PRACTICAL MODELING OF VIBRATION ISOLATION ON WEAK AND FINITE SUPPORTS

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ABSTRACT

An approach to estimating the vibration isolation of machines on weak and finite supports is discussed in principle. The model should also be capable of predicting the forces acting on the foundation from the measured motion; data often requested by constructors for their loading calculations. The novelty lies not, at least not at the first stage, in the utilized theories but rather in the endeavor of creating a base model. The model can be improved along the way and adapted to various situations, thus providing tenable solutions without resorting to constructions with unnecessarily high safety factors. With sufficient knowledge of the support, both the vibrations and the radiated sound can be predicted, provided accurate input data are obtained.

1. INTRODUCTION

A commonly encountered problem within construction and vehicular applications is the influence of machines on the structure. Practically all machines with moving parts will give rise to disturbances, leading to vibrations and ultimately sound. The disturbances are reduced by the introduction of isolators. In the ideal case, the support on which these isolators rest are strong and not resilient. However, in real situations, one cannot count on such conditions. In order to predict the resultant behavior, a good understanding and model of the structure as well as the whole system are necessary. This necessity has lead to the creation of a model which is discussed in this article.

2. THE MODEL AND ITS PARAMETERS

The model relies on the different mobilities inherent to the system, primarily the point mobilities. In their simplest form this involves, respectively, the mobility of the source (normally modeled as a force excited mass), the isolator (usually a spring) and the support (usually a mass spring system). For an existing support, finding the mobility is readily done through an impact excitation. Adopting the experimental data in the model, the resonant characteristics of the support are captured.

For nonexistent structures the mobility needs to be calculated beforehand. Specifying the boundary conditions and characteristics of the structure, and assuming the mode shapes, the modal masses and eigenfrequencies can be calculated-it turns out that in practice it is often the lowest mode that is of interest since the dispersive characteristics of bending waves bring the higher eigenfrequencies well above the excitation frequency for a large range of machines. The eigenfrequencies are obtainable through regular formulas for simple structures that often work well as approximations. Once the eigenfrequency and modal mass are established, the support can be modeled as a simple mass spring system whose mobility shows a resonance dip. Next, setting up the equations for the vibration isolation is a straightforward task.

The vibration isolation is defined as

$$D_{IL} = 20\log\left|\frac{Y_m + Y_i + Y_{ms}}{Y_m + Y_{ms}}\right| \qquad (1)$$

This is equal to the ratio of the force on the support without and with isolators.

Y_m is the mobility of the moving and excited mass, for instance an engine and a fan,

Y_i is the mobility of the isolators and

Y_{ms} is the mobility of the mass spring system representing the support. However, if the machine standing on the support contains a large part that is not isolated, this mass adds to the modal mass of the support. An example of this is a fan with its rotating part encapsulated in a shielding that could weigh many times more than the fan and the engine. In fact, it is not rare that this mass contribution be of the same size as the modal mass of the support, even more so if unluckily placed.

The benefit of performing the whole calculation, instead of simply studying the resonances and comparing them to the excitation frequencies, is that the influence on the vibration isolation of the weak and finite support can be readily obtained, thus yielding the correct vibration isolation, which is always lower than expected on a support of infinite mass or stiffness.

3. INPUT DATA

Undoubtedly, the toughest part in predicting the resulting forces is obtaining input data from manufacturers; this is usually scarce and normally lacks frequency information. Nevertheless, it is essential to know not only the fundamental tone and its amplitude but also the harmonics, especially if the model is to be employed in a noise and vibration context. The general assumption is that most of the disturbances arise from the imbalance of the fan and that the fundamental tone has the same frequency as the RPM of the fan. What appears not to have been thoroughly investigated is the reaction in the structure due to the blade passing frequency; a strong cause of noise.

4. EXTENDING THE MODEL

4.1. Transmissibility of the energy flow to the support

Upon taking the step of the computing the resulting sound and vibrations, the vibration isolation, D_{IL} , is not sufficient; merely the applied force is not enough to know the vibrations of a structure. Rather, turning to the transmissibility function of the energy flow yields more accurate information since it is a measure of the reduction of energy flow to the support without and with isolators. The formula resembles that of equation (1) but also contains a velocity term in both the nominator and denominator.

4.2. Future improvements on the mobilities

There are various aspects related to the mobilities in this model that can be developed further:

Rubber is an often used material for vibration damping. Due to its complex characteristics standard formulas relying on static stiffness is likely to produce large errors. The more so the higher the frequency since viscoelasticity is proportional to the strain rate. A suitable way to model viscoelasticity is through the concept of fractional derivatives that in the frequency domain becomes handy.

A second rubber feature to be incorporated is the amplitude dependent non-linearity referred to as the Fletcher-Gent effect. This phenomenon proves to be especially important for low frequencies where displacement amplitudes are normally high. At large deformations also geometrical non-linearities might have to be accounted for.

In a foreseeable future smart materials should be on the market even for rather simple applications; extending the model to cover such isolators is a natural step as the demand for such solutions will increase.

When structures are complicated the modal mass and stiffness should be evaluated through FEM calculations.

5. CONCLUSION

Built on general formulas a general model for vibration isolation of machines on weak and finite supports is calculated. The basics are presented and some ideas concerning the assumptions that must be made in all practical cases are discussed. This model will be adapted and improved in the future following the shapes of the projects and could cover a large variety of machines and supports. An experimental work is planned where the model and the assumptions will be verified and tested in different practical situations. Especially the influence of the boundary conditions on the mode shapes and the importance of the transfer mobilities will be studied.