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Investigation of the flow part of a high-speed vane pump of a hydraulic unit using numerical hydrodynamic modeling

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Annotation. On this paper was studied the flow part of a high-speed vane pump of a hydraulic unit for pumping fuel of a passenger transport aircraft. To solve the task was used a numerical method for hydrodynamic modeling. It is described the used mathematical model. The obtained pressure and torque graphs on the shaft are given, depending on the feed of the centrifugal pump. The obtained data can be used for further design of the hydraulic unit.

Introduction

The hydraulic unit operation shown in Figure 1 is as follows: the working fluid under pressure, passing through the speed controller (1), enters the discharge cavity of the axial-piston hydraulic motor with an inclined disk (2), as a result of which the cylinder block begins to rotate together with pistons that perform reciprocating motion relative to the cylinder block. As a result of the rotational movement of the block relative to the stationary distributor, the cylinders of hydraulic motors are connected periodically with the suction cavity, then with the discharge cavity.

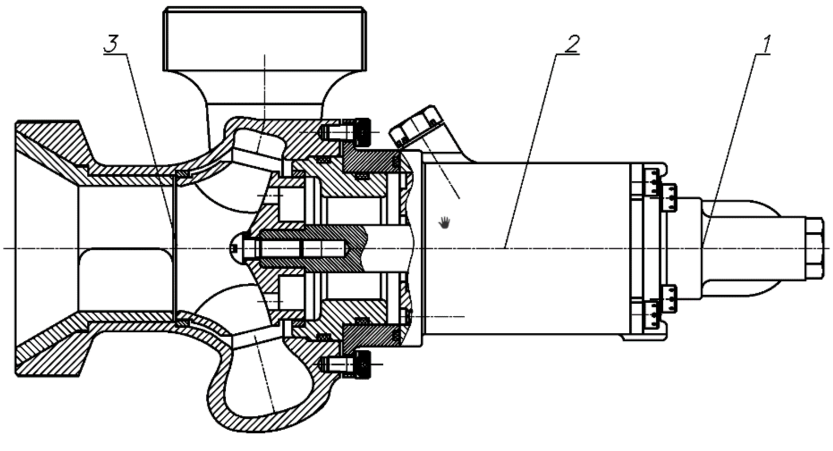

Fig. 1. Diagram of a hydraulic unit for pumping fuel of a passenger transport aircraft

The moment from the pressure forces from the cylinder block is transmitted via a spline connection to the shaft, and then through the spline connection it is transmitted to the impeller of a centrifugal pump (3). The pump casing is completely filled with liquid, and when the impeller rotates, the liquid that is in the channels of the impeller (between its vanes) will be displaced from its center to the periphery under the action of the arising forces. This will lead to the fact that a vacuum is created at the entrance to the impeller, and pressure increases on the periphery. With increasing pressure, fluid from the pump enters the pressure pipe. And under the action of a vacuum in the suction entrance, the liquid enters the pump thought the suction pipe from the feeder tank. Therefore, there is a continuous supply of fluid by a centrifugal pump from the suction to the pressure pipe [1]–[9].

The Centrifugal Pump Specifications are: Q_{nomcp}=4,5 l/s — nominal flow rate, n_{nomcp}=6650 rpm — nominal speed, H_{nomcp}=11 m — nominal head, pumped liquid-aviation kerosene TC-1.

To design the axial-piston hydraulic motor with inclined disk to act as a drive for a vane pump and to study the dynamic characteristics of a hydraulic unit, it is necessary to know the characteristics of the vane centrifugal pump. To obtain more accurate results, it is advisable to use hydrodynamic modeling methods. [10]–[11].

Flow mathematical model

This paper uses a turbulent flow model for an incompressible fluid (ρ=const). The numerical modeling is based on the solution of discrete analogues of the basic equations of hydrodynamics. The calculation is based on a mathematical model of a divided turbulent flow. The developed pump is used in the fuel transfer system of a passenger transport aircraft. It pumps aviation kerosene TS-1. The numerical hydrodynamic modeling was performed in the StarCCM + software package [12]–[15].

