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Computer analysis of the stress-strain state of the centrifugal pump impeller

Метою статті є розрахунок напружено-деформованого стану робочого колеса відцентрового насоса з урахуванням натягу, відцентрової сили і потоку рідини та чисельно оцінити вплив потоку рідини на напружено-деформований стан робочого колеса.

Для досягнення поставленої мети було поставлено дві задачі: 1) розрахувати потік рідини в робочих частинах робочого колеса насоса та визначити поле тиску за допомогою ANSYS; 2) розрахувати напружено-деформований стан робочого колеса з урахуванням натягу, відцентрової сили і потоку рідини.

Наукова новизна - врахування тиску, що створюється потоком рідини при розрахунку напружено-деформованого стану робочого колеса насоса. Практичне значення полягає в аналізі напружено-деформованого стану робочого колеса з урахуванням потоку рідини.

За результатами розрахунку напружено-деформованого стану робочого колеса насоса можна зробити висновок про значний вплив тиску рідини, який впливає на лопаті та диски робочого колеса.

Ключові слова: *робоче колесо насоса, міцність, потік рідини, відцентрова сила, напружений стан, деформований стан, тиск, метод кінцевих елементів, чисельний розрахунок*

Целью статьи является расчет напряженно-деформированного состояния рабочего колеса центробежного насоса с учетом натяжения, центробежной силы и потока жидкости и численно оценить влияние потока жидкости на напряженно-деформированное состояние рабочего колеса.

Для достижения поставленной цели были поставлены две задачи: 1) рассчитать поток жидкости в рабочих частях рабочего колеса насоса и определить поле давления с помощью ANSYS; 2) рассчитать напряженно-деформированное состояние рабочего колеса с учетом натяжения, центробежной силы и потока жидкости.

Научная новизна - учет давления, создаваемого потоком жидкости при расчете напряженно-деформированного состояния рабочего колеса насоса. Практическое значение состоит в анализе напряженно-деформированного состояния рабочего колеса с учетом потока жидкости.

По результатам расчета напряженно-деформированного состояния рабочего колеса насоса можно сделать вывод о значительном влиянии давления жидкости, которое влияет на лопатки и диски рабочего колеса.

Ключевые слова: *рабочее колесо насоса, прочность, поток жидкости, центробежная сила, напряженное состояние, деформированное состояние, давление, метод конечных элементов, численный расчет*

The aim of the article is to calculate the stress-stain state of the centrifugal pump impeller with the tightness, centrifugal force and liquid flow and to evaluate numerically the effect of the liquid flow in the stress-strain state of the impeller.

To reach the aim two problems were stated: 1) to calculate the liquid flow in the running parts of the pump impeller and to determine the pressure field by means of ANSYS, 2) to calculate the stress-strain state of impeller taking into account the tightness, centrifugal force and liquid flow.

Scientific novelty is the incorporation of the pressure that is created by the liquid flow in the calculation of stress-strain state of the pump impeller. Practical value is analysis the impeller strength taking into account the liquid flow.

By results of calculation of the stress-strain state of the pump impeller it is possible to draw a conclusion on significant influence of liquid pressure which affects blades and disks of the impeller.

Key words: *pump impeller, strength, liquid flow, centrifugal force, stress state, strain state, pressure, finite element method, numerical calculation*

1. Research problem definition

The pump impellers are the high-loaded details, often working for a long time. At the same time the impellers are the most critical parts of machines and saving their strength should be provided with complete reliability. All this is defined by special difficulties and responsibilities of the pump impellers strength calculating.

Two problems were stated: 1) to calculate the liquid flow in the running parts of the pump impeller and to determine the pressure field by means of ANSYS, 2) to calculate the stress-strain state of impeller taking into account the tightness, centrifugal force and liquid flow.

As object of the research the impeller of the centrifugal section pump is presented in Fig. 1.

