Abstract: There are situations when certain mechanical systems or buildings may be damaged or destroyed in case of external dynamic loadings from human activities or due to natural hazard (seism). A measure to counteract these negative consequences is including in the mechanical system or to exposed dynamic loadings construction, of some systems for isolation and damping oscillatory movements, such as viscous-elastic systems. These viscous-elastic systems, by prolonged exposure to atmospheric factors and claims arising from anthropogenic environment, change their specific operating parameters leading in certain situations to uncontrolled movements that cannot be anticipated. This situation corresponds to behaviour nonlinear of viscous-elastic system, elastic and damping forces having nonlinear expressions. This paper aims to identify parameters capable of indicating the point from which the viscous-elastic system has a changed behaviour as a nonlinear kind.

Keywords: dynamic, viscous-elastic, nonlinear, antivibration, shock.

1. SYSTEMS FOR ISOLATING THE MACHINE FOUNDATION

Worldwide, efforts have been intensified to control and mitigate the vibration of both the technological machinery and equipments and that of buildings and bridges. Due to the heavy and varied demand regime, to which passive control systems are subjected, they suffer, in time, degradation that can lead to quantitative changes of the dynamic response of heavy industrial machinery that work with impulsive action, [1]. The degradation occurring in the dynamic isolation systems determines the change of the expressions of stiffness and damping forces from linear mathematical relationships into nonlinear dependence. It has experimentally been found that this nonlinear behaviour of vibration isolation systems leads to generating oscillatory movements that sometimes exceed the limit levels required by standards in the field. For this reason, the central objective of this paper is to: theoretical quantify quantitatively the nonlinearity degree of passive control systems, through determining the parameters of influence (control) able to characterize not only the cause but also the effect of nonlinear behaviour. Taking this into consideration, a methodology can be developed, based on experimental measurements, able to evaluate in time, the changes in the dynamic behaviour (response) of shock and vibration generating equipment, according to the change in the characteristics of viscous-elastic systems installed under the foundation of the equipment. The avoidance of the appreciation errors in applying this methodology requires its implementation right from the moment of putting into service of the dynamic isolation system of a viscous-elastic type. The evaluation of the isolators’ degradation is achieved through periodic experimental determinations in order to determine the influence parameters and to analyses them comparatively with their initial values.
Internationally there is a modern identification technique and a quantification of the damages of an antiseismic devices bridge of hydraulic dissipater type. This algorithm was successfully applied for the quantification of the structural integrity of the bridge Vincent Thomas - Los Angeles.

2. ANALYSIS OF DYNAMIC BEHAVIOR OF MECHANICAL SYSTEMS MODELED AS RIGID WITH VISCOUS-ELASTIC TRIORTOGONALE LINKS

The identification and quantification of parameters able to characterize the nonlinear behaviour of viscous-elastic systems for dynamic isolation of mechanical structures, involves at a theoretical level, the development of a physical and mathematical model that can accurately characterize dynamically the behaviour of the impulsively loaded system. That is why a physical model is proposed with a very general nature, namely that of solid rigid leaning on four viscous-elastic triortogonal supports (fig. 1), [2].

Fig. 1 The physical model

The rigid movements depending on the generalized coordinates are defined as follows:

- \( X \) - lateral forced vibration (sliding);
- \( Y \) - forced longitudinal vibration (forward);
- \( Z \) - forced vertical vibration (leaping);
- \( \phi_x \) - forced vibration pitch (galloping);
- \( \phi_y \) - forced vibration turning (return).

This model corresponds to technological equipment such as forging hammers, screw presses, etc., in which rigid solid corresponds to the foundation of machinery. It is considered that the two coordinated planes of symmetry are XOZ and YOZ, in which there are the following terms:

\[
\begin{align*}
\sum k_{ix}x_i &= 0; k_{ix}x_i = 0; \\
\sum k_{iy}y_i &= 0; \sum k_{ix}x_i = 0; \\
\sum k_{iy}y_i &= 0; k_{iy}y_i = 0; \\
\sum k_{iz}z_i &= 0; \sum k_{iz}z_i = 0;
\end{align*}
\]

\[
\begin{align*}
\sum c_{ix}x_i &= 0; c_{ix}x_i = 0; \\
\sum c_{iy}y_i &= 0; c_{ix}x_i = 0; \\
\sum c_{iy}y_i &= 0; c_{iy}y_i = 0; \\
\sum c_{iz}z_i &= 0; c_{iz}z_i = 0;
\end{align*}
\]

where:

