Complex optimization of vaned stators of high-speed centrifugal pumps for thermal stabilization systems

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Abstract. The scope of the pumps for thermal stabilization systems, the main elements of such pumps, their configurations and basic requirements for pumps of this type are considered. In order to increase efficiency, an optimization of the pump vaned stator is carried out. Such simulation results as efficiency and pressure values for each variant are obtained. The data received is analyzed and appropriate conclusion on the efficiency of optimization approach is made.

Introduction

High-speed centrifugal pumps are widely used in thermal stabilization systems in aerospace industry, for computing hardware, as well as in various ground-based installations and radar equipment.

The thermal stabilization system includes a circulation pump (to ensure the reliability of the system two pumps are used in it, one of which acts as a backup), a liquid heat exchanger, a pump control and monitoring unit, and system status sensors (temperature, pressure, etc.). Ethylene glycol coolants (C-40, C-65 and its analogues) are used as a working fluid in the system.

The pump includes an impeller, a vaned stator, a spiral volute with a diffuser and a power end, including a sealing element — pumps with a magnetic coupling and pumps with a lined motor [1] - [3], friction bearings [4], etc. are used. The introduction of the vaned stator to the pump design allows to decrease the radial force on the rotor, which is caused by the violation of velocities and pressure distribution around the circumference of the impeller. Also, by replacing the vaned stator with another one within the same pump, it is possible to change the characteristics of the pump, which in the case of an additional application, for example, frequency regulation of the pump [5] - [8], allows you to get the optimum working point in a wide range of characteristics, using, in fact, one and the same pump. Based on the results of optimization, it is recommended to perform full-scale tests using elements manufactured with additive technologies [9] - [12].

An indisputable fact is that optimization is an important part of pump designing process, especially in terms of impeller geometry. The most successful variants meet all the requirements and allow to achieve the best efficiency. However, the efficiency of the pump is also affected by losses in the stators and volutes. In this article, the optimization of the vaned stator and the expediency of its application is considered. One of the goals of this study is to determine the optimal parameters of a centrifugal pump vaned stator for heat stabilization systems. The parameters of the stator must comply with the requirements for pumps of this type, such as:
1. Minimum pump weight;
2. Minimum pump size;
3. High efficiency.

The optimization is based on parametrized pump geometry and CFD-simulation of the flow inside it. The work flow might be described as follows:
1. 3-D model of the pump is prepared;
2. Geometry is then transferred to the CFD-software;
3. Pump flows simulation is run;
4. Obtained results are analyzed.

**Methods**

The following geometrical parameters of the impeller were considered:
- Number of stator blades (nBl), impeller geometry is not changing and consists of 6 blades;
- Stator blade angle at the inlet (Beta1);
- Stator blade angle at the outlet (Beta2);
- The spacing between impeller outlet and blade leading edge (Inlet);
- The spacing between stator outlet and blade trailing edge (Outlet).

![Fig. 1. Optimization variables](image_url)
The range of optimization parameters values is chosen based on there requirements described above.

The simulation is based on mathematical models describing turbulent flow in pump sand providing their characteristics with out actual field tests. To achieve these values of the turbulent flow it is required to solve a set of Reynolds-averaged Navier-Stokes equations[13]–[20].

Reynolds-averaged Navier-Stokes equation is as follows:

\[
\rho \frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left(T_{ij}^{(v)} - \rho \langle u_i u_j \rangle \right), \quad \text{where (1)}
\]

\( U_i, P \) — averaged velocity and pressure,
\( T_{ij}^{(v)} \) — viscous stress tens or for in compressible fluid,
\( \rho \langle u_i u_j \rangle \) — Reynolds stresses.

Mass conservation equation as follows:

\[
\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial \bar{u}_i}{\partial x_j} = -\frac{1}{\bar{P}} \frac{\partial \bar{p}}{\partial x_i} + j \frac{\partial^2 \bar{u}_i}{\partial x_j^2} \quad \text{(2)}
\]

Introduction of the Reynolds-averaged Navier-Stokes equations makes the system of equations non-closed, since there are additional unknown Reynolds stress. To solve this system the semi-empiricalk-\( \omega \)SS Turbulence model issued. This model includes two additional equations.