The mathematical model consists of a combination of differential and algebraic equations:

1)the mass conservation equation for an incompressible fluid (continuity equation): [16]–[20]

$$div\left(v\right)=0 \quad (1)$$

2)Equations of change in momentum (Navier-Stokes equations averaged by Reynolds): [16]–[20]

$$\rho\left[\frac{\partial\overline{u_i}}{\partial t}+\overline{u_j}\cdot\frac{\partial\overline{u_i}}{\partial x_j}\right]=-\frac{\partial\overline{p}}{\partial x_i}+\frac{\partial}{\partial x_i}\left[T_{ij}^{(v)}-\rho\cdot\overline{\left(u_i^{'}\cdot u_j^{'}\right)}\right] \quad (2)$$

$$Where\, T_{ij}^{(v)}=2\mu S_{ij}\ is\, the\, viscocity\, stress\, tensor\ \ for\, incompressible\ \ fluid$$

$$S_{ij}=\frac{1}{2}\left[\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right]-is\, the\, strain\, rate\, tensor$$

This system of equations is not closed. For its closure was used the $k-\omega SST3$ — semiempirical two-equation turbulence model.

3)The equation of kinetic energy transfer of turbulence:[2]–[9]

$$\frac{\partial K}{\partial t}+\overline{u_j}\cdot\frac{\partial K}{\partial x_j}=P_K-\beta^{*}\cdot K\cdot\omega+\frac{\partial}{\partial x_i}\cdot\left[\left(v+\sigma_K\cdot v_T\right)\cdot\frac{\partial K}{\partial x_j}\right] \quad (3)$$

4)The transport equation for the turbulence kinetic energy relative dissipation rate:[9]–[15]

$$\frac{\partial\omega}{\partial t}+\overline{u_j}\cdot\frac{\partial\omega}{\partial x_j}=\alpha\cdot S^2-\beta\cdot\omega^2+\frac{\partial}{\partial x_i}\cdot\left[\left(v+\sigma_{\omega1}\cdot v_T\right)\cdot\frac{\partial\omega}{\partial x_j}\right]+2\cdot\left(1-F_j\right)\cdot\sigma_{\omega2}\cdot\frac{1}{\omega}\cdot\frac{\partial K}{\partial x_j}\cdot\frac{\partial\omega}{\partial x_j} \quad (4)$$

Construction of the predicted characteristic of Hteor (Q) using the jet theory:

Theoretical head for a finite number of blades:[1]

$$H_T=\frac{\omega}{g}\left(R_2^2 y\omega-\frac{R_2 Q}{2\pi R_{u2} b_2\psi_2 tg\beta_2}\right) \quad (5)$$

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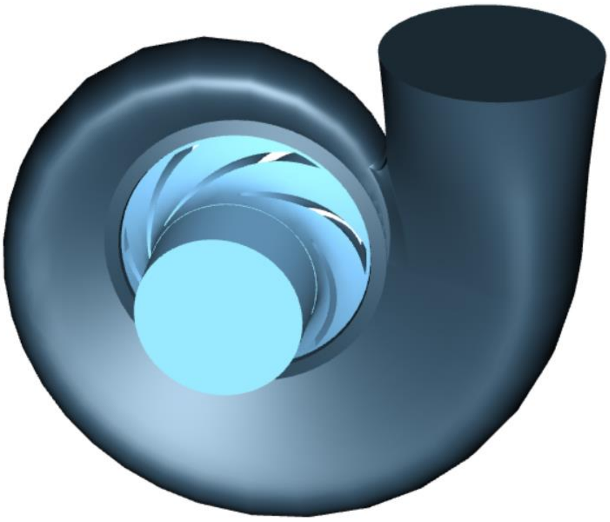

