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Mechanical Design Handbook for Elastomers

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1.0 INTRODUCTION

It is the intent of this handbook to provide design engineers with a comprehensive guide in a convenient format for design of elastomer dampers for application in rotating machinery. Design data has been collected from an extensive group of references and, together with some recent information, assembled into a single volume. This handbook, which is intended to be a self-contained reference, includes some theoretical discussion where necessary to insure that the reader has an understanding of how and why dampers are used for rotating machines. A step-by-step procedure for the design of elastomer dampers is presented along with detailed examples of actual elastomer damper applications. In addition, numerous references are cited for the interested reader. The authors assume that the reader has a basic understanding of rotor dynamics and machinery vibration.

The design information presented is primarily intended for application in rotating machinery. Dampers of this type have been shown to be effective for controlling rotor vibrations; however, an historic lack of design data and designer experience with elastomer dampers has inhibited the expansion of their applications. A primary motivation for this handbook is to provide such data in a compact and convenient form and to encourage the use of elastomer dampers and elastomeric damping treatments.

A portion of the design data contained in this handbook is relevant to many mechanical design problems and applications of elastomers. For example, the elastomer dynamic property data may be utilized in the design of structural supports for shock and vibration isolation and insulation. Some basic information regarding the effect of structural damping is included in this work. However, specific information concerning the design of such structural members is beyond the scope of this handbook, but an extensive bibliography is provided at the conclusion of each chapter to aid the design engineer in finding a solution to the problem.

The term "elastomer" refers to a large variety of synthesized polymers as well as to natural rubber. These materials are capable of relatively large strains with essentially full recovery. This property of high elasticity (from which the term elastomer is derived) is the result of a particular type of molecular structure. As obtained from Reference 1.1, the principal attributes of this type of molecular structure are:

- The molecules must be very long and flexible, or to be more precise, there must be free rotation about most of the bonds joining together neighboring chain atoms;

- The molecules must be attached together here and there by either permanent chemical bonds or linkages, or by mechanical entanglements, either form of attachment being termed a cross-link;

- Apart from these cross-links, the molecules must be free to move readily past one another. This is to say, the intermolecular attractions which exist between all molecules must be small.

In addition to resiliency, elastomers provide energy dissipation which is the key to their ability to convert mechanical energy into heat energy, thereby damping.

Elastomers may be separated into the two general classifications of natural rubber and synthetic polymers. Particular synthetic polymers may be identified either by their generic type (such as fluorocarbon, chloroprene, etc.) or commonly used manufacturer trademarks [1.2]*. Specific compositions of particular synthetic polymers are further identified by surface hardness.® This surface hardness, which is generally measured in durometer (or Shore A hardness), is an indication of the static modulus of elasticity (or of the static shear modulus) of an elastomer. The specific elastomer types for which dynamic properties are presented in this handbook are polybutadiene, 70 and 90 durometer Viton (fluorocarbon elastomer), Buna-N, EPDM, and neoprene. Physical properties of these and many other types of elastomers are also presented in this handbook. These physical properties include all material properties (e.g., density and thermal conductivity) except for the dynamic moduli and dynamic shape factors which are referred to as dynamic properties. An extensive list of references are given from which additional material properties for a large variety of elastomer types may be obtained. Also, test methods are described in detail by which dynamic properties of additional elastomer materials may be determined.

A number of factors must be taken into consideration in elastomer material selection when designing elastomer dampers for rotating machines. The dynamic properties of an elastomer are a function of the strain, temperature, frequency, and preload to which the elastomer is subjected. In addition, many elastomer materials are sensitive to environmental factors, such as exposure to oil or other fluids, exposure to extreme temperatures, and aging. It is important for the designer to recognize that there are inherent variations in elastomer materials of the same type. In particular, there may be considerable variation among the same generic elastomer materials produced by different manufacturers. Even for a particular manufacturer, the batch-to-batch variation in elastomer material may be significant. These variations can have a significant effect on the dynamic properties of the elastomer.

In general, support flexibility and damping are provided in vibratory systems to reduce the transmission of shock and vibration. Of primary interest in this handbook is the reduction of vibrations through the addition of damping. The ultimate aim of vibration reduction is to reduce the potential of machinery failure and safety problems and to reduce the maintenance requirements of machinery. These goals are applicable to rotor vibration as well as structural vibration.

