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

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Abstract

In developing the vane pumps the fundamental basis is experimental research and mathematical modeling of nonstationary hydraulic processes inside the pump, in thrust space and suction and thrust pipeline. By means of experimental research and results of mathematical modeling 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 describe 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: vane pump, mathematical modeling, software, hydrodynamic processes, incompressible, flow-ripple, axial clearance, radial clearance, rotor, stator, experiment.

1. Introduction

Modern methods of designing and constructing the hydraulic pumps can not be done without using the appropriate mathematical models of effects and processes happening in real pump structures. The mathematical model of a process is analytical interpretation of the process with certain assumptions. 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, what is the basis for adopting the assumptions and defining the model equations.

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 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 a great number of researches done in order to define the optimal geometry of working volume when one operating cycle converts into another one [9]. The processes occuring at the area of pressure change and their relation can be researched by experiments and by mathematical modeling 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 5.0.

The increase of rotation number provides better tightness of working chamber at 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, what 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 by 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 $\cdot 10^5$ Pa Specific Flow: 16.5 cm³/ o Number of vanes: 10







Fig.2. In order to gain the instantaneous fluid pressure within chamber in transition region

1.1.Volumetric losses affecting the speed of pressure change in the operating chamber at 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 [8]:

· losses at vane side made by axial clearances Q_{an} (n = 1,2)

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· losses over vane top made by radial clearances $Q_m(n=1,2)$

 \cdot losses made by flow withdrawal Q_{pr}

· losses through the slot at valve plate Q_{DZ}

 \cdot losses through the gap made by the vane in rotor groove $Q_{\mbox{\tiny pc}}$

1.1.1.Losses made by axial clearances

If we assume that the flow is streamline the losses through the axial clearances are:

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

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

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

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

where:

ri-variable radius of the stator;

r-smaller radius of the stator

 Z_{a1}, Z_{a2} -values of axial clearance;

s-vane thickness

η-dynamic viscosity of working fluid;

 p_u -suction pressure

- p_k -pressure in the chamber
- p_r -operating pressure

1.1.2. Losses made by radial clearances

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

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

$$Q_{rl} = \frac{b \cdot z_{rl}^{3}}{12\eta s} |p_r - p_k| sign(p_r - p_k),$$
(3)

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

$$Q_{r2} = \frac{b \cdot z_{r2}^{3}}{12\eta s} (p_{k} - p_{u}),$$
(4)

 z_{r1} , z_{r2} -values of radial clearances

b-*the width of the vane*

1.1.3. Losses made by flow withdrawal

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

$$Q_{\rm pr} = \frac{z_{\rm a1/2} \omega (R+r) (R-r)}{4}$$
 (5)

where are:

 ω - angular speed of rotor;

 ϕ - angle of rotor rotation

R, r - bigger and smaller radius of stator

1.1.4. Losses through the slot at valve plates

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

$$Q_{pz} = \mu A \sqrt{\frac{2}{\rho} (p_r - p_k) sign(p_r - p_k)}$$
(6)

where are:

μ - outflow coefficient;

A - cross-sectional area

 ρ - density of working fluid

1.1.5. Losses made by the vane in rotor groove

If the pressure in the chamber is higher than 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:

$$Q_{pc} = \frac{b z_{pc}^{-3}}{12 \eta l_r} (p_k - p_r)$$
(7)

where are:

 z_{pc} - clearance in the gap;

 $l_r = l - (r_i - r_r)$ - length of the front vane when rotor is in transmission area.

rr-radius of rotor

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

If initial volume V is lowered for $dV = V_1 - V = -(V - V_1)$,

due to pressure rise $dp = p_I - p$ the relative volume - dV/V, calculated per pressure unit:

 $S = -\frac{1}{dp} \cdot \frac{dV}{V},\tag{8}$

is compressibility coefficient.

The reciprocating value of compressibility coefficient is called the compressibility modulus (ε_s) ,

$$\boldsymbol{\varepsilon}_{S} = \frac{1}{S} = -\frac{dp}{dV/V} \tag{9}$$

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

$$-\frac{\Delta V}{V} = \frac{\Delta p}{\varepsilon_s} \tag{10}$$

which represents so called Hooke's law. The marks in previous expression are:

 $\Delta p = p_1 - p$ - pressure increment

 $\Delta V = V_1 - V$ - change of volume

 V_{1} - fluid volume at the pressure p_{1}

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 $n = 1800 \text{ min}^{-1}$ and the angle of the transition regions is $\Delta \phi = 10^{\circ} \sim 12^{\circ}$.