Turbulence kinetic energy transport equation as follows:

\[
\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = P_e - \beta * k \omega + \frac{\partial}{\partial x_j} \left( (v + \sigma_k v_T) \frac{\partial k}{\partial x_j} \right), \quad \text{where (3)}
\]

\( \omega \) — specific dissipation rate,
\( k \) — turbulent kinetic energy,
\( v_T \) — turbulent viscosity,
\( P_e \) — turbulence energy generation.

Specific dissipation rate transfer equation described as follows:

\[
\frac{\partial \omega}{\partial t} + U_j \frac{\partial \omega}{\partial x_j} = \alpha * S^2 - \beta \omega^2 + \frac{\partial}{\partial x_j} \left( (v + \sigma_k v_T) \frac{\partial \omega}{\partial x_j} \right) + 2(1-F_i)\sigma_{\omega \omega} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_j} \quad \text{(4)}
\]

Reynolds stresses are determine during the Boussinesq hypothesis

\[
\rho \langle u_i u_j \rangle = 2\mu_t \left[ 1 \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{1}{3} \frac{\partial U_k}{\partial x_k} \delta_{ij} \right] \frac{2}{3} pk \delta_{ij}, \quad \text{where (5)}
\]

\( \delta_{ij} \) — Kronecker symbol.

AsaCFD-response pump head (H) and efficiency (EFF) are obtained.

Optimization work flow is fully automated: the geometry is changed by the optimization program and is then transferred to the CFD-code, the results are analyzed by the program as well. The decision on geometry changes is made by hybrid adaptive SHERPA algorithm. This method is a combination of several mathematical algorithms, which are used simultaneously and adapt the m selves to the case under consideration. The results are assessed by performance (E) which is the ratio of design under consideration efficiency (\( \eta_d \)) to initial design efficiency (\( \eta_0 \)).
\[ E = \frac{\eta_i}{\eta_0} \quad (6) \]

Fig.2. Velocity field distribution for the optimized design of the 3rd set-up

**Results**

There search includes 4 different set-ups for optimization and comparison, all of the minclude the same values of pump head and volumetric flow rate:

- Head(H)\(\approx\)45m;
- Volumetric flow rate(Q)=4.5m\(^3\)/h.

These setups differ in rotation rate values:

1. Rotation rate(n)=3000(rpm);
2. Rotation rate(n)=8000(rpm);
3. Rotation rate(n)=20000(rpm);
4. Rotation rate(n)=24000(rpm).

125 variants for set-up 1 are considered. The results of top-5 are shown in Table 1.

**Table 1. Optimization results for the 1st set-up**

| Design variant | E     | EFF   | H     | beta1 | beta2 | nBl | inlet | outlet |
|----------------|-------|-------|-------|-------|-------|-----|-------|--------|
| 124            | 1.886 | 58.690| 46.199| 0.0677| 0.25  | 7   | 0.0556| 0.825  |
| 79             | 1.857 | 57.781| 43.800| 0.0633| 0.24  | 7   | 0.055 | 0.839  |
| 71             | 1.857 | 57.780| 43.800| 0.0633| 0.51  | 7   | 0.055 | 0.839  |
| 47             | 1.857 | 57.779| 43.800| 0.0633| 0.2   | 7   | 0.055 | 0.839  |
| 48             | 1.696 | 52.778| 51.448| 0.0988| 0.205 | 7   | 0.0526| 0.849  |

The results obtained show that an initial design is not efficient enough, and optimization helps to improve the pump efficiency.

Then 100 variants for set-up 2 are considered, top-5 are showing in Table 2.
Based on the results of two first set-ups it might be concluded that optimization is an effective tool, that helps improve machine efficiency in a wide range of its operational conditions.

