TEKA Kom. Mot. Energ. Roln. – OL PAN, 2008, 8, 319–325

ANALYTICAL SUBSTANTIATION OF GEOMETRICAL PARAMETERS OF THE VALVE PLATE HYDROMACHINE

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Summary. Analytical methods of definition of geometrical parametres in a variety of distributors have been created. Methods are proved on the basis of numerical simulation of details in the program ANSYS with the help of which dangerous places of pressure concentration have been revealed. Settlement schemes of various designs of distributor have been developed. An analysis of detailed geometrical parameters has been performed; dependences for definitions of pressure in dangerous points have been recieved.

Key words: model, hydromachine, valve plate, finite element method, deformed condition.

INTRODUCTION

One of the important problems of mechanical engineering in Ukraine is its transfer into modern highly effective level which can provide an improvement of existing and creation of essentially new machinery and designs on the basis of the applied resource and energy saving technologies. In the hydraulic drive machinery and equipment most fully satisfied should be requirements of the careful utilization of materials and power resources, and although characterized by small weight and dimensions, the possibilities of parameters regulation provide high efficiency and reliability rate.

The provision of a high technological level at the final stages of machine design and modernisation of the existing designs with the application of numerical methods of calculation and multi- criteria optimisation depend on the achieved parametres in design calculation. In connection with this, the development of calculation techniques in axial-piston hydromachine (APH) of a high technological level by analytical substantiationof rational geometrical and design data is a current scientific problem.

PROBLEM STATEMENT

The speeding up of hydraulic drives in power units is achieved by the nominal freguency of pumps rotation to the frequency of power-driven engine rotation running on optimal modes and in nominal pressure increase to 30 … 45 MPa. This, in turn, leads to the deformation in backlashes increase and requires a build up of hydromechanical losses. Generally, the whole hydromachine efficiency decrease requires working out an extra complex of the design and technological actions, the provision of which can lead to an increase and stabilisation in all the range of operating parametres.

The laws capacity of change losses and consequently the hydromachine efficiency essentially influence its efficiency and technological level. As the efficiency settlement dependence analysis shows, from the similarity theory of volume hydromachines [the Machine-building hydrodrive, 1978] and design treatment on perfection of hydromachine swinging units (SU), the same changes of details and units are directed to volume losses reduction, but at the same time they can speed up an increase of hydromechanical losses. And design improvement directed at a decrease of SU friction in losses can lead to an increase of volume losses. Therefore, for the selection of rational SU design which provides the stabilisation and efficiency increase in the hydromachine as a whole, it is necessary to carry out an analytical estimation of losses components.

The applied calculation methods of capacity losses are based on an assumption of absolute SU rigidity details of a hydromachine, that is the constant to equivalent backlashes size between the mobile details is accepted. Such assumptions lead to the conclusion that it is impossible to choose equivalent backlashes values for the series of geometrically similar piston hydromachines, for which the values of capacity losses factors would be identical. In addition, for a series of geometrically similar hydromachines the universal characteristics configurations and the modes of the greatest efficiency values essentially depend on the operating volumes of hydromachines. These discrepancies have led to the situation that the similarity theory is effectively applied to the capacity losses analysis with the utilization use of experimentally discovered hydromachines characteristics, that is for their verifying calculations.

The principal causes of SU capacity losses are operating fluid leakage in backlashes between mobile details and the fluid friction in them. As the calculations made in the works by [Nikolinko, 2000,2001] show SU details deformations in axial-piston hydromachines in acting the maximum pressure can reach in size the initial backlashes in their connections, that is why the account of deformation backlashes allows to grow the power calculations accuracy of hydromachines.

In the given article the VP settlement schemes of hydromachines allowing to take into account the deformations distributions (DU) units details in calculating the SU capacity lasses of a volume hydromachine are proved.

The creation of new volume hydromachines designs, as well as the modernisation of the produced ones begins with the SU development which is inseparably connected to a rational choice of the DU sizes of design an operating fluid (OF). In all kinds of volume hydromachines a DU is one of the basic parts which is intended for cavity separation of the hydromachine with the high and low pressure from the operating chambers as well as for their periodic connection.