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* Numbers in brackets refer to references in Section 1.1

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The reduction of the transmission of shock or vibration may be referred to as either isolation or insulation. The term "isolation" is used when the object is to minimize the transmission of vibration from a machine to its environment. The term "insulation" is used when the object is to minimize the vibration transmitted from the environment to a machine. When referring to vibration isolation or insulation, as distinguished from vibration reduction, the primary concern is with the transmission media. That is, the object is not the dissipation of vibration energy but rather the reduction of vibration transmission. The purpose of vibration isolation or insulation is to reduce noise, general discomfort, machinery fatigue problems, associated safety problems and maintenance, and ultimately, to reduce operating costs.

The purpose of this handbook is to facilitate the use of elastomer dampers for applications on rotating machinery. To this end, a brief introductory discussion of elastomer dampers intended for those readers who are unfamiliar with elastomer dampers is presented. Currently, elastomer dampers enjoy broad use for control of unidirectional vibrations [1.3]. This is particularly true in the areas of vibration isolation and insulation. However, elastomer dampers have, unfortunately, seen very limited use in rotating machinery. As the necessary design information and new elastomers become more readily available, the use of elastomer dampers in rotating machinery should become much more common.

Elastomer dampers have a number of advantages over other types of dampers. They are easy to design, manufacture, assemble and maintain and have the general attribute of simplicity. They are also relatively inexpensive due to the requirement for less stringent tolerances and the inherent simplicity. Elastomer dampers are very durable and can easily handle shocks, rough handling, and an occasional moderate overload. If designed properly, maintenance is very simple and inexpensive. Elastomer dampers can tolerate substantial misalignment, which is of particular importance in rotor bearing applications. They also provide both support stiffness and damping and thus do not require parallel stiffness elements as conventional hydraulic mounts (squeeze-film dampers).

Elastomer dampers also have certain disadvantages which should be carefully considered during the design process. The inherent property of elastomer dampers of providing relatively low damping at high frequency is not always desirable. This is particularly true in rotating machines which operate at high frequencies. However, even when this condition exists, the level of damping provided by an elastomer damper may be sufficient to control rotor vibrations. Also, elastomer dampers tend to be somewhat sensitive to changes in ambient temperature (some materials more than others) and are generally useful only in a limited temperature range. Elastomers are also sensitive to other environmental factors, such as fluid compatibility. Environmental considerations can generally be handled successfully by an appropriate choice of elastomer material. One of the most formidable obstacles to designing consistent elastomer dampers is the variation that is generally found in the material composition of a single generic compound. This variation is not only observed between manufacturers, but also between batches from a single manufacturer. The designer should be aware that manufacturers may change the material composition of an elastomer between batches for cost or availability reasons. Such variations in the material composition can affect the dynamic properties.

It is useful to make several observations concerning the comparison of elastomer dampers with more conventional dampers (particularly squeeze-film dampers) for rotating machinery. Using proper design data, the predictions of stiffness and damping for elastomer dampers are just as good, or better, than those for squeeze-film dampers. Elastomer dampers are much less expensive and easier to manufacture and maintain than squeeze-film dampers. Elastomer dampers also provide much more adjustment flexibility during assembly and test procedures than do squeeze-film dampers. In particular, an elastomer damper can be designed in a modular fashion so that the elastomer material or configuration can be easily modified in order to change the damper properties. This is especially useful for prototype machines.

This handbook provides dynamic properties of several types of elastomers particularly suitable for use in rotating machinery dampers. (These dynamic properties have been determined for a range of operating conditions which span most practical rotordynamic applications.) The specific steps required for the design of elastomer dampers are reviewed in detail and include the procedures and justifications for the selection of a particular elastomer material and a particular elastomer geometric configuration, as well as the details involved in the design of the actual damper hardware. In order to provide as complete a reference as possible, a full range of physical properties, such as density and thermal conductivity, are also provided for a large variety of elastomer compositions. A discussion of environmental considerations such as fluid, ozone, and adhesive compatibility is also presented along with specific compatibility data for several elastomer materials. For the designer who finds it necessary to determine the dynamic properties for a particular sample of elastomer material, or for an elastomer material for which the dynamic properties are not available, a thorough discussion of the most common dynamic property test methods is presented. This handbook includes a detailed description of the procedure for performing the base excitation resonant mass test, which, of the tests discussed, is the most versatile and least sensitive to instrumentation error. Specific detailed examples of existing elastomer dampers are discussed, along with the steps encountered in the design process, to provide the reader with a complete picture of the design procedure. The design engineer is encouraged to use these elastomer damper designs as a basis for his own damper design.

