

INFLUENCE OF OPERATIONAL PARAMETERS ON CENTRIFUGAL PUMP COUPLING LOADS: A CASE STUDY

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Abstract: An analysis of the operating dynamics of a centrifugal pump driven by an asynchronous electric motor was conducted using a two-mass torsional model incorporating a reduction element as the pump's flexible coupling. Three drive loading scenarios were examined: starting the pump with the discharge valve open (case 1), opening the valve on a running pump (case 2), and starting the pump with the valve closed (case 3). It was demonstrated that breaching centrifugal pump operation guidelines substantially impacts component loading, resulting in considerable overloads.

KEYWORDS: centrifugal pump, dynamic load, coupling, torque; capacity, driving

1 Introduction

Centrifugal pumps are among the most widespread equipment in transportation, including pipeline transport and ships, where they ensure the operation of ballast systems, cooling systems for power plants, cargo systems of tankers, etc. Ensuring pump reliability requires reducing dynamic loads in their drive, which can be achieved by maintaining regulated requirements for the operation and repair of these devices. Operators performing, and students studying, the operation of centrifugal pumps often do not understand how violations of the operational rules of these machines affect the operation and failure of their components. This creates conditions for conducting research on the main operational cases that affect the durability of centrifugal pump components and for developing didactic materials that can be used in training future operators.



Fig. 1 Photo of destroyed elastic spider coupling element [1]

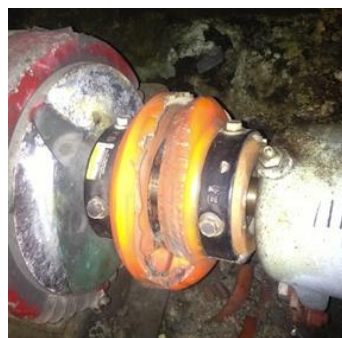


Fig. 2 Photo of destroyed toroidal shell coupling [2]

The weakest links in centrifugal pumps are the main shafts and flexible couplings (Fig. 1, Fig. 2); their replacement is quite resource-intensive, requiring almost complete pump

disassembly, removal of the impeller and bearings. Research [3] based on statistical data for the period 2015-2024, collected from pump failure cases in the oil and gas, chemical, and water supply industries of Ukraine, shows that 78% of pumps failures were related to mechanical damage (including couplings and shafts), 15% to hydraulic issues, 5% to electrical problems, and 2% to corrosion.

The work [4] presents a real case of pump complex failures, where the main failures identified included: misalignment, imbalance, bearing defects, cavitation, mechanical insufficient tightening/looseness of connections, as well as coupling failures.

Materials from work [5] documented several cases of pump coupling failures; Ishikawa analysis revealed the following failure sources:

- abrasive wear of elastomers by sand and scale (particle size 50...200 μm);
- thermal degradation of rubber at temperatures $>80^{\circ}\text{C}$ due to heating of the elastic element resulting from dynamic loads;
- bolt fracture due to overload during startup (torque $>150\%$ of nominal).

In [6], typical mistakes in the operation of flexible couplings were analyzed, with the main ones identified as: misalignment, improper coupling selection by stiffness/dimensions, lack of proper installation, presence of vibrational or torsional loads, and failure to follow maintenance schedules. Article [7] contains descriptions of failure cases in a multi-stage transmission system with coupling and demonstrates the correlation between coupling defects and nonlinear vibration of the system, requiring refinement of calculation models, particularly of the couplings themselves [8, 9]. Work [10] conducted analytical evaluation of natural vibrations in a vertical pump drive, in which one of the factors considered was the motor-pump coupling connection and its possible influence on the vibration regime. In article [11, 12] proposed a multi-field simulation model of the "motor-shaft-pump" system, analyzing torque transmission mechanics, shaft vibrations, pressure, pulsations, and interaction between motor, shaft, and pump. It was shown that coupling/shaft/pump dynamics affect pressure pulsations in the pump.

Violation of centrifugal pump operation rules leads to intense vibrations in the drive and coupling overloads. Among the most dangerous cases are three: starting the pump motor with an open discharge valve (case 1), rapid opening of this valve with the motor running (case 2), and starting the pump without load (case 3).

The influence of coupling stiffness on dynamic loads in the pump drive also has practical consequences, since during repair under practical conditions, couplings not specified by the manufacturer are sometimes installed.

