Theoretical substantiation of the parameters of the vibration protection system for the workplaces of compressor plant operators

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Abstract. The compressor module includes two units – the compressor itself and the energy source, that is, the engine, which come in two types: electric motors for low-power compressors and internal combustion engines. It should also be noted that in almost all sections of compressor stations, sound pressure levels exceed the permissible standard values, and the greater is the power of the compressor unit, the higher is the noise level. The purpose of the carried out theoretical research is to substantiate the parameters of the vibration protection system of the service personnel workplaces.

Keywords: compressor, foundation, vibration protection system, vibration isolation, vibration absorption.

1 Introduction

During the operation of compressor units in the technological mode, vibroacoustic factors arise at the workplaces of the service personnel, which affect the operability [1-4]. In almost all sections of compressor stations, sound pressure levels exceed the permissible standard values. Noise reduction is achieved, among other things, by reducing vibrations by constructive and technological measures. Reducing vibrations at the workplaces of compressor station operators can technically be implemented in two ways [5-7]:

- selection of shock absorbers with the required stiffness parameters;
- designing a foundation with appropriate dimensions and dissipative properties.

The first option is advisable to use for the conditions of compressor stations, when electric motors are used as an energy source. Indeed, for the conditions of such an arrangement of vibration sources, the levels of vibration velocity and vibration acceleration are significantly lower than with internal combustion engines. In addition, vibration dampers can be implemented for this option only for compressors, since their vibration levels are significantly higher than those of electric motors.
2 Main part

The theoretical study of the vibration levels of the foundation excited by the compressor was carried out according to the scheme shown in Fig. 1.

Fig. 1. Calculation scheme of vibrations of an oscillatory system "engine - carrier frame"

\[ P(t) = m_1 \cdot g \cdot K_0 \cdot \sin \omega t \]

\[ K_0 = \text{vibration attenuation coefficient in the presence of vibration isolators;} \]

\[ g = 9.81 \text{ m/s}^2 \text{ - acceleration of gravity;} \]

\[ \omega = \text{circular speed of the crankshaft, rad/s;} \]

\[ m_1 = \text{mass of the power plant, kg.} \]

Then the dependence of the force action will take the form

\[ P(t) = 10 m_1 (1 - \eta) \cdot \sin 0.1nt \]

where

\[ \eta = \text{s the vibrational energy loss coefficient of the vibration isolation system;} \]

\[ n = \text{rotation frequency, rpm.} \]

The validity of this calculation scheme is confirmed by the fact that the mass of the foundation and its geometric dimensions are disproportionately smaller than the mass of the floor and the area of the production room, therefore, in this case, the reaction from the floor to the foundation is not taken into account.

The mass of the foundation is defined as

\[ m = \rho_2 l_1 l_2 h \]

where

\[ \rho_2 = \text{the density of the material, kg/m}^3; \]

\[ l_1, l_2 = \text{the length and width, m;} \]

\[ h = \text{the thickness of the foundation.} \]

The oscillatory system includes a foundation with a mass \( m \) and an engine operating at a rotational speed \( n \) having a mass \( m_1 \), which is mounted on vibration isolators with reduced rigidity \( c_1 \).

The dynamic model of the oscillatory system is described by the differential equation:

\[ m_1 \frac{d^2 z_1}{dt^2} + r \frac{dz_1}{dt} + c_1 z_1 = F_0 \sin 0.1nt \]

\[ m \frac{d^2 z}{dt^2} - r \frac{dz}{dt} - c_1 z_1 + r \frac{dz}{dt} = 0 \]
Let us introduce the notation and to solve it, a system of equations is obtained in the form:

\[ z_1 = F_0[-2c_1^2 m(0,1)n^2 \sin(0,1nt) - c_1^2 m \cdot (0,1)n^2 \sin(0,1nt) + 2c_1 m_1 m (0,1)n^4 \sin(0,1nt) + c_1 m_1^2 (0,1)n^4 \sin(0,1nt) - 2m_2 m^2(0,1)n^6 \sin(0,1nt) - m_1^2 (0,1)n^5 \sin(0,1nt)] - [c_1^2 c_2 - 2c_1 m_2^2 (0,1)n^6 m_1 + m_1^2 (0,1)n^4 c_2^2 + 2 c_1^2 m_1 (0,1)n^4 m_2 - 2 c_1 m_1^2 (0,1)n^6 m_2 + c_1^2 m_2^2 (0,1)n^4 + m^2 (0,1)n^8 m_1^2 + m_1^2 (0,1)n^8]^{-1} \]