1.1 References

1.1 Payne, A. R., and Scott, J. R., Engineering Design with Rubber, Interscience Publishers, Inc., New York, 1960.

1.2 Elastomer Material, Desk-Top Data Book, The International Plastics Selector, Inc., San Diego, California, 1977.

1.3 Göbel, E. F., Rubber Springs Design, John Wiley & Sons, New York, 1974.

2.0 FUNDAMENTALS OF DAMPER DESIGN

An understanding of basic vibration theory is required for the proper design and specification of machinery dampers. For reader convenience, a brief introduction to vibration theory is presented in this chapter. To fully appreciate rotor vibration, an understanding of linear vibration theory is fundamental. Thus, the introduction includes discussions of both unidirectional and rotor-bearing vibration theory. Of course the principal interest here is rotor-bearing system vibrations. A theoretical and historical background is also presented concerning the use of linear damping models, with particular emphasis on rotor-bearing systems. Included is a discussion of currently used types of rotor dampers and a comparison with elastomer dampers.

2.1 Unidirectional Vibration Theory

This section is intended to provide no more than a brief introduction to vibration theory. Many references, such as [2.1]* through [2.3], are available which are much more complete in this regard.

Most real vibratory systems are composed of a continuum with finite mass energy-storage and energy-dissipation characteristics. An ideal vibratory system may be considered to be composed of a number of individual elements, each with the capability to either store potential or kinetic energy or to dissipate energy. The vibration phenomenon is a manifestation of an alternating transfer of kinetic and potential energy. Simple system representations use elements which possess only one of these capabilities. These representations are referred to as lumped parameter models. The number of these elements required to accurately represent a real vibratory system depends on several factors, including the geometry system constraints, traction loads, and the anticipated excitation frequencies. For the purpose of this work, these lumped parameter model elements are assumed to behave linearly. This assumption is a reasonable approximation for most small vibratory motions.

The three fundamental lumped parameter model elements are: the spring, for storing potential energy; the mass, for storing kinetic energy; and the damper, for dissipating energy.

The simplest possible vibratory system representation is a single-degree-of-freedom model for free vibration with no damping as illustrated schematically in Figure 1 where m represents mass and k represents stiffness (or spring constant). The term "free vibration" indicates that the system is provided with a set of initial conditions and then allowed to vibrate freely with no external influences. For the model in Figure 1, a mass is attached to a rigid support by a linear spring which exerts a force proportional to its change in length. This ideal (lumped parameter) spring is assumed massless** so that the forces at the ends of the spring are equal in magnitude and

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* Numbers in brackets indicate references found in Section 2.6.

