Conditions for the Occurrence of Resonant Oscillations of Vertical Booster Pumping Units

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Abstract. Features of vibration diagnostics and causes of vibration of vertical booster pump units are considered in the work. On the basis of experimental data and mathematical modeling, the excitation and natural frequencies of vibration characteristic of the pump unit NPV 3600-90-M are determined. It was established that the reverse frequency of the upgraded pump unit in the present case is resonant with its own frequency of the engine-supporting structure system, which is an above-ground part of the equipment under consideration. Taking into account unacceptable limits of ratio of natural and exciting frequencies, optimal range of natural frequency of above-ground part of pump unit is determined.

1. Introduction

Vibration analysis plays an important role in monitoring the state of centrifugal pumps and is widely used to evaluate the operability of the unit. Among the main sources of vibration of pumping equipment, the causes of hydrodynamic, mechanical or electromagnetic origin can be distinguished. It is the evaluation of the vibration value that indicates whether it will actually lead to a failure.

The reduction of vibration to acceptable values can be achieved by various means and methods, but each case of increased vibration requires individual analysis, and a universal quenching method has not yet been detected. This issue is important for vertical booster pump units of main oil pipelines. The design features of this equipment and the observation in some cases of increased vibration caused by the proximity of the return frequency with the natural frequencies of the pump unit [1, 2] significantly complicate the solution of this problem. The determination of the conditions of occurrence, the possibility of forecasting, as well as the development of methods for reducing the vibration of vertical booster pump units is an urgent task of pipeline transport of oil and oil products.

2. Design and technical characteristics of vertical booster upgraded pump units

Transportation of oil is one of the most important stages on its path from production to the consumer. In this case, the booster pumps provide the necessary support for the cavitation-free operation of the main centrifugal pumps.

Pumps of NPV-M type - centrifugal vertical two-hull section type with pre-connected wheel, impellers of one-way inlet and end seal of cartridge type [3]. Vertical pump unit NPV 3600-90-M (Table 1) is selected for investigation at oil pumping station N, at which vibration is recorded, which exceeds shutdown value or limit values set by contract during start-up and testing both in idling mode and in main modes in operating feed interval.
Table 1. Technical parameters of the pump NPV 3600-90-M [4].

| Parameters of the pump NPV 3600-90-M in the nominal mode |  |
|----------------------------------------------------------|---|
| **Synchronous speed, \( s^-1 \) (rpm)** | **Supply, \( m^3/s \) (m\(^3\)/h)** | **Head, m** |
| 16,7 (1000) | 1,0 (3600) | 90 |

**Technical and energy efficiency indicators**

| Permissible cavitation margin, m, maximum | Efficiency, %, minimum | Power, kW, maximum |
|-------------------------------------------|-------------------------|--------------------|
| 3,1 |

**Main engine characteristics**

| Power, kW | Voltage, V | Weight, kg, maximum |
|-----------|------------|---------------------|
| 1250 |

Vibration readings are one of the pump failure criteria (Figure 1).

![Pump failure criteria]

**Figure 1.** Failure criteria of NPV 3600-90-M pump [4].

For the analysis of the vibration state, the natural frequencies of the engine-support system were measured (Table 2). Of particular importance is the natural frequency of the first order mode. To obtain the results of vibration analysis at characteristic natural frequencies, the engine was subjected to a shock test. Also, vibration measurements were carried out at the operating electric motor, since in this state there is not only vibration resulting from impact loads, but also vibration created by the electric motor itself.

Table 2. Natural frequencies on the reference structure.

| Mode | Frequency [Hz] | Vibration speed [rpm] | Vibration speed [mm/s] |
|------|----------------|-----------------------|------------------------|
| 1    | 16,18          | 970,8                 | 8,1                    |
| 2    | 40,72          | 2443,2                | 3,8                    |

Vibration level on the first mode of natural frequency \( f_01 = 16.18 \) Hz exceeds the permissible value of 4.5 mm/s (Figure 1). When analyzing the vibration state of the booster pump stations, it is necessary to take into account the fact that their rotation speed is significantly lower than that of the main MPA. So, for booster pump unit NPV 3600-90-M with rotor speed of 1000 rpm in idling mode and 991 rpm in nominal mode, the main frequencies are given in Table 3.

The engine speed is \( f_{rev} \), very close to the natural frequency of oscillation, therefore, the engine-support system operates in resonance mode, which is the reason why the permissible level of oscillation of the motor is exceeded. The frequency of the first mode or the reverse frequency (16.5 Hz) and the frequency of the second mode (33 Hz) are the main indicators of the vibration of the system and lead to resonance.
caused by the coincidence of the natural frequencies of the design with the frequencies of disturbing forces.

