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INFLUENCE OF AIR CONTENT ENTRAINED IN FLUID OF A VANE PUMP WITH DOUBLE EFFECT OPERATING PARAMETERS

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Preliminary notes

In developing the vane pumps the fundamental basis is experimental research and mathematical modelling of nonstationary hydraulic processes inside the pump, in thrust space and suction and thrust pipeline. By means of experimental research and results of mathematical modelling and software package KRILP, it is possible to determine the parameters of operating processes of vane pumps precisely enough. This research examines the idealized and actual flow ripple of a high-pressure vane within vane type pump. For the idealized case, a "perfect" pump is examined in which the leakage is considered to be zero and the fluid is considered to be incompressible. Based upon these assumptions, expressions describing the characteristics of the idealized flow-ripple are derived. Next, the actual flow-ripple of the pump is examined by considering the fluid compressibility and for computing these results a numerical program is used. From the idealized analysis it is shown that the idealized flow-ripple is determined by geometrical flow property.

Keywords: *axial clearance, experiment, flow-ripple, hydrodynamic processes, incompressible, mathematical modelling, radial clearance, vane pump*

Utjecaj zraka sadržanog u tekućini krilne pumpe s radnim parametrima dvostrukog učinka

Prethodno priopćenje

Razvoj krilnih pumpi temelji se na eksperimentalnom istraživanju i matematičkom modeliranju nestacionarnih hidrauličkih procesa unutar pumpe, u prostoru potiska i usisnom i potisnom cjevovodu. Pomoću eksperimentalnog istraživanja i rezultata matematičkog modeliranja i programskog paketa KRILP, moguće je dovoljno precizno odrediti parametre operativnih procesa krilnih pumpi. Ovo istraživanje ispituje idealizirani i stvarni protok pumpe pod visokim tlakom među pumpama krilnog tipa. U idealiziranom slučaju ispituje se "savršena" pumpa u kojoj se smatra da je curenje jednako nula, a tekućina da je nestlačiva. Na temelju tih pretpostavki, izvedene su jednadžbe koje opisuju karakteristike idealiziranog protoka. Nadalje, ispitan je stvarni protok pumpe s obzirom na stlačivost tekućine, a za izračunavanje tih rezultata korišten je numerički program. Idealizirana analiza pokazuje da je idealizirani protok određen geometrijskim svojstvom protoka.

Ključne riječi: *aksijalna zračnost, eksperiment, protok, hidrodinamički procesi, nestlačiv, matematičko modeliranje, radijalna zračnost, krilna pumpa*

1 Introduction

Modern methods of hydraulic pumps designing and constructing cannot be done without using the appropriate mathematical models of effects and processes occurring in real pump structures. The mathematical model of a process is analytical interpretation of the process with certain assumptions [1 ÷ 3]. In order to reach the mathematical model it is necessary to make detailed theoretical research based on the laws of fundamental sciences and explanation of processes, which is the basis for adopting the assumptions and defining the model equations [4 ÷ 9].

The balanced vane pump is widely used in many hydraulic power systems, in which low discharge flow ripple and low noise are required. In high-pressure vane pump, the fluid noise which is mainly generated by pressure impact within the transition region, as well as mechanical noise constitutes a key factor in evaluating noise level. The pressure gradient was adopted to measure the rate of pressure impact, the maximum of which determines the level of fluid noise. Diminishing the max. of pressure gradient in transition region is an effective measure to reduce fluid noise. In order to avoid in the pressure impact the pre-loading process in transition regions and silencing grooves are used on bushing in high-pressure vane pump, Which is the process that fluid in discharge port backfill to the chamber through silencing grooves and the fluid in chamber be encapsulated. The air content entrained in fluid is the main influential factor of backfilling flow which affects the discharge flow ripple and the max. of pressure gradient accordingly [10 ÷ 15].

2 Mathematical model of pressure change in the operating chamber

The level of noise made by vane pump with double effect is crucially influenced by pressure rise and fall in the pump chambers in the areas of the change of operating cycles. Constant conversion of thrust pressure into operating pressure in the installation and vice versa is an important assumption for lowering the noise level. There are numerous researches done in order to define the optimal geometry working volume as one operating cycle converts into another one. The processes occurring at the area of pressure change and their relation can be researched by experiments and by mathematical modelling by means of adequate software packages. For these researches the software KRILP has been developed and it has been written in program language Digital Visual Fortran [2].