73 variants for the third setup are considered, the best variants are showing Table 3.

**Table 2. Optimization results for the 2\textsuperscript{nd} set-up**

| Design variant | E   | EFF  | H    | beta1 | beta2 | nBl | inlet | outlet |
|----------------|-----|------|------|-------|-------|-----|-------|--------|
| 43             | 1.267 | 60.092 | 53.726 | 0.240 | 0.805  | 7   | 0.063 | 0.722   |
| 89             | 1.245 | 59.038 | 54.055 | 0.083 | 0.818  | 5   | 0.090 | 0.800   |
| 33             | 1.235 | 58.561 | 54.675 | 0.064 | 0.805  | 5   | 0.063 | 0.821   |
| 77             | 1.231 | 58.396 | 54.665 | 0.064 | 0.851  | 5   | 0.059 | 0.722   |
| 35             | 1.225 | 58.083 | 53.693 | 0.096 | 1.000  | 5   | 0.072 | 0.737   |

**Table 3. Optimization results for the 3\textsuperscript{rd} set-up**

| Design variant | E   | EFF  | H    | beta1 | beta2 | nBl | inlet | outlet |
|----------------|-----|------|------|-------|-------|-----|-------|--------|
| 73             | 1.027 | 72.024 | 48.713 | 0.162 | 0.306  | 7   | 0.067 | 0.719   |
| 72             | 1.025 | 71.902 | 48.268 | 0.162 | 0.205  | 7   | 0.068 | 0.719   |
| 39             | 1.025 | 71.902 | 48.268 | 0.162 | 0.306  | 7   | 0.068 | 0.719   |
| 50             | 1.025 | 71.880 | 48.687 | 0.162 | 0.306  | 7   | 0.070 | 0.719   |
| 63             | 1.023 | 71.779 | 47.836 | 0.143 | 0.306  | 7   | 0.068 | 0.719   |
Fig. 3. Efficiency for the best variant of the 3\textsuperscript{d} set-up

Fig. 4. The change of efficiency for the 3\textsuperscript{d} set-up during optimization

For the forth set-up 78 variant sare considered, the best of the mare showing Table 4.

**Table 4.** Optimization results for the 4\textsuperscript{th} set-up

| Design variant | E | EFF | H | beta1 | beta2 | nB1 | inlet | outlet |
|----------------|---|-----|---|-------|-------|-----|-------|--------|
The analysis of the results obtained shows that optimization is an effective tool helping achieve high efficiency for the pumps of this type only by changing vaned stator geometry, corresponding methods need to be integrated in the design process of such pumps and additional geometry changes of other pump elements should be also considered.

Discussions
1. Achieved results show that optimization improve efficiency values of the pump, e.g. for the first set-up the best design efficiency is 58.7% when initial design value is only 31.1 % (for the second the yare 60.1% and 47.4% respectively, for the third set-up — 72.0 % and 70.1 % respectively and for the forth — 75.1 % and 69.1 % respectively);
2. The comparison (Fig.5) shows that it is preferable to choose pumps with higher rotation speed, but for such cases the irreliability should also be considered;
3. The results obtained call for additional investigations including optimization of all pump elements geometries, such as impeller, volute and diffuser.

References:
[1] Chaburko, P., Kossova, Z. Wet part hybrid optimization method of hermetic pump(2019) IOP Conference Series: Materials Science and Engineering, 492 (1), № 012011.
[2] Guo,B.,Zhang,M.,Jin,F.Use of Magnetic Clutch to Improve Performance of Sucker Rod Pumps (2002) SPEWestern Regional/AAPG Pacific Section Joint Meeting, pp.541–552.
[3] Qiu, S.-J. ,Zhu,J.-H. Magnetic Drive Pump Selection in Small Flow rate and High Head Design Conditions (2018) Petrochemical Equipment,47(2),pp.20–24.
[4] Mahner, M., Bauer, M., Lehn, A., Schweizer, B. An experimental investigation on the influence of an assembly preload on the hysteresis, the drag torque, the lift-off speed and the thermal behavior of three-pad air foil journal bearings(2019) Tribology International, 137, pp. 113–126.
[5] Ni, J., Peng, L., Chen, G. Nonlinear PID synchro control on pattern drawing machine with four cylinders. (2011) Proceedings of 2011 International Conference on Modelling, Identification and Control, ICMIC 2011 5973725, pp. 325-330

