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Methodology for the comprehensive optimization of a centrifugal pump with a magnetic coupling

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Abstract: The article considers the design of a leakproof pump with a magnetic coupling operating on a chemical fluid. In this work the optimization of the pump assembly was carried out, including hydrostatic bearings and a magnetic coupling, using the STARCCM + simulation package. The throttle diameters optimal for this design are calculated. The author minimized the overflows in the auxiliary cooling paths of the coupling and the axial force on the pump rotor.

Introduction

Centrifugal pumps with magnetic coupling are designed for use in various industries for pumping clean, viscous and non-viscous fluids, as well as chemically aggressive and toxic fluids. Most often, they are used in pressure circulating systems designed for pumping hazardous substances.

The tightness of the pumping is due to the absence of mechanical seals in the pump. A magnetic drive installed in the pump allows zero leakage of the pumped product, which has a beneficial effect on the environmental safety of the device.

The main problem with centrifugal sealed pumps is the problem of heat balance in a cooled magnetic coupling. A limitation of the use of a pump with a magnetic coupling for pumping liquids having high temperatures is the limit temperature of the coupling magnets [1–7], which begin to lose magnetic properties when the temperature approaches the Curie point. The use of an integrated coupling cooling system is an effective way to solve this problem [8–20].

Another equally important problem is the calculation of the fluid flow in auxiliary paths [14]. The pressure distribution in the auxiliary paths determines the value of the axial forces on the rotor and the heat balance of the entire pump [13]. To solve this problem, it is necessary to have calculation methods that will take into account the hydrodynamic phenomena that occur in the tracts. Modern packages of hydrodynamic modeling are widely used for these calculations [13].

However, now there are no methods for calculating auxiliary paths that can evaluate the working fluid overheating at problem points, estimate the flow rate of the working fluid in the studied channels, evaluate the axial force and cavitation characteristics at the problem points, and optimize the auxiliary paths taking into account all of the above parameters.

This article describes the method for determining leaks in auxiliary cooling paths and the method for determining the axial force acting on the rotor from the magnetic coupling. A method for checking the achievement of the working fluid of the Curie temperature at a minimum flow rate in the auxiliary paths and a method for checking the bearing capacity of sliding bearings, which satisfies the maximum possible axial force, are also considered.

Below there is a diagram of the auxiliary cooling paths of the magnetic coupling.

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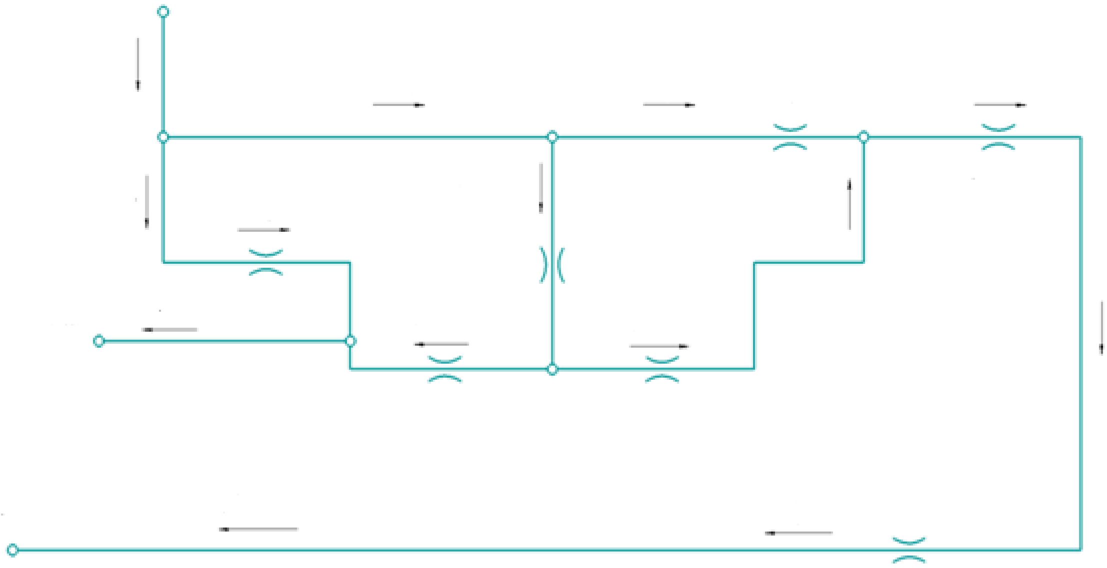

Fig.1. Scheme of auxiliary cooling paths of the magnetic coupling.