** See [2.4] for details on the effect of spring mass on vibratory motion.

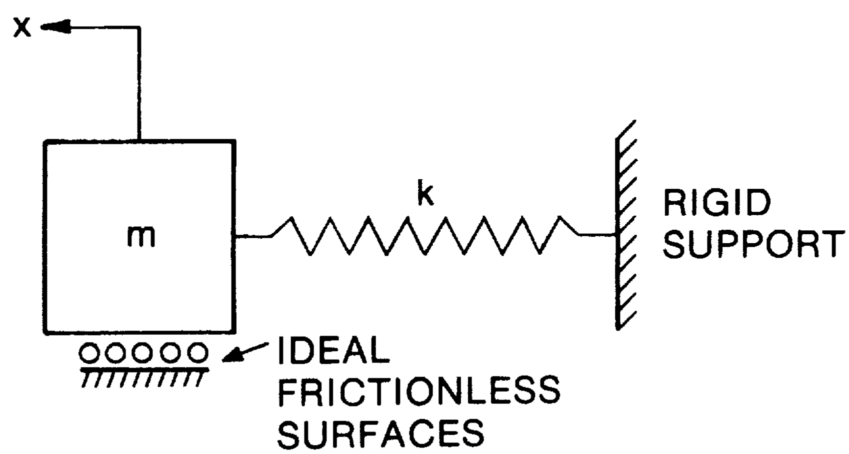

Fig. 1 Single-Degree-of-Freedom Free Vibrational Model With No Damping

opposite in direction. The force exerted by the mass is proprotional to acceleration. The motion of the mass is unidirectional with displacement x, where x is a function of time. The differential governing equation for this system may be obtained through the use of Newton's second law of motion, which may be stated as: the rate of change of momentum $\frac{d}{dt}(m\dot{x})$* is equal to the vector sum of forces exerted on the body and takes place in the direction in which the force (vector sum) acts. If the mass is constant, the rate of change of momentum is equal to the mass times its acceleration ($m\ddot{x}$).

Since the system being considered is unidirectional, this law reduces to

$$m\ddot{x} = \Sigma \text{ (forces in the x-direction)} = -kx \qquad (2\text{-}1)$$

The negative sign indicates that the spring force is a restoring force (resists applied motion). The form of the restoring force indicates that the static equilibrium position is assumed to be at x equals zero.

Rearrangement of equation (2-1) results in the more familiar form of the governing equation given by

$$m\ddot{x} + kx = 0 \qquad (2\text{-}2)$$

The solution to this differential equation is

$$x = A\sin\omega_n t + B\cos\omega_n t \qquad (2\text{-}3)$$

where the constants A and B are determined from the initial conditions of the system and where ω_n, known as the undamped natural frequency or resonance, is given by:

$$\omega_n = \sqrt{k/m} \qquad (2\text{-}4)$$

The units of ω_n are radians per second.

Equation (2-3) may be rewritten in the form

$$X = C\sin(\omega_n t + \theta) \qquad (2\text{-}5)$$

where C is the amplitude of vibration given by:

$$C = \sqrt{A^2 + B^2} \qquad (2\text{-}6)$$

and θ is the phase angle of vibration given by

$$\theta = \arctan(B/A) \qquad (2\text{-}7)$$

The term "phase angle" can have slightly different meanings when used in different contexts. This form of vibration is referred to as simple harmonic motion due to the harmonic (cyclic) form of equations (2-3) and (2-5).

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* Dots over the x represent derivatives with respect to time, i.e., $\dot{x} = \frac{dx}{dt}$

The vibratory system of Figure 1 is made slightly more complicated by the addition of a viscous damping element (the most commonly used model for an energy dissipation element) as shown in Figure 2 where b represents the viscous damping coefficient. The force exerted by a viscous damper is proportional to the relative velocity between the ends of the damper. The ideal damper, like the ideal spring, is assumed massless. Again considering only free vibration, the differential governing equation can be derived from Newton's second law to give

$$m\ddot{x} + b\dot{x} + kx = 0 \tag{2-8}$$

The form of the solution to this differential equation depends on the value of b relative to the values of m and k (assuming all are positive). The criteria for the selection of the appropriate solution are dependent upon the value of the damping coefficient, b, relative to that of the critical damping value, b_c, which is defined by

$$b_c = 2\sqrt{km} = 2\ m\omega_n \tag{2-9}$$

where ω_n is defined in equation (2-4). The ratio of the damping coefficient to the critical damping is referred to as the damping ratio, ζ, such that

$$\zeta = \frac{b}{b_c} \tag{2-10}$$

- For the case in which the damping is less than critical (or underdamped) ($\zeta < 1$), the solution to equation (2-8) is of the form

  $$x = e^{-bt/2m}\ (A\sin\omega_d t + B\cos\omega_d t) = Ce^{-bt/2m}\ \sin\ (\omega_d t + \theta) \tag{2-11}$$

  where A, B, C and θ are constants which depend on the initial conditions, and C and θ are related to A and B by equations (2-6) and (2-7). ω_d is called the damped natural frequency and is given by

  $$\omega_d = \sqrt{k/m}\ (1 - \zeta^2)^{1/2} = \omega_n\ (1 - \zeta^2)^{1/2} \tag{2-12}$$

  and is given in units of radians per second. Note that for unidirectional vibration, the damped natural frequency, ω_d, is less than the undamped natural frequency, ω_n. The vibratory response represented by equation (2-11) is referred to as a transient damped oscillation. A plot of a typical damped oscillation response is presented in Figure 3.