Dynamic models developed by other researchers that would allow studying vibrations and overloads of flexible couplings operating with centrifugal pumps in the mentioned cases could not be found. Analysis of these cases requires developing calculation models and studying pump drive dynamics, which constitutes the goal of this work.

2 Methodology

The scheme of the model pump unit is shown in Fig. 3. When developing and analyzing the dynamic model, we use materials from works describing the basic principles of developing such models [14], mass and force reduction [15], work [16] contains examples of dynamic calculations and reference data. The characteristics of the pump's driving asynchronous motor are determined by quite complex electromagnetic phenomena in its windings [17], but in practice it can be described with sufficient accuracy by the Kloss formula [18]. To calculate losses in the pump, we will use dependencies given in [19],

examples and reference data from [20]. Pipeline losses will be calculated using general provisions of hydromechanics [21] and electromagnetic analogies [22].

We will represent the pump's dynamic scheme as a two-mass system (Fig. 4), where the driven mass J_2 is formed by the pump impeller and the driven half-coupling of the flexible coupling with torsional stiffness C_φ , and the driving mass J_1 consists of the electric motor rotor and the driving half-coupling of the same coupling. When developing the dynamic model, the following assumptions were made:

- pipelines are absolutely rigid;
- elastic elements of couplings have constant stiffness and zero damping;
- hydraulic resistance coefficients in the pipeline and inside the pump do not depend on fluid velocity.

In the general case, the system of motion equations for such a system will have the form [12-16]:

$$\begin{cases} J_1 \cdot \frac{d^2 \varphi_1}{dt^2} + C_\varphi \cdot (\varphi_1 - \varphi_2) = T_m; \\ J_2 \cdot \frac{d^2 \varphi_2}{dt^2} - C_\varphi \cdot (\varphi_1 - \varphi_2) = -T_r, \end{cases} \quad (1)$$

where φ_1 and φ_2 are the angular coordinates of the respective masses; T_m is the electric motor torque; T_r is the resistance torque.

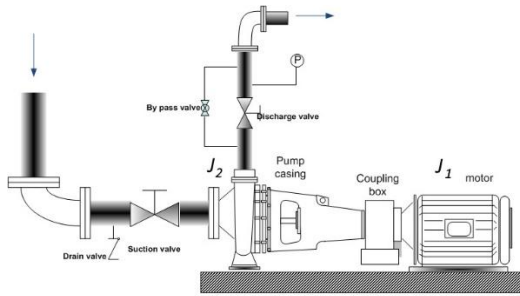


Fig. 3 Schematic of unit with centrifugal pump and asynchronous electric motor [12]

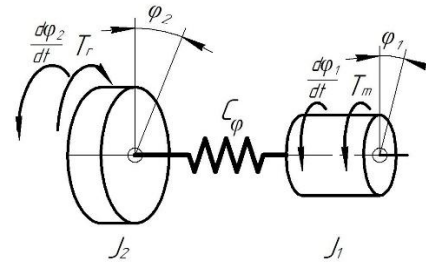


Fig. 4 Two-mass dynamic model of pump unit

The torque of the driving asynchronous motor according to the Kloss formula [15, 16]:

$$T_m = 2T_{cr} \frac{(\omega_s - \omega_{cr}) \cdot \left[\omega_s - \frac{d\varphi_1}{dt} \right]}{(\omega_s - \omega_{cr})^2 + \left[\omega_s - \frac{d\varphi_1}{dt} \right]^2} = \frac{A \cdot \left[\omega_s - \frac{d\varphi_1}{dt} \right]}{B + \left[\omega_s - \frac{d\varphi_1}{dt} \right]^2}, \quad (2)$$

where T_{cr} is the critical motor torque; ω_s is the synchronous rotor rotation frequency; ω_{cr} is the critical rotor rotation frequency.

The resistance torque T_r reduced to the pump shaft contains components from disk friction in the pump T_{df} and load from the pipeline side T_{tb} :

$$T_r = T_{df} + T_{tb}. \quad (3)$$

The disk friction torque of the impeller, taking into account seal losses, is determined by the expression [19, 20]:

$$T_{df} = (2,4 \dots 2,7) c_f \rho R^5 \omega^2 = \chi \omega^2 = \chi \left(\frac{d\varphi_2}{dt} \right)^2, \quad (4)$$

where $2,4...2,7$ is a coefficient that accounts for losses from both impeller ends, increased friction in a real pump compared to a model disk, and friction losses in the seal; $c_f = 0,0465Re^{-0,2}$ is the friction coefficient depending on the fluid flow regime inside the pump and the magnitude of the ratio of the gap between the impeller ends and the housing (Re is the Reynolds criterion); R is the impeller radius.