Fig. 2. Flow 3D-model for the centrifugal pump

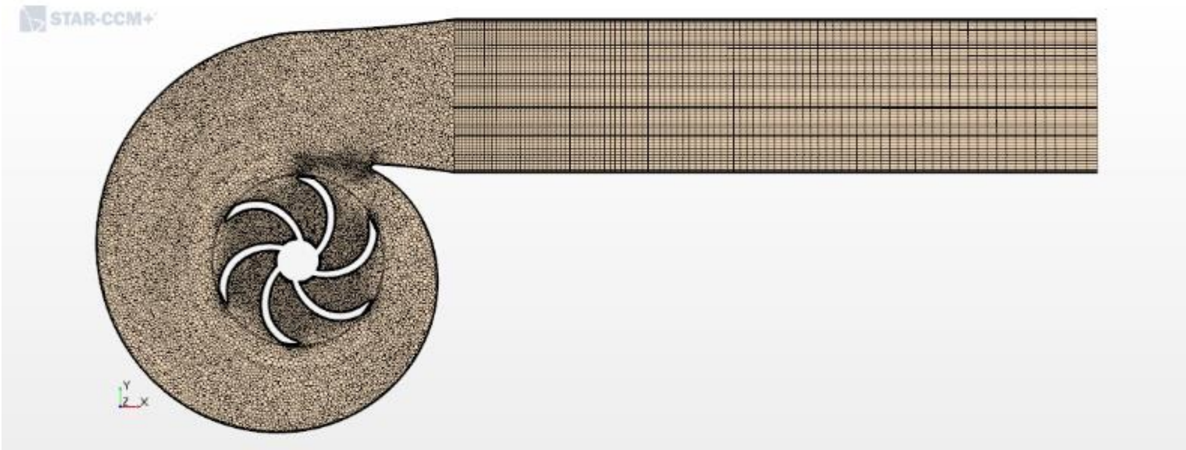

Fig. 3 Extruder grid

Outlet beam:

$$H_T = \frac{Q\omega}{A_s g} \quad (6)$$

Differential pressure on the gap seal:[1]

$$\Delta H = H_{Topt} - \eta_h H_{Topt} \quad (7)$$

Hydraulic loss:[2]

$$\Delta H = h_{fr}\left(Q\right) = H_T\left(1-\eta_h\right)\bar{Q}^2 = \Delta H\bar{Q}^2 \quad (8)$$

Where

$$\bar{Q} = \frac{Q}{Q_{opt}} = \frac{Q}{4{,}5\cdot 10^{-3}} \quad (9)$$

We find the predicted characteristic by subtracting from the theoretical pressure head the hydraulic losses:(10)

$$Hteor(Q) = H_T - h_{fr}(Q)$$

When constructing the predicted characteristics, an assumption was made about the shockless entry of the working fluid into the impeller. [2]

The magnitude of the flow Ql through the impeller seal near to the optimal regime is taken constant. This allows you to find for each point the corresponding value of the pump flow Qp = Q-Ql and build its predicted pressure characteristic Hteor (Q) at a given rotational speed of the impeller. [2]

Calculation results

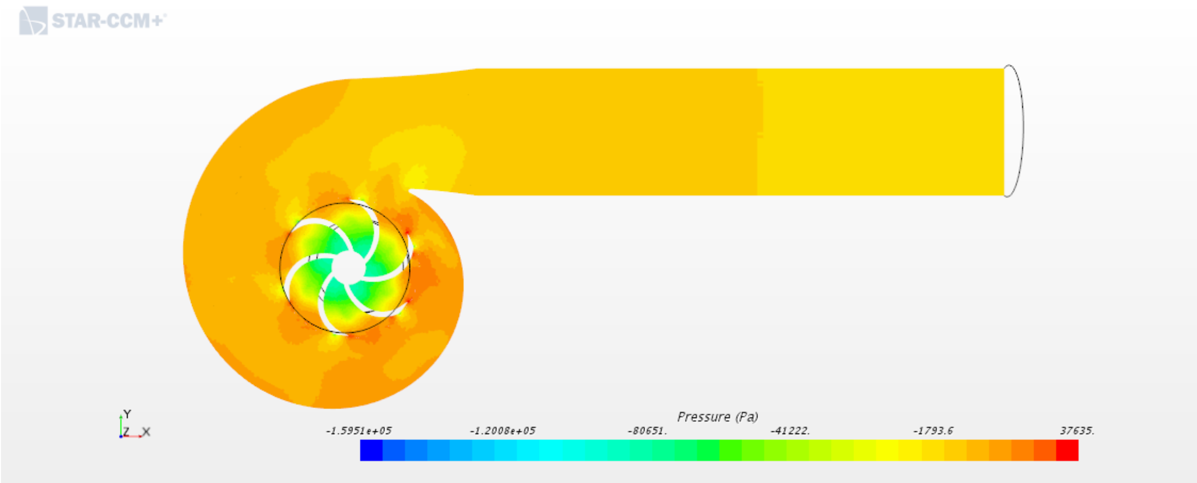

Fig. 4. Scalar field of the fluid pressure in the pump flow part at optimal operation

