Evaluation of the Dynamic Regime Upon Trial of An Elastic Beam Excited by Vibrodyne

Nicuşor DRĂGAN
"Dunărea de Jos" University of Galaţi, Romania, nicusor.dragan@ugal.ro

Gigel Florin CĂPĂŢÂNĂ
"Dunărea de Jos" University of Galaţi, Romania, gcapatana@ugal.ro

Aurora Maria POTÎRNICHE
"Dunărea de Jos" University of Galaţi, Romania, aurora.potirniche@ugal.ro

Abstract: - There are presented the outcomes of the research on the dynamic behavior for an elastic beam dynamically excited by an inertial vibrator placed in the middle of it.
In this case the dynamic load inertial vibrator is called vibrodyne and has a rotating inertial force with known static moment of the eccentric mass of dynamic imbalance.

Keywords: - rigid beam, elastic beam, elastic dynamic loading

1. INTRODUCTION

The testing of the elastic beams in dynamic regime is conducted with the inertial vibrator called vibrodyne which is provided with an eccentric mass m_0 placed at distance r from the axis of rotation. The angular velocity ω can be modified so that the rotational excitation force $F=m_0r\omega^2$ can deform the beam in a significant dynamic regime.

In essence, the reactions R_1 and R_2 will be influenced by both the elasticity of the beam at bending and the action of the force F_0. This study highlights both the inertial dynamic testing when the beam has very high rigidity and the vibrational dynamic testing when the beam has significant elasticity [1-4].

2. EVALUATION OF THE DYNAMIC REACTIONS

For the straight beam simply leaning in Figure 1, it is adopted the model of the constant section bar and the rigidity mode EI with the excitation in the middle so that the inertial force F_0 may be equally divided on the two supports.

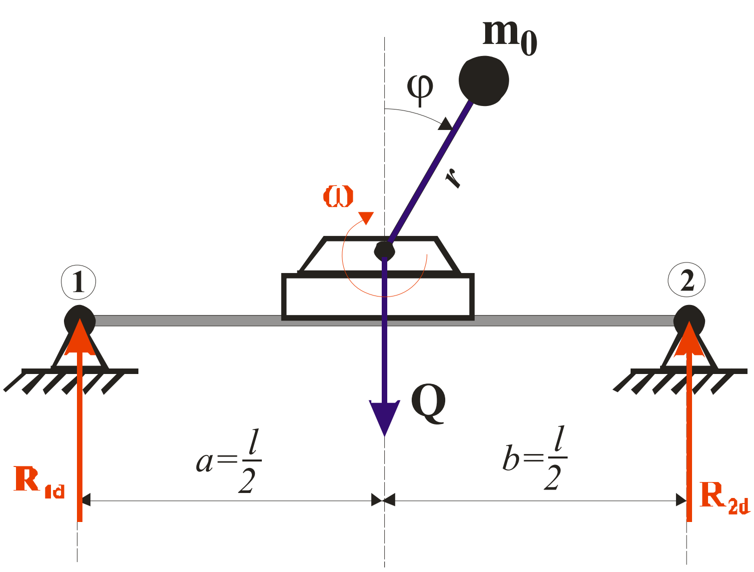

Figure 1. Calculation model

In essence, it is considered the model of the straight bar of length *l*, connected to a rigid and fixed base by joint 1 and the simple support 2. In the middle of the bar it is fixed the inertial vibrator called testing vibrodyne with static moment m_0 r and it own weight Q. Based on this study it is emphasized that the maximum value (amplitude) of the dynamic reactions must be determined on the basis of the following assumptions [5-7]:

a) the bar is absolutely rigid which situation highlights the inertial dynamic load

b) the bar is elastic, which situation highlights the dynamic vibrating load, $E\dot{I}$ is known.