The load on the pump impeller from the pipeline side consists of the torque T_{st} from the static head of the pipeline, the torque from pipeline resistance T_{tr} and the torque T_{in} from fluid inertia in the filled pipe:

$$T_{tb} = T_{st} + T_{tr} + T_{in}. \quad (5)$$

The torque from static head H_{st} will be:

$$T_{st} = \rho g H_{st} \frac{Q}{\omega} = \alpha \cdot \frac{Q}{\frac{d\varphi_2}{dt}} \quad (6)$$

The torque from hydraulic resistance in the pipeline [19]:

$$T_{tr} = \rho g H_{fr} \frac{Q}{\omega} = \frac{8\rho}{\pi^2} \cdot \left\{ \frac{1}{d_{sc}^4} \left[\lambda \frac{l_{sc}}{d_{sc}} + \sum \xi_{sc} \right] + \frac{1}{d_{dc}^4} \left[\lambda \frac{l_{dc}}{d_{dc}} + \sum \xi_{dc} \right] \right\} \cdot \frac{Q^3}{\omega} = \beta \cdot \frac{Q^3}{\frac{d\varphi_2}{dt}}, \quad (7)$$

where d_{sc} is the suction pipeline diameter; d_{ds} is the discharge pipeline diameter; l_{sc} is the suction pipeline length; l_{ds} is the discharge pipeline length; ξ are local resistance coefficients (valves, filters); λ is the Darcy coefficient.

For torque from inertial head, we will have:

$$T_{in} = \rho g H_{in} \frac{Q}{\omega} = \frac{4\rho}{\pi} \cdot \left(\frac{l_{sc}}{d_{sc}^2} + \frac{l_{dc}}{d_{dc}^2} \right) \cdot \frac{Q}{\omega} \cdot Q' = \gamma \cdot \frac{Q}{\frac{d\varphi_2}{dt}} \cdot \frac{dQ}{dt}. \quad (8)$$

Or, finally, taking into account that the pump efficiency η is approximately 80%, we will have an expression for the resistance torque reduced to the driven mass J_2 :

$$\begin{aligned} T_r &= \chi \omega^2 + \frac{1}{\eta} \cdot \left(\alpha \cdot \frac{Q}{\omega} + \beta \cdot \frac{Q^3}{\omega} + \gamma \frac{Q}{\omega} Q' \right) = \\ &= \chi \left(\frac{d\varphi_2}{dt} \right)^2 + \frac{1,25Q}{\frac{d\varphi_2}{dt}} \cdot \left(\alpha + \beta \cdot Q^2 + \gamma \cdot \frac{dQ}{dt} \right). \end{aligned} \quad (9)$$

Taking into account the above, we will have for the developed dynamic scheme the system of equations (10).

$$\begin{cases} J_1 \cdot \frac{d^2\varphi_1}{dt^2} + C_\varphi \cdot (\varphi_1 - \varphi_2) = \frac{A \cdot \left[\omega_s - \frac{d\varphi_1}{dt} \right]}{B + \left[\omega_s - \frac{d\varphi_1}{dt} \right]^2}; \\ J_2 \cdot \frac{d^2\varphi_2}{dt^2} + \chi \cdot \left(\frac{d\varphi_2}{dt} \right)^2 + \frac{1,25Q}{\frac{d\varphi_2}{dt}} \cdot \left(\alpha + \beta \cdot Q^2 + \gamma \cdot \frac{dQ}{dt} \right) = C_\varphi \cdot (\varphi_1 - \varphi_2). \end{cases} \quad (10)$$

Initial conditions:

- for case 1: at $t = 0$; $\varphi_1 = \varphi_2 = 0$; $\frac{d\varphi_1}{dt} = \frac{d\varphi_2}{dt} = 0$;
- for case 2: at $t = 0$; $\varphi_1 = 0$; $\varphi_2 = -\frac{T_{df}}{C_\varphi} = -\frac{\chi \omega_0^2}{C_\varphi}$; $\frac{d\varphi_1}{dt} = \frac{d\varphi_2}{dt} = \omega_0$.
- for case 3: at $t = 0$; $\varphi_1 = \varphi_2 = 0$; $\frac{d\varphi_1}{dt} = \frac{d\varphi_2}{dt} = 0$; $Q = 0$.

