A brief review on determination of viscous, coulomb and particle damping content from the responses of a single degree of freedom system harmonically forced linear oscillator

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KEYWORDS

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ABSTRACT

Mechanical vibration systems with viscous, Coulomb friction and particle damping are of importance in the applications of dynamics and control problems. When a vibrating system is damped with more than one type of models of damping, it is necessary to determine which of these types of damping are more effective to control the resonant response. In such case, it is important to identify damping parameters from the responses of a vibrating system. The problem of identification of content of each type damping in a given system is a major research area. A brief review of some of research papers in the above mentioned area are focused here. Some of techniques which are used to quantify the level of damping in a system are Logarithmic Decrement Method, Hysteresis Loop Method, and Half Power Band-Width Method suggested by most of authors.

1. Introduction

Various forms of passive damping exist, including viscous damping, visco-elastic damping, friction damping, and impact damping etc. Viscous and visco-elastic damping usually have a relatively strong dependence on temperature. Friction dampers, while applicable over wide temperature ranges, may degrade with wear. Mechanical vibration systems with viscous and Coulomb friction are of importance in the applications of dynamics and control problems. Coulomb damping and viscous damping are the two most important sources of energy dissipation in mechanical systems. It is important in practice to examine the dynamic characteristics of a servo-mechanism and a machine tool slide-way considering the influence of the friction. [1]

Two common classes of friction dampers have been considered in past studies. The first is viscous friction in which the frictional force is proportional with velocity of mass or "seismic element". It is still frequently assumed that the response of an actual vibration instrument is identical with the response of an ideal instrument in which damping is entirely "Viscous". However, the actual response may differ significantly from ideal response if the damping varies with other than the first power of the velocity, or is dependent on displacement, or if any appreciable amount of Coulomb damping is present. This last type of damping, sometimes typically known as Dry friction used to describe simple "Coulomb friction". Coulomb friction model assumes that the friction force is constant in magnitude and is directed so as to oppose relative movement of two surfaces in contact.

The amplitude of friction force is proportional to the normal connecting force with a factor μ defined as the coefficient of friction.

Particle damping technology is a derivative of impact damping with several advantages. Particle dampers significantly reduce the noise and impact forces generated by an impact damper and are less sensitive to changes in the cavity dimensions or excitation amplitude. [2] The advantages of impact dampers are that these

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dampers are inexpensive, simple designs that provide effective damping performance over a range of accelerations and frequencies. In addition, impact dampers are robust and can operate in environments that are too harsh for other traditional damping methods. Vibration damping with impact dampers has been used in a wide variety of applications including vibration attenuation of cutting tools, turbine blades, television aerials, structures, plates, tubing, and shafts. [3]

Particle Vibration Damping (PVD for the short) is a combination of impact damping and friction damping. In a PVD, metal or ceramic particles or powders of small size (0.05 to 5 mm in diameter) are placed inside cavities within or attached to the vibrating structure. Metal particles of high density such as lead or tungsten give high damping performance due to dissipation of kinetic energy. Particle Vibration Damping involves the potential energy absorptions and dissipation through momentum exchange between moving particles and vibrating walls, friction impact restitution. [4] Particle Impact Damping (PID for the short) is a means for achieving high structural damping by the use of a particle-filled enclosure attached to the structure in a region of high displacements. The particles absorb kinetic energy of the structure and convert it into heat through in elastic collisions between the particles and the enclosure, and amongst the particles.

In many situations, it is important to identify damping information from a vibration system with Coulomb, Viscous and particle damping sources of damping. Their frequent occurrence in practical engineering has aroused for a long time the interest of many researchers in the vibration field. Friction dampers (with viscous damping as the system damping) are used in gas turbine engines, high speed turbo pumps, large flexible space structures under carriage of railway bogie, vehicle suspension systems etc. These dampers are used to reduce resonant stresses by providing sliding contact between points experiencing relative motion due to vibration, thereby dissipating resonant vibration energy.

2. Literature Survey

Den Hartog [5] has presented an exact solution for the steady-state vibration of a harmonically excited oscillator damped by combined dry and viscous friction. The system, as shown in figure (1),

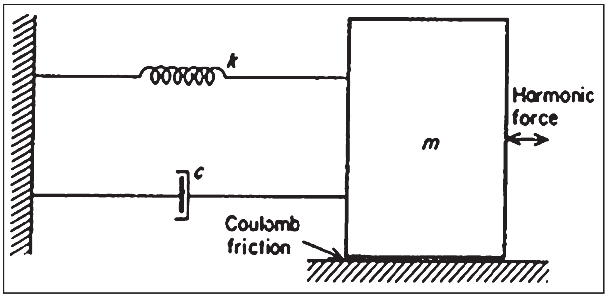

Fig. 1. Schematic representation of the system, analyzed by Den Hartog [5].

