TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics and Mechanics
Vol. 54, Issue IV, 2011

THE OPTIMAL DESIGN OF RUBBER VIBRATION ISOLATORS OF A PLATE COMPACTOR

Radu Mircea MORARIU-GLIGOR

Abstract: *This paper presents a procedure of optimization that can be applied for vibration isolators made of rubber, part of a plate compactor. The dimensioning of rubber vibration isolators is based, first on the calculation, then on choosing of some values or roundness of them, after which the final calculation is made. The proposed dimensioning method removes the drawback of the classic method being also faster.*
Keywords: *optimization, vibration isolators, MathCAD.*

1. INTRODUCTION

It is a fact that the vibrations affect the proper functioning of any machine or installation from the working process. Hence, vibration isolation is required for obtaining the optimal functioning parameters, for reducing the wear of the components as well as for ensuring the safety during the working procedure of the human operator. Starting from the design stage it is necessary to provide a package of technical prescriptions (measures) that ensure that the vibrations regime is kept between the allowable limits imposed by the current regulations.

The 2000/14/EC, 98/37/EC directions (applied in EU countries) the law 90/1996 (republished) and Work safety rules (published in 2002) are establishing the admissible levels for noise and vibrations in case of industrial installations. Likewise, the Romanian standard SR EN 1299 assesses a method for isolating the vibration source.

The vibration isolators from isolation systems are: the metallic springs and elastic elements made of rubber, the last ones offering a better damping, especially for passing through the resonance.

In this paper is presented an optimization method that can be applied for the vibration isolators made of rubber, with round section, submitted to compression (figure 1).

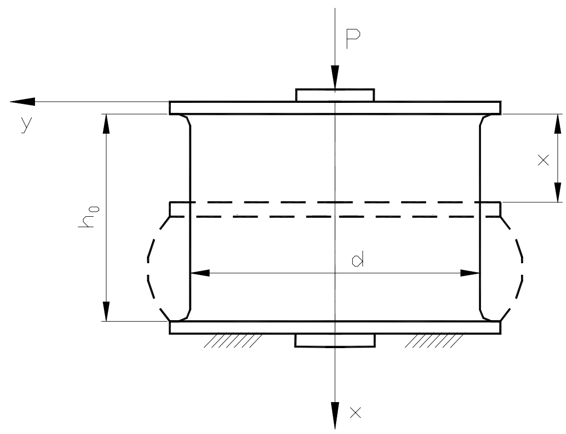

Fig. 1. Vibration isolator submitted to compression

2. ISOLATION SYSTEMS BASED ON RUBBER VIBRATION ISOLATORS

Isolation systems are made of rubber vibration insulators, which can be connected either in series or in parallel so that both, the prescribed geometrical and mechanical functioning conditions are fulfilled. The selection of one isolation system or another must take into account the following aspects: the mounting scheme, the magnitude of the acting forces and first of all, the isolation rate.

The figure 2 presents an assembly which consists of following components: the bed plate of the plate compactor (1) on which are attached (by means of chocks) four vibration isolators (3) that are mounted in the horizontal plane and are subjected to shear. In order to absorb (undertake) the vertical vibrations, was picked a vibration isolator (2) mounted in vertical position relative to the bed plate (1).

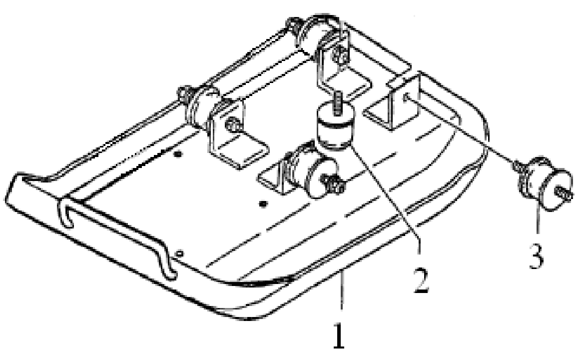

Fig. 2. Isolation system based on four vibration isolator elements with horizontal disposition and one vibration isolator in vertical plane.

3. THE CALCULUS, DIMENSIONING AND TESTING ALGORITHMS

The parameters which are considered in dimensioning and checking of the vibration isolators made of rubber are: the modulus of elasticity (E) and the shear modulus (G), the admissible compression stress (σ) and shearing (τ), the allowable unit strain (ε), the dynamic multiplication coefficient (φ_d), the internal energy loss factor, the mode of operation.

