Precision vibration measurement systems

S Stefanov*, R Hristov and P Toteva
Faculty of Mechanical Engineering and Technologies, Technical University of Varna, 1 Studentska Street, Varna 2010, Bulgaria

* corresponding author: stefanov_48@abv.bg

Abstract. This report examines the design of a vibrostand for precision vibration measurements by comparing the parameters of the transducer with that of a reference transducer. The report gives the methodology for designing a stand for precise vibration measurements. Based on the theoretical calculation, a stand for precise vibration measurements was designed and manufactured. The manufactured stand has the lowest possible natural frequency, a characteristic of the suspension approaching the ideal, the possibility to adjust the stiffness of the suspension and a large load capacity.

1. Introduction
Engineering requires that the values of different parameters be measured. The measured parameters' accuracy depends on the objectives of the tasks. For this purpose, the companies producing different types of measuring instruments produce also transducers converting measurement parameters of different accuracy classes. The measured parameters have heterogeneous nature, such as pressures, temperatures, displacement, velocity, acceleration, etc. These instruments and transducers may, over time, register some deviations from the set parameters [1,2]. Each country has its metrological control service, which verifies the parameters and issues a certificate of serviceability and suitability of measuring instruments. Precise measurements need very accurate equipment because measurements with less accuracy may be incorrect [3,4,5]. Vibration measurements at low frequencies are used in several applications, as for example: environmental monitoring, seismic measurement, human-body vibration assessment, mechanical tests. A comparison calibration system requires a calibrated reference standard to quantify the measured with high accuracy [6].

This report discusses the design of vibrostand for precision vibration measurements, comparing the parameters of the transducer with those of a reference transducer.

2. Explanation
The main objective of project described at this paper is:
1. To design and develop vibration isolation system for precision vibration measurements of piezoelectric and inductive vibration transducers measuring vibration acceleration, vibration velocity and vibration displacement.

2.1. Operating Principle
The system must ensure maximum isolation of the tested vibration transducer from external influences.

After studying similar systems and structures, pneumatic vibration isolation scheme was selected.
The main indicators that the system must meet are: low natural frequency; characteristic of the suspension must approach the perfect one, adjustability of the suspension stiffness, high load capacity and relatively low cost [7].

The general view of the system is shown in figure 1. The two units (left and right) are absolutely identical and can work both together and independently of each other. The pumping of vibration isolators 5 is performed by compressor 2, with nozzle 11 being placed in one of the eight valves. The compressor is switched on by switch 9. The pressure in vibration isolators (0.08 - 0.1) [MPa] is read by pressure gauge 10.

Figure 1. General view of the system

2.2. Theoretical Statement:
Figure 2 shows the vibration insulated plate, which is considered as a solid body with six degrees of freedom. Two coordinate systems are introduced: movable - with the main inertial axes of the plate (x, y, z), which is fixed and coincides with the movable plate at its static equilibrium (X, Y, Z). The plate displacements during oscillations are insignificant. For this reason, the rotation of movable, compared to the fixed coordinate system is set with the elementary angles - φ, ψ, θ.

The foundation free oscillations are described by a system of linear second order differential equations with constant coefficients:

\[ m\ddot{X} + a_{11}X + a_{12}Y + a_{13}Z + a_{14}\phi + a_{15}\psi + a_{16}\theta = 0 \]

\[ m\ddot{Y} + a_{21}X + a_{22}Y + a_{23}Z + a_{24}\phi + a_{25}\psi + a_{26}\theta = 0 \]

\[ m\ddot{Z} + a_{31}X + a_{32}Y + a_{33}Z + a_{34}\phi + a_{35}\psi + a_{36}\theta = 0 \]
\begin{align}
  m \cdot \rho_x^2 \cdot \phi + a_{41} \cdot X + a_{42} \cdot Y + a_{43} \cdot Z + a_{44} \cdot \phi + a_{45} \cdot \psi + a_{46} \cdot \alpha &= 0 \\
  m \cdot \rho_y^2 \cdot \phi + a_{51} \cdot X + a_{52} \cdot Y + a_{53} \cdot Z + a_{54} \cdot \phi + a_{55} \cdot \psi + a_{56} \cdot \alpha &= 0 \\
  m \cdot \rho_z^2 \cdot \phi + a_{61} \cdot X + a_{62} \cdot Y + a_{63} \cdot Z + a_{64} \cdot \phi + a_{65} \cdot \psi + a_{66} \cdot \alpha &= 0
\end{align}

where:

- \( m \) is the foundation mass
- \( X, Y, Z \) is the foundation plate displacement compared to the fixed coordinate system
- \( \phi, \psi, \alpha \) are angles of rotation of the plate relative to \( X, Y, Z \)
- \( a_{ij} \) are constant coefficients reflecting the linear, linear-angular and angular stiffness of suspension. The matrix of these coefficients is symmetric - \( a_{ij} = a_{ji} \).
- \( X, Y, Z \) are suspension points' coordinates in the movable coordinate system
- \( \rho_x, \rho_y, \rho_z \) are inertial radii of mass relative to \( x, y \) and \( z \) axes.
- \( \alpha, \beta, \gamma \) are guidance vectors of the vibration isolators' stiffness axes.

