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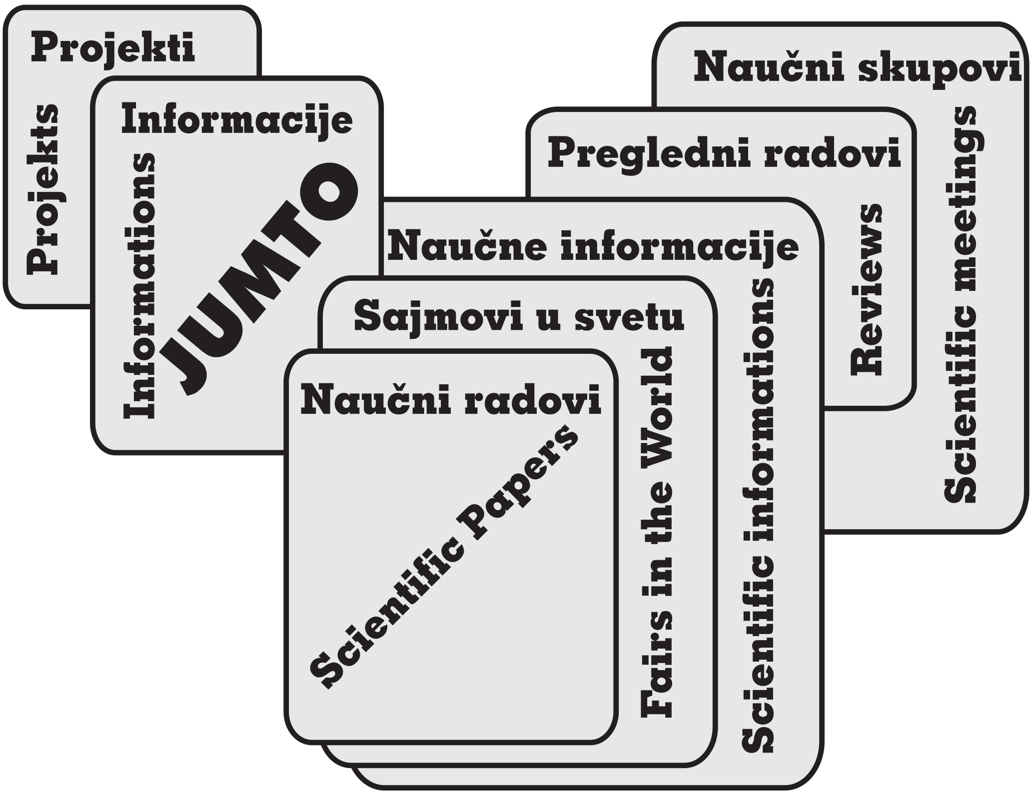

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Scientific paper

OPTIMIZATION OF PARAMETERS ON THE VALVE PLATE FROM AXIAL PISTON PUMPS AND THEIR EFFECT ON THE PRESSURE PULSATION AND NOISE LEVEL

Petrović R.[1], Savić S.[2], Todić N.[2]

SUMMARY

Modern design of the axial piston pump/motor, based on computer aided design, requires description of all processes and parameters in the pump/motor. The hydrodynamic and dynamic processes in the axial piston pump (cylinder, intake and discharge chambers, discharge valve, and high pressure pipe line) are very complex, so they demand a thorough physical and mathematical analysis. Compression losses, which are caused by sudden pressure changes in the cylinders of cylinder block, result in energy loss and increased noise. The compression losses can be decreased by identification of optimal dimensions and shapes of the port at the valve plate.

Key words: Piston pump, geometrical flow section, radial clearance, diameter of cylinder, numbers of cylinders, dynamic viscosity, modulus of elasticity, flow

INTRODUCTION

A review of the axial piston pump with constant pressure variable displacement. These pumps have found wide application in complex hydraulic system in the field of aviation [1]. Demands placed stations in aviation are numerous but the most significant reliability in operation, exceptional efficiency in the supply of highly complex branch system and economical operation and request that the maximum energy saving.

