METHODS AND PRACTICAL APPLICATIONS OF VIBRATION ISOLATION

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It is everywhere apparent that we are becoming more and more noise conscious. The employee no longer tolerates excessive noise or vibration; the consumers of household appliances are demanding quieter operation; and the manufacturer is beginning to realize that silence is salable. This paper concerns one method which is applicable to noise and vibration reduction.

In those cases where it is impossible or impractical to reduce the noise or vibration at its source, it is in many instances feasible to isolate these vibrations so that they will not be transmitted to places where they will be disturbing. A common method used in the isolation of machine vibrations has been to introduce some type of resilient support between the machine and the base on which it is mounted. In order to determine in advance the effect of a resilient support, it has been common practice to make use of calculations giving the "force transmissibility" to a rigid base. The meaning of the term "force transmissibility" may be explained by reference to Figure 1.

In fig. 1a is shown diagrammatically a rigid machine of mass m fastened directly to a rigid base. If an internal or external alternating force, $F_1 \cos 2\pi\nu t$ is acting on the machine, the force amplitude transmitted to the base is F_1. In fig. 1b the same machine is separated from the base by a resilient support having a stiffness factor K and zero damping resistance. If the same alternating force is applied to the machine, the force transmitted to the base in this case is no longer F_1 but some different force amplitude F_2. The ratio of F_2 to F_1 is called the "force transmissibility," ε.

The manner in which ε varies with the frequency of the applied force is shown in fig. 2. Along the abscissa is plotted Ω which is the ratio of the applied force frequency ν to the natural frequency ν_0 of the system of fig. 1b. In order to

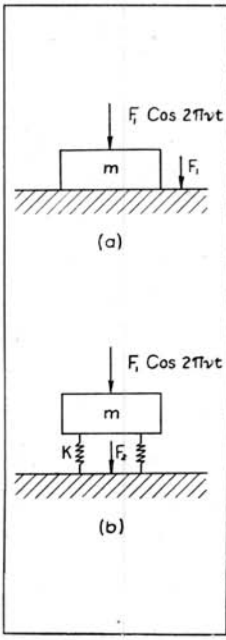

Fig. 1

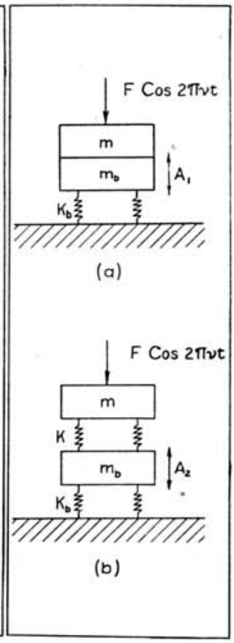

Fig. 3

show the wide range of values of ε most advantageously, $20 \log_{10} \varepsilon$ (the force transmissibility in decibels) has been plotted along the ordinate instead of ε itself. For all applied force frequencies greater than $\sqrt{2}\,\nu_0$, $20 \log_{10} \varepsilon < 0$ or $F_2 < F_1$.

The use of the curve of fig. 2 has in many cases lead to the wrong results because it has been applied to problems where the base on which the machine is mounted is not rigid. In fact if the base were actually immovable there would be no problem of vibration isolation since no machine vibrations could be transmitted to the base regardless of the magnitude of the transmitted force. Thus, in all problems which are of interest in

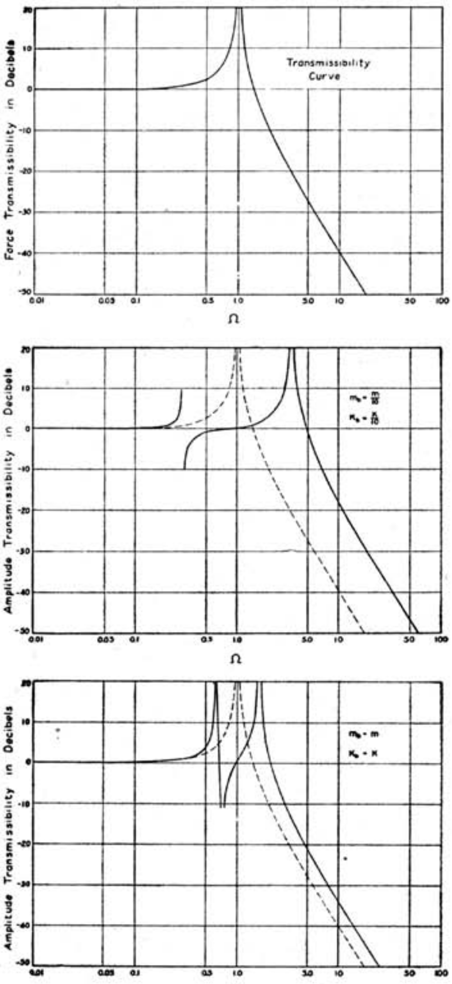

Fig. 2 (top); Fig. 4 (middle); Fig. 5 (bottom)

practice, it is necessary to consider a more complex vibrational system in which the base is free to vibrate.

Fig. 3 shows diagrammatically the case in which the rigid base has been replaced by a movable base of mass m_b and stiffness factor K_b. In Fig. 3a the machine is mounted directly on the movable base and in Fig. 3b it is separated from this base by a resilient support. In this discussion all damping resistances are neglected. If A_1 is the steady state amplitude of motion of the base in Fig. 3a and A_2 the amplitude in Fig. 3b, then the "amplitude transmissibility" ε_a is defined as the ratio of A_2 to A_1 and 20 $\log_{10} \varepsilon_a$ is the amplitude transmissibility in decibels.

The curves of figs. 4 and 5 show how the amplitude transmissibility in decibels varies with Ω which again is ratio of ν to ν_0, the latter being the natural frequency of the system composed of mass m and spring K alone. The force transmissibility curve (the dashed curve) is shown for comparison. The curves in figs. 4 and 5 are for two special cases in which different ratios of m to m_b and K to K_b are used. It is evident for values of m_b of the order of magnitude or less than m that the solid curve differs materially from the dashed curve. However, as the mass of the base, its stiffness factor or both become larger, i.e., as the base becomes more rigid, the solid curve differs from the dashed curve less and less.

Curves similar to those shown in figs. 4 and 5 have been very helpful in the design of resilient supports for vibrating machines mounted on light bases. For example, in a computing machine, mounted directly on a thin metal base which was a part of the case surrounding the machine, the vibrations generated within the machine were being transmitted directly to the base and case. The vibrations of the base and case sent out sound waves which were very disturbing to the operator and to those in the neighborhood of the machine. By the use of a proper resilient mounting, determined by means of the amplitude transmissibility curves, it was possible to reduce the noise sent out by the machine by approximately 10 decibels.

In another case in which the amplitude transmissibility curves have been useful, it was desirous to isolate the low frequency vibrations of a small compressor unit from a flexible wooden floor. The compressor had been mounted on a resilient support which according to the force transmissibility curves should have reduced the force transmitted to the floor. However, the particular resilient support used actually increased the floor vibrations. By the use of a more resilient mounting whose stiffness factor was determined by means of the amplitude transmissibility curves, the floor vibration was reduced to a point where it was no longer objectionable.