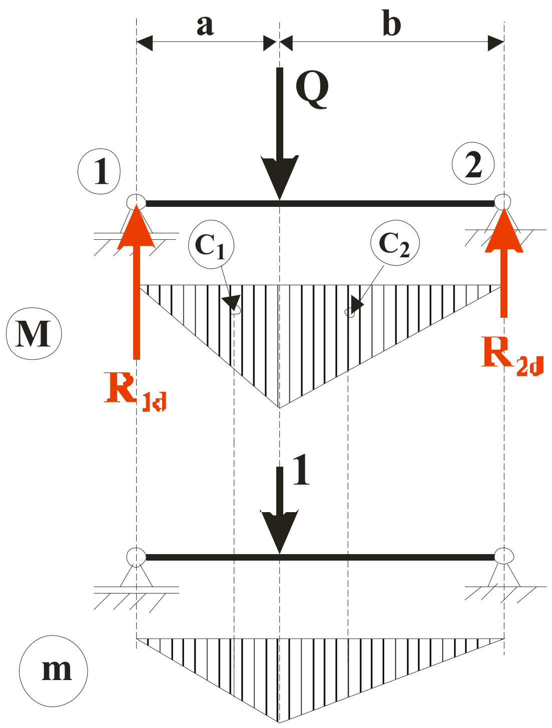

Figure 2. Bending moment distribution

a) In the hypothesis of the rigid bar (Figure 1) we have (for *a*=*b*):

$$\begin{cases} R_1 = R_2 = R_d^i \\ 2R_d^i = m_o r\omega^2 \cos\,\varphi \end{cases} \quad (1)$$

The maximum value for the reaction is

$$R_d^i = \frac{1}{2} m_0 r\omega^2, \quad (2)$$

which corresponds to the inertial dynamic load

b) The elastic bar hypothesis. We determine the static arrow f_s only due to the weight force Q of the vibrator. [6-10]

-the static arrow in the middle of the bar determined with Veresciaghin's rule for the diagram in figure 2 is:

$$f_s = \frac{1}{EI}\left[\frac{1}{2}\cdot\frac{Ql}{4}\cdot\frac{l}{2}\cdot\frac{l}{6}\right]\cdot 2 = \frac{Ql^3}{48EI} \quad (3)$$

- rigidity coefficient of the elastic bar is:

$$k = \frac{Q}{f_s} = \frac{Q}{Ql^3}\cdot 48EI = 48\frac{EI}{l^3} \quad (4)$$

-the eigenpulsation of the bar together with mass $Q/_g$ of the vibrator is:

$$q= \sqrt{\frac{K}{Q}g} = \sqrt{\frac{48EI}{l^3 Q}g} \quad (5)$$

The differential equation of vertical motion (origin in the static equilibrium position) is:

$$\frac{Q}{g}\ddot{x} + kx = m_o r\omega^2 \cos\,\omega t \quad (6a)$$

or

$$\ddot{x} + \frac{k}{Q}gx = \frac{mog}{Q}r\omega^2 \cos\,\omega t \quad (6b)$$

Having the solution in the stabilized regime of the form $x = A\cos\,\omega t\; cu\; \ddot{x} = -A\omega^2 \cos\,\omega t,$ which must check the equation [11-14]:

$$\ddot{x} + p^2 x = \frac{mog}{Q}r\omega^2 \cos\,\omega t \quad (7)$$

Replacing x and $\ddot{x}$ we obtain

$$-A\omega^2 \cos\,\omega t + p^2 A\cos\,\omega t = \frac{mog}{Q}r\omega^2 \cos\,\omega t$$

from where

$$A(p^2-\omega^2) = \frac{m_o g}{Q}r\omega^2 \text{ or } A = \frac{m_o r\omega^2}{p^2-\omega^2}\frac{g}{Q}. \quad (8)$$

The vertical dynamic force $F_d^{\,v}$ (Figure 3) of the beam supports will be given by the dynamic equilibrium equation

$$2F_d^v = \frac{Q}{g}\ddot{x} \text{sau } 2F_d = -\frac{Q}{g}A\omega^2 \cos\,\omega t \quad (9)$$

from where

$$2F_{d_{\max}}^v = -\left|\frac{Q}{g}A_{\omega^2}\right| = \left|\frac{Q}{g}\frac{m_o g r\omega^2}{Q(p^2-\omega^2)}\omega^2\right| \quad (10)$$