Table 3. Main frequencies of vibration of the NPV 3600-90-M booster pump unit.

| Frequency, Hz | Name                          |
|--------------|-------------------------------|
| 16.5         | Main rotor frequency          |
| 33           | Second harmonic of rotor frequency |
| 115.5        | Impeller blade frequency      |
| 231          | Second harmonic of blade frequency |

3. Features of vibration diagnostics of vertical pumps
Vertical pumps differ from similar horizontal ones by more flexible motor housings and pump pressure part, as well as by more flexible attachment of these housings to the foundation (Figure 2).

Figure 2. Vertical booster pump.

For such a design, it is particularly important to provide allowable oscillations of the vertical shaft of the pump-electric drive system in frequency and amplitude. The main sources of vibration in such a
system are unbalanced rotating masses, hydraulic forces in the flow part, alignment of the pump and electric motor shafts, resonant oscillations [1, 2].

Vertical booster pumps, similar to those described herein, are subject to high axial stresses occurring on the impellers of the one-way inlet and increasing due to the weight of the rotor. In practice, this gives some advantage: a strong restoring moment is provided that can suppress the lateral vibrations of the shaft, improve alignment and increase critical speed.

On the other hand, the booster pumps create less pressure than the main pumping units, which reduces the axial force and the resulting restoring moments. This makes such pumps more prone to increased vibration. This problem becomes especially urgent if any of the characteristic vibration frequencies of this unit is close to the intrinsic frequencies of the design.

3.1. Above-ground design of vertical pump
The typical overhead design of a vertical pump with a heavy electric motor at the top and a flow part is quite flexible. The structure can be further weakened by a lightweight support structure with a mass less than the supported equipment. The support part and piping increase the probability that the pump will operate under high vibration conditions.

In practice, there may be cases of coincidence or proximity of the frequencies of the force forces and the natural frequencies of oscillations of the motor with the supporting structure. Prolonged operation of such a system in resonance conditions can lead to failures of the pump unit. However, to date, optimal methods for reducing vibration on the above-ground structure of the vertical pump have not been identified. The difficulty of solving this problem lies in the low knowledge of the possibilities of using vibration isolation to reduce vibration on vertical pumping units.

In addition to oscillations caused by mechanical and electromagnetic causes, hydraulic disturbances can also enhance vibration. Although cavitation is not such a big problem for vertical pumps (since the lower part of the impeller is immersed in the pumped liquid and there is less chance of having high suction energy), it can still increase the vibration of the pump. Pulsations of pressure at suction can also increase vibration in pump [5]. In some cases, increased oscillations of the transfer units may be associated with vibration parameters of the piping. Moreover, this problem is observed both at pump stations [2, 6-8] and gas pumping units [9].

4. Estimation of natural frequencies and results of modal analysis of vertical pump unit
The vibration activity of the engine-support structure system should be considered as a set of interrelating elements, therefore, as a complex polyharmonic process. Reduction of motor vibration reduces vibration of support structure. This dependence is significantly manifested in resonance, when the natural frequencies of the system oscillations coincide with the driving frequencies of the motor.

In this case, the force is an imbalance of the rotating mass of the rotor.

Amplitude of forcing force $F_0$ caused by imbalance of rotating mass of rotor in steady-state mode of pump unit operation is determined by formula:

$$F_0 = m_p \cdot e_p \cdot \omega_p^2 = 161.5 \, N,$$

where $m_p$ – rotor weight;
$e_p$ – rotor eccentricity;
$\omega_p$ – circular rotor speed,
$\omega_p = 2\pi f_p$,
$f_p$ – rotation frequency.

In the absence of a vibration quenching system and under conditions of force force, the vibration of the system in question is described by the following differential equation:

$$m \cdot x'' + a \cdot x' + c \cdot x = F_0 \cdot \cos(f_p \cdot t),$$

where $x''$ – acceleration of moving body, $x'$ – velocity, $x$ – displacement.
where $m$ – weight of the object protected from vibration, kg;
$x$ – coordinate of the protected object, m;
$\alpha$ – friction coefficient, kg/s;
$F_0$ – force amplitude, N;
$t$ – time, s;
$c$ – stiffness factor, N/m.

The calculation was carried out in the Wolfram Mathematica 11.1 software complex. The electric motor is simulated by a body with mass $m$ located on a spring with stiffness factor $c$ (Figure 3).

![Figure 3](image1)

**Figure 3.** Scheme of simulated electric motor without vibration isolators.

The engine-support structure system is in resonance mode with constantly increasing vibration amplitude (Figure 4).

![Figure 4](image2)

**Figure 4.** Vibration amplitude of "engine - support structure" system at absence of vibration damping system.

To obtain a mathematical model of a pump with a real physical ratio of sizes, an automated KOMPAS-3D design program was used, and for modal analysis, the ANSYS software complex. The simulation data coincide with the actual frequencies (Table 2).