Angular velocity $\omega_0 = 156,6 \text{ rad/s}$, which will be established in the system before opening the discharge valve, was found by solving system (10) for steady motion and in the absence of flow – $Q = 0$.

The change in fluid flow rate through the pipeline over time when starting the pump with an open discharge valve (case 1) will be calculated according to expression (11) [21], and when opening the discharge valve on a running pump (case 2), using the analogy [22] with the change in current in an electrical RC circuit - according to the known expression (12).

$$Q(t) = \frac{Q_n}{\omega_n} \cdot \frac{d\varphi_2}{dt} = \psi \cdot \frac{d\varphi_2}{dt}; \quad (11)$$

$$Q(t) = Q_n \cdot (1 - e^{-\frac{t}{\tau}}), \quad (12)$$

where ω_n is the nominal angular velocity of the motor; τ is the relaxation time of dynamic processes in the pipeline.

Modeling will be performed using the example of a pump unit that includes a centrifugal pump with flow rate $Q_n = 400 \text{ m}^3/\text{h} = 0.1111 \text{ m}^3/\text{s}$, driven by an asynchronous motor with power $P = 90 \text{ kW}$ at nominal rotation frequency $n_n = 1440 \text{ rpm}$ through a flexible coupling. The pump operates on a pipeline with diameters $d_{sc} = 0.20 \text{ m}$ and $d_{dc} = 0.15 \text{ m}$, length $l_{sc} = 20 \text{ m}$, $l_{dc} = 45 \text{ m}$, supplying fresh water ($\rho = 1000 \text{ kg/m}^3$) through a filter to a tank at height $H_{st} = 20 \text{ m}$ from the water free surface. The pump impeller has mass $m = 42 \text{ kg}$ and radius $R = 0.203 \text{ m}$.

For the described pump unit, the constants used will have next values: $T_{cr} = 1373,00 \text{ N} \cdot \text{m}$, $\omega_n = 150,80 \text{ rad/s}$, $\omega_{cr} = 129,60 \text{ rad/s}$, $\omega_s = 157,10 \text{ rad/s}$, $A = 75410,0 \text{ W}$, $B = 754,3 \text{ rad}^2/\text{s}^2$, $\alpha = 196200 \text{ Pa}$, $\beta = 30978014 \text{ Pa} \cdot \text{s}^2/\text{m}^2$, $\gamma = 3183099 \text{ kg/m}^4$, $Re = 7 \times 10^6$, $c_f = 0,002$, $\chi = 0,0019 \text{ N} \cdot \text{m s}^2/\text{rad}^2$, $\psi = 0,00072 \text{ m}^3/\text{rad}$, $\tau = 0,05 \text{ s}$. The moments of inertia of masses and torsional stiffnesses will be: $J_1 = 0,55 \text{ kg} \cdot \text{m}^2$, $J_2 = 1,02 \text{ kg} \cdot \text{m}^2$, $C_\varphi = 80000 \text{ N} \cdot \text{m}/\text{rad}$ - for pin-bush coupling, $J_1 = 1,00 \text{ kg} \cdot \text{m}^2$, $J_2 = 1,47 \text{ kg} \cdot \text{m}^2$, $C_\varphi = 8000 \text{ N} \cdot \text{m}/\text{rad}$ - for toroidal shell coupling.

The results of solving the systems of equations for both cases are presented in the form of graphs showing changes in flow rate $Q(t)$, angular velocities $\frac{d\varphi_1}{dt} = f(t)$; $\frac{d\varphi_2}{dt} = f(t)$ of motor rotor J_1 and impeller J_2 , as well as coupling torque $T_{cpl} = C_\varphi \cdot (\varphi_1 - \varphi_2) = f(t)$ and motor torque $T_m = f(t)$.