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

$$S' = S'' \frac{1 + (V_{a0}/V_{f0})(p_0/p)^{2/k}}{1 + (V_{a0}/V_{f0})(p_0/p)^{Vk}(S''/kp)}$$
(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 exsponet, V_{a0} is volume of air at absolute atmosphere, V_{n0} is volume of oil at absolute atmosphere. The volumetric content of air in oil at absolute atmosphere is given by

$$x_0 = \frac{V_{a0}}{V_{a0} + V_{f0}}$$
(12)

So:

$$\frac{V_{a0}}{V_{f0}} = \frac{x_0}{1 - x_0} = x_1$$
(13)

Substituting (12) into (13) yield the fluid bulk moduls S'in following form

$$S' = S'' \frac{1 + x_1 p_o^{2/k} p^{-2/k}}{1 + x_1 p_o^{1/k} p^{-1-1/k} S''/k}$$
(14)

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

$$\Delta p_k = \frac{\varepsilon_s}{V_{k(R)}} (\Delta V)_{k(R)},\tag{15}$$

where are:

 Δp_k - pressure increment in the chamber between the vanes $p_u < p_k < p_p$

 ϵ_s - compressibility modulus of working fluid

 p_u -suction pressure of working fluid

 p_{p} - thrust pressure of working fluid

 $V_{k(R)}$ - volume of the chamber (when the chamber is in the zone of pressure change constrained by angle)(sb- and bigger stator radius R)

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

$$V_{k(R)} = \frac{b}{2} \left(R^2 - r_r^2 \right) \beta - \sigma , \qquad (16)$$

where are:

 $(\beta - \sigma)$ [rad] - angle between two adjacent vanes *b* - vane width

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

$$\frac{\Delta p_k}{\Delta t} = \frac{\varepsilon_s}{V_{k(R)}} \frac{\Delta V_{k(R)}}{\Delta t},$$
(17)

in case when $\Delta t \rightarrow 0$; $\Delta V_{k(R)} \rightarrow dV_{k(R)} \rightarrow 0$ and $\Delta p_k \rightarrow dp_k \rightarrow 0$, previous equation has differential form:

$$\frac{\Delta p_k}{\Delta t} = \frac{\varepsilon_s}{V_{k(R)}} \frac{d}{dt} (\Delta V)_{k(R)}$$
(18)

If we put 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:

$$\frac{dp_{k}}{dt} = \frac{\varepsilon_{s}}{V_{k(R)}} \left(\frac{dV_{k(R)}}{dt} + 2Q_{a1} - 2Q_{a2} + Q_{r1} - Q_{r2} - Q_{pr} + Q_{pr} + Q_{pc} \right)$$
(19)

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

$$\frac{dp_{k}}{dt} = \frac{\varepsilon_{s}}{V_{k(R)}} \left[\frac{dV_{k(R)}}{dt} + 2\frac{(\rho - r) \cdot z_{al}^{3}}{12\eta s} | p_{r} - p_{k}| sign(p_{r} - p_{k}) - \frac{2(\rho - r) \cdot z_{a2}^{3}}{12\eta s} (p_{k} - p_{u}) + \frac{b \cdot z_{r}^{3}}{12\eta s} | p_{r} - p_{k}| sign(p_{r} - p_{k}) - \frac{b \cdot z_{r2}^{3}}{12\eta s} (p_{k} - p_{u}) - \frac{z_{aV2}\omega(R + r)(R - r)}{4} + \frac{4}{12\eta s} + \frac{2}{\rho}(p_{r} - p_{k}) sign(p_{r} - p_{k}) + \frac{bz_{p}^{3}}{12\eta l_{r}} (p_{k} - p_{r}) \right]$$

$$(20)$$

The expression of p is given by

$$\rho \approx \frac{\left[1 + \left(V_{a0}/V_{f0}\right)\right] p_0}{1 + \left(V_{a0}/V_{f0}\right) p_0/p}$$
(21)

Where ρ_0 is the density of pure oil.

Substituting (13) into (21) yield ρ in following form

$$\rho \approx \frac{(1+x_1)p_0}{1+x_1(p_0/p)^{1/k}(S''/kp)}$$
(22)

For simulatiom purposes, the above equation can be rearranged as

$$\frac{dp}{d\varphi} = -\frac{1}{1 - \frac{\Delta p}{S} \frac{dS}{dp}} \frac{S}{V} \frac{dV}{dt} \left(\frac{-B\omega}{2} \left(\frac{R^2 - r_i^2}{r_i^2} \right) + bs \frac{dr_i}{dt} + c_q A_0 \left(2\Delta p/p \right)^{1/2} \right) \frac{1}{\omega}$$
(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 , $n=1800min^{-1}$ and $x_o=1\%,3\%,5\%,7\%$.

Substituting the numerical results of p into Eq.(23) can yield the pressure gradient $dp/d\phi$ vary with the content of air in fluid, as shown in Fig.4.



Fig.3. Instantaneous pressure varying with the air content



Fig.4. Pressure gradient varying with the air content

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

2 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 are shown in Fig.5.



Fig.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.



Fig.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.



Fig.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.



Fig.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.



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

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.



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



Fig.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 modeling only. Accurate working parameters can be reached by combined application of experimental measuring, mathematical modeling 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 modeling, 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.

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