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ANALYSIS OF DYNAMIC VIBRATION ABSORBER EXPOSED TO A NON-PERIODIC COMPLEX EXCITATION

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ABSTRACT
This work contains analysis of a dynamic vibration absorber with damping which is exposed to a non-periodic complex form of excitation. In the beginning an analytical model of the dynamic absorber was developed. After that, the determination of systems main mass dynamics and the corresponding equations of motion for the simple case of harmonic excitation acting on the system were made.
A FORTRAN program was developed that counts the basic movement of the mass of a system which is exposed to a non-periodic complex excitation and which is based on the application of Runge - Kutta and Newmark methods. Using this program we have made the necessary analysis and calculations.
Keywords: dynamic vibration absorber, complex excitation, FORTRAN, numerical analysis.

1. INTRODUCTION
An important segment of modern engineering is the analysis and prediction of dynamic behavior of physical systems. One common form of the dynamic behavior is the vibrational motion, in which the system vibrates around some equilibrium position. This paper is the result of efforts to contribute to a more detailed study of dynamical models of vibration absorbers with damping, the method of modeling through dynamic parameters, in order to assess more fully the character of their actions on the system. Of particular interest is the analysis of vibration behavior of the dynamic absorber system when the system operates non-periodic complex excitation. On this way we would come to the conclusion how to design such a system with the aim to reduce the negative effects of vibration on the reliability of the system.

2. VIBRATION ABSORBER WITH DAMPING
We observe the damped vibration absorber shown in Figure 1, which actually represents a mechanical system in the form of a single-chain embedded translator. Thereby between mass M_1 and M_2 whose sizes are m_1 and m_2, there is a damper whose damping force is proportional to the relative velocity of the mass:

$$F_w = b\left(\dot{x}_2 - \dot{x}_1\right) \qquad \ldots(1)$$

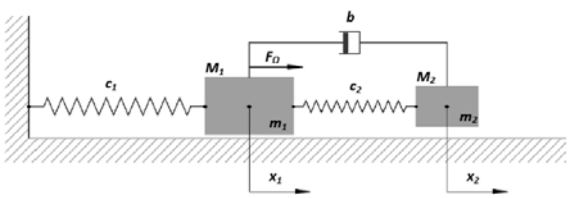

Figure 1.Mechanical system with a damping absorber

Let the mass M_1 be effected by a disturbance force in harmonic form:

$$F_{\Omega} = F_0 \cos \Omega t \ , \qquad \ldots(2)$$

Where: F_0 - amplitude of force, Ω - angular frequency of change of force, t -time.

Differential equations of small vibrations of the mentioned masses have the form:

$$\begin{gathered} m_1\ddot{x}_1 + b\left(\dot{x}_1 - \dot{x}_2\right) + c_1 x_1 + c_2\left(x_1 - x_2\right) = F_0 \cos \Omega t \ , \\ m_2\ddot{x}_2 + b\left(\dot{x}_2 - \dot{x}_1\right) + c_2\left(x_2 - x_1\right) = 0 \, . \end{gathered} \qquad \ldots(3)$$

In order to obtain the dynamic amplification factor η_1 of the primary mass m_1, it is necessary to know the maximum amplitude of forced vibrations of the primary mass, at dynamic acting of the disturbance force $F_{\Omega} = F_0 \cos \Omega t$, which is shown in the expression (4).

$$\eta_1^2 = \frac{\left[\left(b\Omega\right)^2\left(c_1 - \Omega^2 m_1 - m_2\Omega^2\right) + \left(c_1 - \Omega^2 m_1\right)\left(c_2 - \Omega^2 m_2\right)^2 - \left(c_2 - \Omega^2 m_2\right)c_2 m_2 \Omega^2\right]^2 + \left(b\Omega\right)^2\left(m_2\Omega^2\right)^4}{\left\{\left[\left(c_1 - \Omega^2 m_1\right)\left(c_2 - \Omega^2 m_2\right) - c_2 m_2 \Omega^2\right]^2 + \left(b\Omega\right)^2\left(c_1 - \Omega^2 m_1 - m_2\Omega^2\right)^2\right\}^2} \qquad \ldots(4)$$

Previously stated is true only if the excitation has a harmonic form. Since in most cases it is not so, there is a need for more detailed analysis of the dynamic vibration absorber to which acts an excitation of complex form.

