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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 bedescribed as follows:

1. 3-D model of the pump is prepared;
2. Geometry is then transferred to the CFD-software;
3. Pump flows imulationis 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).

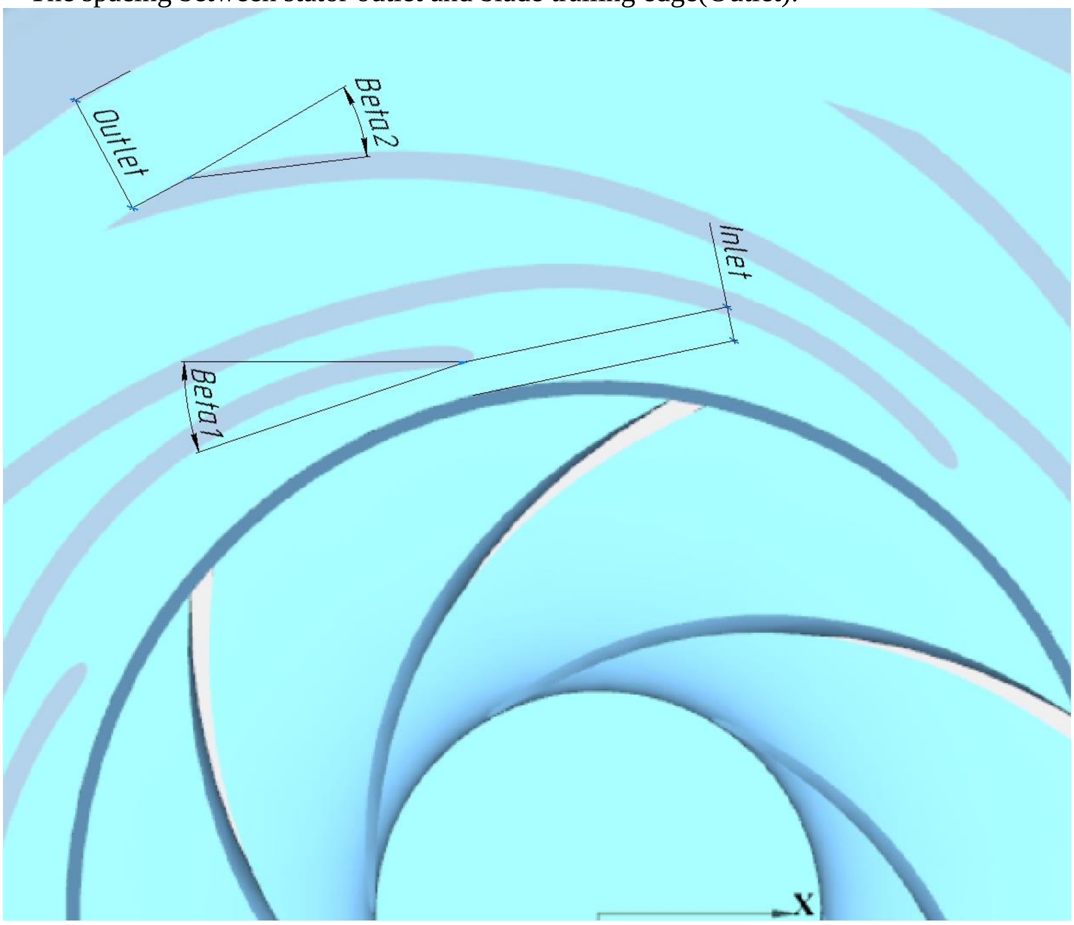

Fig. 1. Optimization variables

The range of optimization parameters values is chosen based on there equirements described above.

The simulation is based on mathematical models describing turbulent flow sin 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-Stokese equation is as follows:

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

U_i , P —averaged velocity and pressure,

$T_{ij}^{(v)}$ — viscous stress tens or forin compressible fluid,

$\rho\left\langle u_i u_j\right\rangle$ —Reynolds stresses.