- \( k_{ix}, k_{iy}, k_{iz} \) are stiffness coefficients of the bearings after three directions,
- \( c_{ix}, c_{iy}, c_{iz} \) are viscous damping coefficients after the three directions,
- \( x_i, y_i, z_i \) are rigid motions.

The equations of forced vibration characterizing the foundation behaviour in steady state are in relation (3):

\[
\begin{align*}
\dot{m} \ddot{X} + 4c_x \dot{X} + 4k_x \dot{X} - 4hk_x \dot{\phi}_x &= 0 \\
\dot{m} \ddot{Y} + 4c_y \dot{Y} + 4k_y \dot{Y} + 4k_y \dot{\phi}_y &= 0 \\
\dot{m} \ddot{Z} + 4c_z \dot{Z} + 4k_z \dot{Z} &= -F_z \\
\dot{J} \ddot{\phi}_x + 4hc_x \dot{\phi}_x + 4(c_x h^2 + c_x n^2) \dot{\phi}_x + \\
+4hk_x \dot{Y} + 4(k_x h^2 + k_x n^2) \dot{\phi}_y &= -c_x F_x \\
\dot{J} \ddot{\phi}_y - 4hc_x \dot{\phi}_y + 4(c_x d^2 + c_x h^3) \dot{\phi}_x - \\
-4hk_x \dot{Y} + 4(k_x d^2 + k_x h^2) \dot{\phi}_y &= c_x F_y \\
\dot{J} \ddot{\phi}_z + 4(c_x n^2 + 2c_x d^2) \dot{\phi}_x + \\
+4(k_x n^2 + 2k_x d^2) \dot{\phi}_y &= 0
\end{align*}
\]

where \( m \) is foundation mass, \( k \) is rigidity of the viscous-elastic element, \( c \) is damping of the viscous-elastic elements, \( J \) is inertia moments of the foundation block.

The main elastic axes of elastic bearing are parallel to the reference axes.
In this case, the movements represented by the variation of coordinates corresponding to the six degrees of freedom, may be decoupled as follows:

- coupled translational motion along the axis X and rotation around axis Y ($X, \varphi_y$);
- coupled translational movement along the Y axis and rotation around the X axis ($Y, \varphi_x$);
- translational movement along the Z axis independent of the other ways;
- rotation around Z axis ($\varphi_z$) independent of the other ways.

Next only the translation movement along the Z axis will be analyzed.

3. CHARACTERIZATION OF FORCE APPLICATION

System excitation is considered as impulsive force $F_z$ eccentrically applied on the axis OZ. This model is specific for technological equipment to which the technology force is impulsively applied in the vertical direction during the working process. The machine-foundation system is considered, stressed by a disruptive force four-pulse train haversine type (fig. 2 a), the time between applying two consecutive pulses being close but uneven, [5]. Haversine impulse type (fig. 2 b) has the following characteristics: amplitude of $9 \cdot 10^6$ N, frequency 7.1 Hz and $\varphi = 3 / 7 \pi$.

4. THE HYPOTHESES FOR CALCULATION

In order to reveal the influence of the nonlinear nature of antivibration system parameters (stiffness and damping) on the dynamic performance of the isolated system, a number of parameters characteristic to vibration are investigated in the following situations:

- elastic and damping forces have linear expressions;
- elastic force in the OZ direction has a nonlinear expression;

$$F_{ez} = k_z (1 + \text{sgn}(\dot{z}) \cdot \beta_1 \cdot z + \beta_2 \cdot z^3) \cdot z,$$

$\beta_1 = 5 \cdot 10^2 \ 1/m^2; \ beta_2 = 2 \cdot 10^6 \ 1/m^2$,

and viscous forces in the direction OZ have linear expression.