Contents of the the paper, in an attempt to provide further improve the performance of the devices from the structure of the hydro system of the aircraft. In particular, these activities are related to the components that generate certain kinds of power , namely, first of all, pumps and motors [2,3].

Besides causing of the compression losses, changes of the pressure inside the cylinder at short intervals are causing hydraulic impacts in the fluid that spread through the housing and cause the noise. There is a tendency of development of new pumps with constant increase of the operating pressure, which results in increase of the efficiency coefficient while increasing the noise. It is interesting to study what possibilities exist to reduce the noise while the pressure

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rises at the same time. One way is selection of optimal angles of suction and suppression openings on valve plate to obtain certain angles – compression angle and expansion angle.

MATHEMATICAL MODELING

Axial piston pumps with constant pressure and variable flow have extraordinary possibilities for controlling the flow by change of pressure [4,5,6]. Owing to pressure feedback, volumetric control of the pump provides a wide application of these pumps in complex hydraulic systems, particularly in aeronautics and space engineering. Some dynamic processes developing in control cycle have been analyzed by mathematical model and a large number of process simulations have been done by means of programming language Matlab. On that occasion some diagrams have been made which have been a basis for analysis of time-constants of transient and it has been concluded that the characteristics comply with the requirements defined by the aeronautics standards.

Figure 1a. shows a schematic view of an axial piston pump with swash plate and with shown angles on valve plate α_k and α_e .

The following equation satisfies the compression and expansion of the fluid:

$$\frac{dV}{dp} = -\frac{V}{E} \tag{1}$$

where:

E – compressibility module

V – volume of the working fluid in the cylinder.

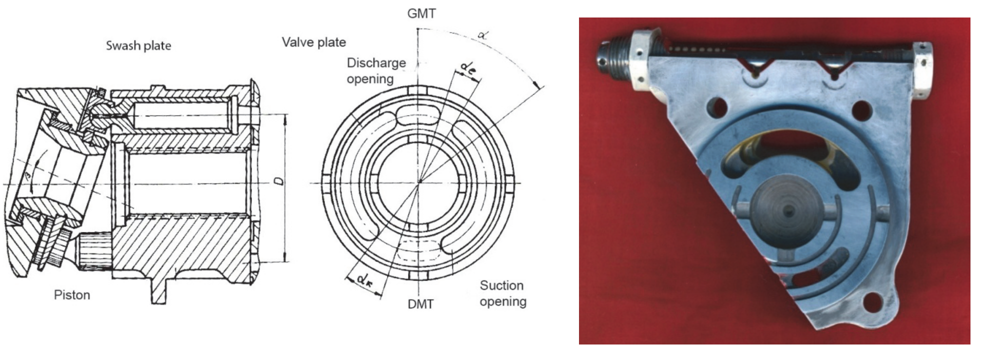

Fig. 1.a) axial piston pump with swash plate, b) detail of valve port plate

The volume of the working fluid in the cylinder of the cylinder block can be expressed by the equations:

$$V = V_0 + RA\,\mathrm{tg}\,\beta_{max} - RA\,\mathrm{tg}\,\beta\ \cos\alpha \tag{2}$$

where: R=D/2- radius of divider circle of the cylinder block

A - surface of the piston

The cycle of compression and expansion of the fluid in cylinders is shown in the diagram, points ABCD. At point B the pressure of the fluid in the cylinder rises rapidly and reaches a discharge value, while at point D the pressure begins to decrease rapidly and reaches a filling value. The shock waves that cause noise occur because of these rapid pressure changes. In the case of right choice of compression angle, fluid in the cylinder is compressed and then

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discharged into suppression opening, point F. The shock wave is avoided in this way since the fluid is compressed to the pressure that exists in the suppression opening. In the same way, by choosing the correct angle of expansion, the pressure in the point D can be reduced to the pressure in the point E. In the first case when $\propto_k = \propto_e = 0$, the required mechanical work is equal to the surface of the rectangle ABCD, and in the second case the required work is equal to the surface EBFD. By analyzing this we find that in both cases the generated hydraulic energy is represented with surface EBFD. In case when $\propto_k = \propto_e = 0$, the higher work is required which is spent on compression losses to a certain extent. Compression losses in the lower dead point (DMT) are shown with surface of the triangle BCF, while losses in the upper dead point (GMT) are shown with surface of the triangle ADE [7].