Mass conservation equationis as follows:

$$\left(\frac{\partial \bar{u}_i}{\partial t}+\bar{u}_j\frac{\partial \bar{u}_i}{\partial x_j}\right)=-\frac{1}{\bar{\bar{p}}}\frac{\partial \bar{p}}{\partial x_j}+\vartheta\frac{\partial^2 \bar{u}_i}{\partial x_j{}^2}\quad(2)$$

Introduction of the Reynolds-averaged Navier-Stokes equations makes the system of equationsnon-closed, since there are additional unknown Reynolds stress. To solve this system the semi-empiricalk-ωSS Tturbulence model isused. This model includes two additional equations.

Turbulence kinetic energy transport equationis as follows:

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

ω —specific dissipation rate,

k — turbulent kinetic energy,

v_T —turbulent viscosity,

P_k —turbulence energy generation.

Specific dissipation rate transferis described as follows:

$$\frac{\partial \omega}{\partial t}+U_j\frac{\partial \omega}{\partial x_j}=\alpha^* S^2-\beta\omega^2+\frac{\partial}{\partial x_i}\left[(v+\sigma_k v_T)\frac{\partial k}{\partial x_j}\right]+2(1-F_1)\sigma_{\omega 2}\frac{1}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_j}\quad(4)$$

Reynolds stresses are determine dusing the Boussinesq hypothesis

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

δ_{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 (η_i) to initial design efficiency (η_0).

$$E=\frac{\eta_i}{\eta_0} \quad (6)$$

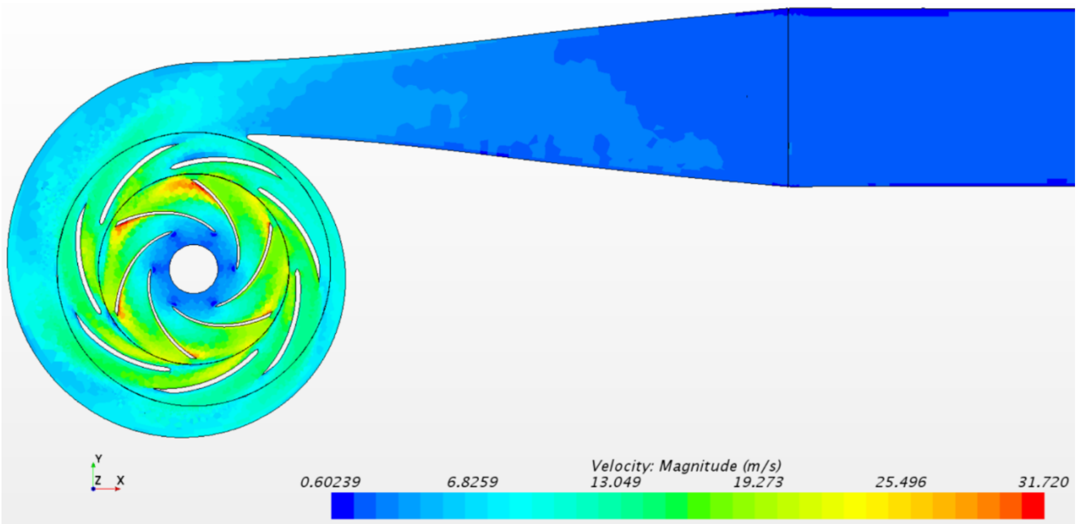

Fig.2. Velocity field distribution for the optimized design of the 3dset-up

Results

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

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

These setups differin 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-up1are considered. The results of top-5areshowninTable1.

Table 1. Optimization results for the1stset-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

Theresultsobtainedshowthatinitialdesignisnotefficientenough,andoptimizationhelpstoimprovethepumpefficiency.

Then 100 variants for set-up 2 are considered, top-5 are showing in Table 2.