BASIC VP MODELS

In the volume hydromachines various designs of distributors which are divided into sliding valve and valve are applied. In power units of a modern hydrodrive, the best application at sliding valve distributors during the direction of the OF parallely to the axes of rotation of the hydromachine shaft are called a valve plate (VP), and during the stream direction perpendicularly to the axis of rotation – a trunnion, one according to the design. On design VP differ with the form and number of windows of distribution, and also the surface form against which the rotor leans. On the number of windows it is possible to distinguish one –, two-, and multi-window distributors. The face surface VP can be both flat and spherical (Fig. 1). For example, in the adjustable axial-piston hydromachines with the inclined block of trimot designs cylinders, one-window VPs are applied to provide an OF suction directly from a casing cavity. Two-window VPs with crescent windows of high and low have also been popular . The sizes of windows and condensing corbels are defined from the condition of the rotor to the distributor guaranteed clip. The basis of the face distributor design is its calculations on durability, rigidity, wear resistance and hydraulic ones. The methods of VP calculation are based on an assumption of its flat deformation, that is the assumption was accepted, that its thickness is infinitely big. In this case all the cross-sections are in the same load stress conditions, and deformations do not depend on longitudinal coordinate, and longitudinal motion is equal to zero.

The deformations of supporting and contacting surfaces of SU mobile details of hydromachines, sufficiently influence their wear resistance and conseguently the serviceability and life as well as the valve capacity losses.

The performance of the exact calculation and analysis of the intense-deformed condition (IDC) of a distributor for definition of zones of the maximum stresses and deformations is complicated, which can be explained by the complexity of the detail design, and also by the character of operating loadings. The direct solution or even the differential equations character of the theory of elasticity for its IDC definition has not been known so far. The existing engineering methods of calculation of this detail are based on the assumptions considerably simplifying and changing the strength stresses and deformations distribution, and therefore cannot be suitable for such an analysis.

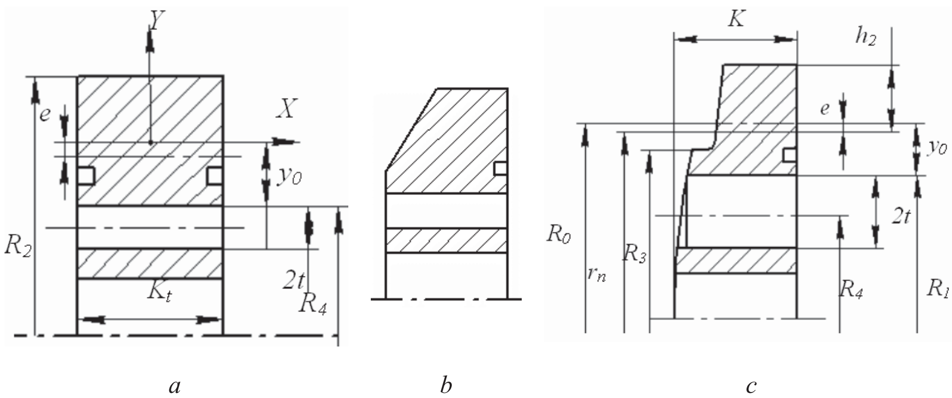


Fig. 1. Channels of cross- sections in a plane of distributors of symmetry: a –of the rectangular form and with a flat end face, b – trapeze form with a flat end face, c – the complex form and a spherical end face

The most complicated facilities manufacturing leads to the necessity of computer aided design as many practical problems cannot be solved analytically owing to the design and boundary conditions complexity. That is why in order to consider the real features of the deformed material, it is necessary to resort to the numerical methods of calculation. Unlike the analytical solution which describes a behaviour system in any point, the numerical solution approximates the exact solution only in discrete points.

With the development of computer engineering the numerical methods of the theory of elasticity for researching the intense conditions of complex form have been widely spread. Among them the most effective one for engineering calculations has been FEM [Kaplun A. B, Olfereva M. A, 2003].

In the present work VP APH IDC complex ANSYS was applied to the research.

The researches were carried out for APH series 300 sizes 25. According to the working drawing of the distributor 300.25.00.200, its model was created, which is a three-dimensional figure, built with the characteristic volumes as shown in Fig. 2.

The model represents a half TP separated along a symmetry axis.

