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On the issue of starting-up overheating of electric motors of centrifugal pumps

A Protopopov[1,2] and D Bondareva[1]

[1]Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation

[2]E-mail: proforg6@yandex.ru

Abstract. In various industries where centrifugal pumps are used, a common problem is the starting overheating of electric motors. Such overheating can lead to motor failure, especially in the case of starting-up the centrifugal pump on the open valve. It happens due to the fact that the starting current is many times greater than the rated current, and the rated current with an open valve usually makes is much more. In this case, complex methods of centrifugal pumps analysis and manuals do not contain any methods of evaluation the magnitude of the starting overheating of centrifugal pump electric motors. In order to fill this lacuna, a mathematical model of the centrifugal pump start-up, which allows to estimate the starting overheating value, is developed in this article.

1. Introduction

Centrifugal pumps are widely used in various industries: mining, chemical industry, municipal engineering, aviation, astronautics, agriculture, robotics, etc. In this case, one of the actual problems is the problem of starting such pumps [1]-[8].

The fact is that at the starting time of the electric motor centrifugal pump its starting overheating occurs. This happens due to the fact that the starting current is many times greater than the rated current. The starting overheating is usually stronger in the case of the pump starting on the open valve, than the rated current. It is known from the characteristics of any centrifugal pump, that the rated current is greater when the valve is open [9]-[15].

Due to this, the specifications for any modern centrifugal pump have recommendations about the desirability of starting it up on a closed valve [16]-[20].

However, there is nothing more detailed except such discrete recommendations like "it is better to start up on a closed valve than to open one" in the literature. It is not known from the literature how much this or that pump overheats and why and to what extent the amount of starting overheating depends. To fill this lacuna, this article proposes a mathematical model of starting up, which is able to estimate the amount of starting overheating.

2. Mathematical model of centrifugal pump starting-up

Based on the theorem of the change in the amount of motion, the moment equation can be written:

$$J \cdot \frac{d\omega}{dt} = M_{dv}(t) - \alpha \cdot M_{rk}(t), \quad (1)$$

where

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J – the moment of inertia of the rotor relative to the axis;
ω – angular rotation speed of the pump shaft;
M_{dv} – motor moment without load;
M_{rk} – impeller moment at starting-up;
α – loss by coupling, bearings, pump seals when the moment is transmitted , $\alpha > 1$.

Let's consider the terms in equation (1).

The moment on the impeller is calculated:

$$M_{rk} = M_c + M_{dt}, \quad (2)$$

where
M_c – centrifugal moment;
M_{dt} – the moment of disk friction.

The impeller centrifugal moment:

$$M_c = \rho \cdot Q \cdot {R_2}^2 \cdot \omega(t), \quad (3)$$

where
ρ – working fluid density;
Q can be determined by the formula

$$Q = \mu_p \cdot \pi \cdot D_1 \cdot a\sqrt{2 \cdot g \cdot H(t)}, \quad (4)$$

where
D_1 – impeller diameter on groove seal;
μ_p– the discharge coefficient in the front axial clearance between the impeller and the pump casing.

The moment of disk friction is equal to:

$$\mathrm{M}_{dt}(t) = \frac{\omega(t) \cdot \pi \cdot \mu \cdot {R_2}^4}{a}, \quad (5)$$

where
a – axial clearance between impeller and pump casing.

Then it turns out that the impeller moment is equal to:

$$M_{rk}(t) = \frac{\omega(t) \cdot \pi \cdot \mu \cdot {R_2}^4}{a} + \rho \cdot Q \cdot {R_2}^2 \cdot \omega(t) \quad (6)$$

The engine moment can be presented in the form of a linear dependence on the angular velocity:

$$M_{dv}(t) = K - K_1 \cdot \omega(t), \quad (7)$$

where
K and K_1 – coefficients of the torque-mechanical characteristics of the electric motor.

Let's make the balance equation for the required pressure:

$$H_H = H_{st} + H_{tr} + h_{in}, \quad (8)$$

where
H_H – pump head required to overcome losses;
H_{st} – static head between the tanks of the feeder and receiver;
H_{tr} – pressure loss in the pipeline;
h_{in} – inertial head.

Let's consider the terms in equation (8).

From the similarity of centrifugal pumps:

$$H_H(Q; \omega) = H_o \cdot \left(\frac{\omega_o}{\omega}\right)^2, \quad (9)$$

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where

$H_н$, ω – head and angular velocity of the pump at starting-up;

H_o, ω_o – head and angular velocity of the pump at initial value.