The increase of rotation number provides better tightness of working chamber in the area of pressure change which can be explained by the increase of centrifugal force acting on the vanes and pressing them against the inner surface of the stator. When the vanes are separated from the operating stator profile, pressure pulsation and amplitude changes are registered. These changes often occur with low number of revolutions and they lead to oil coming back from thrust area towards suction area. When chambers are not sealed tight, i.e. when clearances are large, the pressure does not rise enough in the area of pressure change, which leads to unexpected relation between the chamber and thrust port and also leads to pressure balance. Thus the leakage between the suction and thrust zone is being increased as well as the amplitude of pressure pulsation.

Due to the influence of the clearance on the tightness in the chamber, the pressure change in the chamber should be presented by mathematical model depending on volumetric losses and it is necessary to make certain simulations on the computer. On the basis of data obtained through experiments and simulations we should determine geometries of suction and thrust ports as well as partitions between them at valve plate. The following phases can be distinguished in simulating the pressure change in the chamber while passing over the partition separating suction and thrust zones:
- the chamber is closed, i.e. there is no connection between the chamber and/neither suction zone and/nor thrust zone
- the chamber is connected to thrust port through the slot
- the chamber is fully connected to thrust port.

Vane pump with double effect is shown in Fig. 1.
Technical data:

- Speed: 500 ÷ 3500 min^{-1}
- Pressure: 210×10^5 Pa
- Specific flow: 16,5 cm^3/rev
- Number of vanes: 10.

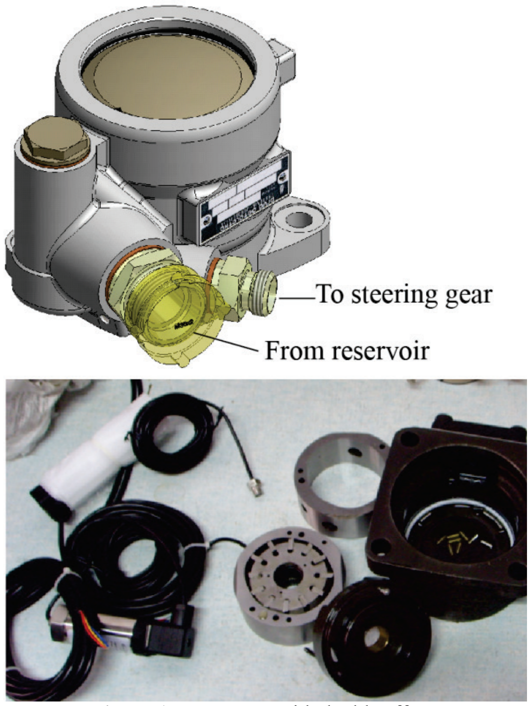

Figure 1 Vane pump with double effect

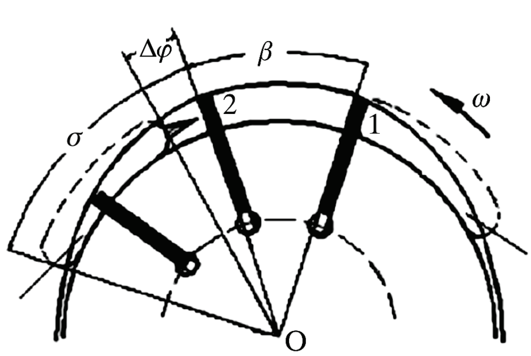

Figure 2 Gaining the instantaneous fluid pressure within chamber in transition region

2.1 Volumetric losses affecting the speed of pressure change in the operating chamber in the zone of pressure change

In order to operate properly the pump must have appropriate clearances between vane rotor and valve plates. There is a certain flow through these clearances. Volumetric losses in the chamber can be classified as follows [14, 15]:

- losses at vane side made by axial clearances Q_{an} (n = 1,2), mm^3/s
- losses over vane top made by radial clearances Q_{rn} (n = 1,2), mm^3/s
- losses made by flow withdrawal Q_{pr}, mm^3/s
- losses through the slot at valve plate Q_{pz}, mm^3/s
- losses through the gap made by the vane in rotor groove Q_{pc}, mm^3/s.