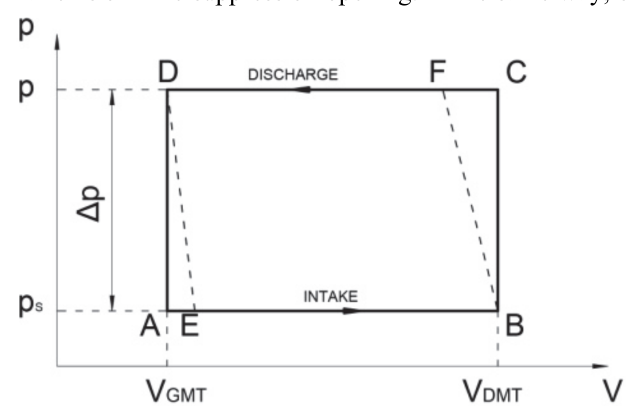

Fig.2. The cycle of compression and expansion of the fluid in cylinders

Using the equations (1) and (2), we get

$$\frac{\Delta p_0}{E} = \frac{V_{GMT} - V_\alpha}{V_{GMT}}; \ \alpha = \pi + \alpha_k \qquad (3)$$

where:

V_{GMT} - The volume of the fluid in the upper dead point.

Δp_0 - The optimum pressure difference at the lower dead point.

V_α - The volume of the working fluid in the cylinder when it is turned for the angle α.

This equation can also be written in another form

$$\Delta p_0 = \frac{ERA\,(1-\cos\alpha_K)}{V + RA\,\mathrm{tg}\,\beta_{max} + RA\,\mathrm{tg}\,\beta} \qquad (4)$$

If the requirement that $\Delta p = \Delta p_b$ is fulfilled, then the losses do not occur in the upper dead point for the expansion angle $\propto_e$.

Analogously to equation (4), the equation (5) is obtained:

$$\Delta p_b = \frac{ERA\,(1-\cos\alpha_e)}{V_0 + RA\,\mathrm{tg}\,\beta_{max} + RA\,\mathrm{tg}\,\beta} \qquad (5)$$

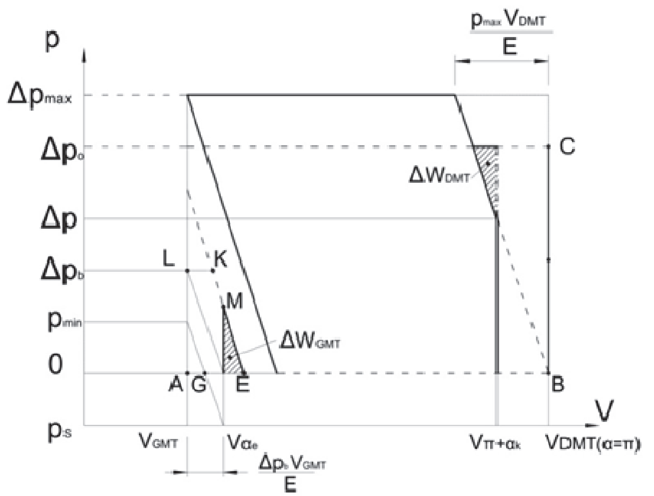

Fig.3.Diagram $\Delta p \geq \Delta p_{min}$

The compression losses

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in the upper dead point for the pressure difference Δp will be:

$$W_{GMT} = \frac{1}{2}(\Delta p - \Delta p_b)^2 \cdot \frac{V_{GMT}}{E} = \frac{1}{2E}(\Delta p - \Delta p_0)^2 \cdot (V_0 + RA\,\mathrm{tg}\,\beta_{max} + RA\,\mathrm{tg}\,\beta\,) \quad (6)$$

We can write the equation of the compression losses in the lower dead point:

$$W_{DMT} = \frac{1}{2E}(\Delta p - \Delta p_0)^2 \cdot (V_0 + RA\,\mathrm{tg}\,\beta_{max} - RA\,\mathrm{tg}\,\beta\,) \quad (7)$$