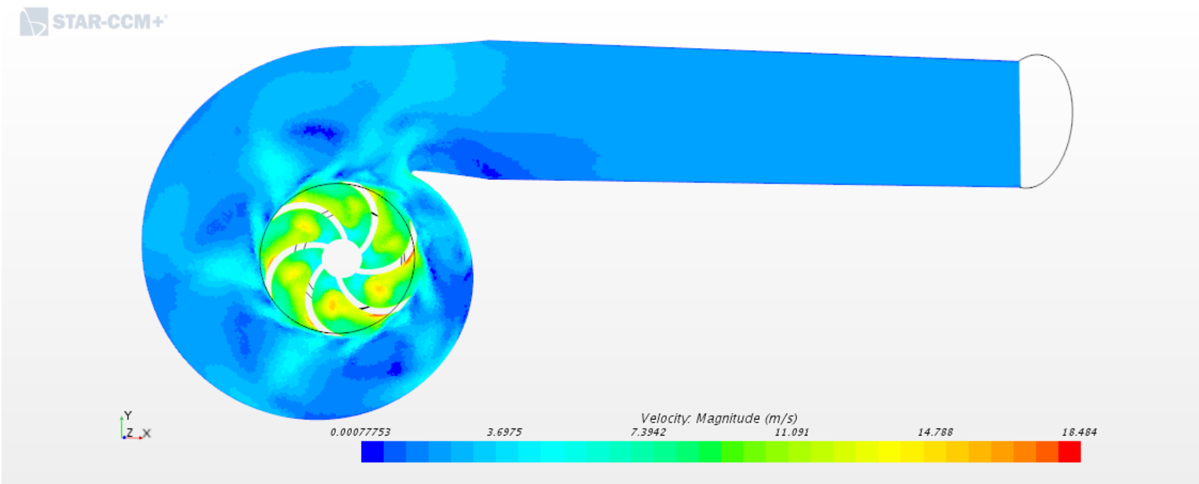

Fig. 5. Scalar field of the fluid velocity in the pump flow part at optimal operation

The calculation points were obtained from the STARCCM + software package, according to which the graphs of the desired dependencies were plotted in Excel, shown at Fig.6 and 7.