2.1.1 Losses made by axial clearances

If it is assumed that the flow is a streamline, the losses through the axial clearances are:

a) volumetric losses for the chamber in front of the vane can be presented by the following equation:

$$Q_{a1} = \frac{(r_i - r)\cdot z_{a1}^3}{12\cdot\eta\cdot s}\cdot\left|p_r - p_k\right|\cdot \mathrm{sign}(p_r - p_k), \tag{1}$$

b) volumetric losses for the chamber behind the vane can be presented by the following equation:

$$Q_{a2} = \frac{(r_i - r)\cdot z_{a2}^3}{12\cdot\eta\cdot s}\cdot(p_k - p_u), \tag{2}$$

where:
r_i − variable radius of the stator, mm
r − smaller radius of the stator, mm
z_{a1}, z_{a2} − values of axial clearance, mm
s − vane thickness, mm
η − dynamic viscosity of working fluid, Pa·s
p_u − suction pressure, MPa
p_k − pressure in the chamber, MPa
p_r − operating pressure, MPa.

2.1.2 Losses made by radial clearances

Between the inside surface of the stator and vane top working flow leaks which can be presented as follows:

a) volumetric losses for the chamber in front of the vane can be presented by the following equation:

$$Q_{r1} = \frac{b\cdot z_{r1}^3}{12\cdot\eta\cdot s}\cdot\left|p_r - p_k\right|\cdot \mathrm{sign}(p_r - p_k), \tag{3}$$

b) volumetric losses for the chamber behind the vane can be presented by the following equation:

$$Q_{r2} = \frac{b\cdot z_{r2}^3}{12\cdot\eta\cdot s}\cdot(p_k - p_u), \tag{4}$$

z_{r1}, z_{r2} − values of radial clearances, mm
b − the width of the vane, mm.

2.1.3 Losses made by flow withdrawal

The mean value of losses made by flow withdrawal at the vane is presented by the expression:

$$Q_{\mathrm{pr}}=\frac{z_{\mathrm{a}1/2}\cdot\omega\cdot(R+r)\cdot(R-r)}{4}, \tag{5}$$

where are:
ω − angular speed of rotor, rad/s
φ − angle of rotor rotation, °
R, r − bigger and smaller radius of stator, mm.

2.1.4 Losses through the slot at valve plates

Losses through the slot in thrust port at valve plate are determined in the following manner:

$$Q_{\mathrm{pz}}=\mu\cdot A\cdot\sqrt{\frac{2}{\rho}\cdot(p_{\mathrm{r}}-p_{\mathrm{k}})\cdot\mathrm{sign}(p_{\mathrm{r}}-p_{\mathrm{k}})}, \tag{6}$$

where are:
μ − outflow coefficient
A − cross-sectional area, mm^2
ρ − density of working fluid, kg/m^3.

2.1.5 Losses made by the vane in rotor groove

If the pressure in the chamber is higher than the working pressure of the pump, there is a gap in rotor groove made by front vane tilting because of tangential load and there is oil leakage which can be presented by:

$$Q_{\mathrm{pc}}=\frac{b\cdot z_{\mathrm{pc}}^{3}}{12\cdot\eta\cdot l_{\mathrm{r}}}\cdot(p_{\mathrm{k}}-p_{\mathrm{r}}), \tag{7}$$

where are:
z_{pc} − clearance in the gap, mm
$l_{\mathrm{r}}=l-(r_{\mathrm{i}}-r_{\mathrm{r}})$ − length of the front vane when rotor is in transmission area, mm
r_{r} − radius of rotor, mm.

2.2 Speed of pressure change in the chamber when suction and thrust zones are being separated

If initial volume V is lowered for $\mathrm{d}V=V_1-V=-(V-V_1)$, due to pressure rise $\mathrm{d}p=p_1-p$ the relative volume -dV/V, calculated per pressure unit:

$$S=-\frac{1}{\mathrm{d}p}\cdot\frac{\mathrm{d}V}{V}, \tag{8}$$

is compressibility coefficient.

The reciprocating value of compressibility coefficient is called the compressibility modulus ε_{s}:

$$\varepsilon_{\mathrm{S}}=\frac{1}{S}=-\frac{\mathrm{d}p}{\mathrm{d}V/V}, \tag{9}$$

which has the same dimension as the pressure.

