Designing parts for a robust behaviour against vibrations. A new concept

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Abstract. The paper presents a new concept on how to design parts as they have a robust behavior against vibrations, that is to say they can remain stable, even if they were subject to vibrations. The new idea is that the value of the own-frequency must be kept away from the vibration frequency the part or assembly it belongs to is subjected. A way to control the own-frequency of a part is to slightly modify geometrical or material features of the part. The paper displays the way the modification of the shape of the part can modify its own-frequency.

1 Introduction

Vibrations are a wide spread phenomenon in nature and in technique. In both areas, they are either welcome, or on the contrary undesirable. In technique/engineering, in most cases vibrations are to be eliminated. Vibrations are not always dangerous, but often they are annoying or even irritating. Note, that vibration are mostly dangerous if their frequency equals or is near to the own-frequency of the body that is subjected to vibrations.

The most used technique to avoid or eliminate vibrations of a certain part or ensemble is to insulate from the source of vibrations by means of dumpers. Literature displays many scientific papers that deal with the subject in various approaches, as it is briefly described below.

In [1] is presented an “electro-hydraulic actuator that attenuates and isolates ground motion to keep dynamic excitations transmitted to machine tools below permissible levels”; [2] describes an absorber, which has “a linear viscous damper connecting the absorber mass directly to the ground instead of the main mass”; in [3] is presented the “optimum design of damped vibration absorber for viscoelastic bladed disk assemblies”; in [4-7] different type of non-conventional types of damping systems (“with material-inlaid plates”, “a class of piecewise smooth vibration isolation system”, other).

Other scientists propose different means to reduce/avoid vibrations of mechanical systems. E.g. selecting optimally the nut parameters in case of ball screw driven machine tools [8].

The present paper proposes a new approach of the problem: preventing, or at least decreasing effect of vibrations of slim parts, by means of adequate adjusting of geometrical features. None of the cited works refer to such a manner to solve the problem.

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2 The concept

It is known that thin parts have a specific behaviour under the stress of vibration. One of the particularities is that they display big displacements in specific points. This happens only at some certain frequencies of the vibrations. The key point of the concept here proposed is to simulate by means of specific software the behaviour of the part subjected to a large range of frequencies vibrations. In this way the “bad” frequencies, those which cause big displacements of part’s sides, are identified. If any of these frequencies fits to the own-frequency of the part, it becomes a dangerous one.

Adding to the part some specific geometrical features, the “bad” values of the frequencies, as they were defined above, can be pushed to upper or lower values, which do not overlap to the dangerous ones. Of course, the geometry alterations have to be slight, and they may not affect the correct work of the part. In the next sections of the paper will be demonstrated how different specific geometrical features can (sometimes significantly) move the bad frequencies away.

3 Carrying on the researches

Simulation of behaviour under stress of vibration was carried on using COMSOL software [9]. A special designed sample part and some variants of it were used for simulations.

3.1 The sample part and its variants

A thin part was used as standard and some variants of it were designed to study the modifications in terms of frequencies that affect it significantly (figures 1 and 2). Apart from standard, the variants were obtained by adding some geometric features: a holes pattern, a nervure, and holes inside the part. The holes inside the part are an axial one in the median horizontal plane, and a bended one in the same plane. For the last two, in figure 2 a detail and a section through median plane are displayed. These models have no modification in terms of external geometry compared to the standard.

The bended hole in the part aims to emphasize how the dissymmetry of the part affects the way it distorts under the stress of vibrations. It is important to note that even if the external geometry of the part is symmetrical, in terms of mass properties, it is not.

Fig. 1. Standard part and variants with holes pattern and nervure, respectively.

Fig.2. Variants with internal holes, having the same external geometry as the standard.

3.2 Finding the frequencies that affect significantly the parts

Research was carried on by means of simulating the behaviour of the parts with COMSOL software. The mathematical model is described below. All the notations and formula above are according to [9].

$$s^{ij} = s_0^{ij} + C^{ijkl}(\varepsilon_{kl} - \alpha_{kl}(T - T_{ref}) - \varepsilon_{0kl}) \quad (1)$$

Formula for elasticity tensor component:

$$C^{ijkl} = \lambda g^{ij} g^{kl} + \mu(g^{ik} g^{jl} + g^{il} g^{jk}) \quad (2)$$

For the research model equation (3) was used:

$$s = S_0 + C : (\varepsilon - \varepsilon_0 - \varepsilon_{inel}) \quad (3)$$

Where, s - total stress, S_0 - initial stress, C - elasticity tensor

The total strain tensor is written in terms of the displacement gradient (4).