The inertial head is determined by the acceleration or deceleration of the fluid flow, therefore, to find it, the second Newton law for the element of the stream of an ideal incompressible fluid, will be used and the desired equation will be obtained:

$$h_{in} = \frac{1}{g}\int_{l_1}^{l_2}\frac{\partial V}{\partial t}\cdot dl = \frac{j}{g}\cdot l, \quad (10)$$

where

j – the fluid flow acceleration;

l – the pipeline length.

The acceleration of the fluid flow will be obatined by differentiating the flow formula j in time t:

$$\frac{dV}{dt} = j = \frac{1}{F}\cdot Q' \quad (11)$$

The pressure losses in the pipeline are the sum of friction losses along the length and losses in local resistances, expressed in terms of flow. Based on these conditions, we'll write out the general loss formula:

$$H_{tr} = \left(\frac{\lambda\frac{l}{d}+\xi(t)}{2g\cdot F^2}\right)\cdot Q^2(t), \quad (12)$$

where

λ – friction resistance coefficient;

ζ(t) – the aggregated coefficient of local resistance;

l,d –the length and diameter of the pipeline;

$F = \frac{\pi d^2}{4}$ – the pipeline cross-sectional area;

$Q(t)$ – flow rate through the section;

$$\xi(t) = K_2 - K_3\cdot t, \quad (13)$$

where

K_2 and K_3–coefficientsdescribing the linear law of the resistance coefficient changing.

The initial conditions for the task are the following conditions:

$$\omega(0) = 0 \quad (14)$$

$$Q(0) = 0 \quad (15)$$

Thus, the mathematical model of the pump starting-up process is as follows:

$$\begin{cases} H_o\cdot\left(\frac{\omega_o}{\omega}\right)^2 = H_{st} + \left(\frac{\lambda\frac{l}{d}+\xi(t)}{2g\cdot F^2}\right)\cdot Q^2(t) + \frac{Q'(t)\cdot l}{F\cdot g} \\ J\cdot\frac{d\omega}{dt} = \left(K - K_1\cdot\omega(t)\right) - \alpha\cdot\left(\frac{\omega(t)\cdot\pi\cdot\mu\cdot {R_2}^4}{a} + \rho\cdot Q\cdot {R_2}^2\cdot\omega(t)\right) \\ \omega(0) = 0 \\ Q(0) = 0 \end{cases} \quad (16)$$

Let's solve the system of equations (16). To do this, we'll rewrite the system of equations (16) in the form:

$$\begin{cases} Q'(t) = \frac{F \cdot g}{l} \cdot \left(H_o \cdot \left(\frac{\omega_o}{\omega} \right)^2 - H_{st} - \left(\frac{\lambda \frac{l}{d} + \xi(t)}{2g \cdot F^2} \right) \cdot Q^2(t) \right) \\ \omega'(t) = \frac{(K - K_1 \cdot \omega(t)) - \alpha \cdot \left(\frac{\omega(t) \cdot \pi \cdot \mu \cdot R_2{}^4}{a} + \rho \cdot Q \cdot R_2{}^2 \cdot \omega(t) \right)}{J} \\ \omega(0) = 0 \\ Q(0) = 0 \end{cases} \quad (17)$$

3. The results of mathematical modeling

The system of equations (17) is solved in the Mathcad system using the 4th order Runge–Kutta method. The graph of the angular velocity versus time is obtained (figure 1).