It is considered a vibration isolator made of rubber, of full circular section, height h_0 and diameter *d*.

The input data are: the axial force P, the maximum deformation x_{max}, the hardness of the rubber, the operating conditions.

The dimensioning to compression implies the following steps:

A. The selection of the shape coefficient *Φ* from one of the values: 0,25; 0,50; 0,75; 1,0;
B. The assignment (based on tables) of a value for the modulus of elasticity (E), ;
C. The calculus of the preliminary diameter *d* of the vibration isolator, using the formula:

$$d = \frac{P}{\pi \cdot x_{\max} \cdot E_{st} \cdot \left[\frac{2}{3} \cdot \Phi + 2 \cdot \Phi^3\right]} \tag{3}$$

The obtained value is rounded to integer.

D. The height h_0 of the vibration isolator is determined in the following way:

$$h_0 = \frac{d}{4 \cdot \Phi} \tag{4}$$

E. The admissible strength of the vibration isolator element is verified using one of the following mathematical relations:

$$\sigma^{st} = \left[\frac{4 \cdot x_{\max} \cdot E_{st}}{d}\right] \cdot \left[\left(\frac{2}{3}\right) \cdot \Phi + 2\Phi^3\right] \le \sigma_a^{st} \tag{5}$$

for static loading, respectively:

$$\sigma^{din} = \left[\frac{4 \cdot x_{\max} \cdot \varphi_d \cdot E_{st}}{d}\right] \cdot \left[\left(\frac{2}{3}\right) \cdot \Phi + 2\Phi^3\right] \le \sigma_a^{din} \tag{6}$$

in case of dynamic loading.

F. Check the specific admissible deformation:

$$x^{st} = \frac{P}{\pi \cdot d \cdot E_{st} \cdot \left[\frac{2}{3} \cdot \Phi + 2\Phi^3\right]} \le h_0 \cdot \varepsilon_a \tag{7}$$

in static conditions, or dynamic conditions:

$$x^{din} = \frac{P}{\pi \cdot d \cdot \varphi_d \cdot E_{st} \cdot \left[\frac{2}{3} \cdot \Phi + 2\Phi^3\right]} \le h_0 \cdot \varepsilon_a \tag{8}$$

G. If one of the conditions, mentioned above is not verified, the diameter *d*, the height h_0 or the shape coefficient as well as the type of rubber must be reconsidered and the calculations are done again.

4. THE OPTIMAL DESIGN OF A RUBBER VIBRATION ISOLATOR, USING MATHCAD

For the optimal design of rubber vibration isolators with rounded section, subjected to virtual compression using MathCAD software, the mass of the isolator element was considered the objective function that must be minimized.

The constraints that are imposed refer to check on the admissible strength and admissible unit deformation. A supplementary condition requires that the deformation of the vibration isolator to be less than the maximum deformation.

Hence, the implementation of the problem of the optimal dimensioning in MathCAD is:

Input data: axial force, maximum deformation, rubber strength, working conditions, modulus of elasticity, admissible resistance, admissible unit deformation, shape coefficient, rubber hardness (ρ).

Objective function:

$$M(d) = \rho \cdot \frac{\pi \cdot d^3}{16 \cdot \Phi} \qquad (9)$$

is subjected to the following constrains:

$$\left[\frac{4 \cdot x_{\max} \cdot \varphi_d \cdot E_{st}}{d}\right] \cdot \left[\left(\frac{2}{3}\right) \cdot \Phi + 2\Phi^3\right] \le \sigma_a^{din} \qquad (10)$$

$$\frac{4 \cdot x_{\max} \cdot \Phi}{d} \le \varepsilon_a \qquad (11)$$

$$\frac{P}{\pi \cdot d \cdot E_{st} \cdot \left[\frac{2}{3} + 2 \cdot \Phi^2\right]} \cdot \frac{1}{\Phi} \le x\max \qquad (12)$$

The called function *Minimize* is:

F:= Minimize(M, d)

The returned value of the *Minimize* function represents the optimal value of the diameter *d*.

4. NUMERICAL EXAMPLE

For exemplifying the optimal dimensioning method, it is considered a vibration isolator element that is subjected to an axial force P = 1960 [N], the maximum deformation having the value of 4 mm, the rubber hardness is 60 [°Sh (A)], and the work regime is "dynamic and short termed".

Are also selected: the value for the shape coefficient Φ = 0,50, the modulus of elasticity, E = 5,4 [N/mm^2], admissible strength σ_a = 1,385 [N/mm^2], as well as the admissible unit deformation ε_a = 0,15 (or 15%).