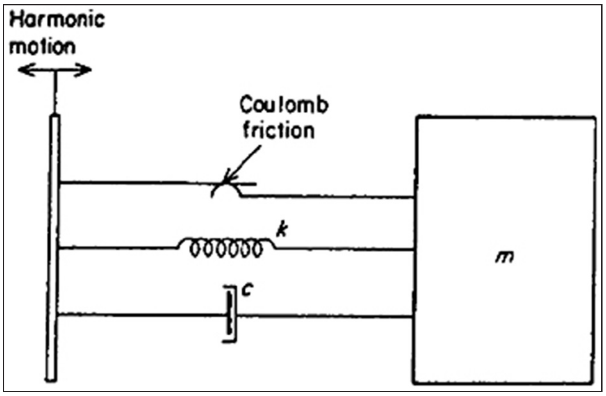

Fig. 2. System with Coulomb friction between mass and ground, with harmonic excitation of the base analyzed by Hundal [6].

consists of a forced excited mass with friction forces acting between it and the ground. The several experimental tests to verify solutions have been performed to find out the forced response of a single-degree-of-freedom system with both viscous and dry friction damping. The results of analysis included the analytical solutions of periodic non-stop and stick-slip motions, and illustrations of the frequency response curves for different values of viscous damping, dry-friction damping have been presented.

Hundal [6] studied a base-excitation frictional oscillator as shown in figure (2), in which close form analytical solutions of the equation of motion were obtained. Results have been presented in non dimensional form as magnification factors versus frequency ratios as functions of Viscous and Coulomb friction parameters. It has been shown that the mass motion may be continuous or one stop during each cycle, depending upon system parameters. The response of a single degree of freedom spring-mass system with Viscous and Coulomb friction, with harmonic base excitation, has been determined.

Levitan [7] analyzed the motion of a system with harmonic displacement of the base, as shown in figure (3). The friction forces in his model act between the base and the mass. An analytical solution for the response of the support-excited system has been presented. The solution to the equation of motion has been developed through the application of a Fourier series to represent the frictional force opposing the relative motion between mass and supporting structure. It has been assumed that no stop occur during any portion of the steady state oscillation. The results have been presented in a form of convenient for observing the influence of system parameters.

Perls and Sherrard [8] have extended the results of Den Hartog through the ranges applicable to inertial instruments as accelerometers and jerkmeters. They obtained the curves with analog computers for the magnification factor verses frequency ratio of second order systems with combined Coulomb and Viscous damping. The figure (4) shows a typical vibration instrument with its frame rigidly attached to a sinusoidally vibrating structure having a motion $X \cos \omega t$.

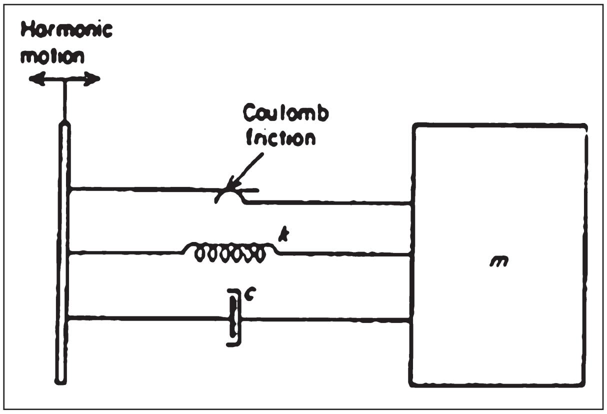

Fig. 3. Schematic representation of the system, analyzed by Levitan [7].

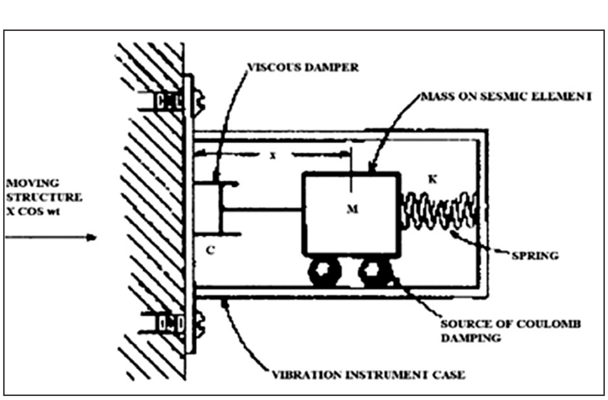

Fig. 4. Schematic representation of the system, analyzed by Perls and Sherrard [8].