In previous expressions the minus sign shows that pressure rise corresponds to volume decrease and vice versa. The previous expression can be also presented in the following form, in case of final changes of pressure and volume:

$$-\frac{\Delta V}{V}=\frac{\Delta p}{\varepsilon_{\mathrm{S}}}, \tag{10}$$

which represents Hooke's law. The marks in previous expression are:
$\Delta p=p_1-p$ − pressure increment, MPa
$\Delta V=V_1-V$ − change of volume, mm^3
V_1 − fluid volume at the pressure p_1, mm^3.

In this study, the compression of oil in the transition regions is considered as adiabatic change. For the transition time is about 1ms at the speed of rotation n = 1800 min^{-1} and the angle of the transition regions is $\Delta\varphi$ = 10° ÷ 12°.

Mathematically the fluid bulk modulus varying with the content of air is expressed by:

$$S'=S''\cdot\frac{1+\left(\frac{V_{\mathrm{a}0}}{V_{\mathrm{f}0}}\right)\cdot\left(\frac{p_0}{p}\right)^{2/k}}{1+\left(\frac{V_{\mathrm{a}0}}{V_{\mathrm{f}0}}\right)\cdot\left(\frac{p_0}{p}\right)^{1/k}\cdot\left(\frac{S''}{k\cdot p}\right)}, \tag{11}$$

where S' is the bulk modulus of oil containing air, S'' = 1660 MPa is the bulk modulus of pure oil, p is the absolute pressure of oil, p_0 is absolute atmosphere, k = 1,4 is adiabatic exponent, $V_{\mathrm{a}0}$ is volume of air at absolute atmosphere, $V_{\mathrm{f}0}$ is volume of oil at absolute atmosphere [10, 14]. The volumetric content of air in oil at absolute atmosphere is given by:

$$x_0=\frac{V_{\mathrm{a}0}}{V_{\mathrm{a}0}+V_{\mathrm{f}0}}. \tag{12}$$

So:

$$\frac{V_{\mathrm{a}0}}{V_{\mathrm{f}0}}=\frac{x_0}{1-x_0}=x_1. \tag{13}$$

Substituting (12) into (13) gives the fluid bulk modulus S' in the following form:

$$S'=S''\cdot\frac{1+x_1\cdot p_0^{2/k}\cdot p^{-2/k}}{1+x_1\cdot p_0^{1/k}\cdot p^{-1-1/k}\cdot\left(\frac{S''}{k}\right)}. \tag{14}$$

The pressure increment in the working chamber of vane pump with double effect can be obtained from the following expression:

$$\Delta p_{\mathrm{k}} = \frac{\varepsilon_{\mathrm{S}}}{V_{\mathrm{k}(R)}} \cdot (\Delta V)_{\mathrm{k}(R)}\,, \tag{15}$$

where:
Δp_{k} − pressure increment in the chamber between the vanes, $p_{\mathrm{u}} < p_{\mathrm{k}} < p_{\mathrm{p}}$
ε_{S} − compressibility modulus of working fluid
p_{u} − suction pressure of working fluid, MPa
p_{p} − thrust pressure of working fluid, MPa
$V_{\mathrm{k}(R)}$ − volume of the chamber (when the chamber is in the zone of pressure change constrained by angle $(\beta-\sigma)$ and bigger stator radius R), mm^3.

Volume $V_{\mathrm{k}(R)}$ is calculated like this:

$$V_{\mathrm{k}(R)} = \frac{b}{2} \cdot (R^2 - r_{\mathrm{r}}^2) \cdot (\beta - \sigma), \tag{16}$$

where are:
$(\beta - \sigma)$ − angle between two adjacent vanes, rad
b − vane width, mm.

The speed of pressure change in the chamber is obtained by differentiating the expression (15) with respect to time t:

$$\frac{\Delta p_{\mathrm{k}}}{\Delta t} = \frac{\varepsilon_{\mathrm{S}}}{V_{\mathrm{k}(R)}} \cdot \frac{\Delta V_{\mathrm{k}(R)}}{\Delta t}, \tag{17}$$

in case when $\Delta t \to 0$; $\Delta V_{\mathrm{k}(R)} \to \mathrm{d}V_{\mathrm{k}(R)} \to 0$ and $\Delta p_{\mathrm{k}} \to \mathrm{d}p_{\mathrm{k}} \to 0$, previous equation has differential form:

$$\frac{\Delta p_{\mathrm{k}}}{\Delta t} = \frac{\varepsilon_{\mathrm{S}}}{V_{\mathrm{k}(R)}} \cdot \frac{\mathrm{d}}{\mathrm{d}t} (\Delta V)_{\mathrm{k}(R)}. \tag{18}$$

