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DESIGN AND DEVELOPMENT OF AN ECCENTRIC TYPE CENTRIFUGAL PENDULUM LINEAR VIBRATION ABSORBER

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Abstract: The use of centrifugal pendulum for dynamic vibration absorber design is a proven method for reducing undesired torsional vibrations in rotating systems. These devices are in use for many years, most commonly in light aircraft engines, helicopter blade rotors etc. These devices have also been reported for reduction of rectilinear vibrations. Pendulum type dynamic vibration absorbers use impact forces for effective reduction of rectilinear vibrations describing modelling method and transient state analysis in view of spring impact absorbers, floating impact absorbers and pendulum impact absorbers. Bond graph technique for modelling of single degree of freedom (SDOF) vibrating system excited by rotation unbalance at the sprung mass using a pendulum type dynamic vibration absorber is reported in the recent literature. This paper deals with the modelling and design procedure for a centrifugal pendulum type dynamic vibration absorber (CPVA) subjected to base excitation. It presents a detailed analysis and experimental investigations of the effect of various parameters affecting the motion transmissibility of the sprung mass such as size of the pendulum mass, eccentricity of the pendulum pivot with respect to axis of rotation of the pendulum assembly, mass ratio (ratio of the pendulum mass to the sprung mass), gear ratio (the ratio of the pendulum rotational frequency to the excitation frequency) and the frequency ratio (the ratio of the excitation frequency to the natural frequency of the SDOF system). It has been proved that the CPVA is effective in reduction of motion transmissibility of the sprung mass of the SDOF system with the proper selection of the affecting parameters.

Keywords: Centrifugal Pendulum Vibration Absorber (CPVA), Motion transmissibility, single degree of freedom SDOF, Mass ratio, Gear ratio, Frequency ratio.

I. INTRODUCTION

Vibration is an integral part in the life of human being. Many human activities involve vibration in one form or other. Though vibration can be utilized profitably in several consumer and industrial applications, it's transmission to human being results in discomfort and loss of efficiency. Control of undesired vibration continues to be a practical concern in many system designs. Several techniques are utilized to either limit or alter the vibration response of such systems. A vibration control system, either as an isolator or an absorber, is said to be active or passive, depending on the external power required for the system to perform its function. A passive vibration control unit consists of a resilient member (stiffness) and an energy dissipater (damper) to absorb vibratory energy. Active modes of vibration control use external power to perform its function and is particularly helpful when there is variation or uncertainty in disturbance frequencies and/or magnitudes. CPVA is such a device which falls in the category of active vibration control techniques. The use of CPVA is a proven method for reducing torsional vibrations in rotating systems. They are used to suppress torsional vibrations in rotating and reciprocating machinery and are widely employed in camshafts driven by internal combustion engines, light aircraft engines, reciprocating pumps, helicopter rotors, high-performance automotive racing engines and other places where torsional vibrations at a given order are troublesome.

II. RELATED WORK

This work is motivated by such a problem, with constraints on the solution requiring a compact, relatively low- mass actuation scheme that would be effective in minimizing low frequency linear vibrations. The need for a practical and simple actuation scheme satisfying these constraints led to a detailed study of a novel vibration suppression device categorized in the literature as Centrifugal Pendulum Vibration Absorber. It is a novel device because here it is used for controlling rectilinear or translational vibrations. It is one of the simplest controllable absorbers, which can be mounted

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on an unbalanced machine to reduce the translational vibrations of the primary system. One of the beneficial capabilities of this vibration suppression device is its ability to tune into changes in the dominant disturbance frequency. It has a wide range, although it is not equipped with a special damping device. Its automatic tuning to the excitation or disturbing frequency is performed by a simple mechanical connection [2].

Aside from this benefit, this device is generally more efficient for a given mass ratio (μ=absorber mass/primary system mass). In an active vibration control scheme, the rotational speed of the hub, ω_d, would be adjusted to "track" the dominant frequency of the disturbances. This speed directly determines the characteristics of inertial neutralizing forces induced by the pendular motion when the system is subjected to disturbance power via translational motion. In this way, vibration suppression is made possible by reflection of disturbances back to their source. This device, in particular, gives promising results in minimizing low frequency vibrations.