[6] Zadorozhnaya, N.M., Lisitsyn, A.N. Automatic Tuner for PID Controllers with Elements of Artificial Intelligence. (2018) IOP Conference Series: Earth and Environmental Science 194(2), 022046 DOI:10.1088/1755-1315/194/2/022046

[7] Bogdanova, Y., Guskov, A. Synergetic synthesis of control laws for left ventricular assist device rotor on magnetic suspension. (2016) Proceedings of 2016 International Conference "Stability and Oscillations of Nonlinear Control Systems" (Pyatnitskiy's Conference), STAB 2016 7541168. DOI: 10.1109/STAB.2016.7541168

[8] Bushuev A.Yu., Ivanov M.Yu., Korotaev D.V. Minimization of amiss match time of movement of actuators of a throttle synchronization system. (2018) Journal of Physics: Conference Series 1141(1), 012090DOI:10.1088/1742-6596/1141/1/012090

[9] Azarov, A.V., Antonov, F.K., Golubev, M.V., Khaziev, A.R., Ushanov, S.A. Composite 3D printing for the small size unmanned aerial vehicle structure (2019) Composites Part B: Engineering, 169, pp. 157–163.

[10] Sanders P., Young A.J., Qin Y., Fancey K.S., Reithoffer M.R., Guillet-Nicolas R., Kleitz F., Pamme N., Chin J.M. Stereolithographic 3D printing of extrinsic callly self-healing composites (2019) Scientific Reports, 9(1), #388.

[11] Li J., Rudykh, S. T Unable microstructure transformations and auxetic behavior in 3D-printed multiphase composites: The role of inclusion distribution (2019) Composites Part B: Engineering, 172, pp. 352–362.

[12] Lengauer, W., Duretek, I., Fürst, M., Schwarz, V., Gonzalez-Gutierrez, J., Schuschnigg, S., Kukla, C., Kitzmantel, M., Neubauer, E., Lieberwirth, C., Morrison, V. Fabrication and properties of extrusion-based 3D-printed hardmetal and cermet components (2019) International Journal of Refractory Metals and Hard Materials, 82, pp. 141–149.

[13] Lomakin, V.O. Investigation of two-phase flow in axial-centrifugal impeller by hydrodynamic modeling methods (2015) Proceedings of 2015 International Conference on Fluid Power and Mechatronics, FPM 2015, № 7337302, pp. 1204–1206.

[14] Lomakin, V., Cheremushkin, V., Chaburko, P. Investigation of vortex and hysteresis effects in the inlet device of a centrifugal pump (2018) 2018 Global Fluid Power Society PhD Symposium, GFPS2018, №8472374

[15] A Gouskov et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012013

[16] A Protopopov and V Vigovskij 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012003

[17] Sun-Sheng, Y., Singh, P., Zhang, H. Flow investigations of reverse running volute pumps with backward vanes in comparison to forward type turbine vanes (2019) Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 233 (1), pp. 111–131.

[18] Katalazhnova, I. Correlation and regression method of centrifugal pump geometry optimization (2019) Lecture Notes in Mechanical Engineering,(9783319956299), pp.1839–1845.

[19] Guleren, K.M. Automatic optimization of a centrifugal pump based on impeller–diffuser interaction (2018) Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 232(8), pp.1004–1018.

[20] Siddique, M.H., Afzal, A., Samad, A. Design Optimization of the Centrifugal Pumps via Low Fidelity Models (2018) Mathematical Problems in Engineering, 2018, №3987594.

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