3. NUMERICAL ANALYSIS AND THE RESULTS

Tests and calculations were done in a specific order and methodology. First, we select a shape that represents the excitation force and acts on the primary mass, then the force is reduced to a mathematical form that is useful for numerical analysis.

As an example of a non-periodic (complex) excitation acting on the primary mass, an excitation is taken that is shown in figure 2. First, it was necessary to reduce this excitation in mathematical form that is acceptable for analysis. This is done so that the specified function is approximated using a polynomial of 10-th order.

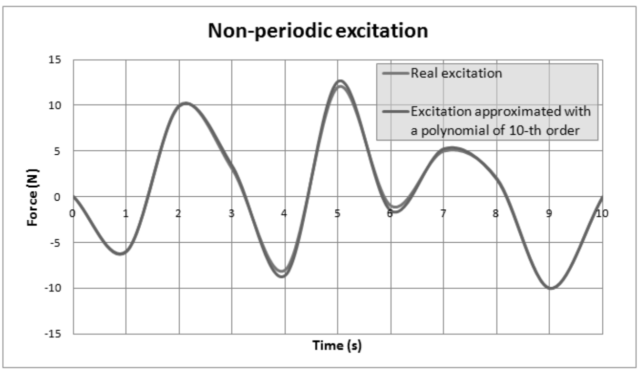

Figure 2. Non-periodic excitation approximated with a polynomial of 10-th order

As seen in the picture, we got a very good approximation with the polynomial of 10-th order. The polynomial describing the force is shown in the following form:

$$F(t) = 0,01 \cdot (-1.554087151 \cdot 10^{-3} \cdot t^{10} + 7.773322397 \cdot 10^{-2} \cdot t^{9} - 1.663876623 \cdot t^{8} + \\ +19.90282262 \cdot t^{7} - 145.6982264 \cdot t^{6} + 671.5820673 \cdot t^{5} - 1929.607463 \cdot t^{4} + \\ +3289.489236 \cdot t^{3} - 2969.011872 \cdot t^{2} + 1058.985062 \cdot t + 2.901648185 \cdot 10^{-2}) \quad \ldots(5)$$

After that it was necessary to select the appropriate calculation method for movements of the basic mass of the system to which acts the above stated excitation. We applied two methods to solve the problem, method of type Runge-Kutta of fourth order and the Newmark method. For numerical analysis programs are written in FORTRAN with which we can count on the movements of the basic mass depending on the time. We have analyzed a system to which acts the named excitation with the following parameters:

$m_1 = 20 \quad kg$ - main system mass, $m_2 = 1 \quad kg$ - mass of the dynamic absorber, $c_1 = 200 \quad N/m$ - stiffness of the main system (system spring), $c_2 = 10 \quad N/m$ - stiffness of the dynamic absorber (stiffness of the absorber). Moving of basic mass is shown in the following figures:

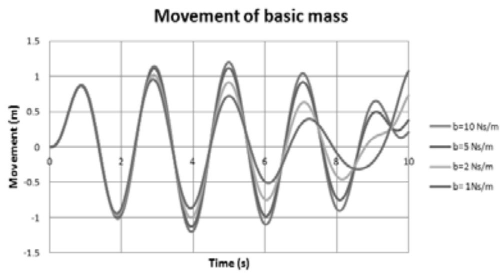

Figure 3. Movement of basic mass - Runge-Kutta fourth order

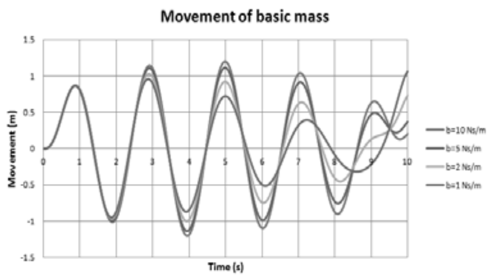

Figure 4. Movement of basic mass – Newmark method

If we compare the movement obtained by the Runge-Kutta fourth order method (figure 4) and Newmark method (figure 5), we see that we gets very good matching results. The maximum displacement for $b = 10 \quad Ns/m$ obtained using the Runge-Kutta fourth order method is $x_1 = 0,01204536$ *m*, and for Newmark method it is $x_1 = 0,01204356$ *m*. We see that the results differ after the fifth decimal. Verification of the results for these methods was made by comparison with a problem that can be solved analytically. We concluded that the Newmark method is more accurate than Runge-Kutta method. The reason for this is that the error using Runge-Kutta method increases with the number of steps, i.e. with decreasing the time interval Δt.