After that the dimensioning calculus was conducted, it resulted a diameter of 49,515 [mm], the determination and testing for all considered elements, this value being rounded to an integer value, that is 50 [mm]. It can be noticed that the obtained element do not verify nor the strength condition neither the admissible unit stress, so, the dimensioning calculus must be performed from the beginning.

The optimal dimensioning, using MathCAD software, assumes making of the objective function and restrictions as previously described.

So, it result an optimum diameter of 54,591 [mm]

By performing a test for the vibration isolator element, using the rounded value of 55 [mm] it resulted that it keeps in with all the constraints.

In Table 1 are presented the dimensional and design characteristics of the vibration isolators:

Table 1.

The dimensional and design characteristics of the vibration isolators

Item	Classical design calculus	Optimization calculus
Preliminary diameter [mm]	49,515	54,591
Selected diameter [mm]	50	55
The height of vibration isolator [mm]	25	27,5
Strength test	**NO**	**YES**
Admissible deformation test	**NO**	**YES**

It is noticed that after performing the dimensioning calculus, it was obtained a diameter that don't verify the admissible strength and compression tests.

Thereafter, the calculus must be performed again by considering higher values for the vibration isolator diameter or modified values of shape coefficient or rubber strength (high strength rubber).

This type of calculations are time consuming and are purely empiric, the designer don't have the certainty that he selected the best variant of vibration isolator element.

5. CONCLUSIONS

Due to elastic properties, characterized by equivalent stiffness coefficient, these elements are determining the working regime as the vibration isolator regime for the components that must be protected against vibrations, especially in case of passing through resonance regime.

The vibration isolator elements are very important in the well functioning of machines and installations. The dimensioning of these elements can became very difficult and may require a lot of calculus.

The author propose a method of dimensioning that bases on the optimization techniques and uses a dedicated soft, the results obtained validate the proposed method.

The optimization process, by introducing some constrains, leads to a solution that satisfies the strength and deformation requirements, by a reduced material consumption, the total time necessary to design the vibration isolator being shortened.

Although, this method presents a great disadvantage: the selection of a large number of input data that implies a problem with a single unknown: the diameter of the vibrating isolator element.

This paper can offer a simple alternative to those who desire to dimension vibration isolators but don't own enough knowledge about other optimization techniques.

6. REFERENCES

[1] Bratu, P. – Sisteme elastice de rezemare pentru maşini şi utilaje, Editura Tehnică, Bucureşti, 1990, 288 pag.

[2] Buzdugan, Gh. – Izolarea antivibratorie a maşinilor, Bucureşti, Editura Academiei, 1980.

[3] Davey, B.A., Payne, R.A. – Rubber in Engineering Practice, New York, 1984.

[4] Gafiţanu, M., ş.a. – Organe de maşini, Volumul I, Editura Tehnică, Bucureşti, 1981.

[5] Lambrescu, I. – Calcul numeric ingineresc – MathCAD, Editura Universității din Ploieşti, Ploieşti, 2004.

[6] Lindley, P. - Engineering design with natural rubber. Tehnical Bulletin The Malayesian Rubber Producers Research Association – MRPRA, 1973.

[7] Maxfield, B. – Essential MathCAD For Engineering, Science and Math, Academic Press, 2009, ISBN: 978-0-12-374783-9.

[8] Ursu-Fischer, N. – Vibrațiile sistemelor mecanice. Teorie şi aplicații, Editura Casa Cărții de Ştiință, Cluj-Napoca, 1998.

PROIECTAREA OPTIMALĂ A ELEMENTELOR ANTIVIBRATILE DIN CAUCIUC DIN COMPONENȚA UNEI PLĂCI COMPACTOARE

Rezumat: Lucrarea prezintă o modalitate de dimensionare optimală a elementelor antivibratile din cauciuc, supuse la compresiune centrică, din componența plăcilor vibratoare. Modalitatea de dimensionare propusă înlătură dezavantajele metodei clasice, care se bazează pe un calcul de predimensionare urmat de alegerea unor valori standardizate sau rotunjirea valorilor parametrilor obținuți.

Radu Mircea MORARIU-GLIGOR, Lector, Technical University of Cluj-Napoca, Department of Mechanical Systems Engineering, Faculty of Machine Building, rmogli70@yahoo.com, Ariesului 102/107, Cluj-Napoca, 0743-120463.