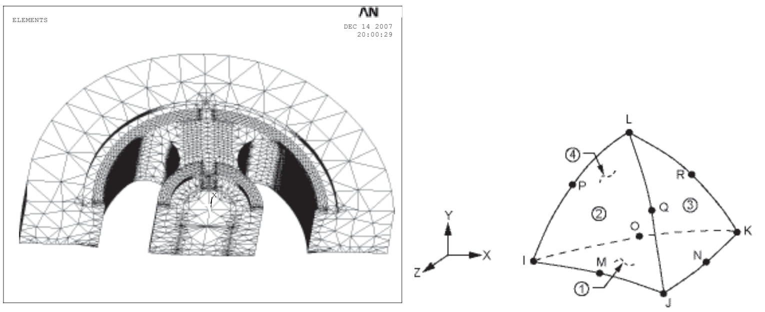
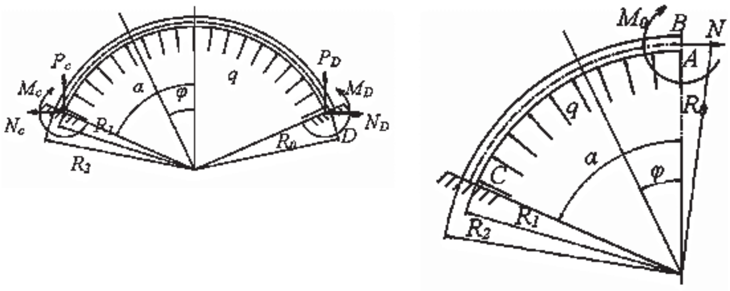


Fig. 2. Three-dimensional model VP with splitting into final elements and its geometry

For splitting, the element SOLID92 was applied, suitable for the irregular nets simulation. The element SOLID92 is a pyramid determined by ten units which have three degrees of freedom in every unit: translation in the directions of axes X, Y, Z of unit coordinate system. [Basses K.A, 2005].

As a result, the distributor model was broken into more than 160000 finite elements. When constructing the net, the algorithm of rational choice of marking-out element was used. It allows to build the net of elements taking into account the model surface curvature as well describing its real geometry in the best way.



a *b*

Fig. 3. VP Settlement scheme: a – statically not definable; b- equivalent system calculation

The obtained results allow to define the adequacy of results according to the offered calculation scheme. The analysis of intense-deformed IDC VP as a result of simulation in program complex ANSYS has shown, that the maximum stresses in it operate within the high pressure channel in a zone of its beginning and on an axis of symmetry [3] [Zheglova V.M., Nikolenko I.V., 2007]. In other zones, cross-section VP of pressure is less essential. Therefore, the VP strength and rigidity is defined by stresses and deformations within the distributor channel wall in which the pressure operates. The analysis of IDC VP has given a chance to accept a number of assumptions simplifying its settlement scheme. Half VP with the high pressure channel is considered since the stress deformation in a zone of the low pressure channel is of an essentially low stress deformation. The external wall deformations are directed basically in the radial direction, replaced with a curvilinear bar with cross-section section equal the one of the distributor channel external wall. The obtained settlement scheme of VP wall is presented in Fig. 3a. [Zheglova V.M., Nikolenko I.V., 2007]. In points of the bar fixation C and D, P_c and P_D operate diametrical forces, N_c, N_D – longitudinal forces, M_c, M_D – bending moments. Therefore in this scheme the amount of reactions in bar support is three times greater than the balance equations, therefore it is not statically defined.

Taking into account the symmetry VP, we receive the equivalent system presented in Fig. 3b, where the diametrical forces in cross-section along a symmetry axis are absent. The considered curvilinear bar of equivalent system is operated: on N – longitudinal force, M_0 – the bending moment, q – the distributed load arising from the pressure WF action. Taking into account that the chosen cross-section for the equivalent scheme does not turn and is not displaced, we make the system of equations for translation of cross-section AB (Fig. 3b) for the chosen scheme in a longitudinal direction:

$$\begin{cases} \delta_{11}X_1 + \delta_{12}X_2 + \Delta_{1q} = 0 \\ \delta_{21}X_1 + \delta_{22}X_2 + \Delta_{2q} = 0. \end{cases} \quad (1)$$

Where: X_1, X_2 – unknown power factors in cross-section; δ_{11}, $\delta_{12,}$ $\delta_{21,}$ δ_{22}-specific translations from the stated load X_i; Δ_{1q}, Δ_{2q} – the translation from pressure action q.

As to geometrical reasons we defined a total area S_Σ of bar cross-sections as the sum of the areas of including figures [Zheglova V.M., Nikolenko I.V., 2007].

The stresses in the points of a curvilinear bar cross-sections are defined in the form of the sum of stresses from the action of the bending moment and stretching force. We determine in dangerous points of the cross-section VP: A – in contour of a high pressure window in its axis of symmetry, B – on the external surface VP in its axis of symmetry, C – on a contour of a high pressure window in its initial point.