If we place the expressions for volumetric losses (1) to (7) into the expression (18) for speed of pressure change in the chamber, we obtain the following expression:

$$\frac{\mathrm{d}p_{\mathrm{k}}}{\mathrm{d}t} = \frac{\varepsilon_{\mathrm{S}}}{V_{\mathrm{k}(R)}} \cdot \left(\frac{\mathrm{d}V_{\mathrm{k}(R)}}{\mathrm{d}t} + 2Q_{\mathrm{a1}} - 2Q_{\mathrm{a2}} + Q_{\mathrm{r1}} - \right. \\ \left. - Q_{\mathrm{r2}} - Q_{\mathrm{pr}} + Q_{\mathrm{pz}} + Q_{\mathrm{pc}} \right). \tag{19}$$

After replacing the values for volumetric losses we get the required expression for speed of pressure change in relation to the clearance in the chamber of vane pump with double effect:

$$\begin{aligned} \frac{\mathrm{d}p_{\mathrm{k}}}{\mathrm{d}t} = \frac{\varepsilon_{\mathrm{S}}}{V_{\mathrm{k}(R)}} \cdot \Bigg[& \frac{\mathrm{d}V_{\mathrm{k}(R)}}{\mathrm{d}t} + 2\frac{(r_{\mathrm{i}} - r)\cdot z_{\mathrm{a1}}^3}{12\cdot\eta\cdot s} \cdot \left| p_{\mathrm{r}} - p_{\mathrm{k}} \right| \cdot \mathrm{sign}(p_{\mathrm{r}} - p_{\mathrm{k}}) - \\ & - 2\frac{(r_{\mathrm{i}} - r)\cdot z_{\mathrm{a2}}^3}{12\cdot\eta\cdot s} \cdot (p_{\mathrm{k}} - p_{\mathrm{u}}) + \frac{b\cdot z_{\mathrm{r1}}^3}{12\cdot\eta\cdot s} \cdot \left| p_{\mathrm{r}} - p_{\mathrm{k}} \right| \cdot \mathrm{sign}(p_{\mathrm{r}} - p_{\mathrm{k}}) - \\ & - \frac{b\cdot z_{\mathrm{r2}}^3}{12\cdot\eta\cdot s} \cdot (p_{\mathrm{k}} - p_{\mathrm{u}}) - \frac{z_{\mathrm{a1/2}}\cdot\omega\cdot(R+r)\cdot(R-r)}{4} + \\ & + \mu\cdot A\cdot\sqrt{\frac{2}{\rho}\cdot(p_{\mathrm{r}} - p_{\mathrm{k}})\cdot\mathrm{sign}(p_{\mathrm{r}} - p_{\mathrm{k}})} + \frac{b\cdot z_{\mathrm{pc}}^3}{12\cdot\eta\cdot l_{\mathrm{r}}} \cdot (p_{\mathrm{k}} - p_{\mathrm{r}}) \Bigg]. \end{aligned} \tag{20}$$

The expression of ρ is given by:

$$\rho \approx \frac{\left(1 + \frac{V_{\mathrm{a0}}}{V_{\mathrm{f0}}}\right)\cdot\rho_0}{1 + \left(\frac{V_{\mathrm{a0}}}{V_{\mathrm{f0}}}\right)\cdot\left(\frac{p_0}{p}\right)^{1/k}\cdot\left(\frac{S''}{kp}\right)}, \tag{21}$$

where ρ_0 is the density of pure oil.