III. METHODOLOGY

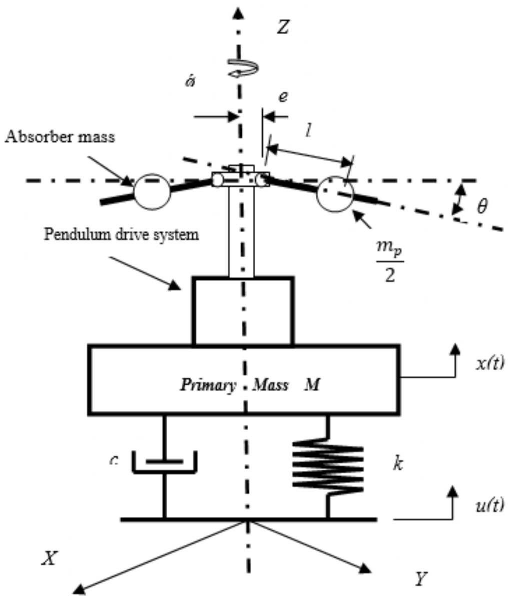

Figure 1: Schematic representation of the CPVA.

Figure 1 shows a schematic representation of a SDOF system associated with a pendulum type dynamic vibration absorber. It shows a SDOF vibrating system which may be a simple model of any real physical system subjected to base excitation *u(t)* resulting sprung mass displacement *x(t)*. Sprung mass *M* (the total mass i.e., sprung mass in addition to the mass of pendulum assembly rotating drive system) is supported by suspension spring having spring rate *k* and system damping having coefficient of viscous damping *c*.

The pendulum assembly consists of a pair of absorbing masses (*mp/2*) each suspended from the top of the vertical rotor free to rotate about Z- axis. A speed-controlled dc motor drives the vertical shaft carrying pendulum assembly through at a controlled angular speed $\dot{\phi}$ such that $\dot{\phi} = \eta\, \omega$ where η is the gear ratio. The system has a mass ratio μ i.e., ratio of absorber mass (*mp*) to the main mass *M*. The pivots supporting the pendulums have an eccentricity *e* with respect to the Z-axis. Pendulum masses (*mp /2*) each are fixed diametrically opposite at a distance *l* from the center of the pivot in each case. The masses (*mp /2*), eccentricity *e* and length *l* are variables.

There are two possible states i) $\dot{\phi} = 0$ (the vibration absorber is ineffective) and ii) $\dot{\phi} \neq 0$ (the vibration absorber is effective). The system is analysed for tuning the angular rotational speed of the pendulum assembly so that motion transmissibility of the sprung mass is controlled in the neighbourhood of natural frequency of the system.

Equations of motion have been developed as follows,

1) At mass *M*, the equation for the force balance is as under,

$$M\ddot{x} + 2\left(\frac{m_p}{2}\right)(x + l\sin\theta)^{\ddot{}} = -c(\dot{x} - \dot{u}) - k(x - u)$$

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$$\text{Let } \frac{m_p}{M} = \mu \ , \ \frac{k}{M} = \omega_n^2 \ \& \ \frac{c}{M} = 2\zeta\omega_n$$

Where θ is the angle made by the pendulum with respect to the horizontal during pendulum rotation. For small values of θ, $sin\ \theta \sim \theta,\ cos\ \theta \sim 1$

$$\{(1+\mu)D^2 + 2\zeta\omega_n D + \omega_n^2\}x + \mu l D^2\theta = (2\zeta\omega_n D + \omega_n^2)u \tag{1}$$

2) At the pivot 'O', the equation for torque balance is as under,

$$2\left[\frac{I}{2} + \frac{m_p}{2}l^2\right]\ddot{\theta} = -2\left(\frac{m_p}{2}\right)\dot{\phi}^2(e + l\cos\theta)l\sin\theta - 2\left(\frac{m_p}{2}\right)\ddot{x}l\cos\theta + 2\left(\frac{m_p}{2}\right)gl\cos\theta - Tfr \tag{2}$$

where *Tfr* is frictional torque developed during pendulum oscillation process. Assuming *Tfr = 0.*

$$\theta = \frac{m_p g l - m_p l D^2 x}{(I + m_p l^2)D^2 + m_p\dot{\phi}^2(e+l)l} \tag{3}$$

Substituting equation (3) in equation (1) and simplifying we obtain,

$$[b_4D^4 + b_3D^3 + b_2D^2 + b_1D^1 + b_0D^0]x = [a_3D^3 + a_2D^2 + a_1D^1 + a_0D^0]u \tag{4}$$

