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RANDOM VIBRATION OF HOUDAILLE SHOCK ABSORBER

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Abstract *Random vibrations are extensively used in transportation, wind and earthquake. Exact solutions for random excitations are very limited, particulary when the material behavoir is hysteretic, having a multi-value forced deformation pattern with non conservative energy dissipation. The various techniques available for the analysis of nonlinear systems subjected to random excitations are briefly introduced and an overview of the progress which has been made in this area of research is presented. Formulation of the equivalent linearization has been used to analyze system using differential mathematical models [2,3,12] with approximed solutions. In equivalent linearization method the governing set of nonlinear differential equations are replaced by an equivalent set of linear equations, and the difference between the sets being minimized in some appropriate sense. The results obtained from the method are validated by simulation results.*

Keywords: *Dynamic equilibrium, random vibration, the power spectral density, linear equation*

1. INTRODUCTION

The environment may be transformed into an equivalent spectral density of forced as a function of frequency. The upper frequency limit beyond which such forces are unimportant has not been defined. The spectral density of the forces is continuous, the forces are not those which might be produced by the sum of finite number of vibration source. A Houdaille damper [1,4,5] is used I rotating devices such as engine crankshaft where absorption is needed over a wide range of speeds, The damper is inside a casing attached to the end of the shaft. The casing contains a viscous fluid and a mass that is free to rotate in the casing.

2 SYSTEM MODEL

Unawarded viscous vibration absorber (damper Houdaille) is used in some internal combustion engines to limit the vibration amplitudes torsion over a wide speed range. It is constructed of a rigid disc that can freely rotatable in the cylindrical cavity [4,5] filled with a viscous fluid. The engines this vehicle is located at the end of the crankshaft in wheel drives fan belt. The crankshaft is modeled simplified as a bar cantilever stiffness K. Torsional damper attached to the free end has a housing with time mass inertia J in which a disc can rotate freely with the moment of inertia J_m mass, upon which a pair of damping proportional to speed the relative angle between the housing and the disc.
Using the principle of d'Alembert [1,2,7,8], if the damper is called a couple harmonic exterior random equations of motion can be written

$$\begin{pmatrix} J & 0 \\ 0 & J_m \end{pmatrix}\begin{pmatrix} \ddot{-}_e \\ \ddot{-}_m \end{pmatrix}+\begin{pmatrix} 2®«\left(\dfrac{bR_2^3}{h_2}+\dfrac{R_2^4-R_1^4}{2h_1}\right) & -2®«\left(\dfrac{bR_2^3}{h_2}+\dfrac{R_2^4-R_1^4}{2h_1}\right) \\ -2®«\left(\dfrac{bR_2^3}{h_2}+\dfrac{R_2^4-R_1^4}{2h_1}\right) & 2®«\left(\dfrac{bR_2^3}{h_2}+\dfrac{R_2^4-R_1^4}{2h_1}\right) \end{pmatrix}\begin{pmatrix} \dot{-}_e \\ \dot{-}_d \end{pmatrix}+ \quad (1)$$

$$+\begin{pmatrix} K & 0 \\ 0 & 0 \end{pmatrix}\begin{pmatrix} -_e \\ -_d \end{pmatrix}+\sim\begin{pmatrix} K & 0 \\ 0 & 0 \end{pmatrix}\begin{pmatrix} -_1^3 \\ -_2^3 \end{pmatrix}=\begin{pmatrix} M(t) \\ 0 \end{pmatrix}$$

The nonlinear factor $\sim$ controls the type and degree of nonlinearity [3,9] in the system.

The linear equation [3,7,8] can be write

$$\begin{pmatrix} J & 0 \\ 0 & J_m \end{pmatrix}\begin{pmatrix} \ddot{¯}_e \\ \ddot{¯}_m \end{pmatrix}+\begin{pmatrix} 2®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right) & -2®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right) \\ -2®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right) & 2®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right) \end{pmatrix}\begin{pmatrix} \dot{¯}_e \\ \dot{¯}_d \end{pmatrix}+~\begin{pmatrix} K_{ech} & 0 \\ 0 & 0 \end{pmatrix}\begin{pmatrix} ¯_e \\ ¯_d \end{pmatrix}=\begin{pmatrix} M(t) \\ 0 \end{pmatrix} \quad \textbf{(2)}$$

where $¯_e$ is the rotation of the housing and $¯_d$ is turning the interior. The coefficient of damping has the expression (Harris and Crede, 1968)

$$c=2®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right) \quad \textbf{(3)}$$

