Assessment of units vibration state and water supply path of Amuzang-2 pumping station

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Abstract. This article presents separate results of studies devoted to the study and analysis of vibrational phenomena occurring on pumping units and along the line of the pressure pipeline of the surveyed pumping stations, associated with flow pulsation in the water supply path and cavitation manifested on pumping units. As is known, the operation of pumps is accompanied by hydrodynamic oscillatory phenomena, which are expressed in the non-stationarity of the field of velocities and pressures of water at the outlet of the pump. This is especially typical for powerful centrifugal pumps with high volumetric flow rates and pressure drops on the impeller, which took place at Amuzang-2 pumping station. Thus, the analysis of the results of vibration tests of pipelines allows us to draw the following conclusions: 1. The vibration state of all pipelines is unsatisfactory. 2. The cause of increased vibration is an unfavorable combination of design, installation, and operational factors.

1 Introduction

The operation of pumps is accompanied by hydrodynamic oscillatory phenomena, expressed in the non-stationarity of the field of velocities and pressures of water at the outlet of the pump[1–3],[4–9]. This is especially true for powerful centrifugal pumps with high volume flows and pressure drops.

The main reasons for the unsteady pulsating flow are inherent in the very nature of the centrifugal pump, its design features, and the physical processes that occur when water flows through the elements of the flow path[10–19].

According to the main frequencies of manifestation, the following hydrodynamic oscillations are distinguished:

- low-frequency pulsations generated by the impact on the flow structure of fixed elements, for example, a spiral volute, as a result of which the axial symmetry of the flow is violated;
- vane vibration and its higher harmonics, caused by asymmetry and hydraulic imbalance of the turbulent flow in the impeller with a finite number of blades;
- high-frequency vibration that accompanies cavitation in the working parts of the pump.

The pulsating flow, interacting with the pressure pipeline, causes forced vibrations of

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the latter, which are dangerous due to their cyclic loading of the thin-walled pipeline shell.

2 Research methods and results

The parameter limiting the safe flow pulsation in [20–29] is the recommended maximum amplitude of pressure pulsation in the pipeline, determined by the formula:

$$\Delta P = (0.20...0.25)\ P_{st} \qquad (1)$$

where: P_{st} is static pressure in the desired section of the pipeline.

This parameter (pressure fluctuations) is not correct enough for inclined pressure pipelines of the pumping station, where it is not entirely logical to obtain a decrease in the allowable pulsation along the pipeline route. In this work, instead of ΔP, the second pressure-related parameter is used - the flow rate, which is unchanged along the length of the pipeline.

Figure 1 shows a graph of changes in the cross-sectional average instantaneous flow velocity in pipelines, measured with a maximum resolution of an ultrasonic flow meter of - 5 seconds over a period of 2 minutes. (for conduit No. 1 - 8 minutes).

As shown by instrumental measurements, the pulsation has a different character for different aggregates, does not depend on the absolute velocity, and after merging into a common conduit, does not go out but somewhat increases.

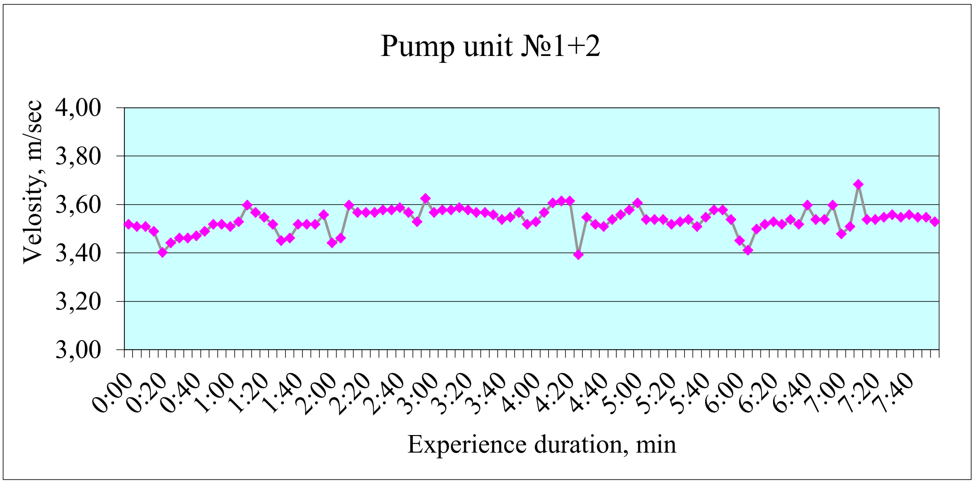

Fig. 1. Graph changes in the average cross-sectional instantaneous flow velocity in pipelines.