and

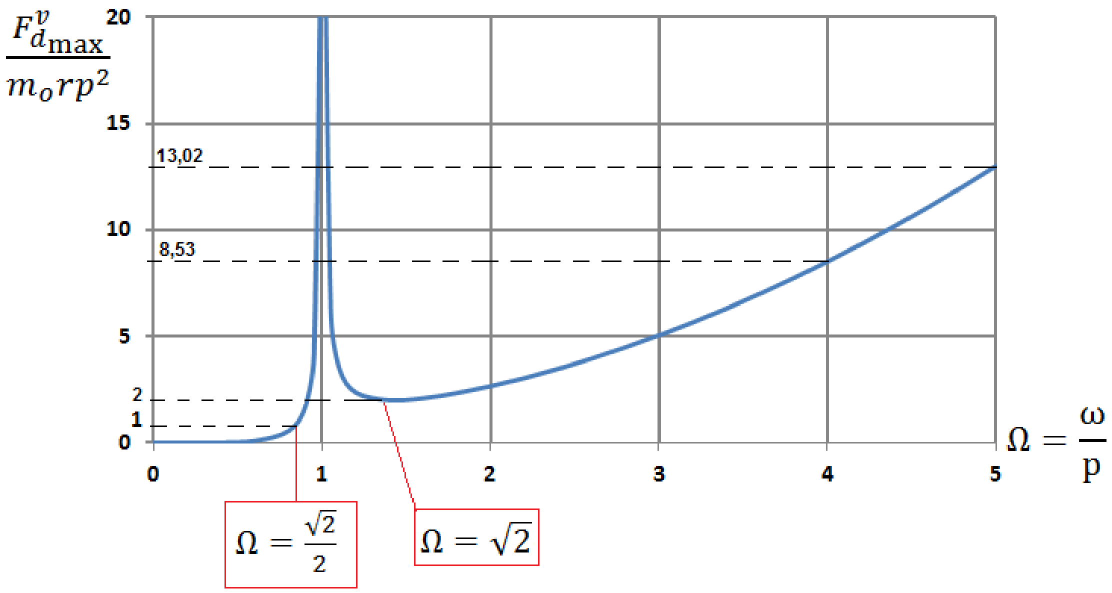

Figure 3. The force on the supports dependent on the pulsation of the excitation

$$F^v_{d\max} = \frac{1}{2}\left|\frac{m_o r\omega^2}{p^2-\omega^2}\omega^2\right| \quad (11a)$$

$$F^v_{d\max} = \frac{1}{2}m_o r p^2\left|\frac{\omega^4}{p^2(p^2-\omega^2)}\right| = \frac{1}{2}m_o r p^2\left|\frac{\frac{\omega^4}{p^4}}{1-\frac{\omega^2}{p^2}}\right|$$

$$F^v_{d\max} = \frac{1}{2}m_o r p^2\frac{\Omega^4}{|1-\Omega^2|} \quad (11b)$$

$$\frac{F^v_{d\max}}{m_o r p^2} = \frac{1}{2}\frac{\Omega^4}{|1-\Omega^2|} \quad (11c)$$

where

$\Omega = \frac{\sqrt{2}}{2}$ is the relative pulsation

It is found that in dynamic regime the dynamic reactions differ depending on the rigid or elastic nature of the bar. Thus, both the dynamic regime and the degree of elasticity of the bar can distinctly generate inertial dynamic loads or vibrating dynamic loads [15-17].

From the relation emerges that in post-resonance rigidity $k=m\omega^2$ is very high, that is for $\omega \rightarrow\infty$ we have $k\rightarrow\infty$, which makes $F^{din}_{\omega\rightarrow\infty} = m_o r\omega^2$ this being precisely the inertial force. In this case the vibratory load can be evaluated based on the dynamic regime [18-21].

3. CONCLUSIONS

When the beam has significant elasticity with appreciable deformations to the dynamic action given by vibrodyne, two dynamic regimes are observed, namely:

a) the ante-resonance regime which for $\omega<p$ makes the reactions dependent on its own pulsation p, that is on the elasticity of the beam.

b) the post-resonance regime shows a significantly increased rigidity with what makes the dynamic force and respectively the reactions depend on the ω pulsation.

In conclusion, for the dynamic testing of the elastic beam, the dynamic testing regimes must be established for the fair assessment of the dynamic reactions at the points of support of the beam.

REFERENCES

[1] Bratu P., Rheological Model of the Neoprene Elements used for Base Isolation Against Seismic Actions, *Materiale Plastice*, no. 3, 2009, pp 288-294.

[2] Bratu P., Analysis of the natural angular frequencies variation upon soil stiffness, during dynamic soil compacting, *Proceedings of the Romanian Academy, Series A*, no. 4, 2010, pp. 380-386.

[3] Bratu P., Dynamic Parameters Optimization for the Vibrating Sieve with Two Granular Matrial Sizing Units, Working in Resonance, *Revista de Chimie*, vol. 62, iss. 8, 2011, pp 832-836.