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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^{d}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^{d}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

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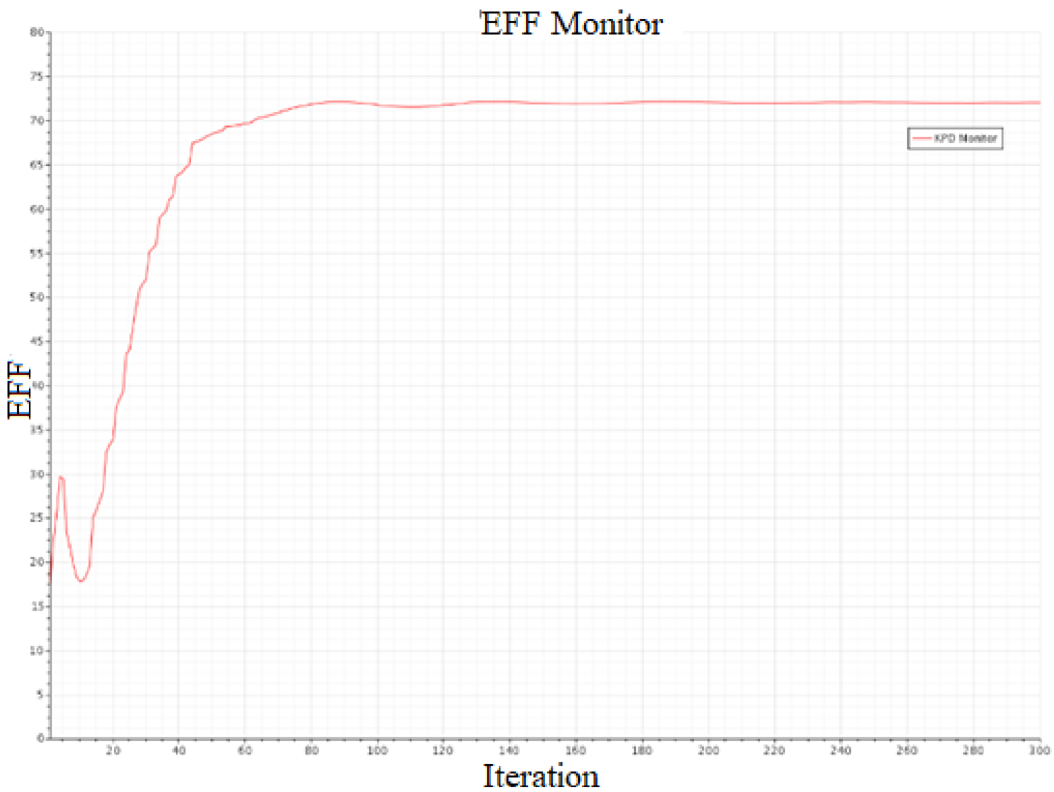