Under operating conditions, the magnitude and nature of the pulsation for each unit are determined by the non-stationarity of the flow during the passage of the water supply path at the water intake, the axial asymmetry of the flow in the spiral chamber of the pump, and vacuum fluctuations in the siphon outlet.

The maximum amplitude of the velocity pulsation ΔV about the average velocity Vav in the pipeline according to the test data is:

Table 1. The results of measuring the flow rates on pumping units

	PA№1	**PA№2**	**PA№3**	**PA №5**	**PA№6**	**PA №7**	**PA№8**	**PA№1+2**
V_{mean}, m/sec	3.24	3.14	3.45	3.78	3.79	3.60	3.71	3.54
ΔV, m/sec	0.07 V	0.05 V	0.04 V	0.03 V	0.03 V	0.06 V	0.07 V	0.08 V

Thus, flow pulsations in pipelines do not exceed the recommended limit values.

When pumps operate in non-optimal modes with reduced tailpipe levels, cavitation occurs in them, which worsens the energy and operational parameters and is a dangerous phenomenon that is unacceptable during normal operation.

In large centrifugal pumps, two main types of cavitation are distinguished - gap cavitation in seals and profile cavitation on the impeller blades.

External manifestations of cavitation in pumps are:

- an increase in hydraulic losses, accompanied by a drop in efficiency and a decrease in supply;
- cavitation destruction, primarily of the streamlined surfaces of the impellers;
- the occurrence of hydrodynamic noise and high-frequency vibrations in the flow part of the pumps.

The increase in hydraulic losses due to cavitation occurs with sufficiently developed cavitation, which makes it difficult to recognize at earlier stages of its development. With joint cavitation and abrasive action on the metal surfaces of the working bodies, the latter is of priority. Cavitation destruction forms a spongy structure of the surface layer of the metal, which, due to more active abrasion, loses visual signs of cavitation.

Vibrational manifestations of cavitation in pumps have been studied only theoretically so far. Generalizing full-scale tests, as well as any norms, are currently absent. It is only known that cavitation impacts are high-frequency processes (1...12 kHz).

With such a state of knowledge of cavitation, the data of single full-scale vibration tests on stable operating modes of the NS are only a preliminary structural assessment.

Measurements show that the maximum vibration value at the suction pipe is noted at an average octave frequency of 2000 Hz (from 110 m/s2 on unit No. 8 to 310 m/s2 - on unit No. 3). Except two cases (units No. 7 and 8), in the vertical (B) direction, the vibration is greater than in the horizontal (P). The dependence of vibration activity on the actual feed (for a lower feed, a smaller over-cavitation head is required) and the operating time of the units (with a lower operating time, the roughness of the blades and the gap in the seals are smaller) does not have an explicit expression. So, on unit No. 3, a new impeller is installed, and the vibration is high. Unit No. 5, with the highest operating time and supply, is the least vibroactive.

High-frequency vibration in the flow is well damped and at a small distance - already on the pump bearing housing, it is reduced by ten times.

Thus, the data presented are of a registration nature, reflecting the magnitude of the cavitation vibration activity of pumps of this design and speed in the absence of fairings at the inlet to the impeller and the observed backwater.

During visual inspection of the impeller of unit No. 5, no pronounced traces of cavitation damage in the form of cavitation cavities and spongy structure of the blade surface were found. This indicates that cavitation is at a subcritical safe stage when the destruction of the blades does not yet occur.