3 Results and discussion.

Fig. 5-8 present the corresponding graphs for case 1 when using the basic pin-bush coupling and the flexible toroidal shell coupling. In the second case, the larger moment of inertia of masses in the system with toroidal shell coupling increases the pump acceleration time from 1.3 to 1.5 s. The lower (by a factor of 10) stiffness of the toroidal shell coupling allows reducing the oscillation frequency and maximum torque values on the coupling - from 13 kN·m to 4.8 kN·m, which corresponds to reducing the dynamic coefficient (ratio of peak torque to nominal $K_d = T_{max}/T_n$) by almost 3 times - from 22:00 to 8:00. The presented values of dynamic coefficients for both variants are threatening from the standpoint of strength of all drive elements, which emphasizes the danger of the first considered case in practice and is important for training relevant specialists.

In Fig. 9-12 present graphs for case 2, which is more common in practice. Analysis of the graphs demonstrates that when opening the discharge valve, installing a more compliant toroidal shell coupling gives the opposite effect - the maximum load on the coupling increases from 1.6 kN·m for the basic coupling to 1.95 kN·m for the toroidal shell coupling (dynamic coefficient K_d increases from 2.67 to 3.25) with decreased oscillation frequency, and the oscillation damping time in the drive increases - from 0.3 to 0.4 s. Further increase in coupling torsional stiffness contributes to reducing the dynamic coefficient K_d . Thus, installing a more compliant coupling in the centrifugal pump drive is not advisable, which emphasizes trends in modern pump manufacturing toward installing couplings with high torsional stiffness.

For case 3, the graphs shown in Fig. 13 and Fig. 14 additionally illustrate the advisability of starting the pump with a closed discharge valve, since for the coupling in this case, achievable

dynamic coefficients are $K_d = 2.83$ when using a pin-bush coupling and $K_d = 2.25$ for a toroidal shell coupling.

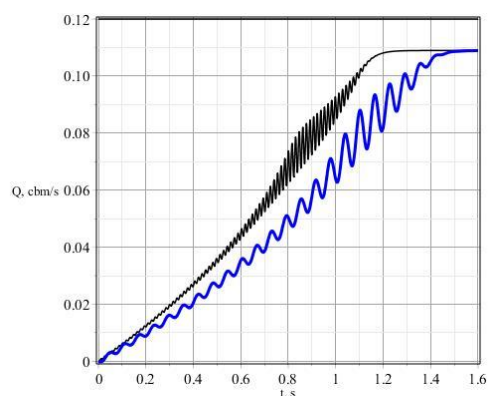


Fig. 5. Graph of flow rate change through pipeline (case 1): for pin-bush coupling (black) and toroidal shell coupling (blue)

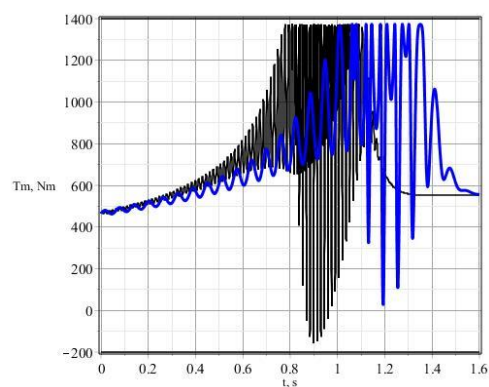
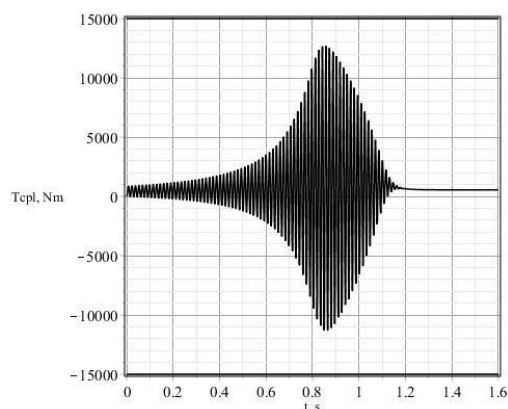
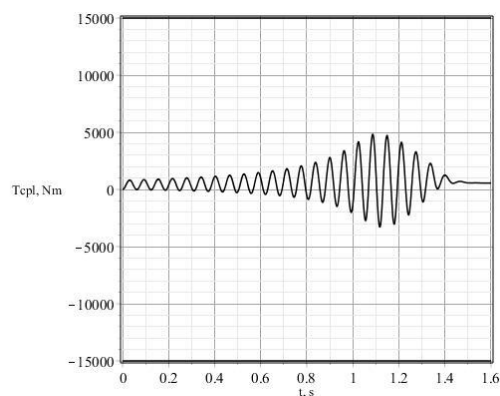