Substituting (13) into (21) yields ρ in following form:

$$\rho \approx \frac{(1 + x_1)\cdot\rho_0}{1 + x_1\cdot\left(\frac{p_0}{p}\right)^{1/k}\cdot\left(\frac{S''}{kp}\right)}. \tag{22}$$

For simulation purposes, the above equation can be rearranged as:

$$\begin{aligned} \frac{\mathrm{d}p}{\mathrm{d}\varphi} = & -\frac{1}{1 - \frac{\mathrm{d}p}{S'}\cdot\frac{\mathrm{d}S'}{\mathrm{d}p}} \cdot \frac{S'}{V} \cdot \frac{\mathrm{d}V}{\mathrm{d}t} \cdot \left(\frac{-B\cdot\omega}{2}\left(R^2 - r_{\mathrm{i}}^2\right) + \right. \\ & \left. + b\cdot s\cdot\frac{\mathrm{d}r_{\mathrm{i}}}{\mathrm{d}t} + c_{\mathrm{q}}\cdot A_0\cdot\left(2\frac{\mathrm{d}p}{p}\right)^{\frac{1}{2}} \right)\cdot\frac{1}{\omega}. \end{aligned} \tag{23}$$

Eq. (23) can be solved numerically for the instantaneous pressure of the chamber in transition regions. Example for one type high-pressure vane within vane pump, Fig. 3 shows p/p_{s} vary with the configuration of silencing groove under condition of p_{s} = 17,5 MPa , n = 1800 min^{-1} and x_0 = 1 %, 3 %, 5 %, 7 %.

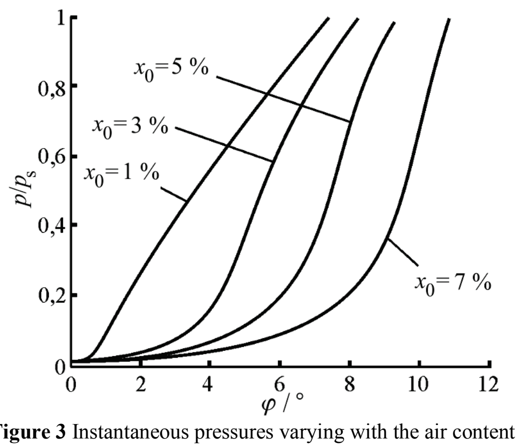

Figure 3 Instantaneous pressures varying with the air content

Substituting the numerical results of p into Eq. (23) can yield the pressure gradient dp/dφ vary with the content of air in fluid, as shown in Fig. 4. Additionally, the same figure presents the pressure change in the vane pump drive shaft angle for different percentages of gas content in the working fluid. Experiments show that characteristics of two-phase fluid must be considered in mathematics modelling since the content of the fluid considerably influences the pressure gradient.

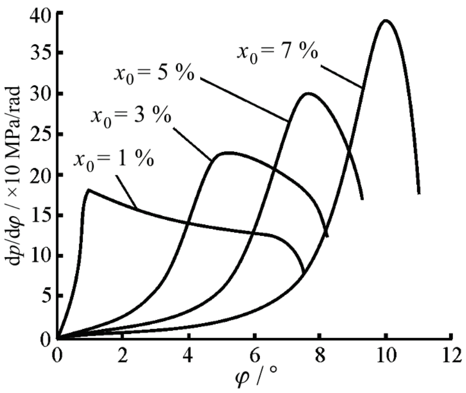

Figure 4 Pressure gradients varying with the air content

Next section shows diagrams of pressure change speed depending on the clearance in the operating chamber of the vane pump with double effect. They are obtained by simulation of expression (20) by means of software package KRILP.

3 Results of simulating the pressure change in the chamber of vane pump with double effect

Results of simulating the expression for speed of pressure change in the chamber for various values of slot cross section [2] are shown in Fig. 5 which also presents the pressure change in the function of time for one-phase fluid which leads to conclusion that the change is influenced by the clearance size in the vane pump rotor grooves. These changes can be related to the volume losses of the vane pump in the function of clearance size.

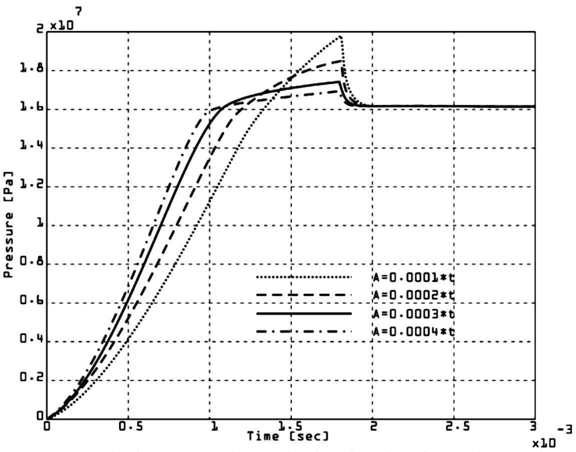

Figure 5 Speed of pressure change in the chamber for various values of slot cross section

Results of simulating the expression for speed of pressure change in the chamber for various values of axial clearance at the first vane are shown in Fig. 6.