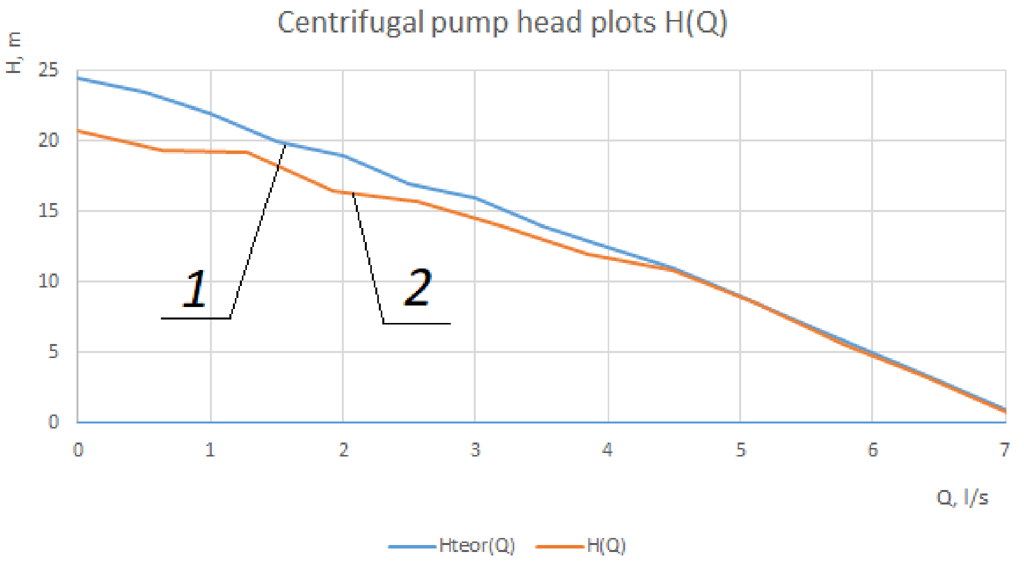

Fig. 6. Graph Head vs pump flow

Predictive characteristic of the Hteor (Q) vane pump, obtained using the jet theory;

1. Characteristic H (Q) of a vane pump obtained by hydrodynamic modeling in the STARCCM + software package.

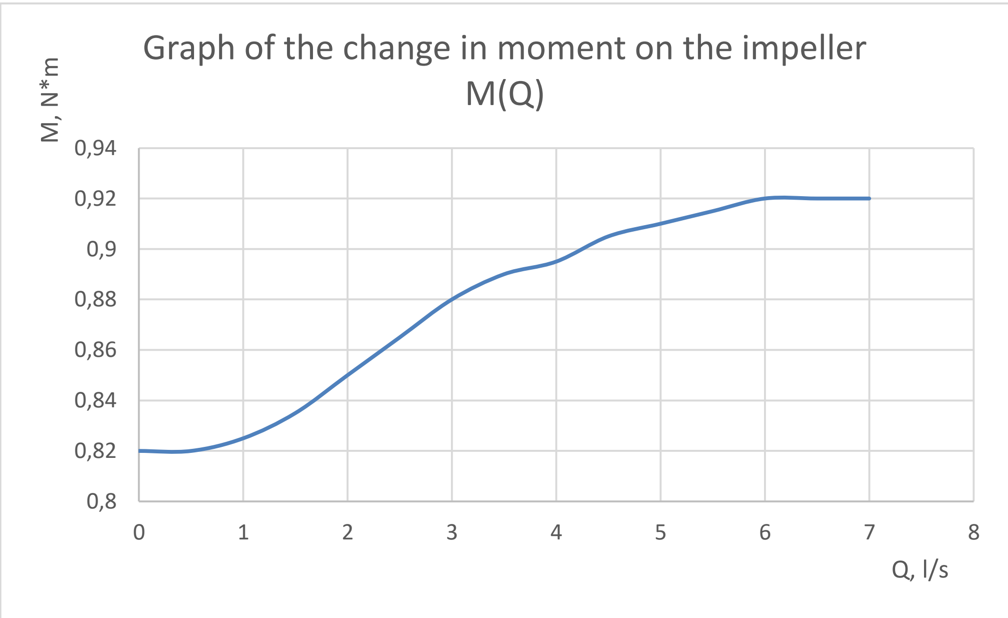

Fig. 7. Graph of the impeller torque based on pump flow

Conclusions

The analysis showed the effectiveness of this mathematical model. The calculation showed that the characteristic of the impeller pump H (Q) near the optimal operation of the hydraulic unit calculated using hydrodynamic modeling tools has a minimum discrepancy with the predicted characteristic Hteor (Q), calculated using the jet theory. Also were obtained the results for designing an axial piston hydraulic motor with an inclined disk as a vane pump drive and the dynamic characteristics of a hydraulic unit.