The difference between the nonlinear stiffness and linear stiffness terms [3,6,7] is

$$e=K¯_e(t)+~K¯_e^3(t)-~K_{ech}¯_e(t)\ . \quad \textbf{(4)}$$

The value of K_{ech} can be obtained by minimizing [2,3,7,8]] the expectation of the square error

$$\frac{dE\{e^2\}}{dK_{ech}}=0\ , \quad \textbf{(5)}$$

Neglecting very small terms we get

$$K_{ech}=K(1+3~±_{¯_1}^2)\ , \quad \textbf{(6)}$$

Using the Fourier transform of equation [1,2,4] and having the relations

$$F(\ddot{¯}(t))=-µ^2\bar{¯}(µ)\ ,\ F(M(t))=\bar{M}(µ) \quad \textbf{(7)}$$

obtain for the response [6,7,8]

$$\bar{¯}_1(µ)=\frac{JJ_mµ^2-2iµ(J+J_m)®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)}{D}\bar{M}(µ),\quad \bar{¯}_2(µ)=\frac{JJ_mµ^2-6µ®«i\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)J}{D}\bar{M}(µ) \quad \textbf{(8)}$$

where

$$D=JJ_mµ^4-µ^2J_mK_{ech}+4®^2«^2µ^2\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)^2+2iµK_{ech}®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right). \quad \textbf{(9)}$$

The frequency response function [5,6] of the system is give by equation

$$\bar{H}_1(µ)=\frac{JJ_mµ^2-2iµ(J+J_m)®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)}{D} \quad \textbf{(10)}$$

and

$$\bar{H}_2(µ)=\frac{JJ_mµ^2-6µ®«i\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)J}{D}\ . \quad \textbf{(11)}$$

The mean square value for the displacement [1,2,3] of the system is given by equation

$$±_{¯_1}^2=R_{¯_1}(0)=\int_{-\infty}^{\infty}\left|H_1(µ)\right|^2J^2S'_0dµ=S'_0\int_{-\infty}^{\infty}\frac{\left(JJ_mµ^2\right)^2+\left[2µ(J+J_m)®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)\right]^2}{m^2+n^2}dµ \quad \textbf{(12)}$$

and for the second structure

$$±_{¯_2}^2=R_{¯_2}(0)=\int_{-\infty}^{\infty}\left|H_2(µ)\right|^2S'_0dµ=S'_0\int_{-\infty}^{\infty}\frac{\left(JJ_mµ^2\right)^2+\left[6µ®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)\right]^2}{m^2+n^2}dµ\ , \quad \textbf{(13)}$$

where

$$m=JJ_mµ^4-µ^2\left[J_mK_e+12®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)^2\right],\ n=2µ®«\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)K_{ech} \quad \textbf{(14)}$$

The power spectral density of response [3,5,6,7] for the the housing in $rad^2 \cdot s$ is given by equation

$$S_i(\mu)=\left|H_i(\mu)\right|^2 S_M=\left|H_i(\mu)\right|^2 JJ_m S_0^{'}(\mu)=\left|\frac{1}{J}H_i(\mu)\right|^2 JJ_m S_0^{'}(\mu)=\left|\bar{H}_i(\mu)\right|^2 S_0^{'}(\mu),\ i=1,2. \quad , \tag{15}$$

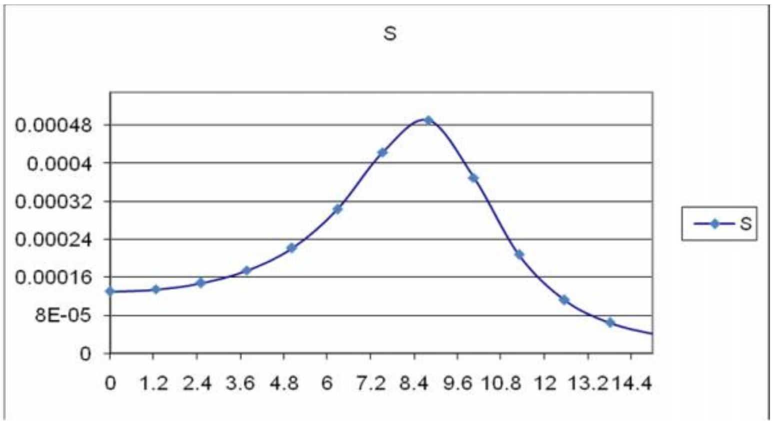
So, the power spectral density of response [3,11,12] will be