For an example of numerical calculation the following parameters of the pump are selected: excessive pressure of liquid on an input in the impeller of the pump of 17 MPa, the mass flow through a stage of 25 kg/s, the mass flow through multistage of 1 kg/s, operating frequency of rotation 3000 rpm. As the transferred liq-

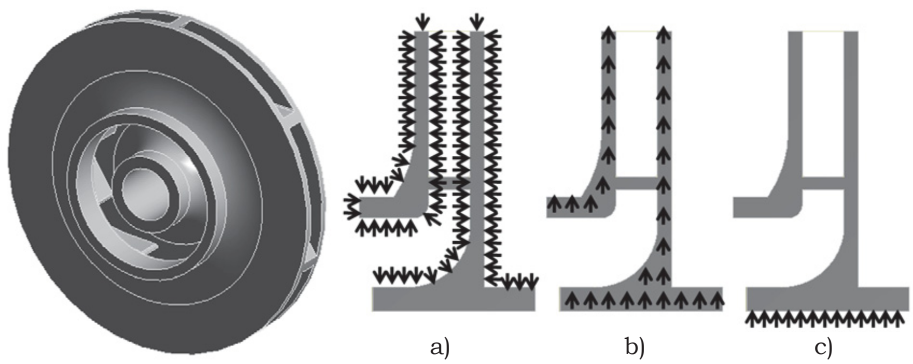

Fig. 1. 3D impeller model and loads acting on it:
a) fluid pressure; b) centrifugal forces; c) preliminary tightness

uid the water was used (ρ=1000 kg/m3). Preliminary tension of the impeller on a shaft was accepted by 0.04 mm.

2. Analysis of recent research and publications

Modern centrifugal pumps and compressors which working parameters constantly grow and can be estimated in tens of thousands of revolutions per minute and pressure up to 50 MPa, are exposed to a number of essential loadings.

One of the most loaded and responsible details of the centrifugal pump is the impeller which is exposed to the considerable static and dynamic loads. Centrifugal forces, forces of preliminary tension and pressure force refer to static loads. The loadings arising because of instability of the flow of transferred liquid refer to dynamic loads.

Due to growth of working parameters of pumps and compressors there is a need to consider the connection between the strength and hydrodynamic calculations.

It is necessary to emphasize that a current in flowing part both of a step, and separately of the impeller has a difficult character. Therefore for studying of similar currents the methods of numerical modeling are more and more widely used.

In work [1] the method is described and the results of calculations of the opened and closed wheels of centrifugal machines in elastic and elasto-plastic areas under the influence of centrifugal forces are given. The tension of the impeller was supposed to be axisymmetric what is justified for impellers with number of blades more than 12. The impeller was conditionally segmented into disk and hub parts. The round three-layer plate or a flat cover with elastic filling was model of disks with blades. At the same time for deformations of the bearing spheres Kirchhoff-Lyava's hypothesis, and for a middle layer (blades) – a hypothesis of shift deformation distribution, uniform on width, is fair. The ring or an isotropic disk was a model of hub part of a impeller. The main equations are received by a variation method, and their decision is consolidated to the solution of the integrated equations.

The finite-element method [2] is more effective and universal. The impeller is modelled by a set of terminal elements, the assumption of an axisymmetry of its tension isn't necessary. Loadings from centrifugal forces, pressure forces, etc. are easily applied.

In this operation for creation of three-dimensional (3D) models of the impeller and flowing part the software product of SolidWorks was used. For calculation of parameters of a current of liquid and the stress-strain state of the impeller the program ANSYS CFX and ANSYS Mechanical complexes respectively were used.

3. The aim of the article

The aim of the article is to calculate the stress-stain state of the centrifugal pump impeller with the tightness, centrifugal force and liquid flow and to evaluate numerically the effect of the liquid flow in the stress-strain state of the impeller.

4. The solution of the task on the flow of liquid in flowing part of the impeller

For carrying out a numerical research of distribution of liquid pressure in flowing part of the impeller the software product of ANSYS CFX was used.