The optimization of auxiliary paths is carried out according to two criteria:
• Volumetric efficiency
• Rotor unloading from axial force

Usually, the temperature at the exit from the gap of the magnetic coupling or the maximum value of the axial force on the rotor is set as a limitation, then optimization will be carried out only by the volumetric efficiency.

Plan for calculating the task:

1) Calculate the plain bearings, gap seal and radial clearance of the coupling in the STARCCM + package to obtain the dependence of the flow rate on the differential pressure. As a result, obtain the flow rate characteristic of all small elements of the auxiliary path.
2) Build a picture of the flow of the working fluid in all channels of the auxiliary path, as well as a picture of the distribution of pressures and speeds.
3) Get a thermal picture in all the channels of the auxiliary path by the method of hydrodynamic modeling, introducing the temperature of the working fluid at the entrance to the path and the heat in the magnetic coupling.
4) Estimate the pressure margin to the Curie point.
5) Find the maximum temperature at the problem point and estimate the pressure margin to the boiling point at this temperature using the method of hydrodynamic modeling.
6) Find the axial force acting on the rotor.
7) Calculate the diameters of the two throttles by the Sobol method for the subsequent optimization of auxiliary paths.
8) Build a compromise curve volumetric flow - axial force.
9) Choose the optimal pair of throttles that create a support for the best cooling of the magnetic coupling and the minimum axial force on the rotor.

The object of study is a magnetic coupling with a sealing titanium screen, designed for power transmission of 195 kW at 2900 rpm. The fluid from the discharge pipe enters the cooling circuit, then is discharged to the inlet of the pump unit. When passing through the cooling circuit, the fluid heats up from the titanium screen. The dimensions of the magnetic coupling, the number of magnets, and their arrangement are selected according to the method of S P Subbotin [3]. The fixed titanium screen is in a rotating magnetic field. Losses on screen heating are determined by the formula

$$P_s = 31 \cdot \frac{B_a^2 \cdot D_s^3 \cdot n_{nom}^2 \cdot B \cdot N_{mag} \cdot \delta_s}{\rho},$$

where $B_a = \beta_a{*}K_i{*}B_r$ is the value of the current induction in the gap [3];

D_s is the diameter of the screen along the midline;

n_{nom} is the nominal frequency of rotation of the magnetic coupling;

B is the length of the magnet [4];

N_{mag}— the number of rows of magnets in the coupling halves;

δ_s— screen wall thickness;

ρ is the electrical resistivity of titanium [5];

Initial data for the design of the cooling circuit of the magnetic coupling:

V_{in} = 1.003 m/s — pressure at the inlet to the cooling circuit;

p_{out} = 0 Pa — pressure at the outlet of the pump shaft;

t_{out} = 60 °C— inlet temperature;

Methods

The method of numerical simulation is based on solving analogues of the basic equations of hydrodynamics and heat and mass transfer.

Mass conservation equation:

$$\frac{\partial \tilde{u}_j}{\partial x_j} = 0,$$

Where $\tilde{u}_j$ — the average value of the fluid velocity in the projection onto the j-th axis (j = 1, 2,3)

The equation of change in momentum in a stationary setting:

$$\rho\left[\tilde{u}_j \frac{\partial \tilde{u}_j}{\partial x_j}\right] = -\frac{\partial \tilde{p}}{\partial x_j} + \frac{\partial}{\partial x_j}\left[T_{ij}^{(v)} - \rho < u_i u_j >\right],$$

where $\tilde{p}$ — averaged pressure value;

$\tilde{T}_{ij}^{(v)} = 2\mu \tilde{s}_{ij}$—viscous stress tensor for incompressible fluid;

$\tilde{s}_{ij} = \frac{1}{2}\left[\frac{\partial \tilde{u}_i}{\partial x_j} + \frac{\partial \tilde{u}_j}{\partial x_j}\right]$ — strain rate tensor;

$\rho\langle u_i u_j \rangle$— Reynolds stresses, which are modeled on the basis of the k-ωSST – turbulence model;

Optimization of auxiliary pump paths.

The optimization problem is reduced to finding the extremum of the objective function with given constraints. It is necessary to determine the value of the vector of variables $x=(x_1,x_2,\ldots,x_m)$ that satisfy the constraints of the form $g_i(x_1,x_2,\ldots,x_m) \le b_i$ for all *i=1, ..., k*, and at which the minimum of the objective function is achieved $f(x_1,x_2,\ldots,x_m)$:

$$f(x_1,x_2,\ldots,x_m) \to (\max,\min)$$

The final solution to the problem is a pair *(x *, f * (x *)),* consisting of the optimal solution and the optimal value of the objective function.