Fig. 6. Graphs of motor torque change (case 1): for pin-bush coupling (black) and toroidal shell coupling (blue)

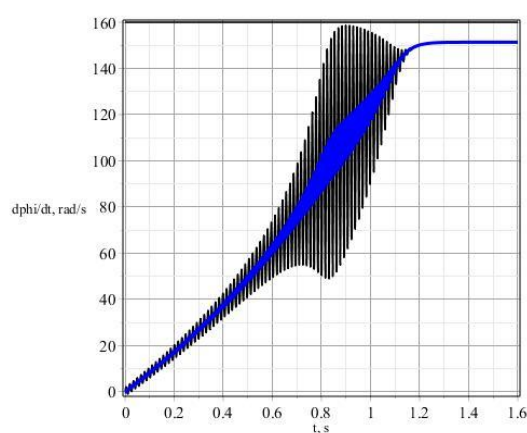


a)

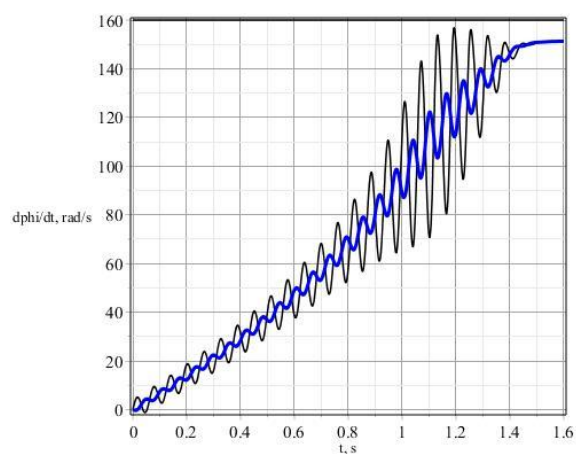


b)

Fig. 7. Graphs of torque change on coupling (case 1): for pin-bush coupling (a) and toroidal shell coupling (b)



a)



b)

Fig. 8. Graphs of angular velocity changes of rotor (black) and pump impeller (blue) (case 1): for pin-bush coupling (a) and toroidal shell coupling (b)

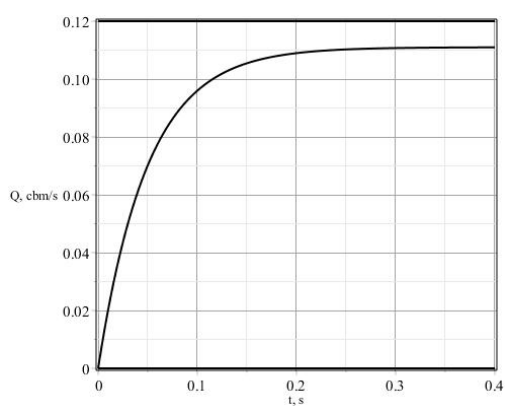


Fig. 9. Graph of flow rate change through pipeline (case 2) for pin-bush coupling

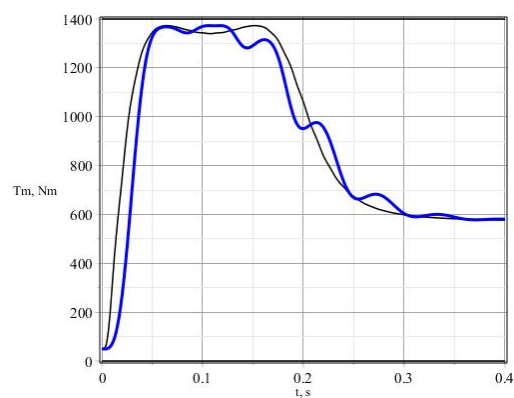
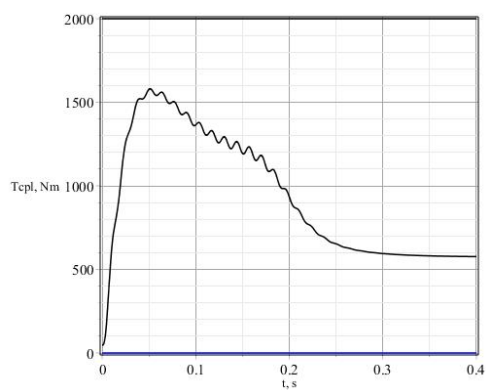
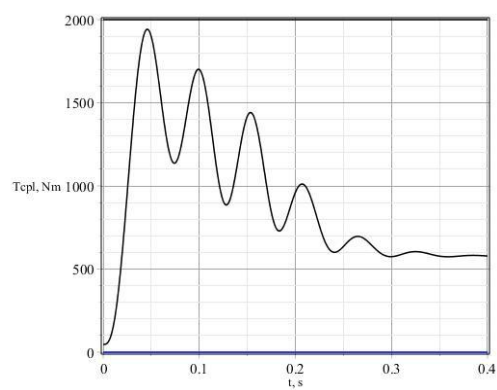