Where
$b_4 = [(1+\mu)(I + m_pl^2) - \mu m_pl^2]$, $b_3 = [2\zeta\omega_n(I + m_pl^2)]$, $b_2 = [\omega_n^2(I + m_pl^2) + (1+\mu)m_p\dot{\phi}^2(e+l)l]$, $b_1 = [2\zeta\omega_n m_p\dot{\phi}^2(e+l)l]$, $b_0 = [m_p\omega_n^2\dot{\phi}^2(e+l)l]$ and $a_3 = [2\zeta\omega_n(I + m_pl^2)]$, $a_2 = [\omega_n^2(I + m_pl^2)]$, $a_1 = [2\zeta\omega_n m_p\dot{\phi}^2(e+l)l]$, $a_0 = [m_p\omega_n^2\dot{\phi}^2(e+l)l]$

Assuming harmonic solution of the type *x(t) = Xe^iωt* & *u(t) = Ue^iωt* and following the regular procedure for solution,

$$M_t = \frac{X}{U} = \frac{[-a_2\omega^2 + a_0] + i[-a_3\omega^3 + a_1\omega]}{[b_4\,\omega^4 - b_2\,\omega^2 + b_0] + i[-b_3\omega^3 + b_1\omega]}$$

The motion transmissibility *Mt* is expressed as,

$$|M_t|^2 = \frac{[-a_2\omega^2 + a_0]^2 + [-a_3\omega^3 + a_1\omega]^2}{[b_4\,\omega^4 - b_2\,\omega^2 + b_0]^2 + [-b_3\omega^3 + b_1\omega]^2} \tag{5}$$

Setting the numerator of equation (5) equal to zero we obtain,

$$\dot{\phi} = \omega\sqrt{\frac{(I + m_pl^2)}{m_p(e+l)l}} \tag{6}$$

Equation (6) provides a tuning rule for nullifying the motion transmissibility *Mt*. Equation (7) represents the simplified form of equation (5).

$$M_t = \sqrt{\frac{(F-A)^2(1 + 4\zeta^2\lambda^2)}{\left\{\left[\lambda^2(AB - \mu m_pl^2 - BF) + (F-A)\right]^2 + [2\zeta\lambda(F-A)]^2\right\}}} \tag{7}$$

$$\text{Where, } \lambda = \frac{\omega}{\omega_n},\ F = m_p\eta^2(e+l)l,\ \ A = (I + m_pl^2),\ \ B = (1+\mu)$$

From equation (7) it can be concluded that in the parametric analysis, the parameters affecting the motion transmissibility *Mt* are system damping ratio ζ, frequency ratio λ gear ratio η, eccentricity e, length l and mass ratio μ.

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IV. EXPERIMENTAL RESULTS

Figure 2 shows the experimental set-up designed, developed and used for experimental investigation of effectiveness of the developed CPVA for a SDOF vibrating system subjected to harmonic base excitation. The sprung mass *M* (1) consisting of the mass of the pendulum, pendulum drive D.C. motor (7) and the sliding base (3) supported on two helical springs (2). The suspension springs (2) are supported on a sliding base assembly (4) and the entire system is excited with harmonic excitation at its base (4) by an electro-magnetic vibration exciter (5). The CPVA consisting of a pair of two pendulums with masses *mp/2* (6) each is attached to the shaft of pendulum drive D.C. motor (7). The entire system is mounted on a massive concrete foundation (8). The measurement of experimental values of motion transmissibility *(Mt)* was carried out with the help of FFT analyser (Multi-purpose data acquisition system) for various values of frequency ratio (λ) in the neighbourhood of natural frequency of the sprung mass system.

Figure 2: Experimental Set-up.

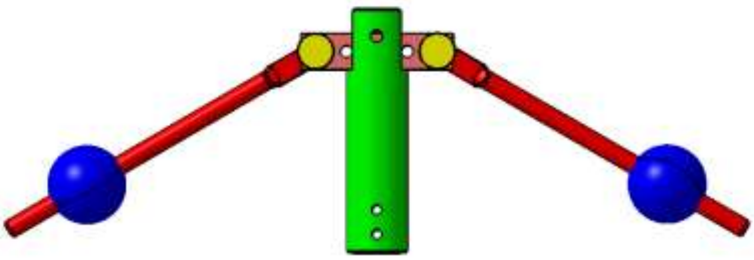

Figure 3: Exaggerated view of eccentric centrifugal pendulum arrangement.

Experimental investigations

Table I and Table II show the details of the SDOF vibrating system subjected to harmonic excitation at the base and of pendulum system respectively.

Table I: Details of SDOF system under investigation.