- **LP τ search**

To search for the extremum of the function, the LPτ sequence method was chosen, since it has the ability to search for the global extremum of the function.

Throttle optimization was performed using the LPT-search program. With the number of design points of 32, the following results were obtained: d_{thr1} = 2...7 mm, d_{thr2} = 2...7 mm.

- **Calculation of auxiliary cooling paths**

The mathematical apparatus used for research is STARCMM +, Mathcad. The auxiliary path model contains clearances in the plain bearings and clearance in the coupling, which complicates the calculation. To simplify the calculation the model was divided into partsin the Solidworks program: a groove seal, two plain bearings and a radial clearance of the coupling.

Groove seal calculation:

It is necessary to obtain the dependence of the differential pressure on the velocity at the inlet to the groove seal. To do this, you need to configure the calculation model in the STARCCM + program, get a picture of the pressure and velocity distribution in the cross section. Next, use Mathcad to get the dependency function.

Calculation of plain bearings:

The calculation is made similarly to the calculation of the groove seal.

Determination of temperature in a dangerous section of a coupling:

The tuning in STARCCM + of the calculation model is similar to the tuning of the groove seal model. It is necessary to set the equation of fluid energy for temperature and heat gain from the coupling, calculated according to the Subbotin formula.

Results

In the process of modeling the auxiliary cooling paths of the magnetic coupling, a graph of the dependence of the flow velocity on the pressure drop in the gap seal was obtained.

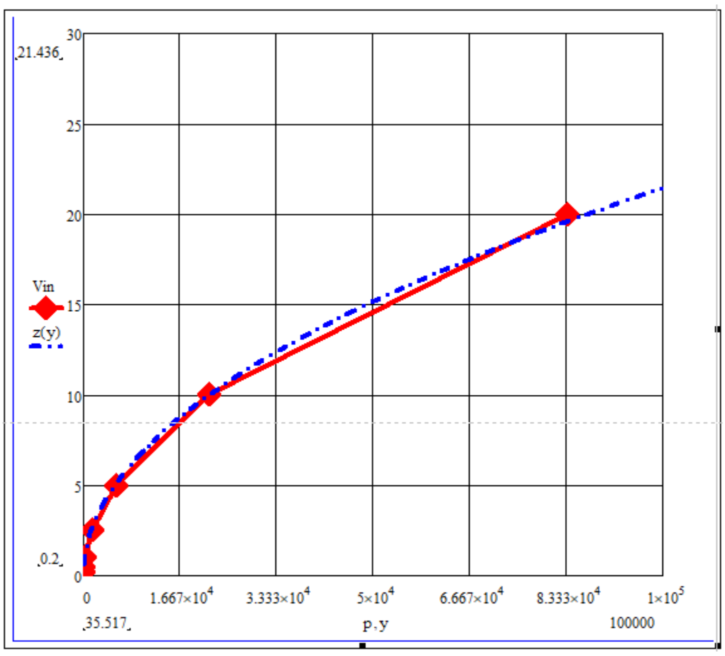

Fig.2. Graph of speed versus differential pressure in a groove seal.

Analyzing the obtained graph, we can conclude that the obtained function, taking into account the errors, fully corresponds to the radical dependence $Q = K * \sqrt{\Delta p}$. Therefore, the desired function is $Q = 0,06779 * \sqrt{\Delta p}$.

During the calculation of plain bearings, the dependence of speed on the differential pressure was obtained, according to which a graph was built in Mathcad. Analyzing the graph, we can conclude that the desired function is $Q = 0,0012 * \sqrt{\Delta p}$.

A set of methods has been obtained that makes it possible to evaluate the energy efficiency of sealed centrifugal pumps with a magnetic coupling. In particular, to evaluate leakage in hydrostatic bearings and fluid flow through auxiliary paths for cooling the magnetic coupling. An assembly optimization method has been developed, which includes hydrostatic bearings and a magnetic coupling.

The ideal sizes of throttles for this task are determined: d_{thr}= 2...7 mm.

Conclusions

1. The design of a leak free pump with a magnetic coupling that operates on a chemical fluid is developed.
2. A 3d-modeling of the design of the pump was carried out.
3. The optimization of the unit, which includes hydrostatic bearings and a magnetic coupling, was carried out using computer simulation methods STARCCM +.
4. The simulation of the magnetic coupling for guaranteed cooling was carried out.

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