The analysis of the equation system shows that the action of force or torque on the system causes oscillations in the six main coordinates and that the foundation will perform interconnected oscillations. The theory of oscillations shows that, other things being equal, their interconnection leads to expansion of the spectrum of free oscillations, and that with its increase the difference between lower and higher frequencies of free oscillations increases. Removing oscillations' interconnection is equal to separating oscillating motions into directions along and around the \( X, Y, \) and \( Z \) axes. Complete separation can be obtained if the matrix of coefficients \( a_{ij} \) is diagonal (\( a_{ij} = 0 \) for \( i \) other than \( j \)).

2.3. Design of Precision Vibration Measurement System

The scheme of the system is shown in Figure 3. The axes of fixed and movable coordinate systems coincide with the main inertial axes of the foundation plate. The vibration isolators are located...
symmetrically to XZ and YZ planes and lie in XY plane. Their main stiffness axes are parallel to the coordinate axes. After removing the zero terms from the differential equation system, the natural oscillations of this suspension system are described by the following expressions:

\[
\begin{align*}
\text{m.} \ddot{X} + a_{11}X &= 0 \\
\text{m.} \ddot{Y} + a_{22}Y &= 0 \\
\text{m.} \ddot{Z} + a_{33}Z &= 0 \\
J_x \phi + a_{44} \phi &= 0 \\
J_y \phi + a_{55} \psi &= 0 \\
J_z \phi + a_{66} \psi &= 0 \\
\end{align*}
\]

(2)

where:

- \(J_x, J_y, J_z\) are mass inertial torques of the plate relative to the coordinate axes.

The calculations are made by optimising the parameters in order to obtain compactness of the structure.

1. Number of vibration isolators on one stand – \(N = 4\)
2. Mass of one stand, together with the vibrotable and sensor, at the time of measurement - \(m = 180 \text{ kg}\).
3. Working absolute pressure in the vibration isolators [8]

\[
 p = \frac{m \cdot g}{4 \cdot F} + 0.981 \cdot 10^5 = 1.86 \cdot 10^5 \text{[N/m}^2\text{]} 
\]

(3)

where: \(g = 9.81 \text{ [m/s}^2\text{]}\) - gravity variations;

\[
F = \frac{\pi \cdot D_a^2}{4} \text{ - active area of a vibration isolator.}
\]
Da = 0.1 [m] - vibration isolator diameter

4. Volume of one vibration isolator.

\[ V_a = \frac{\pi D_a^2}{4} \cdot h = 1.414 \cdot 10^{-3} [m^3] \]  \hspace{1cm} (4)

where: h = 0.18 [m] - vibration isolator height

5. Volume of the receiver to the vibration isolator.

\[ V_p = 4 \cdot V_a = 5.655 \cdot 10^{-3} [m^3] \]  \hspace{1cm} (5)

\[ V_p = \frac{\pi D_p^2 h_p}{4} = 5.67 \cdot 10^{-3} [m^3] \] - volume, at \( D_p = 0.16 [m] \) and height \( h_p = 0.282 [m] \)

6. System natural frequency.

\[ \omega_n = \sqrt{\frac{k_p F_2}{m V_p}} = 5.087 [s^{-1}] \]

\[ f_n = \frac{\omega_n}{2 \pi} = 0.80 [Hz] \]  \hspace{1cm} (6)

7. System resonant frequency at optimal damping.

\[ \omega_c = \omega_n \sqrt{\frac{2N}{N+2}} = 5.87 [s^{-1}] \]

\[ f_c = \frac{\omega_c}{2 \pi} = 0.93 [Hz] \]

\textbf{Figure 4.} Characteristic of this system
8. Dimensions of the throttle capillary tube connecting the vibration isolator and receiver.

\[ d = \left( \frac{128 \mu_L F^2}{2 \pi \omega_c D_{\text{a,m}}} \right)^{\frac{1}{2}} \]

where:
\[ \mu_g = 1.81 \times 10^{-5} \text{ kg/s.m} \]  

\[ d = 1.7 \times 10^{-3} \text{ [m]} \text{ at } L = 0.05 \text{ [m]} \]

The characteristic of this system - \( T_z(\gamma) \) is given in figure 4 and is subject to the following dependence:

\[ T_z(\gamma) = \frac{1+4 D_0^2 \gamma^2}{\sqrt{4 D_0^2 \gamma^2 (1 - \gamma^2)^2 + (1 - \frac{1+N}{N} \gamma^2)^2}} \]

where: \( \gamma = \frac{\omega}{\omega_c} \); \( D_0 = 0.968 \)

**Conclusions**

Based on research that has been done, the following conclusions can be made:

- An appropriate methodology has been selected and the precision vibration measurement system has been calculated.
- According to the obtained results, the precision vibration measurement system is made.
- It is used successfully in Varna Quality Centre.

**References**

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