The classic cavitation criterion for pumps is the NPSH, and the condition for the absence of cavitation is represented by the relation:

$$H_s \leq \frac{\boldsymbol{P_{atm}}}{\rho g} - \Delta h_{perm} - \frac{P_{\boldsymbol{liquid\ vapor}}}{\rho g} \qquad (2)$$

where: H_s is suction lift (water level head in the suction chambers above the mark of the axis of the pump impeller), m; Δh_{perm} is permissible cavitation reserve, m (graph Δh_{perm} = f(Q) is plotted on the factory characteristic of the pump according to TU 26-06-913-74); $h_{suction}$ is pressure loss in the suction line of the pump, m; $P_{atm}/\rho g = (P_{atm})$ norm. - ∇/900 , m is atmospheric pressure at the downstream level, m; $P_{liquid\ vapor.}/\rho g$ - water vapor pressure, m.

For the conditions of PS Amu-Zang 2:

$$P_{atm}/\rho g = 10.3 - 318/900 = 9.95 \text{ m}; \quad P_{atm}/\rho g = 25*1000/1000*9.81 = 0.25 \text{ m};$$

$h_{suction}$ is for large pumps with a chamber supply; no more than 0.5 m is estimated.

Δh_{perm} is taken according to the characteristic for the maximum fixed flow (17.4 m^3 / s for unit #6) and is 15 m.

After substitution, we obtain the permissible suction height:

$$H_s = 9.95-15-0.5-0.25 = -5.8 \text{ m}.$$

The actual suction lift is:

$$(H_s)_{fact} = \nabla O_{impeller} - \nabla_{downstream\ water\ level}, \text{ m}$$

where: $\nabla O_{impeller}$ is the level of the axis of the pump's impeller, according to the drawing - 318.88 m; $\nabla_{downstream\ water\ level}$ is water level of the lower pool, during testing - 325.6 m.

$$(H_s)_{fact} = 318.88 - 325.6 = -6.72 \text{ m}.$$

The condition for the permissible suction lift has been fulfilled. The overwhelming supra-cavitation head is:

$$6.72-5.8 = 0.92 \text{ m}.$$

For other units with a lower flow, the value of the over-cavitation pressure is even greater.

Thus, the NS units are operated with sufficient support that prevents the development of cavitation. The recorded vibration level corresponds to moderate, subcritical cavitation.

There are only three real, in specific operating conditions, ways to reduce cavitation:

- maintenance of the maximum water level of the lower pool under any water supply regimes;
- restoration of lost fairings at the impeller inlet;
- improving the quality of impeller surfaces during surfacing and processing during repairs.

In addition to the constant static load, which leads to plastic deformation of the shell, pressure pipelines also experience dynamic variable loads, which are dangerous for fatigue failure due to a large number of loading cycles, even in cases where static stresses in the shell material are not too large.

A feature of pipelines as oscillatory systems is the close interaction between the vibration of pipelines and pulsating flow. The propagation of waves through water leads to the fact that vibrations of pipelines can appear at a considerable distance from the source of disturbances, where, when the frequencies of forced and natural vibrations coincide, resonance occurs - a sharp increase in the range of mechanical vibrations under the action of variable pressure in the flow of pumped water. In this case, the frequency of forced oscillations of the pipeline is equal to the frequency of the disturbing forces.

The most active source of disturbing forces in the pressure pipeline is the asymmetry and hydraulic imbalance of the turbulent flow in the impeller with a finite number of blades.

For a 2000 V 16/63 pump, the main frequency of disturbances in the flow is vane (the product of the number of impeller blades and the speed of its rotation)

$$f_l = 6*250/60 = 25Hz,$$

The unevenness of the flow from the inlet to the impeller coincides with the turnover frequency:

$$f_o = 250/60 = 4.16 \text{ Hz}$$

In the presence of a two-volute partition in the pump cochlea:

$$f_a = 25*2 = 50 \text{ Hz}$$

Behind the discharge pipe, guide vanes are installed in the elbow, which divide the flow into six parts:

$$f_n = 25*6 = 150Hz.$$

Resonance phenomena can also occur at higher harmonics of the above frequencies.