In a basis of this software product the method of the numerical solution of fundamental laws of hydromechanics is put [3]: motion equations of viscous liquid together with a continuity equation. It is a sufficient condition of validity of application of results of numerical research. The current in flowing parts of hydromachines, as a rule, is turbulent. Simulation of terbulent flows was executed with use of the equations of Reynolds (1) and a continuity equation (2):

$$\frac{\partial}{\partial t}\left(\rho\bar{u}_i\right)+\frac{\partial}{\partial x_j}\left(\rho\bar{u}_i\bar{u}_j\right)+\frac{\partial}{\partial x_j}\left(\rho\bar{u}_i^{/}\bar{u}_j^{/}\right)= =-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial\bar{u}_i}{\partial x_j}+\frac{\partial\bar{u}_j}{\partial x_i}\right)\right]+f_i \qquad (1)$$

$$\frac{\partial\rho}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j\right)=0 \qquad (2)$$

where: ρ - density of liquid, x_i, x_j - coordinates, P - pressure, $\bar{u}_1$, $\bar{u}_2$, $\bar{u}_3$ - values of speeds, average on time, $\bar{u}_1^{/}$, $\bar{u}_2^{/}$, $\bar{u}_3^{/}$ - the pulsation components of speeds.

Density liquids is accepted by a constant. Currents in the rotating working organs of hydromachines are considered in the relative frame of reference. The member of f_i in the right part of the equations (1) expresses action of centrifugal and Coriolis forces:

$$\vec{f}_i=-\rho\left(2\vec{\varpi}\times\vec{u}+\vec{\varpi}\times\left(\vec{\varpi}\times\vec{r}\right)\right) \qquad (3)$$

where: ϖ - angular speed of rotation, r - the radius vector which module is equal to distance from this point to a spin axis.

As boundary conditions, the condition of "adhesion" on solid walls (speed equals to zero), distribution of all components of speed in input section and equality of the zero first derivatives (in the direction of a current) component speeds in the initial section is set.

According to a hypothesis of Bussinesk, members with speed pulsations in the equation (1) are connected with average streaming parameters by the following ratio:

$$\rho\bar{u}_i^{/}\bar{u}_j^{/}=-\mu_t\left(\frac{\partial\bar{u}_i}{\partial x_j}+\frac{\partial\bar{u}_j}{\partial x_i}\right)+\frac{2}{3}\rho\delta_{ij}k \qquad (4)$$

where: μ_t - coefficient of turbulent viscosity, $k=0.5\left(\bar{u}_i^{/}\bar{u}_j^{/}\right)$ - kinetic energy of turbulence, $\delta_{ij}=1$ in case of $i=j$, $\delta_{ij}=0$ in case of $i\neq j$.

For closing of the equations of Reynolds in ANSYS CFX a row of models of turbulence is used. The complete list of opportunities of this software product put in it mathematical apparatus and basic models of hydrodynamics can be found in documentation on this software product [4].

Calculation of a current was carried in stationary setting.

The work fluid (water under normal conditions) was supposed to be incompressible, the flow model – turbulent. For closing of the equations of Reynolds standard k-ε turbulence model was used.

When carrying out numerical research the assumption was accepted that the flow on an input to the estimated area is axisymmetric.

The algorithm of the solution of the task on a flow of liquid assumes the following stages: determination of the main assumptions (analysis type selection, turbulence model choice), preprocessing (creation of geometrical model, creation of an estimated finite and element grid, enter of properties of materials, enter of boundary conditions, choice of calculated parameters), calculation, postprocessing (viewing of results, check of reliability of the decision).

At the first stage of preparation of liquid three-dimensional geometrical model which implementation was carried out in a software product of SOLIDWORKS geometrical models which imitated liquid volume in channels of flowing part and the impeller were created, value of gaps made 3 mm, value of a gap in slit-type multiplexing made 0.5 mm (Fig. 2).

After creation of liquid models the estimated grid (Fig. 3) with use for its creation of the generator of grids of ICEM CFD was created. It allows to regulate compulsorily density of a grid, condensing it in necessary places (for example, on input and output edges of blades) and integrating it where high density of a grid isn't required. Such approach allows to save the computers resources and to receive sufficient density of a grid in the researched part of the estimated area.