In case $\alpha_e = \alpha_k = 0$, losses can be expressed as:

$$W = W_{GMT} + W_{DMT} = \frac{\Delta p^2}{E}(V_0 RA\,\mathrm{tg}\,\beta_{max}) \quad (8)$$

The relative compression loss in percentage can be expressed by equation:

$$\Delta W[\%] = \frac{\Delta W}{P_{ABCD}}100\% = \frac{1}{2E}\Delta p\left(1 + \frac{V_0}{RA\,\mathrm{tg}\,\beta_{max}}\right) \quad (9)$$

where:

P_{ABCD} - surface defined in Figure 2.

If the pressure difference Δp is below Δpmin, where Δpmin=Δpb-ps, and ps is supply pressure, the under-pressure must not occur in cylinder .

The expansion losses in the lower dead point can be expressed by the following equation:

$$\Delta W_{DMT} = P_{PORS} = \frac{1}{2E}(\Delta p_o - \Delta p)^2 \cdot V_{DMT} - \frac{1}{2E}(\Delta p_b - \Delta p - \mathrm{p_s}) \cdot V_{DMT} = \frac{1}{2E} \cdot V_{DMT} \cdot \mathrm{p_s}(2\Delta p_b - 2\Delta p - \mathrm{p_s}) \quad (10)$$

Diagram of the losses depending of the compression and expansion angles can be obtained by these equations. The pressure differences Δp_o and Δp_b are calculated first, and then $\mathrm{W_{GMT}}$ and $\mathrm{W_{DMT}}$.

EXPERIMENTAL RESULTS

Technical data of axial piston pump with variable flow which has the swash plate with these properties:

$D = 43 \cdot 10^{-3}\ m$ - diameter of divider circle of the cylinder block

$V_0 = 0{,}581 \cdot 10^{-6}\ m^3$ – minimum volume of fluid in the cylinder

$\beta = 19°$, -angle of swash plate

$A = 0{,}785 \cdot 10^{-4}\ m^2$ - surface of the piston

$E = 1{,}542 \cdot 10^3\ MPa$ - compressibility module of fluid AMG10 at temperature t=60° and pressure p=20MPa.

Figure 5 shows a diagram of changes of some losses depending on the change in fluid pressure in the pressure pipe. Compression losses and losses due to the leakage are linearly increasing with the increase of the pressure. On the other hand, the mechanical losses are firstly reduced to a certain limit and then are increasing with increase of the pressure [8,9,10].

Compression losses depend largely on the following values:

- Compressibility module of the working fluid
- The pressure difference
- The minimum volume of the cylinder

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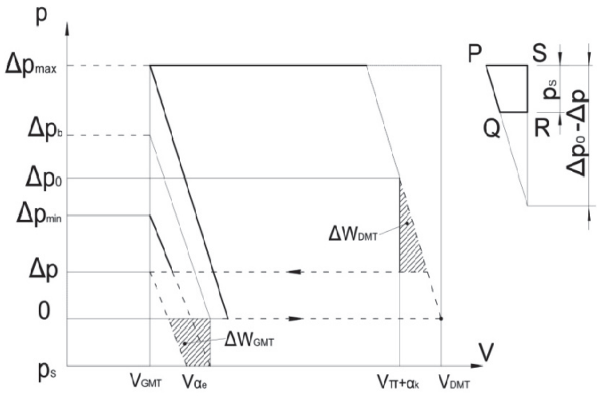