Fig. 10. Graphs of motor torque change (case 2): for pin-bush coupling (black) and toroidal shell coupling (blue)

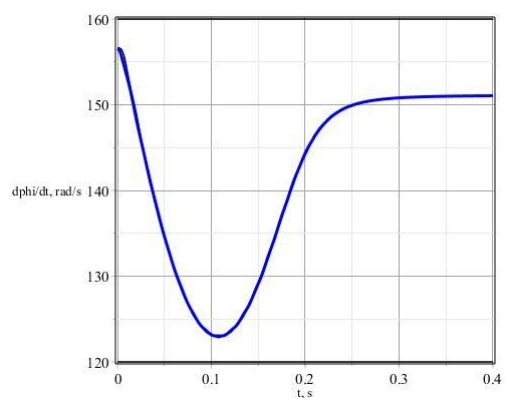


a)

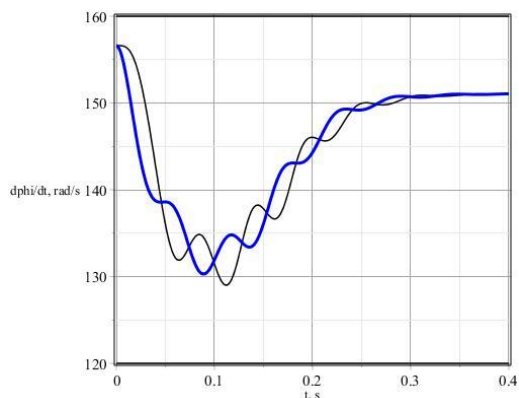


b)

Fig. 11. Graphs of torque change on coupling (case 2): for pin-bush coupling (a) and toroidal shell coupling (b)

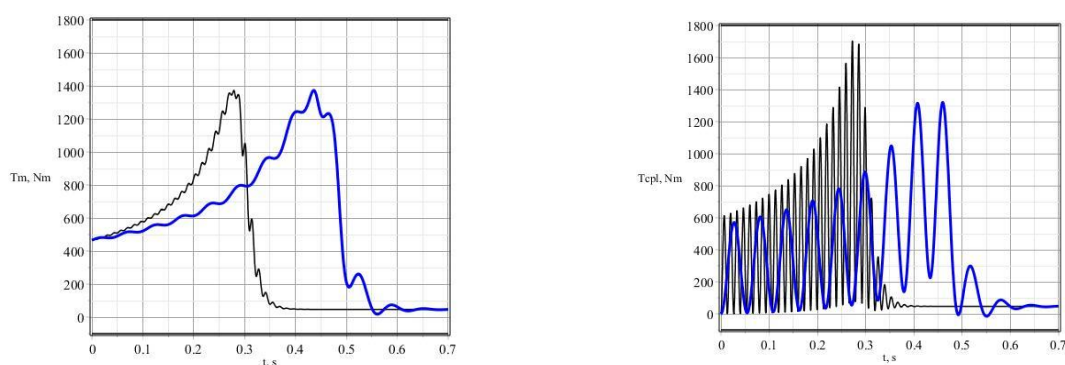


a)



b)

