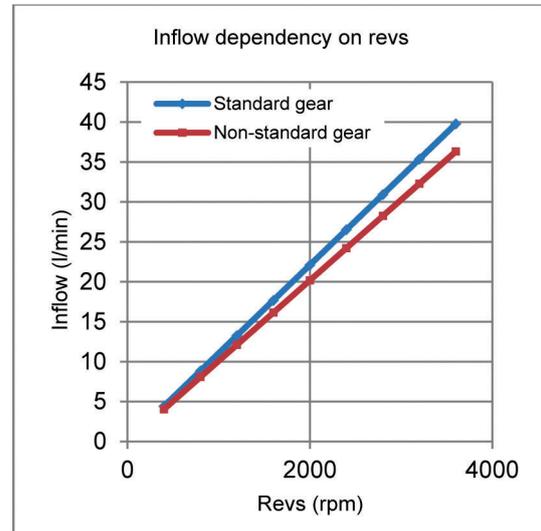
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Figure 3 Inflow dependency on the number of cogs



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Figure 4 Inflow dependency on the modulus



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Figure 5 Inflow dependency on cogging width



Source: author
Figure 6 Inflow dependency on revs

The pulsation inflow factor for an adjusted cogging N with an equal number of cogs within the wheel and sprocket is given by [5.9]:

$$\delta_V = \frac{\pi^2 \cos^2 \alpha}{4 \left[\left(mz + mx + mx_{\Sigma} - \frac{mz}{2} \right)^2 - \left(m \frac{z + x_{\Sigma}}{2} \right)^2 - \frac{m^2 \pi^2 \cos^2 \alpha}{12} \right]} \quad (17)$$

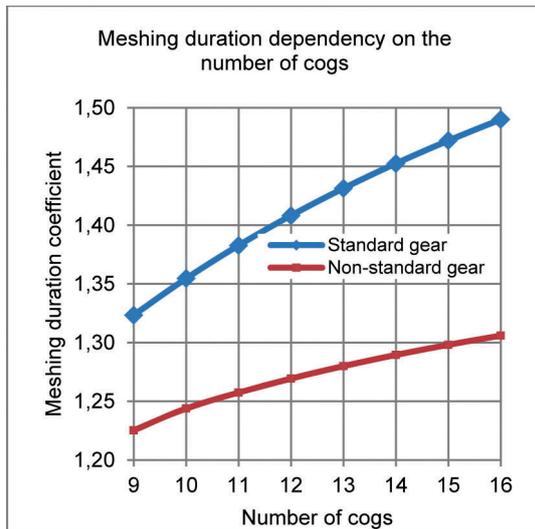
3. RESULTS

A standard gear with parameters $\alpha = 20^\circ$, $x_1 = x_2 = 0$ will be considered in the analysis. The non-standard gear has the following parameters: $\alpha \neq 20^\circ$, $x_1 \neq x_2$, $x_{\Sigma} \neq 0$, $z_2 = 12$, $x_1 = x_2 = -0.08$.

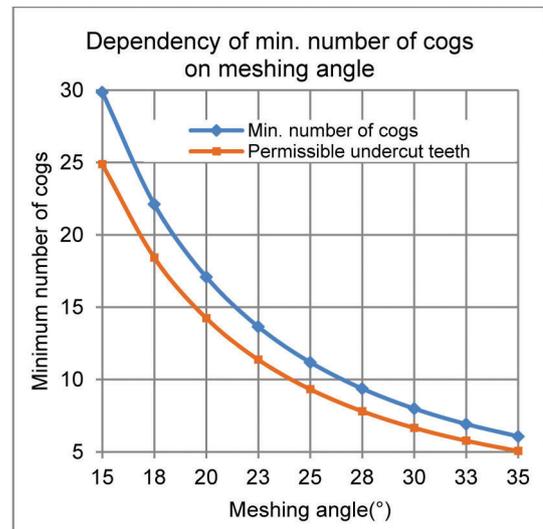
The calculation and graphic interpretation of the parameters for the inflow of the hydraulic liquid Q : the number of cogs, modulus, the width of the cogging and revs, has been depicted in the Figures 3 and 6. The inflow $Q = 30 \text{ l/min}$ complies with

the number of cogs – 13. The following calculations consider cogging $z_1 = 13$ and $z_2 = 12$ adjusted by $x_1 = x_2 = -0.08$ and maximum revs $n = 3000 \text{ rpm}$. The inflow of 30 l/min complies with the modulus $m = 2.5 \text{ mm}$ where the inflow dependency Q is non-linear (see Figure 4). The inflow $Q = 30 \text{ l/min}$ corresponds to the width of the cogging $b = 20 \text{ mm}$ (see Figure 5). The sprocket revs $n = 3000 \text{ rpm}$ corresponds to the hydraulic liquid inflow $Q = 30 \text{ l/min}$ (see Figure 6).