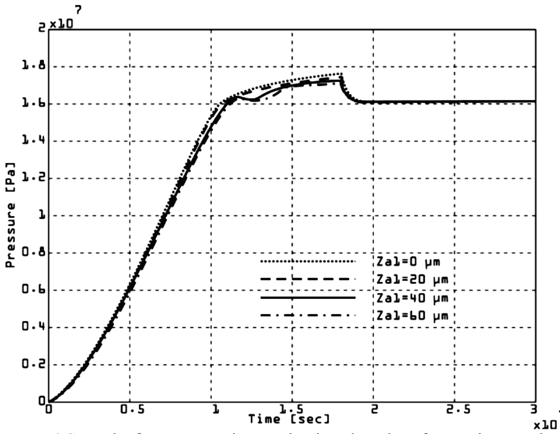

Figure 6 Speed of pressure change in the chamber for various values of axial clearance at the first vane

Results of simulating the expression for speed of pressure change in the chamber for various values of axial clearance at the second vane are shown in Fig. 7.

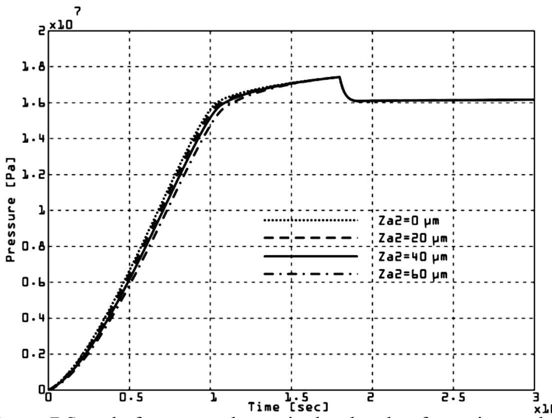

Figure 7 Speed of pressure change in the chamber for various values of axial clearance at the second vane

Results of simulating the expression for speed of pressure change in the chamber for various values of radial clearance at the first vane are shown in Fig. 8.

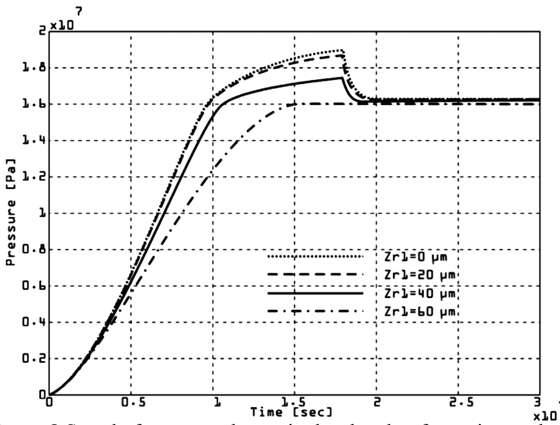

Figure 8 Speed of pressure change in the chamber for various values of radial clearance at the first vane

Results of simulating the expression for speed of pressure change in the chamber for various values of radial clearance at the second vane are shown in Fig. 9.

Results of simulating the expression for speed of pressure change in the chamber for various values of the gap in rotor groove are shown in Fig. 10.

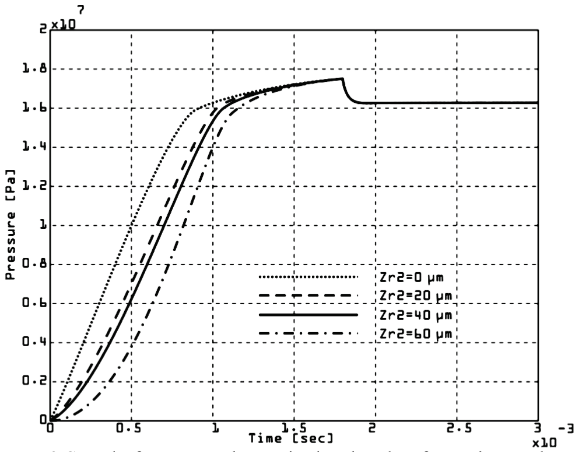

Figure 9 Speed of pressure change in the chamber for various values of radial clearance at the second vane