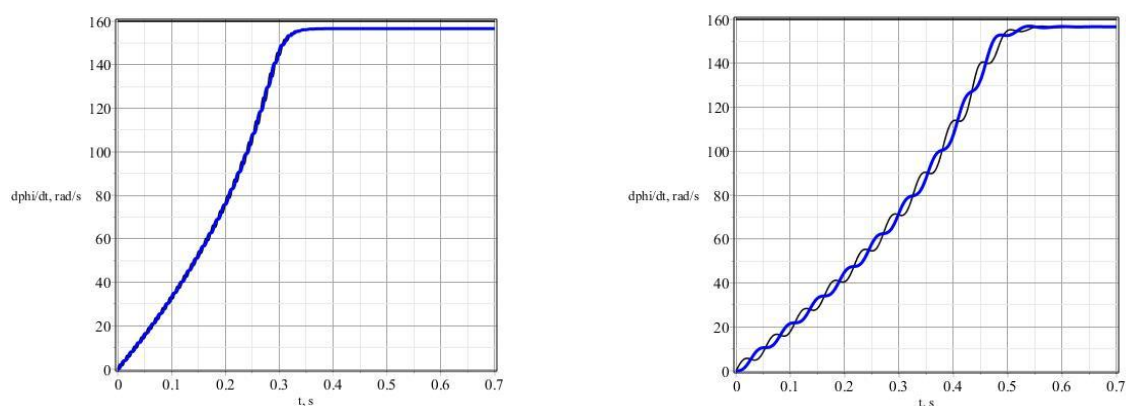
Fig. 12. Graphs of angular velocity changes of rotor (black) and pump impeller (blue) (case 2): for pin-bush coupling (a) and toroidal shell coupling (b)



a)

b)

Fig. 13. Graphs of motor torque change (a) and torque on coupling (b): for pin-bush coupling (black) and toroidal shell coupling (blue) when starting pump with closed discharge valve



a)

b)

Fig. 14 Graphs of angular velocity changes of rotor (black) and pump impeller (blue) (case 3): for pin-bush coupling (a) and toroidal shell coupling (b)

CONCLUSION

1. An analysis of centrifugal pump operation dynamics with an asynchronous drive electric motor was performed based on a two-mass torsional model with a reduction link in the form of a pump flexible coupling. Three cases of drive loading were considered: starting the pump with an open discharge valve (case 1), opening this valve on a running pump (case 2), and starting the pump with a closed valve (case 3). It was shown that violation of centrifugal pump operation rules significantly affects the loading of their components, causing significant overloads.

2. It was shown that starting the pump with an open discharge valve is most dangerous. Overloads of pump drive elements in this case can reach more than $K_d = 20$ times compared to nominal load, which can cause failure of drive components. Startup is accompanied by the occurrence of high-frequency torsional vibrations, which reduce the durability of connecting couplings.

3. Starting the pump, accelerating it to operating frequency followed by opening the discharge valve is safer. Thus, when starting the pump motor at no-load, the coupling dynamic coefficient can reach $K_d = 2.83$, and when opening the valve on a running pump –

$K_d = 2.67$. The obtained results are practically significant for designers when selecting a coupling for a pump unit with a centrifugal pump.

4. Replacing the flexible coupling with a more compliant one (for example, the basic pin-bush coupling with stiffness $C_\varphi = 80000 \text{ N}\cdot\text{m/rad}$ with a toroidal shell coupling with stiffness $C_\varphi = 8000 \text{ N}\cdot\text{m/rad}$) will have a positive effect in case 1 - the overload can be reduced from $K_d = 22.00$ to $K_d = 8.00$. However, for case 2, the overload will conversely increase from $K_d = 2.67$ to $K_d = 3.25$, and for case 3 - will decrease from $K_d = 2.83$ to $K_d = 2.25$. The presented results illustrate the inadvisability of changing the coupling specified by the manufacturer in the pump design to a more compliant one.

5. The considered cases are illustrative didactic materials that demonstrate current pump operation rules and are successfully used by the authors in training operating engineers, particularly when studying the structure and operation of centrifugal pumps to illustrate the impact of violating these rules on their operation.

6. The obtained results create conditions for more detailed analysis of dynamic loads on the pump shaft, which requires studying its drive as a three-mass system, as well as reducing the number of neglects and assumptions when developing calculation models.

7. Practically significant is the study of means for reducing dynamic loads in centrifugal pump drives, particularly through the use of couplings with variable, specifically quasi-zero stiffness.

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