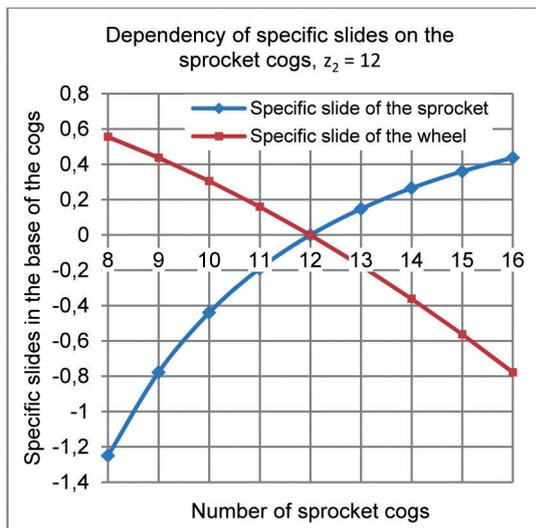
When comparing the inflow dependency Q of the hydraulic liquid to the parameters z , m , b , n , it is evident that the standard gear mechanism N has a heavier inflow than that of the non-standard one. This statement is further testified by the negative adjustment V (machine tool is inserted into the workpiece during production).



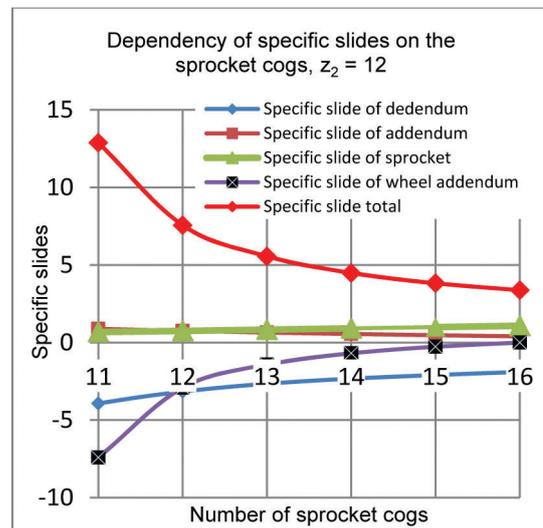
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Figure 7 Meshing duration dependency on the number of cogs



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Figure 8 Dependency of min. number of cogs on meshing angle



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Figure 9 Dependency of specific slides sprocket cogs



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Figure 10 Dependency of specific slides on the sprocket cogs

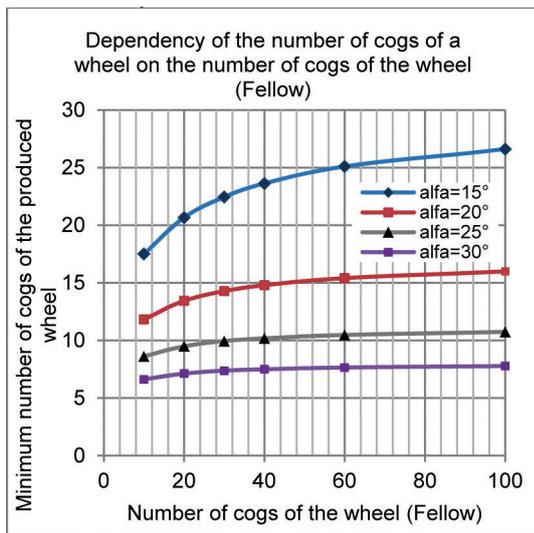
Resources claim that the minimal hypothetical number of cogs for $x_1 = x_2 = 0$ and $\alpha = 20^\circ$ equation of a non-adjusted gear mechanism N amounts to $z_{1min} = 17$ where the minimal actual number amounts to $z'_{1min} = 14$. However, in this case the number amounts to $z_1 = 13 < z'_{1min}$ otherwise undesirable wear of the cogs is likely to happen causing a reduction in the base strength of the cog. The dependency of the meshing angle on the hypothetical number of cogs z_{1min} and the actual number $z'_{1min} = 14$ with the acceptable wear is depicted in Figure 8.

The specific slides in the base of the sprocket ϑ_{A1} , in the head of the wheel ϑ_{A2} , in the head of the sprocket ϑ_{B1} in the base of the wheel ϑ_{B2} and the total of the specific slides $\sum|\vartheta|$ are depicted in Figure 9.

The specific slides in the base of the sprocket ϑ_1 and the wheel ϑ_2 according to the number of cogs of the sprocket z_1 on the invariable number of cogs of the wheel $z_2 = 12$ are depicted in Figure 10.