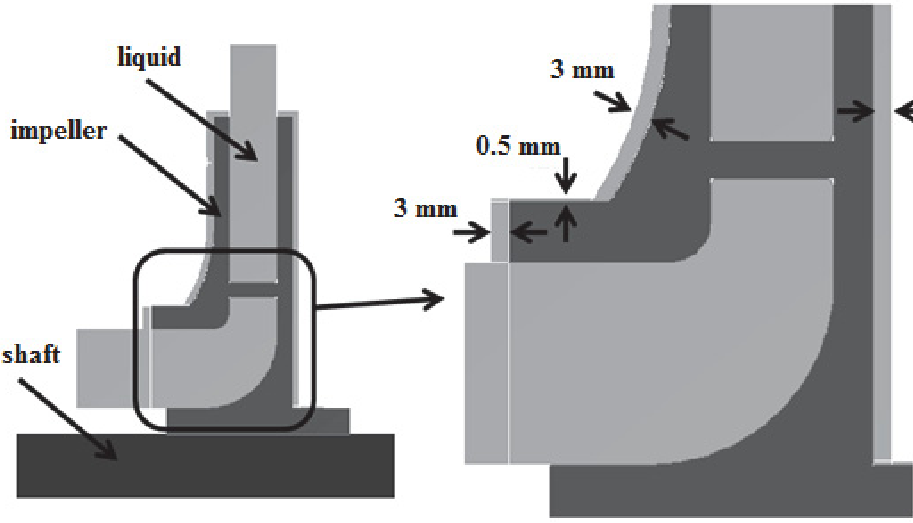

Fig. 2. Gaps in the flowing part between the impeller and the housing

As boundary conditions the total pressure on an input in the impeller and the mass expenditure on an output from the impeller and on an output from slit-type multiplexing, rotation at operating rate 3000 RPM and an adhesion condition on all walls was set.

Viewing, processing and the analysis of results was executed by means of a software of CFX-Post which has ample opportunities on visualization and an assessment of characteristics of the calculated current. The total of numerical calculation was obtaining the instantaneous values of speeds and pressure in each cell of an estimated grid.

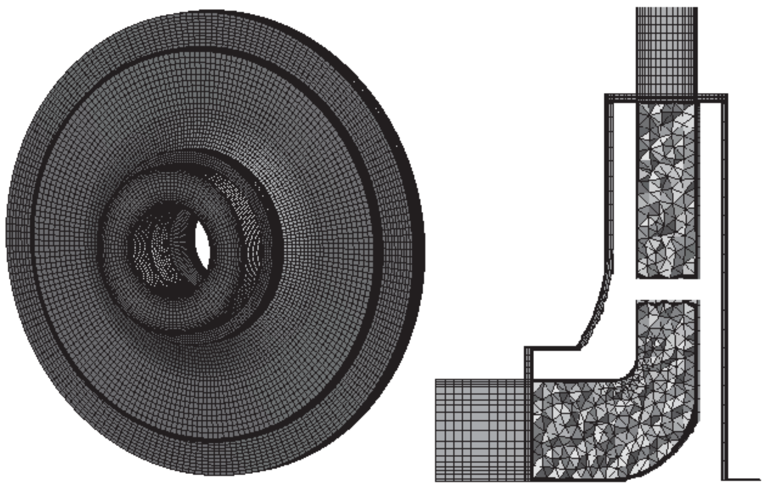

Fig. 3. Gaps in the flowing part between the impeller and the housing

5. The solution of the task on the stress-strain state of the impeller of the centrifugal pump

For carrying out numerical research the stress-strain state of the impeller of the centrifugal pump the software product of ANSYS Mechanical was used. In a basis of this software product the finite-element method is put.

Carrying out numerical modeling within this research consisted of several stages: determination of the main assumptions (analysis type selection, choice of contact model, type selection of elements), preprocessing (transmission of geometrical model, setting of properties of materials, generation of a grid, loading and fixing of construction, choice of calculated parameters), calculation, post-processing (viewing of results, check of reliability of the decision).

The shaft and the impeller are made of steel. Physicomechanical characteristics of material: the boundary of the fluidity of 400 MPa, elastic modulus is 210000 MPa, density is 7800 kg/m^3, Poisson coefficient is 0.3.

The grid which consists of 800000 prismatic elements was constructed, the characteristic size of an element is 2.5 mm (Fig. 5).