[4] Bratu P., The Variation of the Natural Frequencies of Road Vibrator-rollers, as a Function of the Parameters of Neoprene Vibration Isolation Elements, *Materiale Plastice*, no. 2, 2011, pp 144-147.

[5] Dobrescu C., Analysis of the dynamic regime of forced vibrations in the dynamic compacting process with vibrating rollers, *ACTA TECHNICA NAPOCENSIS - Applied Mathematics, mechanics and engineering*, vol. 62, no. 1, 2019, pp 71-76.

[6] Bratu P., Hysteretic Loops in Correlation with the maximum dissipated energy, for linear systems, *Symmetry*, vol. 11(3), 2019, doi.org/10.3390/sym11030315.

[7] Bratu P., Dobrescu, C., Dynamic Response of Zener-Modelled Linearly Viscoelastic Systems under Harmonic Excitation, *Symmetry*, vol. 11(8), 2019, doi.org/10.3390/sym11081050;

[8] Bratu P., Buruga A., Chilari O., Ciocodeiu A., Oprea I., Evaluation of the linear viscoelastic force for a dynamic system (m, c, k) excited with a rotating force, *Romanian Journal of Acoustics and Vibration*, vol. 16, no. 1, 2019, pp 39-46.

[9] Bratu P., The innovative impact on acoustics, vibrations and system dynamics, *Romanian Journal of Acoustics and Vibration*, vol. 16, no. 1, 2019, p2.

[10] Bratu P., Dynamic Rigidity of The Linear Voigt - Kelvin Viscoelastic Systems, *Romanian Journal of Acoustics and Vibration*, vol. 16, no. 2, 2019, pp. 184-188.

[11] Drăgan N., A numerical approach of power analysis for the elastical mechanical systems with nonlinear damping, *Romanian Journal of Acoustics and Vibration*, Volume V, no. 1, 2008, pp 17-22.

[12] Drăgan N., Composite materials used for decreasing thenoise level inside the cabins of the mobile technological equipment, *Annals of the Oradea University - Fascicle of Management and Technological Engineering*, vol. IX (XIX), No. 1, 2010, pp 3.73-3.82.

[13] Drăgan N., Nonlinear approach of the systems between the dynamic modelling difficulties and the advantages of using it in mechanical designing, *Romanian Journal of Acoustics and Vibration*, Volume 2, Issue 1, 2010, pp 2.

[14] Drăgan N., Considerations on the nonlinearity grade of the nonlinear mechanical elastic systems, *The Annals of "Dunarea de Jos" University of Galați - Fascicle XIV Mechanical Engineering*, Volume 1 Issue XVIII, 2012, pp 13-18.

[15] Drăgan N., Considerations on the composite neoprene vibration isolators used for the mechanical systems bearings. The dynamics of the non-linear models, *Annals of the Oradea University - Fascicle of Management and Technological Engineering*, vol. XI (XXI), NR2, 2012, pp 4.35-4.41.

[16] Vasile O., Active Vibration Control for Viscoelastic Damping Systems under the Action of Inertial Forces, *Romanian Journal of Acoustics and Vibration*, vol. 14, issue 1, 2017, pp. 54-58.

[17] Bratu P., Vasile O., Spânu G.C., The analysis of insulation systems based on Hooke - Voigt kelvin dynamic rheological model, *Journal of Vibrational Engineering and Technologies*, Vol. 5, issue 3, 2017, pp 255- 261.

[18] Vasile O., Predoi M.V., Dragomir M., Tiron A., Furdui ,H., Vibration isolation analysis of electric motors for essential dynamic regimes, *Journal of Vibrational Engineering and Technologies*, Vol. 5, issue 3, 2017, pp. 239- 245.

[19] Vasile O., Dynamic behaviour analysis of Machines in transient regime on torsional vibration stresses, *Romanian Journal of Acoustics and Vibration*, vol. 10, issue 2, 2013, pp 129-134.

[20] Iancu V., Vasile O., Gillich G.R., Modelling and Characterization of Hybrid Rubber-Based Earthquake Isolation Systems, *Materiale Plastice*, vol. 49, Issue 4, 2012, pp. 237-241.

[21] Bratu P., Vasile O., Assessment of Dissipated Energy to Harmonic Cycles of Displacement for Visco-Elastic Elastomeric Anti-Seismic Insulators, *Romanian Journal of Acoustics and Vibration*, vol. 9, issue 2, 2012, pp. 83-89.