Ferri and Dowell [9] have investigated the vibration response of both single and multi degree-of-freedom systems with combined dry friction and viscous damping. Jacobsen and Ayre [19] have developed an approximate scheme for estimating both viscous and dry friction quantities from the free-vibration decrements by noting that the viscous friction dominates in the large-amplitude responses, and that Coulomb friction dominates in the small-amplitude oscillations. As such, they exploited the exponential and linear decay of a free vibration Viscous or Coulomb-friction damped system. The methods above, however, rely on sufficient excursion magnitudes of the free-vibration response. If enough damping is present, such responses may not be possible. To this end, it makes sense to develop methods for identifying friction parameters in forced oscillators. Tomlinson and Hibbert [10] have carried out further studies, including modal parameter identification via frequency response functions by using the power dissipation to estimate Coulomb and hysteretic damping coefficients. The effect of coulomb friction on the Kennedy and Pancu vector plot of a single degree-of-freedom system has been analyzed by using the method of harmonic balance. It has been shown that the resulting diagram no longer conforms to a locus of a circle in the resonant region, which restricts the usual methods of analysis. A technique, based upon the in-phase and quadrature power dissipated when exciting a normal mode, has been presented which allows the magnitude of the non-linear friction force and the hysteretic damping constant to be evaluated. The technique has been also applied to systems having several degrees-of-freedom and it has been shown that it is possible to identify the characteristics of a single non-linear coulomb device situated within a structure, but in the case of more than one device, the technique has some restrictions. The theoretical results have been compared with experimental data from a structure containing a non-linear coulomb device.

Tomlinson [11] has used the distortions in the complex receptance plots to estimate damping parameters. An analysis of the vector plot responses of lightly damped single degree-of-freedom systems with Coulomb damping has been made. The vector plots, as derived by using both an exact and an approximate method (the method of harmonic balance) have been compared and it has been shown that the distortion of the normally circular vector locus has been due to the Coulomb damping. Although the vector plots of such systems have been distorted it has been also shown that the frequency gradient criterion has been still applicable for location of a natural frequency even when the frictional force levels approach the excitation force levels. To permit estimation of the modal damping of these systems a criterion by means of which the limits of the useful frequency range can be specified has been suggested. The criterion, which has been based upon the quadrature input power necessary to excite the mode of vibration, is found to be equivalent to that obtained from the half power point theory when applied to linear systems.

Chen and Tomlinson [12] have proposed estimating damping parameters in nonlinear oscillators by using the displacement, velocity and acceleration output and formulating the output in terms of series of frequency response functions. A new type of time series model, the AVD model which accommodates the Acceleration, Velocity and Displacement simultaneously, has been used to estimate the characteristics of nonlinear single and multi-degree of freedom systems with dry Coulomb friction, Viscous damping and nonlinear stiffness. Numerical results via simulation are compared with those from a Fourier transform method, which has been suggested that the AVD model is a powerful technique for nonlinear system parameter identification.

Iourtchenko et al. [13] have applied a harmonic balance analysis based on the technique of Dimentberg [20] to generate identification equations. A non-linearly damped Single-Degree-Of-Freedom (S.D.O.F. for the short) system under broadband random excitation is considered. A procedure for in-service identification of the damping characteristic from measured stationary response is described. The procedure has been based on the stochastic averaging method. The explicit analytical solution has been obtained for the integral equation, which relates the desired damping characteristics to the apparent force in the shortened equation for the slowly varying response amplitude, and thus to the measured probability density of the amplitude. The approach has been of a non-parametric nature, which makes it convenient for testing hypotheses of damping mechanisms from measured random vibration data. Extensive results of numerical tests for the procedure have been presented.

Marui and Kato [14] have worked out a brief analytical technique for the behavior of the

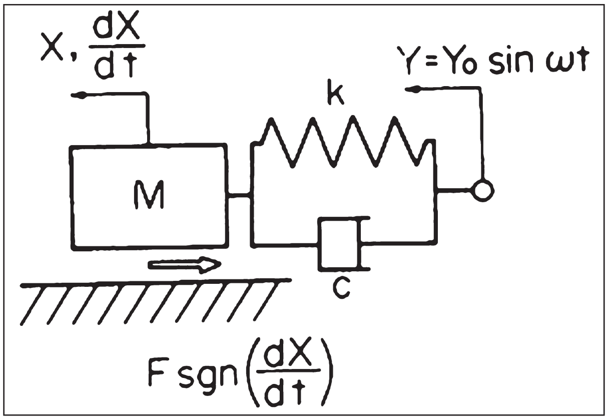

Fig. 5. Forced vibratory system with coulomb friction [14].

linear forced vibratory system under the influence of a Coulomb friction force, as shown in figure (5). The analysis has been based on the new simple idea of stopping region. Using this technique, the behavior of the system in the low exciting frequency range, where the remarkable influence of friction easily develops, has been examined and the results have been compared with the experimental ones. The behavior of the system has been completely determined by the three non-dimensional parameters of non-dimensional frictional force, frequency ratio and damping ratio.