Fig.3. Efficiency for the best variant of the 3d set-up

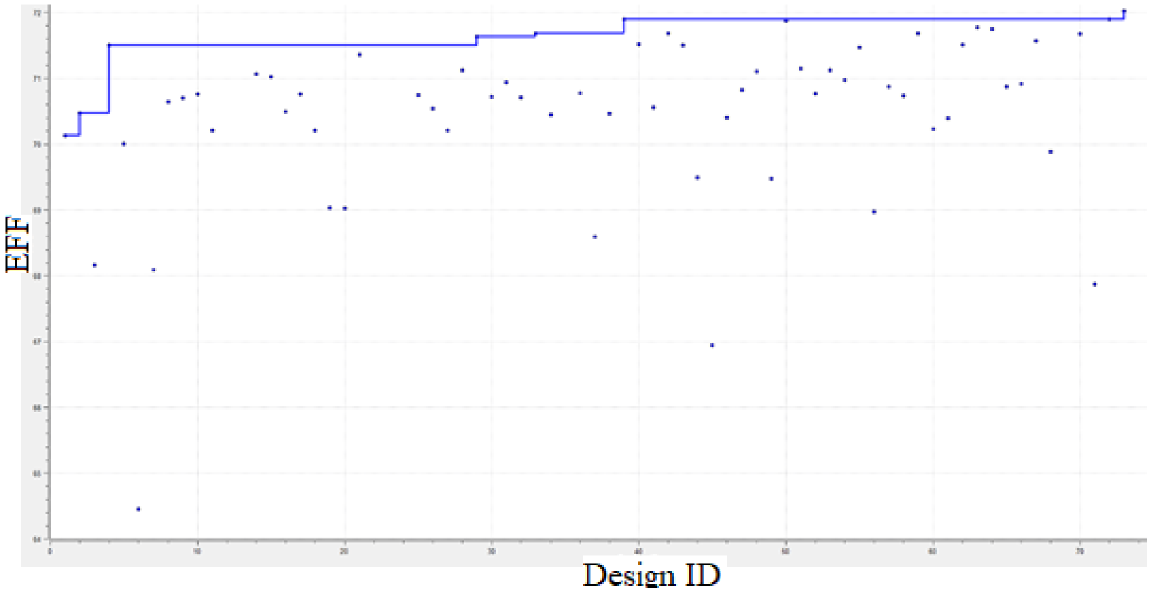

Fig.4. The change of efficiency for the 3dset-up during optimization

For the forth set-up 78 variant sare considered, the best of the mare showing Table4.

Table 4. Optimization results for the 4thset-up

Design variant	E	EFF	H	beta1	beta2	nBl	inlet	outlet

65	1.087	75.125	47.384	0.242	0.188	5	0.072	0.682
62	1.087	75.124	47.269	0.242	0.385	5	0.072	0.682
71	1.086	75.090	47.821	0.242	0.385	7	0.072	0.682
61	1.085	75.007	47.192	0.242	0.188	5	0.072	0.656
18	1.085	74.991	47.176	0.242	0.385	5	0.072	0.656

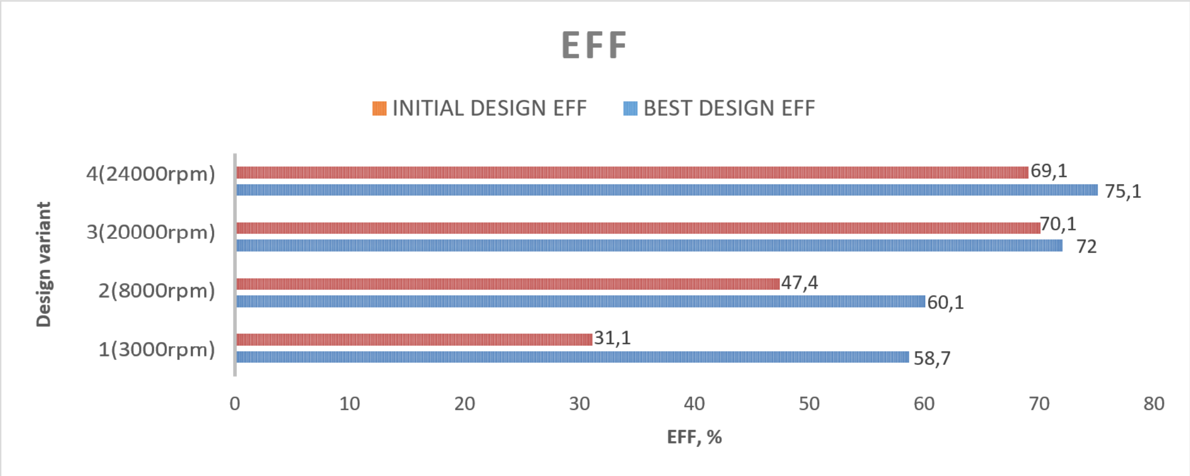

Fig.5. Comparis on of efficiency for the best configurations of different set-up swith initial values

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 hig her 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.

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