Sr. No.	Parameter	Magnitude
1	Sprung mass *(M)* in kg.	2
2	Damping ratio (ζ) of the primary system (determined by experimental half power point method)	0.068
3	Effective spring rate of suspension springs *k* used in the primary system in N/m	2855.38
4	Undamped frequency of the SDOF system $\omega_n = \sqrt{k/M}$ in Hz.	6

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Table II: Details of Pendulum mass under investigation for eccentricity *e* = 20 mm.

Set No.	Radius (r) of each spherical ball (mm)	Mass of each spherical ball i.e. (m_p/2) in kg	Mass ratio (μ)
I	10	0.035	0.035
II	13	0.07	0.07

Verification of tuning relation $\dot{\phi}$

Table III shows the experimental values of *Mt* for observing the effectiveness of the developed CPVA for variation in excitation frequency ratio λ in the neighbourhood of resonance frequency (0.833, 1.00, 1.166 Hz) and the corresponding pendulum assembly rotational speed $\dot{\phi}$ in rpm for theoretical and experimental comparison purpose when eccentricity e and radius r are 20 mm and 10 mm respectively.

Table III: CPVA with *r* = 10 mm & *e* = 20 mm.

Frequency ratio (λ)	Pendulum length (mm)	M_t CPVA Off	M_t CPVA On	Theoretical speed (rpm)	Experimental speed (rpm)
0.833	50	1.475	1.224	255	245
	70	1.780	1.244	266	250
	90	1.250	1.024	272	256
1.000	50	4.083	1.413	307	315
	70	3.148	0.754	319	325
	90	5.230	1.370	326	323
1.166	50	0.866	0.750	358	375
	70	1.160	1.003	372	395
	90	1.304	0.939	381	402

Table IV shows the experimental values of M_t for observing the effectiveness of the developed CPVA for variation in excitation frequency ratio λ in the neighborhood of resonance frequency (0.833, 1.00, 1.166 Hz) and the corresponding pendulum assembly rotational speed $\dot{\phi}$ in rpm for theoretical and experimental comparison purpose when eccentricity *e* and radius *r* are 20 mm and 13 mm respectively.

Table IV: CPVA with *r* = 13 mm & *e* = 20 mm

Frequency ratio (λ)	Pendulum length (mm)	M_t CPVA Off	M_t CPVA On	Theoretical speed (rpm)	Experimental speed (rpm)
0.833	50	1.222	0.895	257	238
	70	1.197	1.012	266	242
	90	1.279	0.882	272	262
1.000	50	5.523	1.205	308	315
	70	5.444	1.054	320	310
	90	4.842	0.740	327	315
1.166	50	1.163	0.652	360	365
	70	1.347	0.710	373	385
	90	1.221	0.657	381	395

V. CONCLUSION

The objective of this paper is to present modelling, design and development of a centrifugal pendulum type dynamic vibration absorber (CPVA) to suppress rectilinear vibrations of a SDOF vibrating system subjected to base excitation. The expression for motion transmissibility (Mt) has been derived in terms of the pendulum parameters such as size of the pendulum mass, eccentricity of the pendulum pivot with respect to the axis of rotation of the pendulum assembly, mass ratio, gear ratio and the frequency ratio. The expression for the rotational speed of the pendula (i.e. tuning speed

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of the pendulum) has also been derived. This tuning relation matched with the relation found by the research scholars earlier by different methods.

The expression for (Mt) has been analysed to study the effect of pendulum parameters on motion transmissibility (Mt). This analysis is quite useful in the component selection and in the design of CPVA system. Beside this it also proves that for a given excitation frequency, the pendulum parameters have great effect in suppression of vibrations. Experimental investigations are further carried out in the neighbourhood of the natural frequency of the sprung mass for the proof of theoretical analysis. Some assumptions are made while deriving the expression for (Mt) such as the friction in pendulum pivot is assumed to be zero which may not be exactly valid in actual practical conditions. It is seen that the vibration suppression of the sprung mass system is strongly dependent on the tuning speed of the pendula and even little variation in its speed may not give the desired results.

While performing the experimentation work the speed of the dc motor is controlled manually which is bit complicated task therefore the speed of the motor is adjusted in such a manner that the values for motion transmissibility (Mt) of sprung mass system are reduced in a range of 60 to 80% as compared to CPVA off conditions. The actual rotational speed of the pendula is found to be in a close agreement range of theoretical values and a substantial reduction of motion transmissibility (Mt) of the sprung mass system is found to be quite impressive especially in the neighbourhood of resonance conditions with the proper selection of the affecting parameters.

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BIOGRAPHY

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