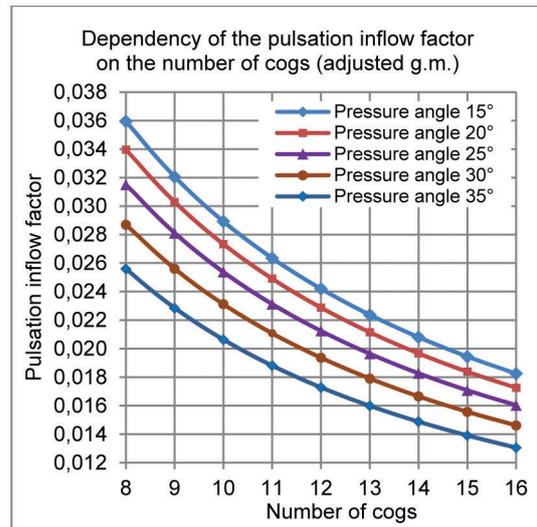
Screw cutters or rack-shaped machine tools are used for cogging production. Since the tools for the production of external cogging are commonly used in practice, the minimum number of cogs must be determined according to this kind of production. Figure 11 shows the dependency of the number of cogs of the machined wheel on the number of cogs of the wheel (Fellow) for pressure angle $\alpha = (15^\circ, 20^\circ, 25^\circ, 30^\circ)$. The figure asserts that it is possible to reduce the number of cogs of the production wheel by increasing the meshing angle. However, it is even possible to reduce the number of cogs of the sprocket.

The pump is characterised by high noise levels and a pulsating variable inflow. Figure 12 shows the dependency of the pulsation inflow factor on the number of cogs for varying meshing angles. The procedure confirms that the number of cogs can be doubled. Afterwards, the pulsation inflow factor decreases by 20% - 30%.



Source: author

Figure 11 Dependency of the number of cogs a wheel the number of cogs of the wheel (Fellow)



Source: author

Figure 12 Dependency of the pulsation inflow on of factor on the number of cogs

4. SUMMARY

According to the methodology and the analysis, rotary front cogging can be applied to a gear pump in order to obtain optimal parameters. The analysis was processed using Excel. The results have been presented numerically and graphically. It is therefore an effective tool with which to achieve good results, and in which alternative parameters can be processed.

5. CONCLUSION

The aim of this article was to define the optimal parameters for cogging. The analysis was carried out on the basis of accurate calculations in the DS TECHNIK program. The test proportions of the cogging, as well as the limit deviation of the sprocket and wheel were prepared in accordance with ISO 1328-1 and -2, measuring cogs, pellets, cylinders. The following elements of the stress on the sprocket and wheel were calculated: touch stress, touch stress within the one-time, heaviest strain, flection stress within the one-time, heaviest strain. When taking these parameters into consideration, the cogged gear mechanism meets the requirements.

REFERENCES

[1] Vochozka, M. Development of Methods for Comprehensive Evaluation of Business Performance. *Politická ekonomie*, 2010, (5), 675-688.

[2] Boháček, F. et al. Části a mechanismy strojů III. Převody. VUT Brno publishing, 55-618-87, p. 267 (1987).

[3] Saleem, A.M. Effect of Tooth Geometry on Gear Pump Performance. *Tikrit Journal of Eng. Sciences/Vol.16/No.4/December 2009*, (20-27).

[4] Kapelevich, A.L., Kleiss, R.E. Direct Gear Design for Spur and Helical Involute Gears. *Gear Technology*, September/October 2002, (29 – 36).

[5] Klarecil, K., Rabsztyń, D., Hermańczyk, M. Influence of setting the selected parameters of hydraulics systems on pressure pulsation gear pumps. *Diagnostyka*, Vol. 16, No. 2, p. 49-54 (2015).

[6] Dalpiaz, G., A. Fernández del Rincón, Mucchi, E., Rivola, A. Model-based analysis of dynamic phenomena in gear pumps. *Proceedings of 5th. International Conference on Acoustical and Vibratory Surveillance Methods and Diagnostic Techniques*, Senlis, France, (2004).

[7] Málík, L. et al. Části a mechanismy strojov. ALFA publishing, Bratislava, Slovak Republic, ISBN 063-784-88, p. 420, (1989).

[8] Ivantysyn, J., Ivantysynova, M. *Hydrostatic Pumps and Motors*, Tech Books Int., 2003, New Delhi, India.

[9] Huang, K.J., Chen, C.C. Kinematic Displacement Optimization of External Helical Gear Pumps. *Chung Hua Journal of Science and Engineering*, Vol. 6, No. 2, pp. 23-28 (2008).

[10] Shipley, J.E., Mishke, C.R., Budynas, R.G. *Mechanical engineering Design*. Seventh Edition, 2004, The McGraw-Hill, Inc., 2004, p. 1159.

[11] Noah D. Manring, Suresh B. Kasaragadda. Suresh B. Kasaragadda. *Journal of Dynamic Systems, Measurement, and Control*. Vol. 125, September 2003 (396-404).

[12] Casoli, P., Vacca, A., Berta, G. L. Optimization of Relevant Design Parameters of External Gear Pumps. *Proceedings of the 7th JFPS International, Symposium on Fluid Power, Toyama 2008*, September 15-18, 2008, p. 277-282, ISBN 4-931070-07-X. <http://dx.doi.org/10.18267/j.polek.754>

[13] Lazarz, B., Perun, G. Influence of construction factors on the vibrational activity of the gearing. *Transport Problems*, 2012, Vol. 7, Issue 2, pp. 95-102. ISSN 1896-0596.