- For the case in which the damping is exactly critical (or critically damped) ($\zeta=1$), the solution to equation (2-8) is of the form

  $$x = (A = Bt)e^{-bt/2m} = (A + Bt)e^{-\omega_n t} \tag{2-13}$$

  where A and B are constants which depend on the initial conditions. For this case, a typical response plot is presented in Figure 4.

- For the case in which the damping is greater than critical (or overdamped) ($\zeta > 1$), the solution to equation (2-8) is of the form

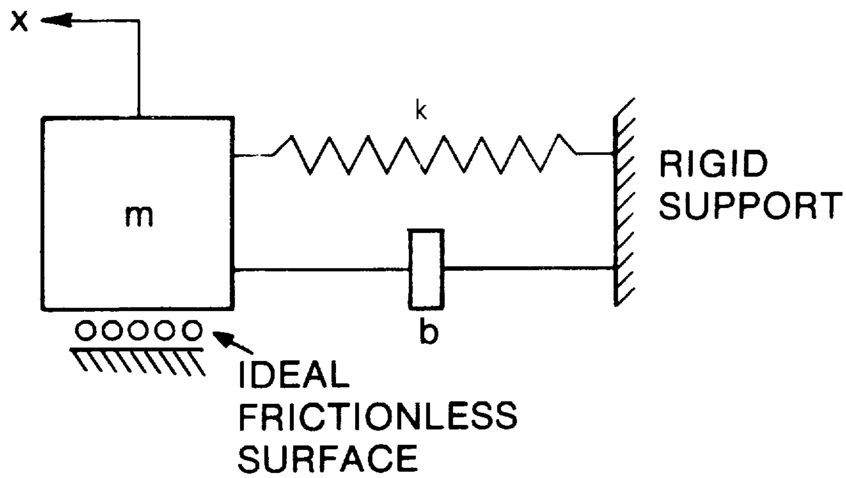

Fig. 2 Single-Degree-of-Freedom Free Vibrational Model With Viscous Damping

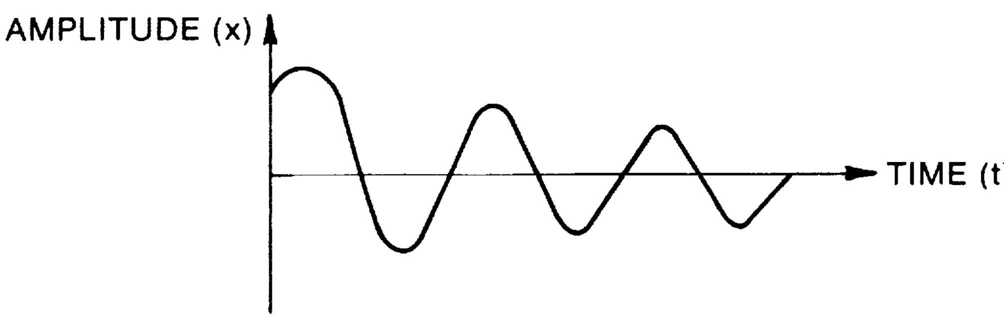

Fig. 3 Typical Damped Oscillation Response for a Single-Degree-of-Freedom Free Vibration System With Viscous Damping; $\rho < 1$ (Undamped)

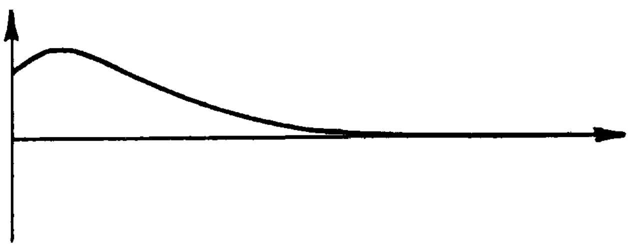

Fig. 4 Typical Response for Single-Degree-of-Freedom Free Vibration System With Viscous Damping; $\rho = 1$ (Critically Damped)