$$S_1(\mu)=\frac{\left(JJ_m\mu^2\right)^2+\left[2\mu(J+J_m)\text{®«}\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)\right]^2}{m^2+n^2}S_0^{'}(\mu), \tag{16}$$

$$S_2(\mu)=\frac{\left(JJ_m\mu^2\right)^2+\left[6\mu\text{®«}\left(\frac{bR_2^3}{h_2}+\frac{R_2^4-R_1^4}{2h_1}\right)\right]^2}{m^2+n^2}S_0^{'}(\mu). \tag{17}$$

3. NUMERICAL RESULTS

As a numerical example, considered
$J=27kg\cdot m^2$, $K=7\cdot10^6\ N\cdot m/rad, h_1=0,005m, h_2=0,005m\ R_1=0,2m, R_2=0,4m,\ S_M=1\,N^2\cdot s, b=0,10m$, the damping factor $\text{¶}=0,25$, with the nonlinear factor to control the type and degree of nonlinearity $\sim=20m^{-2}$, and $S_0^{'}=0,52\frac{rad^2}{s^3}$, wich means that the power spectral density of excitation $S_F=m^2 S_0^{'}=2,08\cdot10^{10}N^2\cdot s$.



$\frac{\mu}{2\text{®}}$

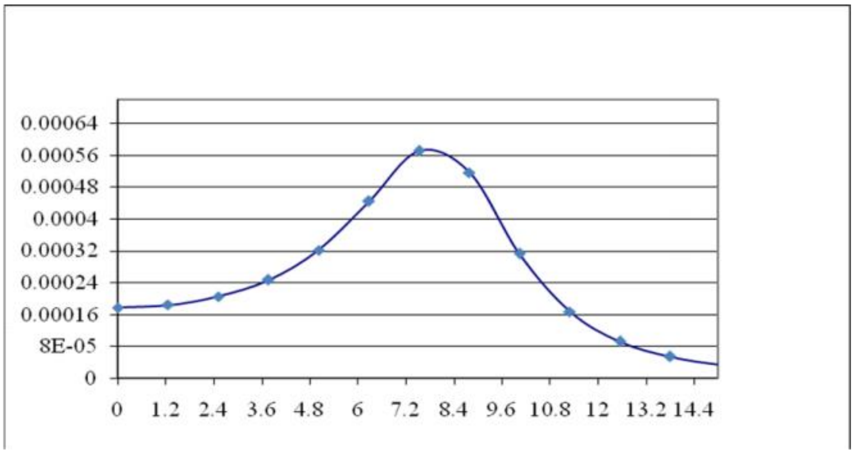
Figure1: The power spectral density of the response $S_1(\mu)\ [rad^2\cdot s]$.

The value of K_{1e} and K_{2e} are

$$K_{1e}=7,315\cdot10^6\ N\cdot m/rad,\ K_{2e}=7,572\cdot10^6\ N\cdot m/rad \tag{18}$$

The power spectral density of the response for the first and second structure, in this case, is

$$S_1(\mu)=\frac{\left(JJ_m\mu^2\right)^2+\left[\mu(J+J_m)c\right]^2}{m^2+n^2}S_0^{'}(\mu),[m^2\cdot s],\ S_2(\mu)=\frac{\left(JJ_m\mu^2\right)^2+9\mu^2c^2}{m^2+n^2}S_0^{'}(\mu)\ [m^2\cdot s] \tag{19}$$



$\frac{\mu}{2®}$

Figure2: The power spectral density of the response $S_2(\mu)\ [rad^2 \cdot s]$

4. CONCLUSION

The method has been proposed for determinate the power spectral response is based on considering the non-linear system to behave as a linear system having varyng natural frequency. In this paper, an iterative method of statistic linearization (IMSL) is presented to solve non-linear stochastic vibration equations. This method represents an improvement over the classical linearization method. The method uses the solution of the corresponding linear vibration equation as an initial value in an iterative procedure.

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