\[ z = F_0[-c_1^2 m_1 (0,1)n^2 \sin(0,1nt) + +c_1 c_2 m_1 (0,1)n^2 \sin(0,1nt)] - [-2c_1 m_2 m_1 (0,1)n^6 + +m_1^2 c_2^2 (0,1)n^4 + +2 c_1^2 m_1 m(0,1)n^4 - 2c_1 m_1^2 c_1 (0,1)n^2 + 2 c_1 m_2^2 c_2 (0,1)n^4 - -2c_1 m_2^2 m(0,1)n^6 + c_1^2 m_1^2 (0,1)n^4 + m^2 (0,1)n^8 m_1^2 - 2 c_1 m_2^2 m(0,1)n^6]^{-1} \]

\[ -c_1^2 m(0,1)n^2 - c_1^2 m(0,1)n^2 + 2c_1 m_1 m (0,1)n^4 + c_1 m_1^2 (0,1)n^4 - m m_2^2 (0,1)n^6 = \alpha \]

\[-2c_1 m_1 m^2(0,1)n^6 + m_1^2 c_2^2 (0,1)n^4 + 2 c_1^2 m_1 m(0,1)n^4 - 2c_1 m_1^2 c_1 (0,1)n^2 - 2c_1 m_2^2 m(0,1)n^6 + c_1^2 m_1^2 (0,1)n^4 + m^2 m_2^2 (0,1)n^6 = \gamma \]

\[ z_1 = F_0 \frac{a \sin(0,1nt) + b \cos(0,1nt)}{\sqrt{a^2 + b^2}} \]

\[ z_1 = F_0 \frac{\sqrt{a^2 + b^2}}{\gamma} \sin \left( 0,1nt + \arctg \frac{b}{a} \right) \]

\[ V_1 = \frac{dz_1}{dt} = \frac{m_1 n(1-\eta) \sqrt{a^2 + b^2}}{\gamma} \]

\[ L_\psi = 20 \log \frac{V_1}{5 \times 10^{-9}} = 20 \log \frac{m_1 n(1-\eta)}{\gamma} + 10 \log (a^2 + b^2) + 146, \text{dB} \]

\[ L_\alpha = 20 \log \frac{a^2}{3 \times 10^{-4}} = 20 \log F_0 + 40 \log n + 10 \log (a^2 + b^2) - 20 \log \gamma + 110, \text{dB} \]

\[ -c_1^2 m_1 (0,1)n^2 + c_1 m_1 m (0,1)n^4 - c_1^2 m(0,1)n^2 = \lambda \]

\[ m_1 m r(0,1)n^5 = \eta; \]

\[ -2c_1 m^2 m_1 (0,1)n^6 + m_1^2 c_2^2 (0,1)n^4 + 2 c_1^2 m_1 (0,1)n^4 - 2c_1 m_1 c_2 (0,1)n^2 - -2c_1 m_2 m(0,1)n^6 + c_1^2 m^2 (0,1)n^4 + m \cdot m_1^2 (0,1)n^8 = \nu \]
For the foundation, the maximum value of the vibration velocity and vibration velocity level are determined as:

\[ V_2 = \frac{dz}{dt} = \frac{m_1 n (1 - \eta) \lambda}{\gamma} \]

\[ L_{v_1} = 20 \log \frac{V_1}{5 \cdot 10^{-8}} = 20 \log \frac{m_1 n (1 - \eta) \lambda}{\gamma} \]

On the foundation, the level of vibration velocity should not exceed the allowable values. In this case,

\[ L_C = 20 \log \frac{m_1 n (1 - \eta) \lambda}{\gamma} + 146 \]

Based on this ratio, the parameters of the foundation itself and the vibration isolation system are determined

\[ z = \frac{F_0}{\nu} \sin \left(0.1 nt + \arctan \frac{\eta}{\lambda}\right) \]

In the theory of vibrations, the quality of the vibration protection system is determined by the force transfer coefficient \( K_v \), which is the ratio between the amplitudes of the forces transmitted to the base \( R_{\text{max}} \) and the disturbing \( F_0 \).

\[ K_v = \frac{R_{\text{max}}}{F_0} \]

\[ R = c_1 z_1 = \frac{c_1 F_0 \sqrt{\alpha^2 + \beta^2}}{\gamma} \sin \left(0.1 nt + \arctan \frac{\beta}{\alpha}\right) \]

\[ R_{\text{max}} = \frac{c_1 F_0 \sqrt{\alpha^2 + \beta^2}}{\gamma} \]

\[ K_b = \frac{c_1 \sqrt{\alpha^2 + \beta^2}}{\gamma} \]
shaft or its mass, and variation in the mass of the foundation is in principle possible, but undesirable, therefore, the system of restrictions will take the following form:

\[
K_n \rightarrow \min
\]

\[
c_{1\min} \leq c_1 \leq c_{1\max}
\]

\[
c_{2\min} \leq c_2 \leq c_{2\max}
\]

\[
r_{\min} \leq r \leq r_{\max}
\]

\[
m_{\min} \leq m \leq m_{\max}
\]

(11)

The range of stiffness variation is limited on the basis of the condition that the natural vibration frequencies of the engine on vibration dampers do not coincide with the engine shaft speed, and the drag coefficient \((r)\) must be less than the critical on e. As a result of such a selection, one of the parameters is on the boundary of limitations and a new approximation is set within technically permissible limits.