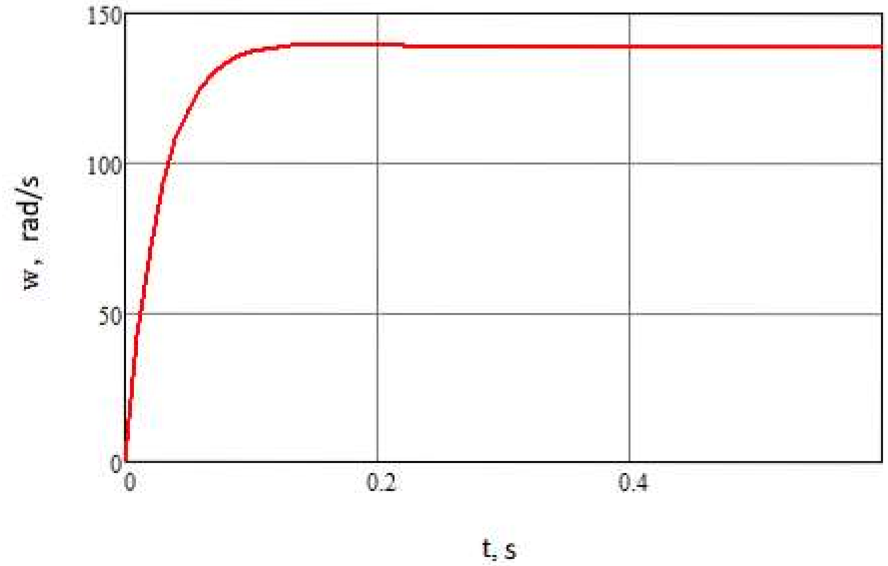

Figure 1. The graph of rotor angular velocity versus time.

The graph of flow versus time (figure 2):

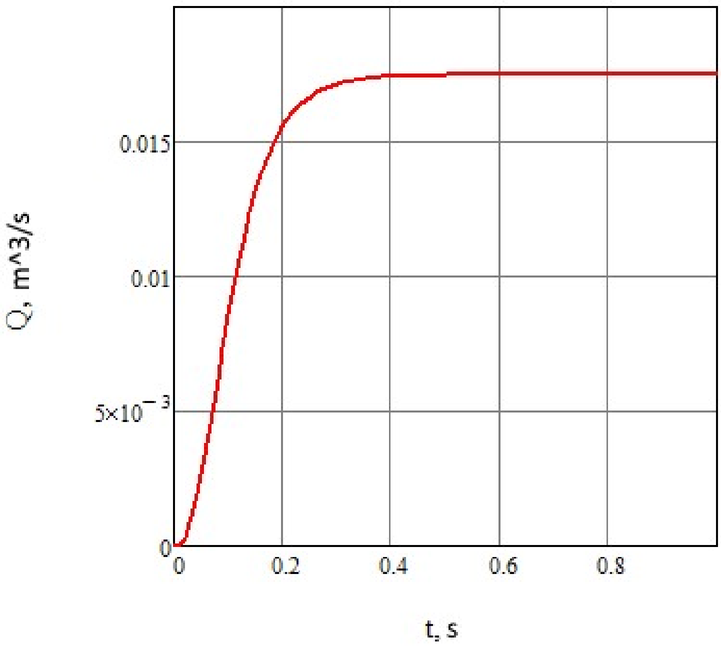

Figure 2. Graph of Q flow versus time.

We need both of these graphs to determine the time of the transition process, which we can substitute into the Joule-Lenz law.

Let's write the law of Joule–Lenz for the amount of heat:

$$\left(I_{nom} \cdot K_p\right) \cdot U \cdot t_{pp} = C_{medi} \cdot m_{prov} \cdot \Delta T, \quad (18)$$

where

I_{nom} – current intensity;
K_p – starting coefficient;
t_{pp} – transition time;
U – electric potential;
C_{medi} – copper specific thermal capacity;
m_{prov} – conductor mass;
ΔT – motor overheating temperature.

Electric current power in the motor winding:

$$P = I_{nom} \cdot U = \frac{\rho \cdot g \cdot Q \cdot H}{\eta}, \quad (19)$$

where

η – pump efficiency.

Let's express the value of the current I_{nom} from the equation (19)

$$I_{nom} = \frac{\rho \cdot g \cdot Q \cdot H}{\eta \cdot U}, \quad (20)$$

The general equations for temperature from equations (18) and (20)

$$\Delta T = \frac{\left(I_{nom} \cdot K_p\right) \cdot U \cdot t_{pp}}{C_{medi} \cdot m_{prov}} = \frac{\rho \cdot g \cdot Q \cdot H \cdot K_p \cdot t_{pp}}{C_{medi} \cdot m_{prov} \cdot \eta}, \quad (21)$$

Thus, we obtain the calculation of the temperature of the motor overheating during starting-up.

4. Conclusion

A mathematical model of starting-up of the centrifugal pump rotor with an asynchronous electric motor is obtained.

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The described method takes into account such factors as the pressure loss in the pipeline H_{tr}, the inertia pressure h_{in} that occurs during the pump operation, the moment of the impeller M_{rk} and electric motor moment M_d.

The obtained method of dynamic analysis allows to estimate the overheating of the electric motor of a centrifugal pump depending on various factors, and as a consequence, to predict its possible failure.

In the above model, a number of assumptions was made, in particular, it was assumed that during the transition process the starting current is constant and multiply more than the rated current with the same parameters.

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