References

[1] Methodical manual on course design of vane pumps / Rudnev S. S., Matveev I. V.; Bauman higher technical school: Moscow, 1975. - 78 p.
[2] Centrifugal and axial pumps: theory, design and application / Stepanov L. I., Polyakovsky V. I.: Publishing house: State scientific and technical publishing house of machine-building literature, 1960.-468 PP.
[3] Ageev S., Druzhinin E., ESP with increased gas content at the inlet. Drilling and oil, 2003.
[4] Lomakin, V.O., Kuleshovav, M.S., Bozh'eva, S.M. Numerical Modeling of Liquid Flow in a Pump Station (2016) Power Technology and Engineering, 49 (5), pp. 324–327. DOI: 10.1007/s10749-016-0623-9
[5] Lomakin V.O. Investigationoftwo-phaseflowinaxial-centrifugalimpellerby hydrodynamic modeling methodsProceedings of 2015 International Conference on Fluid Power and Mechatronics, FPM 2015 no 7337302 pp 1204–6. DOI: 10.1109/FPM.2015.7337302
[6] Lomakin V O, Petrov AIandKuleshova M S2014Investigationof a two-phase flowinan axialcentrifugalwheelbymethodsofhydrodynamicmodelingScienceandeducation: electronic scientific edition of Bauman MSTU (9) pp 45–64 (in Russian)
[7] Brennen C.E. Fundamentals of Multiphase Flows. Cambridge: Cambridge University Press, 2005. 410 p.
[8] Wilcox D.C. Turbulence Modeling for CFD. 3rd ed. DCW Industries, 2006. 515 c.
[9] Ding H., Visser F.C., Jiang Y., Furmanczyk M. Demonstration and Validation of a 3D CFD Simulation Tool Predicting Pump Performance and Cavitation for Industrial Applications // Journal of Fluids Engineering. 2011. Vol. 133, no. 1. P. 011101-011101-1 DOI: 10.1115/1.4003196
[10] Lomakin V.O. Numerical simulation of flow parts of pump models and verification of simulation results by comparison of obtained values with experimental data. Naukaiobrazovanie MGTU im. N.E. Baumana = Science and Education of the Bauman MSTU, 2012, no. 5, pp. 52–62. DOI: 10.7463/0512.0356070 (in Russian).
[11] Lomakin V.O., Petrov A.I. Verification of the calculation results using hydrodynamic modeling package STAR-CCM + for flow channel of the centrifugal pump AX 50-32-200. Izvestiyavysshikhuchebnykhzavedeniy. Mashinostroenie [Proceedings of Higher Educational Institutions. MachineBuilding], 2012, no. S, pp. 6–9 (inRuss.).
[12] ZhuJ,ZhuH,Zhang Jand ZhangH-Q2019Anumericalstudyonflowpatternsinsidean electrical submersible pump(ESP) and comparisonwithvisualization experiments Journal of Petroleum Science and Engineering 173 pp 339–350 DOI: 10.1016/j.petrol.2018.10.038
[13] TimushevSF,KnyazevV A,PanaiottiSS,SoldatovVA andRohatgiUS2009 Numerical procedureforassessmentofcentrifugalpumpcavitationerosionASMEInternational Mechanical Engineering Congress and Exposition, Proceedings 14 pp 73–78 DOI: 10.1115/IMECE2008-67177
[14] Arefyev, K.Y., Voronetsky, A.V., Suchkov, S.A., Ilchenko, M.A. Computational and experimental study of the two-phase mixing in gas-dynamic ignition system (2017) Thermophysics and Aeromechanics, 24 (2), pp. 225–237. DOI: 10.1134/S086986431702007X
[15] Lozovetskii, V.V., Pelevin, F.V. Resistance of spherical fills in the case of flow of single- and two-phase media (2009) Journal of Engineering Physics and Thermophysics, 82 (2), pp. 277–282. DOI: 10.1007/s10891-009-0194-9
[16] Ivanovsky V.N., Sabirov A.A., Donskoy U.A. The effect of dissolved carbon dioxide on carbonate precipitation during oil production using electric centrifugal pumps. Oil industry, 2010
[17] Bellarya S.A.I., Husain A., Samadc A., KanaiaR.A.. Performance Optimization of Centrifugal Pump for Crude Oil Delivery. TheJournalofEngineeringResearch (TJER), Vol. 15, No. 1 (2018) 88–101

[18] Antonio C. Bannwart. Experimental Study of Centrifugal Pump Handling Viscous Fluid and Two-Phase Flow. SPE production&operations 30(02), 2013
[19] Larry Bachus, Angel Custodio. Know and Understand CentrifugaI Pumps. ElsevierInc., 2003, 270 p.
[20] Gülich J.F. Centrifugal Pumps. Springer-Verlag Berlin Heidelberg, 2010, 964 p.