Adjustment of natural frequencies $f_0$ from frequencies of excitation loads $f_p$ is the main means of ensuring vibration resistance of the structure. Condition [10] must be met:

$$f_0 / f_p \leq 0.75 \text{ and } f_0 / f_p \geq 1.3.$$

(3)
The first mode of natural frequencies of the support structure practically coincides with the reverse frequency of the pump unit (Table 4), causing an increased level of oscillations of the entire structure.

Table 4. Ratio of excitation frequencies to natural frequencies of electric motor oscillations.

| Natural frequencies, Hz | Excitation frequencies, Hz | Ratio | Vibration speed, mm/s |
|-------------------------|----------------------------|-------|----------------------|
| 16.18                   | 16.5                       | 0.98  | 8.1                  |
| 40.72                   | 33                         | 1.23  | 3.8                  |

Since a change in rotor speed is not foreseen for these pumps, the best option to disengage from resonance may be to change the natural frequency of the reference structure to optimize system operation. The desired frequency ratio is shown in Table 5.

Table 5. Ratio of natural and excitatory frequencies.

| Excitation frequencies, Hz | Ratio | Natural frequencies, Hz |
|----------------------------|-------|-------------------------|
| 16.5                       | 0.75  | 22                      |
| 16.5                       | 1.3   | 12.7                    |
| 33                         | 0.75  | 44                      |
| 33                         | 1.3   | 25.38                   |

Taking into account unacceptable limits according to condition (3), the range of natural frequencies relative to the reverse frequency of the electric motor \( f_0 \geq 22 \) Hz and \( f_0 \leq 12.7 \) Hz, relative to the double reverse frequency - \( f_0 \geq 44 \) Hz and \( f_0 \leq 25.38 \) Hz. The optimum natural frequency range of the engine-support system is thus in the range of 22 to 25.38 Hz.

5. Conclusions

Based on the analysis of the causes of vibrations of vertical pumping units, it was found that the above-ground structure of the vertical pump, with a heavy electric motor at the top and a flow part, tends to be quite flexible. The support structure with the mass is much smaller than the supported equipment. The reasons for the high vibration readings in some cases are the resonance between the forcing forces and the intrinsic oscillation frequencies of the motor with the supporting structure. The difficulty of solving this problem lies in the low knowledge of the use of vibration isolation to eliminate the source of vibration on vertical pumping units. So, on the vertical pump unit NPV 3600-90-M operation in resonance mode is detected. The motor has a speed of \( f_{dv} = 16.5 \) Hz very close to the natural frequency \( f_0 = 16.18 \) Hz. Taking into account unacceptable frequency limits according to condition (3), the optimal range of natural frequencies relative to the back-up frequency of the electric motor is determined in the range from 22 to 25.38 Hz. You can reach these values by changing the stiffness of the support structure.

6. References

[1] Gumerov A G, Gumerov R S, Akberdin A M 2001 Operation of pumping stations equipment (Moscow: Nedra-Businesscenter LLC) p 475

[2] Gumerov A G, Gumerov R A, Iskhakov R G 2008 Vibration-isolating compensation system of pump-power units (Ufa: SUE "IPTER") p 328

[3] GIDROMASHSERVIS JSC 2012 Pumping equipment for pipeline transportation of oil and petroleum products (Moscow: GIDROMASHSERVIS JSC)

[4] COMPANY STANDARD TRANSENF TU 28.13.14-002-32570437-2016 Vertical oil retaining pumps of the NPV type and electric pump units based on them Technical conditions
[5] Bloch Heinz P, Budris Allan R. 2010. Pump user's handbook: life extension, third edition (Fairmont: Fairmont Press) p 500

[6] Tokarev A, Zotov A, Valeev A. 2017. The application of passive vibroprotective systems having power characteristics with hysteresis loops of rectangular shape for the main pumping units Proceedings of the 3rd International Conference on Dynamics and Vibroacoustics of Machines (DVM2016) DOI: 10.1016/j.proeng.2017.02.279

[7] Tokarev A, Valeev A, Zotov A. 2019. Use of Vibration Isolation Systems with Negative Stiffness on the Basis of Special Shaped Guides to Reduce Pump Piping Vibration. Proceedings of the 5th International Conference on Industrial Engineering (ICIE 2019) Lecture Notes in Mechanical Engineering DOI: 10.1007/978-3-030-22041-9_97

[8] Samarin A A. 1979. The vibrations of pipelines of power equipment and methods of their elimination (Moscow: Energia) p 228

[9] Buranshin A R, Godovskiy D A and Tokarev A P. 2019. Elimination of dead-end oscillations of compressor manufactory piping in operating conditions Bulletin of the Tomsk Polytechnic University Geo Assets Engineering DOI: 10.18799/24131830/2019/9/2268

[10] STATE STANDARD GOST 32388-2013. Technological pipelines Norms and methods of strength analysis, vibration and seismic effects