Fig.4. Diagram for $\Delta p \leq \Delta p_{min}$

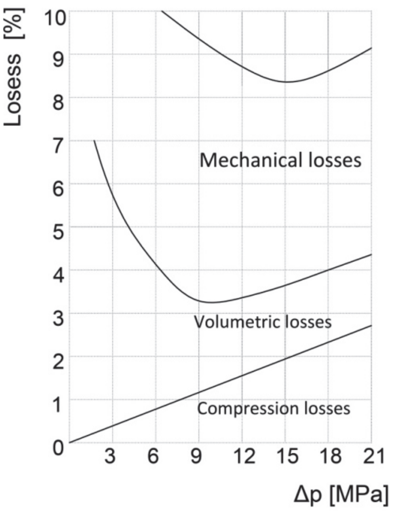

Fig.5. Diagram of some losses of axial piston pump

Minimum volume and compressibility module are taken as a constant, although the compressibility module depends on the temperature and pressure. It is noted that at constant values of the angle of swash plate inclination β and the pressure difference Δp, compression losses can be reduced to minimum by proper choice of angles $\propto_k$ and $\propto_e$. The problem is that inclination angle β and pressure difference Δp are constantly changing during the operation of the pump, so the optimum pre-compression and pre-expansion are not done at certain angles $\propto_k$ and $\propto_e$.

Diagrams ΔWDMT and ΔWGMT for one cylinder and one revolution depending on the angles $\propto_k$ i $\propto_e$ are shown in Figures 6 and 7, as a result of these calculations. It should be noted that any change in the inclination angle of the swash plate affects the change of losses ΔWDMT and ΔWGMT. By doing the analysis it can be determined that the compression loss in GMT is much smaller than in the DMT at higher values of the angle $\propto_e$. The optimal compression angle $\propto_k$ for a specified working range can be chosen from the diagram

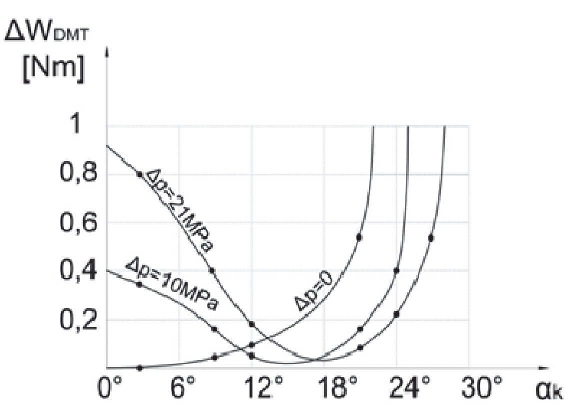

Fig.6.Diagrams of compression losses in DMT for one cylinder and one revolution

shown in Figure 6.

For example, if it is assumed that the angle β=19° and the pressure difference is 0<p<21MPa, then $\propto_k$(opt) =14. For the angle $\propto_k$ =14° compression loss per cylinder and revolution is maximum 0.13Nm, if the pressure difference Δp varies from 0 to 21 MPa. However, if the another angle β is chosen or another range of the pressure difference Δp, then, from the diagram, we can find the optimal value of the compression angle $\propto_k$. Table I shows the compression losses for various values of the inclination angle β and compression angle $\propto_k$. By analyzing the mean values of losses it can be concluded that for certain angles of inclination the compression angle $\propto_k$ has an optimum value which is 14°.

Tab. 1.the compression losses for various values of the inclination angle β and compression angle $\propto_k$

β \ $\propto_k$	10°	11°	12°	13°	14°	15°	16°	17°
19°	0,29	0,24	0,20	0,15	0,13	0,19	0,22	0,29
15°	0,31	0,27	0,22	0,20	0,14	0,12	0,14	0,22
10°	0,31	0,29	0,27	0,24	0,20	0,18	0,19	0,21
5°	0,33	0,33	0,31	0,29	0,28	0,27	0,25	0,22
Average value	0,31	0,28	0,25	0,22	0,18	0,19	0,20	0,23

Figure 7 shows a diagram of losses depending on expansion angle $\propto_e$, at particular inclination angle β, supply pressure ps and minimum volume of the cylinder Vo. It can be seen from the diagram that for large enough expansion angle $\propto_e$ the losses ΔWgmt can be neglected. The optimum angle of expansion $\propto_e$ for above mentioned data is in the range from 12 to 16°. This paper also considers the noise problems and the tests were performed on the same pump where the compression losses were measured. Noise level was measured using a device for measuring the noise with C filter. The microphone was attached to the stand at a distance of 5cm from the frontal area of the valve plate. This place was chosen because the loudest noise occurs in the area of the valve plate, and to rule out the effect of noise from other devices.