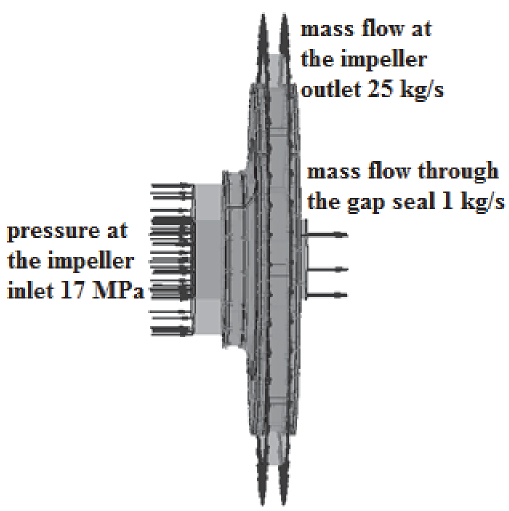

Fig. 4. The boundary conditions in the fluid flow calculation

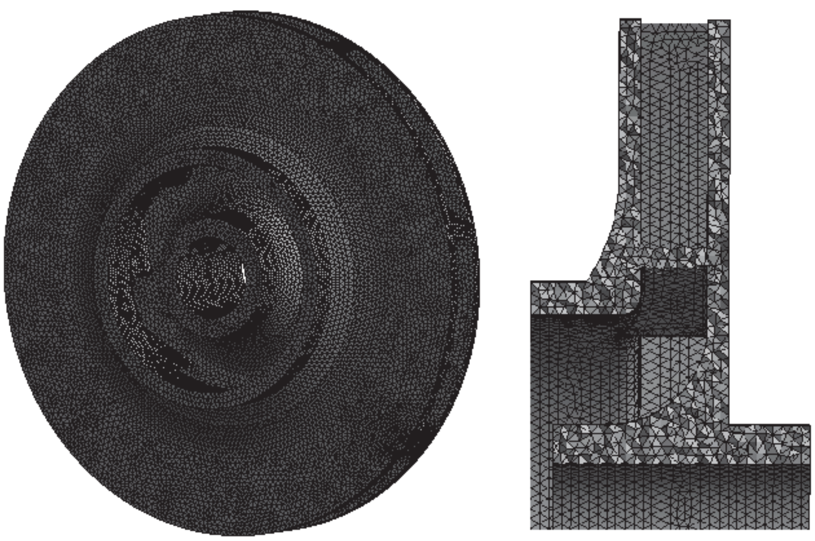

Fig. 5. Calculated mesh for impeller

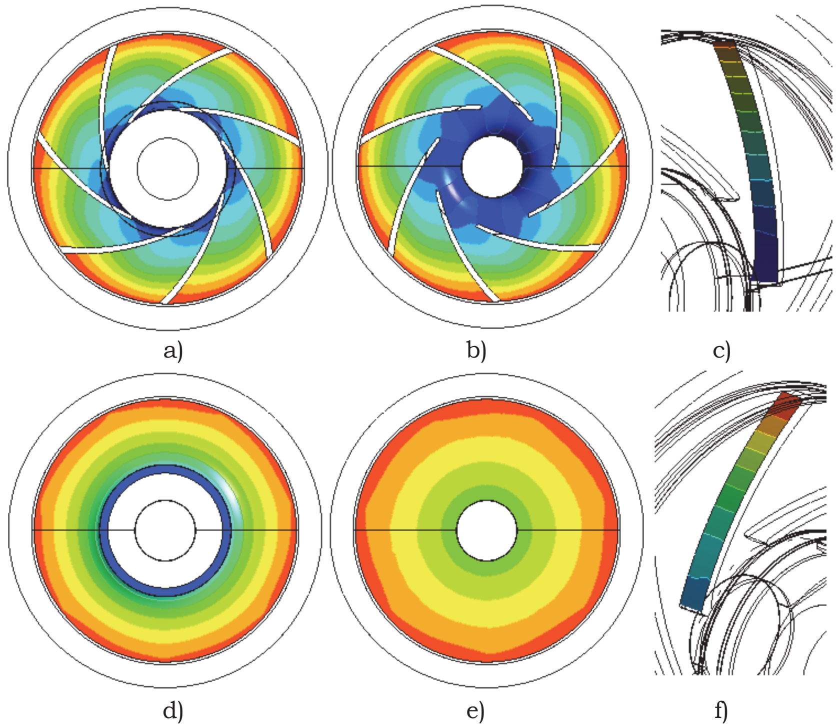

Fig. 6. Distribution of pressure flowing part of the impeller:
a) inner side of the coating disc; b) inner side of the main disc; c) inner side of the blade; d) outer side of the coating disc; e) outer side of the main disc; f) outer side of the blade