Stanway et al. [21] have proposed a nonlinear least-squares estimator scheme which involves the online solutions to determine the parameters of viscous damping and Coulomb from the harmonic response of a vibrant system. Yao et al. [18] have obtained the Coulomb and viscous friction parameters by using a recursive nonlinear least-squares algorithm. Liang and Feeny [22] have proposed a simple identification algorithm for estimating both Viscous and Dry friction in harmonically forced single degree of freedom mechanical vibration systems. The method has been especially suitable for the identification of systems for which the traditional free-vibration scheme is difficult to implement. Numerical simulations have been included to show the effectiveness of the proposed algorithm. A numerical perturbation study has been also included for in sight on the robustness of the algorithm.

Liang and Feeny [1] have presented a method for estimating Coulomb and Viscous friction coefficients from responses of a harmonically excited dual-damped oscillator with linear stiffness. The identification method has been based on existing analytical solutions of non-sticking responses excited near resonance. The method has been applicable if the damping ratio of viscous component can be considered small. The Coulomb and Viscous friction parameters can be extracted from two or more input-output amplitude pairs at resonance. The method has been also tested numerically and experimentally. Experimental results have been cross checked with estimations from free-vibration decrements and also from friction measurements. A schematic diagram depicting a SDOF oscillator with viscous, Coulomb friction and base excitation has been shown in figure (6).

Cheng and Zu [16] have studied a mass-spring oscillator damped with both Coulomb and Viscous friction and subjected to two harmonic excitations with different frequencies. By employing an analytical approach, closed form solutions for steady state response have been derived for both non-stop and one-stop motion. From numerical simulations, it has been found that near the resonance, the dynamic response due to the two-frequency excitation demonstrates characteristics significantly different

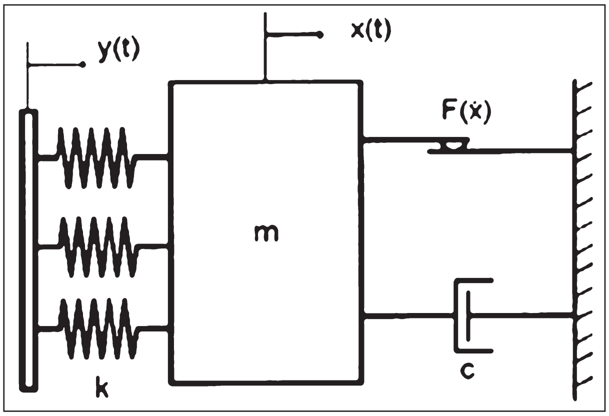

Fig. 6. Schematic diagram depicting a SDOF oscillator with viscous, Coulomb friction and base excitation [1].

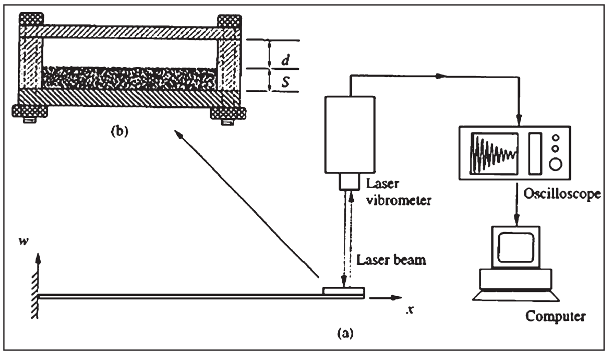

Fig. 7. (a) Schematic of the experimental arrangement (b) A magnified view of the particles inside the enclosure [17].

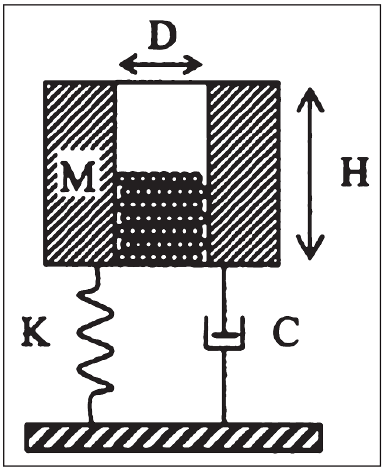

Fig. 8. The model of a single degree of freedom system of a particle vibration damper [18].

from those due to a single frequency excitation. In addition, the one-stop motion has been demonstrated peculiar characteristics, different from those in the non-stop motion.