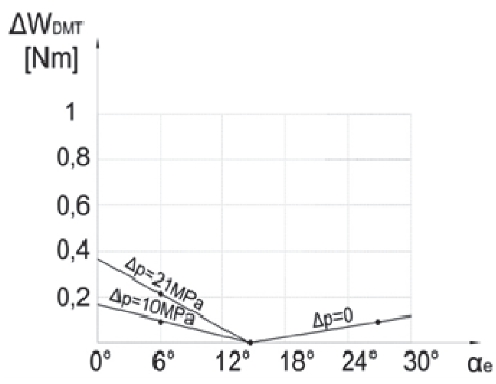

Fig.7.Diagrams of compression losses one cylinder and one revolution

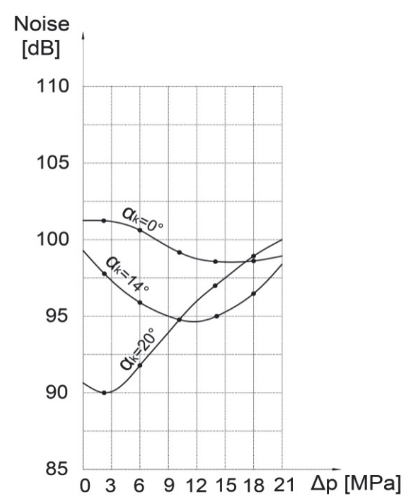

Fig.8.Results of measurements of noise depending on in GMT for the pressure n=1980 min^{-1}, β=19°,$\propto_e$= 0

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Figure 8 shows results of measurements of noise depending on the pressure of the fluid in the pressure pipe. In this case, the expansion angle is $\propto_e = 0$, rotation frequency n=1500 min-1, inclination angle of the swash plate is β=19°, while the compression angle was changed and had values $\propto_{k1} = 0, \propto_{k2} = 14, \propto_{k3} = 20$. It can be concluded that at the lowest pressure and the highest compression angle $\propto_k = 20°$ noise is greater than with the first pump, in which compression angle was $\propto_k = 0°$. If the compression angle is $\propto_k = 14°$, reduction of noise will occur at the pressure of 6MPa.

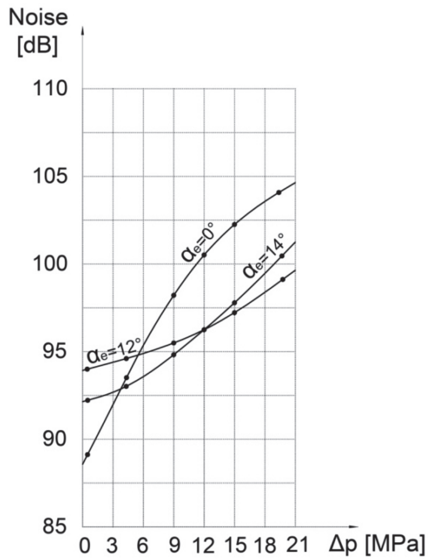

Fig.9. Results of measurements of noise depending on the pressure n=1980 min^{-1}, β=19°,$\propto_k = 0$

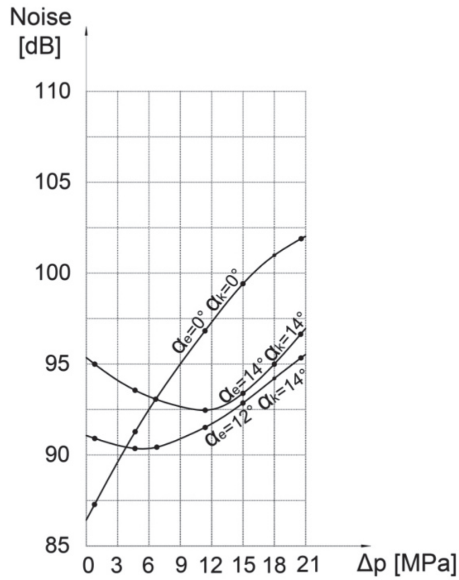

Fig.10. results of measurements of noise depending on the pressure n=1500 min^{-1}, β=19°

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Figure 9 shows results of noise measurements under the same conditions as in the previous case, with the exception in compression and expansion angles $\propto_k$ and $\propto_e$. The compression angle was $\propto_k = 0°$, and the expansion angle was changed and had values $\propto_{e1} = 0, \propto_{e2} = 12, \propto_{e3} = 14°$. By analyzing a diagram it can be concluded that for these test conditions the optimal expansion angle is $\propto_e = 12°$.