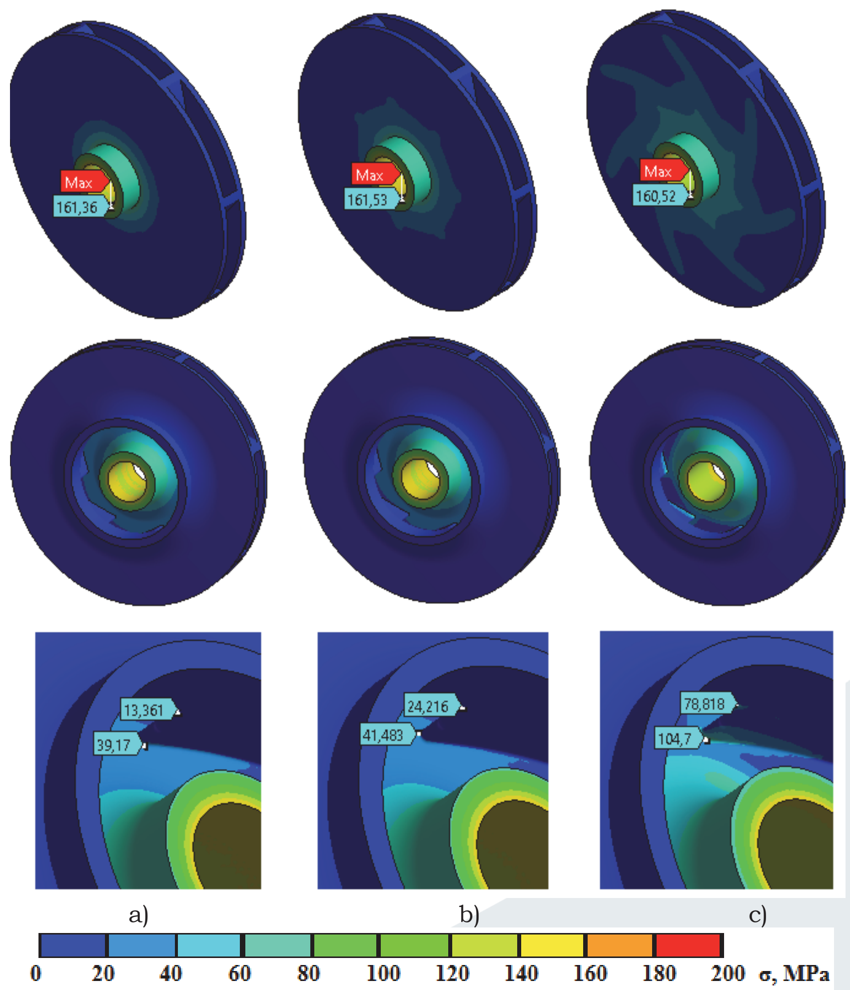

Fig. 8. Equivalent stress in the impeller:
a) accounting preliminary tightness; b) accounting preliminary tightness and centrifugal forces; c) accounting preliminary tightness, centrifugal forces and fluid pressure

At the first stage of the decision only preliminary tension was considered. At the second stage action of centrifugal forces by means of the rotational velocity function was added and rotating speed 3000 rpm was set. At the third stage the field of pressure which beforehand was received in the solution of the task on a current of liquid was imported. One end of a shaft was fixed.

6. Results of calculations and their analysis

The received results of distribution of pressure in flowing part of the impeller are provided in Fig. 6. Minimum pressure makes 17 MPa and is near an input in the impeller, maximum pressure of 18 MPa is on an output from the impeller.

Results of calculations have showed that the most dangerous are two places: on a contact surface of the impeller with shaft and on a blade edge on an input in the impeller (Fig. 7). The equivalent stress was determined by the fourth (energetic) failure theory.