Friend and Kinra [17] have studied PID measured for a cantilevered aluminum beam with the damping enclosure attached to its free end. Lead particles have been used in the study. PID is a form of impact damping in which particles of various shapes, sizes, and materials are inserted into an enclosure attached to a structure. The unique aspect of PID is that high damping has been achieved by absorbing the kinetic energy of the structure as opposed to the more traditional methods of damping where the elastic strain energy stored in the structure is converted to heat. The effect of acceleration amplitude and clearance inside the enclosure on PID has been presented. PID has been found to be highly non-linear. A schematic of the test set-up and a magnified view of the particles inside the enclosure are shown in figure (7).

Mao et al., [18] have examined the characteristics of Particle damping with respect to a simple single-mass impact damper and a dry-friction damper. The analysis of the damping characteristics of particle vibration dampers based on the 3D discrete element method has been carried out. The model of a single degree of freedom system of a particle vibration damper is as shown in figure (8).

An efficient procedure for the dynamics of a large number of particles without excessive computing complexity has been developed. A particle vibration damper and the particle contacts are as shown in figure (9).

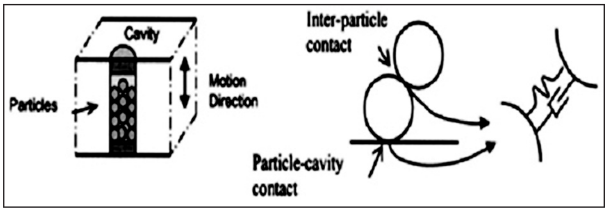

Fig. 9. A particle vibration damper (left) and the particle contacts (right) [18].

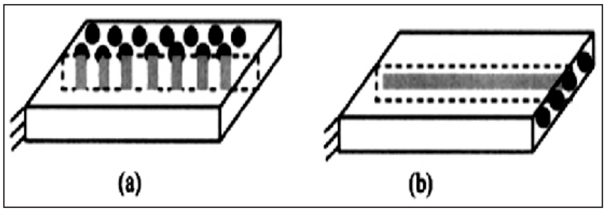

Fig. 10. Schematic of a beam with transverse particle dampers (a) or longitudinal particle dampers (b). [18].

Particle damping is a derivative of single-mass impact damper that has been thoroughly studied on the influence of mass ratio, particle size, particle/slot clearance, excitation levels, and direction of excitation. Figure (10) shows two common orientations of the container holes for a beam structure. The distribution and arrangement of the multiple particle dampers on the structure usually have a significant impact on the damping effect of the dampers.

3. Discussion

It is seen that the identification and estimation of the numerous forms of damping present in a dynamic system is helpful in improving the control of resonant response. When a vibrating system is damped with more than one type of models of damping, it is necessary to determine which of these types of damping are more effective to control the resonant response. In such case, it is important to identify damping parameters from the responses of a vibrating system. Therefore when a system is damped due to Coulomb friction, viscous friction and Particle damping, it is necessary to develop theoretical & Experimental methods for identification of these damping parameters from the responses of the vibrating system.

4. Conclusion

A comprehensive review of determination of damping content in spring mass system presented in the past decade has been attempted. It can be seen that the area of damping content identification is active and continuously evolving. Newer techniques are emerging as further in sight is gained in the fields of vibrations, simulation and condition monitoring. The future points to a flexible, multidisciplinary and robust identification methodology for different types damping parameters in forced oscillator system. This will go a long way to increasing the overall reliability and safety of dynamic control systems areas in general. The methodologies have some specific advantages over the other. Now, there is a choice before the dynamics people to adopt proper techniques. Some of techniques which are used to quantify the level of damping in a system are Logarithmic Decrement Method, Hysteresis Loop Method, Half Power Band-Width Method suggested by most of authors.

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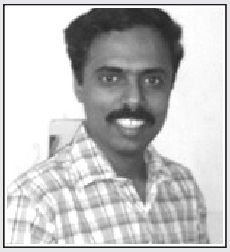

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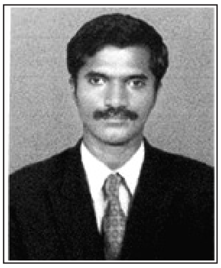

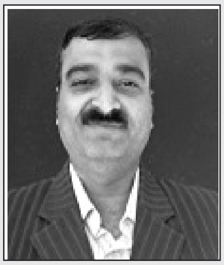

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