Figures 10,11 and 12 show the results of noise measurements depending on the pressure for different rotation frequencies n1=1500min-1,n2=1980min-1,n3=3000min-1 and for the different expansion angles $\propto_{e1} = 0°, \propto_{k1} = 0°, \propto_{e2} = 12°, \propto_{k2} = 14°, \propto_{e3} = 14°, \propto_{k3} = 14°$.

Since the optimal value of the expansion angle $\propto_e = 12°$ is near the optimal value of the compression angle $\propto_k = 14°$,the case when $\propto_e = \propto_k = 14°$ is considered.Symmetry of openings of the pump distribution is present in this case, so the pump can be the two-way with the same valve plate, which reduces the production cost with a slight increase in noise.

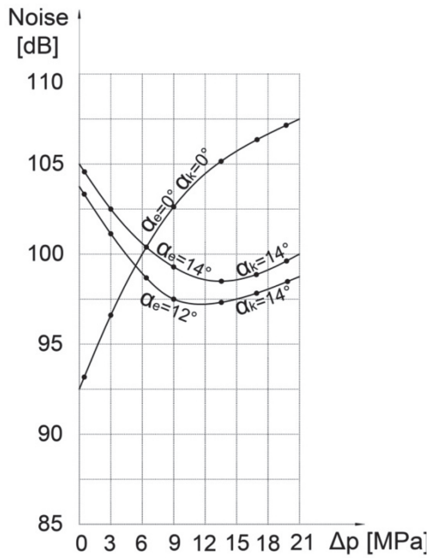

Fig.11 Results of measurements of noise depending on the pressure n=1980 min^{-1}, β=19°

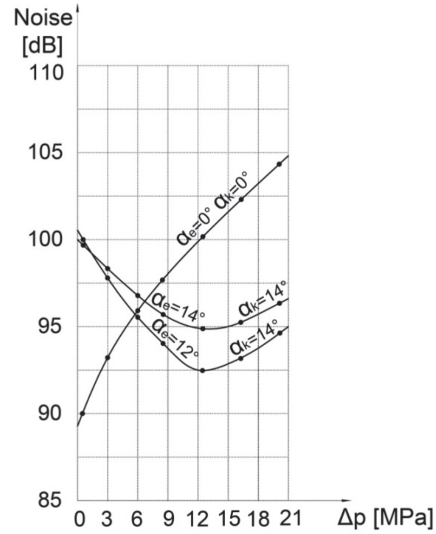

Fig.12.Results of measurements of noise depending on the pressure n=3000 min^{-1}, β=19°

CONCLUSION

The basic fact that producers and users of axial piston pumps must accept is that every increase in the operating pressure leads to the increase in pump noise. The greater influence on the noise level has the increase of rotation frequency of pump's drive shaft than the increase of pressure in the pressure pipe. Since the rotation frequency has a large impact on noise, it is often reduced in order to neutralize the noise caused by increasing pressure. If the rotation frequency is reduced, we have to use larger pump that is heavier and more expensive to remain the same flow. In case of increasing flow two times (which means that the rotation frequency can be reduced two times), the pump price increases by 30%. In addition, the weight increases by approximately 50% and the larger space is required for installation due to increase of built-in measures.

Reducing the rotation frequency in order to reduce the noise has a negative effect also, but if the pump is assembled in places where noise must be as small as possible, then this solution is used.

Cavitations that occur as a result of insufficient supplying of the pump can affect the noise level. Bearings and gears also contribute to the increase of noise. It should also be noted that the choice of appropriate materials can attenuate certain vibrations that also lead to increase of the noise. The pump noise can be minimized if all these factors are taken into account.

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