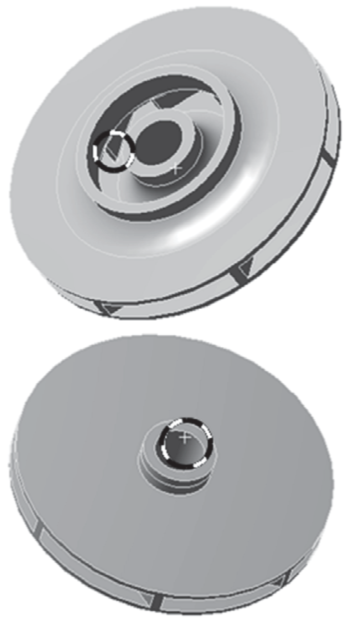

Fig. 7. Potentially dangerous places in the pump impeller

In Figs. 8, 9 the equivalent stress and deformations in the impeller are shown. Maximum stress makes 161.5 MPa and is caused by preliminary tension of the impeller on a shaft. It is necessary to mark that influence of centrifugal forces in this case is not significant, however, when calculating a rather small rotating speed of 3000 rpm was set. In case of the accounting of pressure created by a liquid flow in flowing parts the new potentially dangerous place on a joint of a blade and the main disc at an input of the impeller appears. Stress significantly increases in this place and in this case is equal to 104.7 MPa.

In the analysis of deformations of the impeller it is visible that the accounting of liquid pressures leads to essential increase in deformations on an output from the impeller. In Fig. 9 the deformations of the impeller with a scale of deformation 1000:1 are shown, that is relocation were increased by 1000 times, at the same time the sizes of the impeller didn't

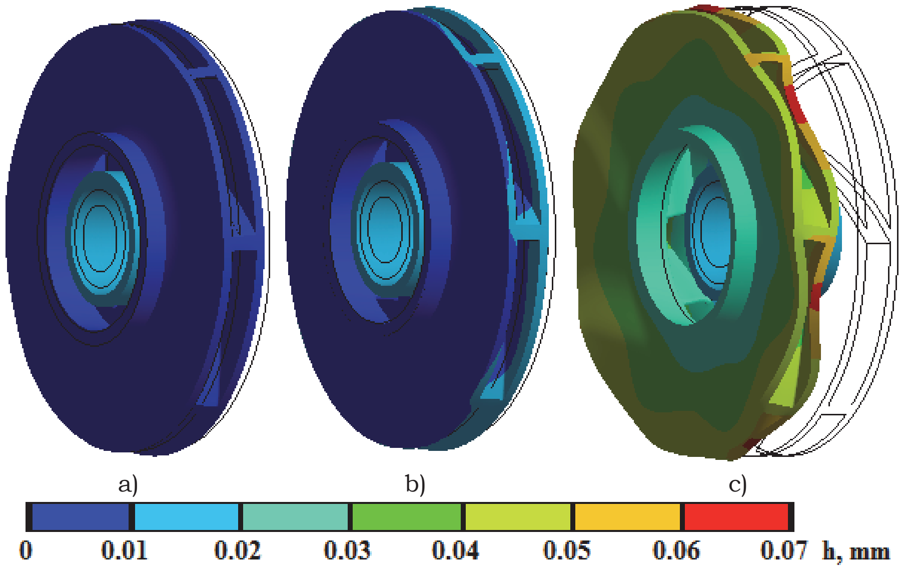

Fig. 9. Deformations of the impeller:
a) accounting preliminary tightness; b) accounting preliminary tightness and centrifugal forces; c) accounting preliminary tightness, centrifugal forces and fluid pressure

change.

7. Conclusions

The potentially dangerous places of the impeller have been found on the contact surface of the impeller with shaft and on a blade edge on an input in the impeller where blades joint to discs.

By results of calculation of the stress-strain state of the impeller of the centrifugal pump as elastic body it is possible to draw a conclusion on significant influence of liquid pressure which affects blades and disks of the impeller. The results are also the evidence of a potential danger of destruction and of need of more detailed further theoretical and experimental study of stress distribution and margins of safety in the impeller of the centrifugal machine.

As a result of the calculation pressure distribution on all surfaces of the impeller was determined. After summation of forces acting on the surface of the impeller, axial force can be calculated. Calculation of axial forces is an important task for the selection thrust bearing or hydraulic balancing device.

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