In points A and C of a high pressure window VP there is a complex IDC. Apart from the stresses operating in the cross-section, the pressure WF acts on the channel surface, and upon its face surface, the pressure WF operates as a backlash between VP and the contacting details. The equivalent stresses in these points are defined according to the fourth theory of strength which is applicable for the ductile materials equally resisting to a stretching and compressing:

$$\sigma_4 = \sqrt{\frac{1}{2}\left[(\sigma_i - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_i)^2\right]}, \quad (2)$$

$\sigma_{i,}$ – stresses in a required point; σ_2, σ_3 – stresses caused by pressure WF [Zheglova V.M., Nikolenko I.V., 2007].

THE RESULTS OF CALCULATIONS

Whe have chosen the pressure p = 25 MPa in accounts. The material of VP was steel of P25. The analysis of results allowed to determine the value of maximal stress intensity which arise in the beginning of a window and on the distributor channel axis. The most rigid is the rectangular cross-section distributor, the results of its calculation are represented in Fig. 4.

At pressure action p = 25 MPa the maximum stress in trapeze cross-section VP in the symmetrical position have made 188 MPa, in the rectangular one-158 MPa, and in the complex one 180 MPa. The maximum translations arise on distributor lateral faces, which increase from a face surface turned to BC to a face surface turned to a casing cover (Fig. 5) and have reached in trapeze cross-section 27, and in rectangular one 23 microns, and in complex one 22 microns.

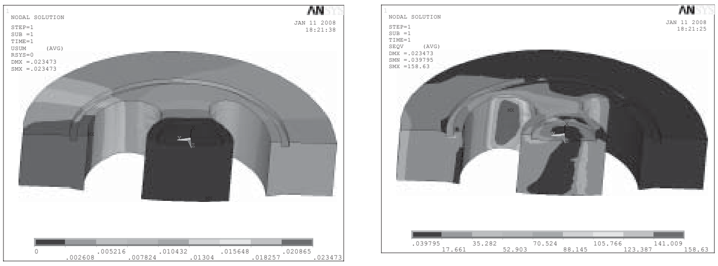


Fig. 4. The results of calculations in rectangular cross-section: a – total translations, b – fields of stresses intensity

As a result of calculations performed with the offered technique, the stresses have reached in trapeze cross-section VP 190 MPa, in rectangular one 165 MPa, and in complex one 190 MPa. The maximum translations reach at trapeze cross-section 24 microns, at rectangular one 23 microns, and at complex one 25 microns.

CONCLUSION

The calculation methods of power losses in SU volume hydromachines have been analyzed. The numerical volume models for the determination of stresses and deformations in VP with various cross-section have been developed. Calculation of IDC VP has been executed in program complex ANSYS, an analysis of the obtained the results carried out and zones with the maximum deformations and stresses in VP have been revealed while operating at the nominal pressure.

The mathematical models VP of the volume hydromachine, on the basis of its representation in the form of a flat curvilinear bar have been developed. The analytical dependences have been found between the determination of stresses and deformations in the distributor dangerous points. The calculations of stresses in dangerous points VP have been carried out. Serial rotary piston hydromachine 310.112 has been tested at pressure 25 MPa. The error of calculation FEM results and the analytical method does not exceed 7 %. The obtained results allow to calculate the losses of power in hydromachine SU in relation to the volume of a hydromachine, for the purpose of an analytical substantiation of its rational design data.

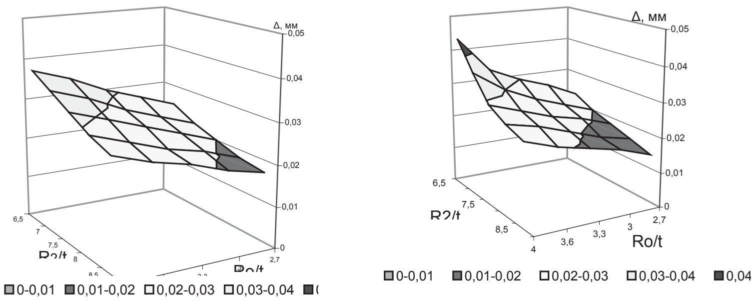


Fig. 5. Dependences of deformations on relative radius of an axis of the channel (R_4/t) and relative radius of an external surface (R_2/t): a – $z_ц$ = 5, b – $z_ц$ = 9

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