Determining the natural oscillation frequencies, it is allowed to consider the engine as a solid undeformed body, representing an oscillatory system with six degrees of freedom. However, such an assumption is possible when determining the natural vibration frequencies and the corresponding vibration levels in the frequency range from 4 to 125 Hz.

Thus, the initial data in the calculation of the vibration protection system are the mass of the engine and the moments of inertia about the main axes. Oscillations of an elastically suspended solid undeformed body are described by differential equations:

\[
m \frac{d^2 x}{dt^2} + P_x = 0
\]

\[
l_x \frac{d^2 \theta}{dt^2} + M_x = 0
\]

\[
m \frac{d^2 y}{dt^2} + P_y = 0
\]

\[
l_y \frac{d^2 \psi}{dt^2} + M_y = 0
\]

\[
m \frac{d^2 z}{dt^2} + P_z = 0
\]

\[
l_z \frac{d^2 \xi}{dt^2} + M_\xi = 0
\]

\[
I_x, I_y, I_z, 0, \psi, \xi
\]

\[
\theta, \psi, \xi
\]

\[
q_\xi = A_\xi \sin (0,1nt + \phi),
\]

\[
(K_1 - mp^2)x = 0
\]

\[
(K_2 - mp^2)y = 0
\]

\[
(K_3 - mp^2)z = 0
\]

\[
(K_4 - p^2I_x)\theta = 0
\]

\[
(K_5 - p^2I_y)\psi = 0
\]

\[
(K_6 - p^2I_z)\xi = 0
\]

\[
(K - Mp^2) \cdot Q = 0
\]
where $K$ is the matrix of stiffness coefficients; $M$ is the matrix of inertial coefficients; $Q$ is the matrix of coordinates; $p$ are natural oscillation frequencies, $\text{r/s}$.

**Stiffness coefficient matrix**

\[
K = \begin{bmatrix}
K_{11} & K_{12} & K_{13} & K_{14} & K_{15} & K_{16} \\
K_{21} & K_{22} & K_{23} & K_{24} & K_{25} & K_{26} \\
K_{31} & K_{32} & K_{33} & K_{34} & K_{35} & K_{36} \\
K_{41} & K_{42} & K_{43} & K_{44} & K_{45} & K_{46} \\
K_{51} & K_{52} & K_{53} & K_{54} & K_{55} & K_{56} \\
K_{61} & K_{62} & K_{63} & K_{64} & K_{65} & K_{66}
\end{bmatrix}
\]

**Inertia coefficient matrix**

\[
M = \begin{bmatrix}
m & 0 & 0 & 0 & 0 & 0 \\
0 & m & 0 & 0 & 0 & 0 \\
0 & 0 & m & 0 & 0 & 0 \\
0 & 0 & 0 & I_x & 0 & 0 \\
0 & 0 & 0 & 0 & I_y & 0 \\
0 & 0 & 0 & 0 & 0 & I_z
\end{bmatrix}
\]

**Coordinate matrix**

\[
Q = \begin{bmatrix}
x \\
y \\
z \\
\theta \\
\psi \\
\xi
\end{bmatrix}
\]

When solving a system of differential equations, it is necessary that the determinant of the system be equal to 0, that is,

\[
|K - M p^2| = 0,
\]

which makes it possible to determine six values of natural vibration frequencies.

### 3 Conclusion

The vibration protection system includes vibration isolation and vibration absorption methods, and the definition of its parameters has the following algorithm:

- a dynamic model of the system is created, given by equations (3) and (4), and an equation is made with respect to $z_1$ (5);
- the force transfer coefficient is determined, from expression (10) and in general form a system of restrictions is formed to minimize it (11);
- to solve the minimization of the force transfer coefficient, an appropriate software package (MATLAB) is used. In addition, if the parameter values are at the limits of the restrictions, the calculation procedure should be repeated within technically acceptable limits;
- when solving the task to determine the permissible values of vibration velocity (6) and vibration acceleration (7) during operation, their permissible values are substituted into the left side of the expression instead of the actual values of the vibration transmitted to the foundation.

The data obtained during the calculations make it possible to determine the natural vibration frequencies of the engine and the normalized frequency ranges in which they fall, as well as the vibration levels of the foundation at its attachment points.

Comparison of the values of vibration levels obtained by calculation with normalized values makes it possible to determine the required efficiency of vibration protection systems.

### References