The equation of motion is then like relation (4):

$$m \ddot{Z} + 4c_z \dot{Z} + 4k_z Z = -F_z$$

(4)

where: $m$ is foundation mass, $k_z$ is rigidity of the viscous-elastic element on OZ direction, $c_z$ is damping of the viscous-elastic elements on OZ direction, $Z$ displacement on OZ direction and $F_z$ is excitation force.

Graphical representations were performed using Matlab, [4] and [6].
5. FORCED LEAP VIBRATION

A. The response in time of the motion on the OZ axis

The representation in time of the movement on OZ axis, highlights its amplitude decrease for nonlinear elastic force, to a value of $3.8 \times 10^{-4} \text{ m}$ compared with $4.5 \times 10^{-4} \text{ m}$ in the linear case, fig. 3.

![Fig. 3 Movement on OZ axis](image)

B. The response in frequency of movement on OZ axis

The frequency responses of the displacement of the system in the OZ direction for the two considered cases are similar in values of dominant frequency spectral components, fig. 4.

![Fig. 4 Frequency spectrum](image)

C. Energy dissipated by viscous friction

The value of dissipated energy through viscous friction in one period of movement, decreases slightly in the nonlinear case $W=930.76 \text{ J}$ compared to the linear one $W=934.79 \text{ J}$, the difference being explained by reducing the movement on axis OZ, fig. 5.

![Fig. 5 Energy dissipation](image)
b. \( F_{ez} = k_z (1 + \text{sgn}(\dot{z}) \cdot \beta_1 \cdot z + \beta_2 \cdot z^2) \cdot z \)

W = 930.76 J

Fig. 5 The hysteretic loop

In the nonlinear case, hysteresis curve changes its shape, its median curve being the stiffness characteristic of the isolation systems.

E. Response in time of acceleration on OZ direction

From the representation in time of the system acceleration on the OZ direction one can observe a significant increase in amplitude for the nonlinear case to the value 32 m/s\(^2\) in comparison with 22 m/s\(^2\) value in the linear case, fig. 6. The explanation of the change of the linear behaviour of viscous-elastic systems into a nonlinear one is the modification of the structural integrity because of their wear or due to external agents such as temperature, humidity and others.

Fig. 6 Acceleration on OZ axis

This phenomenon increases the harm degree of the vibrations from the technological equipment towards the environment. The nonlinear behaviour of viscous-elastic systems, according to their structural changes, should be closely monitored over time through certain specific parameters, due to the chaotic motion that may occur during the operation of industrial equipment.

F. Frequency response of acceleration in the OZ direction

This representation identifies, for the nonlinear case, the two significant areas of spectral components with central values in 32.8 Hz and 103.4 Hz, compared to the linear case in which the dominant area is centered on the value of 28 Hz, fig. 7.

a. \( F_{ez} = \) linear function; \( F_{vz} = \) linear function
b. \( F_z = k_z (1 + \text{sgn}(z) \cdot \beta_1 \cdot z + \beta_2 \cdot z^2) \cdot z \)

Fig. 7 Frequency spectrum of acceleration on the OZ axis

6. CONCLUSION

Moulding the foundations of the machines as rigid with six degrees of freedom leaning on viscous-elastic triorthogonal systems, presents a character of generality, this model can thus be simplified according to the characteristics of the equipment or of the dynamic isolation systems.

From the analysis of the results the following aspects can be highlighted:

- among the cinematic parameters of the system vibration disturbed by impulsive forces, the variation of acceleration is the most sensitive in order to highlight the presence of nonlinearity in the viscous-elastic system
- the presence of elastic and/or dissipative nonlinear forces leads to super-harmonic and sub-harmonic vibrations, which increases the possibility of the resonance phenomenon
- the energy dissipated by viscous friction decreases for nonlinear vibration, which means that the industrial vibration propagated have a greater amount of energy that they transmit to the environment.

In this way, this paper has established and characterized a series of cinematic parameters, the displacement, acceleration and energy dissipated by hysteresis, which through monitoring in time can characterize the degree of nonlinearity of dynamic isolation systems, and consequently the degree of degradation of their viscous-elastic links, [3].

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7. REFERENCES


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