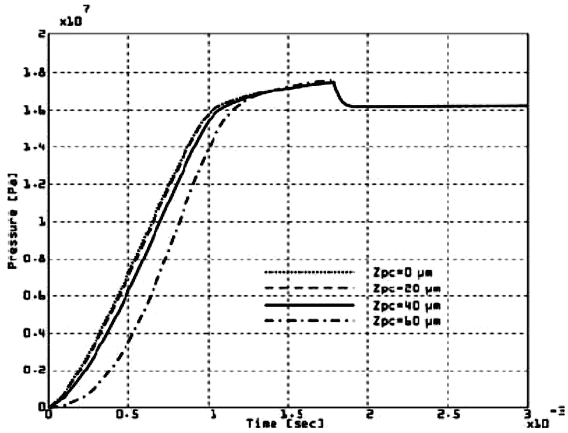

Figure 10 Speed of pressure change in the chamber for various values of the gap in rotor groove

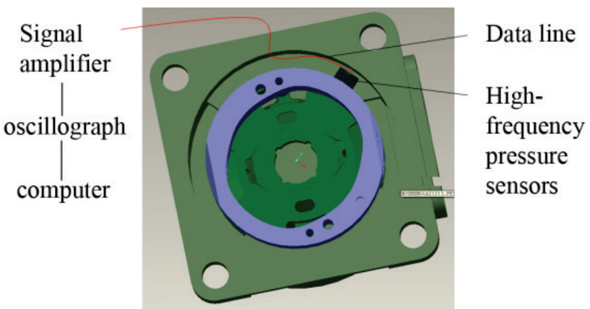

Figure 11 The overall structure of experiment

4 Conclusion

It is not possible to determine precisely enough the parameters of hydrodynamic process of the vane pump with double effect neither by experiments nor by mathematical modelling only. Accurate working parameters can be reached by combined application of experimental measuring, mathematical modelling of hydrodynamic process and nonlinear optimization but at the same time the system errors of measuring and unknown parameters can be determined.

The program KRILP developed for mathematical modelling, identification and optimization of vane pumps provides developing of a whole family of vane pumps in further research of hydrodynamic processes along with the analysis of advantages and disadvantages of vane pumps with double effect.

Nomenclature

Q_{an} − losses at vane side made by axial clearances, mm^3/s
Q_m − over vane top made by radial clearances, mm^3/s
Q_{pr} − losses made by flow withdrawal, mm^3/s
Q_{pz} − losses through the slot at valve plate, mm^3/s
Q_{pc} − losses through the gap made by the vane in rotor groove, mm^3/s
r_i − variable radius of the stator, mm
r − smaller radius of the stator, mm
z_{a1}, z_{a2} − values of axial clearance, mm
s − vane thickness, mm
η − dynamic viscosity of working fluid, Pa·s
p_u − suction pressure, MPa
p_k − pressure in the chamber, MPa
p_r − operating pressure, MPa
z_{r1}, z_{r2} − values of radial clearances, mm
b − the width of the vane, mm
ω − angular speed of rotor, rad
φ − of rotor rotation, °
R, r − bigger and smaller radius of stator, mm
μ − outflow coefficient,
A − cross-sectional area, mm^2
ρ − density of working fluid, kg/m^3
z_{pc} − clearance in the gap, mm
$l_r = l - (r_i - r_r)$ − length of the front vane when rotor is in transmission area, mm
r_r − radius of rotor, mm
$\Delta p = p_1 - p$ − pressure increment, MPa
$\Delta V = V_1 - V$ − change of volume, mm^3
V_1 − fluid volume at the pressure p_1, mm^3
Δp_k − pressure increment in the chamber between the vanes $p_u < p_k < p_p$, MPa
ε_S − compressibility modulus of working fluid, −
p_u − suction pressure of working fluid, MPa
p_p − thrust pressure of working fluid, MPa
$V_{k(R)}$ − volume of the chamber (when the chamber is in the zone of pressure change constrained by angle $(\beta - \sigma)$ and bigger stator radius R), mm^3
$(\beta - \sigma)$ − angle between two adjacent vanes, rad
b − vane width, mm.

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