To consider the changes in intensity of amplitude of the main mass at the various parameters of the system, we've analyzed the change in intensity of amplitude at constant attenuation coefficient ($b = const.$) and varied the other characteristics of the system.

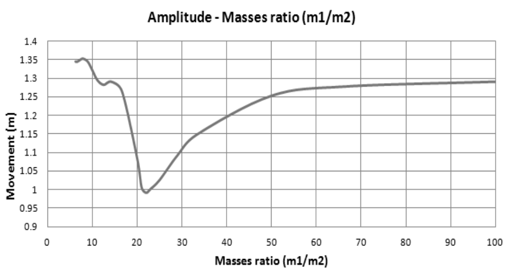

Figure 5. Moving of basic mass depending on the mass ratio (m_1/m_2)

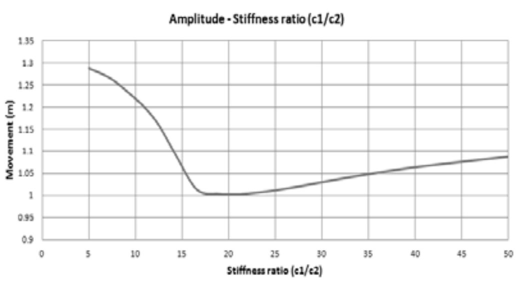

Figure 6. Moving of basic mass depending on the stiffness ratio (c_1/c_2)

Taking into account the above, the parameters of the system for which we get the minimum amplitudes independent of the absorber damping, are following: $m_1 = 20\ \ kg$, $m_2 = 0,87\ \ kg$, $c_1 = 200\ \ N/m$ and $c_2 = 9,756\ \ N/m$.

As next we have analyzed a system wit above stated parameters whereby we change the attenuation coefficient of the absorber b. Figure 7 shows the changes of amplitude of basic mass depending on the attenuation coefficient of the absorber.

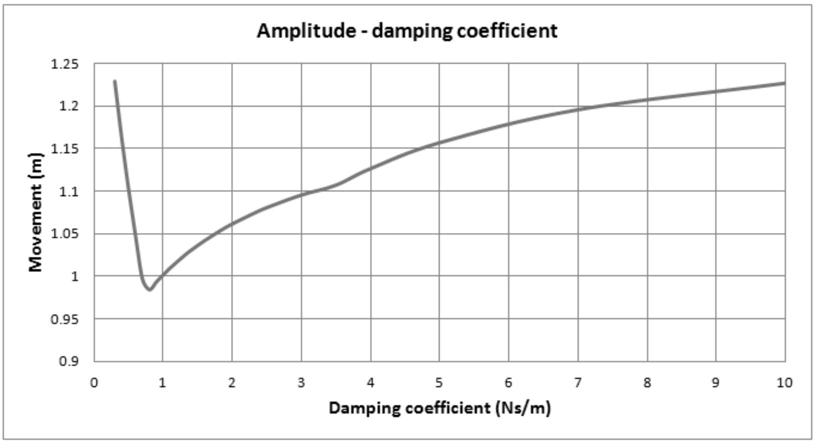

Figure 7.Moving of basic mass depending on the attenuation coefficient

It was found that the smallest amplitude of the basic mass is obtained for $b = 0,8\ \ Ns/m$.

When we take into account all the previous analysis for the case of excitation, we can conclude that the ideal parameters are as follows: $m_1 = 20\ kg$, $m_2 = 0,87\ kg$, $c_1 = 200\ \ N/m$, $c_2 = 9,756\ \ N/m$ i $b = 0,8\ \ Ns/m$.

4. CONCLUSION

With this we have shown a way for analysis of a system with dynamic damper on which is acting a non-periodic complex excitation. It should be noted that in this case we can get the ideal parameters of the absorber only by numerical or experimental means. We see that the amplitude of basic mass, i.e. its movement depends on the damper characteristics and the shape and type of excitation. This procedure could be used for some other form of excitation, in order to determine the ideal parameters of absorbers.

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