The parameter characterizing the dynamic state of pipelines with a known vibration spectrum is the root-mean-square vibration velocity in octave bands of the main (4 ... 150 Hz) range of flow oscillations.

A VShV-003-M2 noise and vibration meter with a DN-3-M1 transducer were used to register the vibration activity of pipelines. Relative error of the device - 10 ... 20% (depending on the frequency range of measurements).

To clarify the direction and form of vibration, measurements were made in the vertical (B) and horizontal (P) directions and to register vibration activity along the pipelines - in the vertical direction.

The vibration state of pipelines based on the test results is presented in Appendix 7 and in Figures 4.21….4.23.

Test data show that the most active frequencies range from 8 to 31.5 Hz. Outside this range, oscillations present a growing or fading insignificant background (hydrodynamic noise).

Increased vibration is observed on all water conduits without exception.

The main form of pipeline oscillations is bending in the cross-sectional plane (simultaneous flattening and expansion of the shell along different diameters), acting in the radial from the axis, including in the vertical and horizontal directions.

In the spans between the supports, a beam shape is added, bending in the plane of the axis of the pipelines (only works in the vertical direction).

Thus both shapes have a vertical component, and the vibration level increases vertically.

When inspecting the water conduits, it was found that the supporting surfaces of the intermediate supports were mounted at different heights from the horizontal axis of the cross-section of the pipelines. With such support, a more complex form of oscillation appears - torsional-bending, in which the pipelines "roll over" from one support leg to another.

As a result of a combination of various forms of vibration, pipelines perform complexcombined vibrations.

For a section of increased vibration activity, for example, between AO3 and AO4, the natural frequency of the beam vibrations is found by the formula:

$$f_c = \frac{a}{L^2}\sqrt{\frac{E\times J}{d}}, \text{Hz}$$

where *a* is coefficient, for a beam freely lying on two supports equal to 49.2; *E* is modulus of elasticity, for steel E = 2.1 * 106 kg / cm^2, J = (π / 64) * (D^4-d^4), cm^4 is moment of inertia of the cross-section of the pipeline, *D* and *d* are outer and inner diameter of the pipeline, cm, *d* is own weight of 1 running cm of the pipeline (including water), kg / cm^2, *L* is span length between supports, cm.

After substitution

$$f_c = \frac{49.2}{1600^2}\sqrt{\frac{2.1\times 10^6 \times 13.2 \times 10^6}{92.4}} = 8.3Hz$$

Resonant frequency range:

$$0.5\ f_c < (f_c)_p < 1.5\ f_c$$
$$(f_c)_p = (4.15...12.45)\ \text{Hz}$$

From a comparison of the frequencies of natural and forced vibrations, it follows that the conditions for resonance are present only at the reverse and double reverse frequencies up to 12.45 Hz. However, in reality, the rigidity of the pipelines is sufficient to extinguish it. The actual vibration carrier is the vane frequency of 25 Hz.

In general, the level of vibration in water conduits, even without considering peak points, is high. In the absence of pipeline vibration safety standards, comparing the results obtained with data on other similar NS is advisable. For example, on Amu-Zang 1 pumping station, the overall vibration level is an order of magnitude lower and does not exceed 10...15 mm/s.

3 Conclusions

Thus, the analysis of the results of vibration tests of pipelines allows us to draw the following conclusions:

1. The vibration state of all pipelines is unsatisfactory.
2. The cause of increased vibration is an unfavorable combination of design, installation, and operational factors, including:

-double trimming of the impellers, due to which the velocity field in the spiral outlet with a two-volute baffle is distorted;

- design tees and sharp turns, the curvature of the longitudinal axis of pipelines, causing a restructuring of the velocity diagram after each next resistance and additional flow irregularity;
- asymmetry of the blade array of impellers;
- asymmetrical laying of intermediate supports.

To reduce the vibration activity of pipelines, it is necessary, first of all, to adjust the height of the intermediate supports, install the inlet fairings of the impellers, and during repairs, ensure the restoration of the symmetry of the profile of the impellers and the two-volute baffle.

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