$$\varepsilon = \frac{1}{2}(\nabla u + (\nabla u)^T) \quad (4)$$

The gradient of the displacement, which occurs frequently in the following theory, is always computed with respect to material coordinates in 3D:

$$\nabla u = \begin{bmatrix} \frac{\partial u}{\partial X} & \frac{\partial u}{\partial Y} & \frac{\partial u}{\partial Z} \\ \frac{\partial v}{\partial X} & \frac{\partial v}{\partial Y} & \frac{\partial v}{\partial Z} \\ \frac{\partial w}{\partial X} & \frac{\partial w}{\partial Y} & \frac{\partial w}{\partial Z} \end{bmatrix} \quad (5)$$

$$\sigma \cdot n = F_A \quad (6)$$

$$F_A = \frac{F_{tot}}{A} \quad (7)$$

F_{tot} - the force applied on the Y axis, A - surface area, n - coefficient of deformation, σ - stress.

To perform simulation, some parameters related to part’s material had to be set in the software. They are according to those presented in Table 1.

Table 1. General properties of the used material.

Property	Symbol	Value	Unit
Density	rho	1020	Kg/m^3
Young’s module	E	2300	MPa
Poisson’s ratio	nu	0.24	1

The standard and its four variants were subjected to a continuous range of frequencies from 0 to 120 Hz. For each part, some frequencies caused significant distortions (1 mm) in certain zones of the sample. The values of those frequencies are displayed in Table 2. They are graphically represented in figure 3. It shows that always Variant 2 (that reinforced with a nervure) reacts at lower values of frequency, and Variant 4 (that having a bended hole inside) is affected rather by bigger values of frequency. The way a variant behaves compared to the standard is the same for each “bad” frequency.

Table 2. Frequencies that caused significant distortions to standard and variant parts.

Part name	Standard	Variant 1	Variant 2	Variant 3	Variant 4
Frequencies	22.74	21.42	17.49	23.22	24.42
	28.87	26.52	21.64	28.14	29.88
	61.82	57.38	45.26	60.74	64.36
	88.36	82.23	55.72	92.03	94.86
	103.82	96.69	65.36	102.96	109.67

Fig.3. Groups of “bad” frequencies, and the way they affect different samples.

3.3 Discussion on results of simulation

A very good view of the influence of different frequencies on the samples can be gathered from figures 4 and 5. Figure 4 shows clearly how a certain group of frequencies (those displayed on the third line of Table 3) causes similar distortions on the samples. On contrary in figure 5 one can see how another group of frequencies (displayed on the second line of Table 3) cause different types of distortions on the samples, depending on their particularities/geometrical features. These observations reveal that in design stage, if a part needs to modify its features to be free of danger of vibrations, the modifications must be

carefully chosen, taking into account the frequency of the vibration the part will be subjected to.

Fig.4. Group of frequencies that produce similar distortions on parts, regardless to their features.

Fig. 5. Group of frequencies that produce different type of distortion on parts, depending on their features.

If the geometry of the part is dissymmetric, the distortion is as well, dissymmetric as one can see in figure 6. This means, that if the specific features which modify the shape of the part are clever chosen, a desired effect can be obtained. That is to say that if designer is skilled, he can insulate the side that distorts maximum in an area of the part where it should not affect the correct work of the whole.

Fig. 6. Dissymmetry of the part distributes unevenly the sides with maximum magnitude of distortion.

Note that even a small difference of frequency pushes the areas with maximum magnitude of distortion to opposite sides. This shows that any particular part and its modifications for avoiding the bad influence of vibrations must be judged in a strict correlation with the vibration frequency analysed.

Series of the “bad” frequencies values follow similar patterns of increasing, regardless to the shape of the sample, as one can see in figure 7.

Fig. 7. Patterns of increasing “bad” frequencies.

4 Conclusions

Parts are different stressed by vibrations depending on the frequency they are subjected to. Some frequencies make the part to distort significantly (with big magnitude) – those presented in Table 1 – they were called “bad” frequencies. Other frequencies affect only slightly, or not at all the parts.

The geometry of the part may be modified by adding or removing some features. These modifications should have as effect a better behaviour of the part under the stress of vibrations. Such modifications might not change the external shape of the part, but only the mass distribution of the part. In this way, either the value of a “bad” frequency is pushed to another value that does not affect negatively the work of the part, or moves the side with big magnitude of distortion to another area of the part, as it no more has a negative impact on the way the part works. However, any modification in terms of geometry should be applied, it may not affect the functionality of the part or of the assembly it belongs to.

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