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Centrifugal Pendulum Dynamic Vibration Absorber for Helicopter Blades

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Abstract. A large part of the engineering problems in rotating machines are related to vibration. This shortcoming causes discomfort, wear and fatigue, leading to structural failures if they are not properly controlled. The main techniques used to reduce the amplitudes of vibration include the installation of Dynamic Vibration Absorbers (DVAs) which vary roughly from passive, semi-passive, semi-active and active, differing mainly in two aspects: *i)* frequency range where it is effective and *ii)* external energy consumption. In rotating machines such as helicopter blades, a type of DVA historically used is the Centrifugal Pendulum DVA (CPDVA), the use of this passive DVA is justified due to the frequency of rotation of the helicopter rotor remains almost constant during its operation and for the complexity of installing an active system on the blade. In this work, we will design a numerically passive CPDVA. The blade and the absorber are modelled as beams using Finite Element Method (FEM) and validated with analytical previous studies. Results show a good agreement with reality and theoretical expected behaviour, the CPDVA attended the function it was designed.

Keywords: Centrifugal Pendulum Dynamic Vibration Absorber, Finite Element Method, Passive vibration control.

1 Introduction

Beam vibration is an important and interesting topic. Rotating beam-like structures are widely used in various engineering applications, such as compressors, wind turbines and turbo-machinery [1]. Continuous structural beam systems are also widely used in many engineering fields, such as structural engineering, transportation engineering, mechanical engineering, and aerospace engineering. In the last one, rotating beams serve as important mechanical components such as helicopter blades. Vibrations in helicopters are undesirable and cause failure to the structure and discomfort to the people.

According to [2], a significant amount of vibration in helicopters is due to the strong aeroelastic interactions between the highly unsteady aerodynamic environment and rapidly rotating flexible blades. One of the advantages of the main rotor blades of a helicopter is its constant rotational speed, so that, the source of mechanical vibrations is known. Conventional helicopters produce their lift and propulsion with a main rotor. Main rotor and tail rotor are driven by a turbine with constant speed while power transmission is conducted by a constant transmission gearbox. Using constant rotor speeds during operation requires a less complex blade design for this single design point [3].

Equations of motion of cantilever blades mounted on a disk rotating at a constant speed have been considered by various authors such as [4], and [5]. When a beam is excited by a steady-state harmonic load, it vibrates at the same frequency as the frequency of the applied harmonic load. When the applied loading frequency equals to one of the natural frequencies of the system, large oscillation occurs, which can cause large beam deflection. In order to contour this problem, helicopter vibration reduction can be achieved by passive structural changes and/or active control strategies. The passive ones include the Centrifugal Pendulum Dynamic Vibration Absorber (CPDVA) studied in the present work. The CPDVAs are used in helicopter models: Bell-412, V-22 among others where are called as pendular absorbers or out-of-plane pendulum as known in H145 (BK117) helicopter as shown in Fig. 1.

CPDVAs have been devised and studied in several works. [6] investigated the dynamic response of a rotor

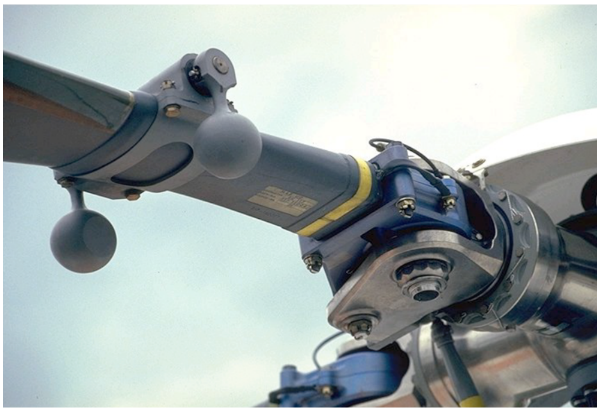


Figure 1. Pendular Absorber - BK117 (H145) [2]

system with two pendulum absorbers with different non-linear behaviours. It was shown that these absorbers have several advantages in using counterweights like ease of parameter control, little or no addition of mass to the rotors, and less dissipation of energy. [7] investigated vibrations suppression characteristics of pendulum absorbers in helicopter blades considering non-linear characteristics of pendulums, they considered only the first elastic mode using a three Degrees of Freedom (DOF) model. It was concluded that the suppression effect is improved by tuning the pendulum natural frequency to a value slightly higher than the excitation frequency. A similar work was carried out by [8], various parametric studies were performed to investigate the effect of variation of natural frequencies with rotational speed, the location, length and mass of the centrifugal absorbers and the effect of centrifugal vibration absorbers on reducing the transverse vibration response of the beam.

Despite some efforts performed in order to analytically derive the equations of motion of the beam and CPDVA problem as in the works of [9], [10] and [7]. There is a lack of detailed study in the field of Finite Element Method (FEM) as an alternative way to explain the behaviour of these devices. In the present work, it is made a contribution in this area in order to show the modelling and tuning of CPDVA and demonstrate the effectiveness of the developed finite element formulation by validating the results comparing with previous works results.

2 Centrifugal Pendulum Dynamic

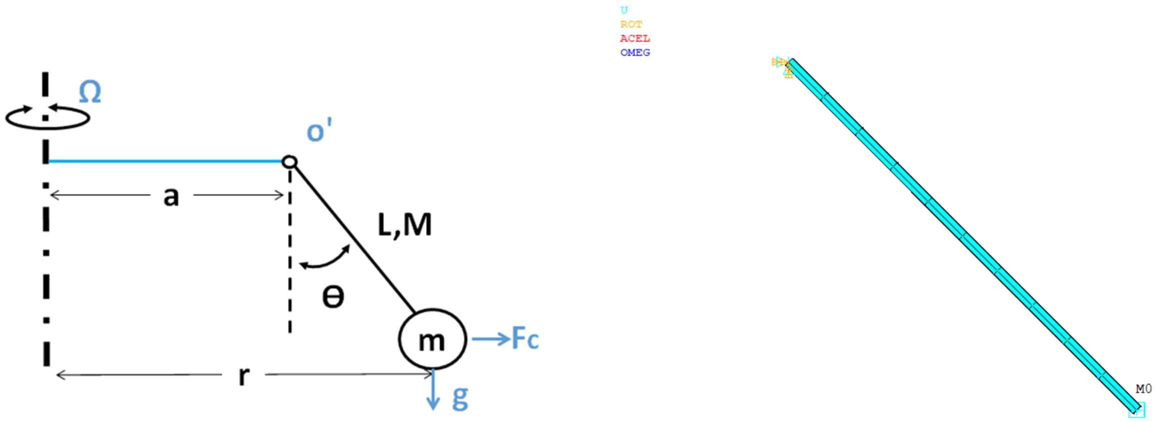
2.1 Analytical model

Pendulum vibration absorbers have been used in several configurations for various applications which include Tuned Mass Dampers (TMDs) installed in tall buildings to mitigate the wind induced swaying or seismic protection [11]. A vast number of works studied this pendulum TMDs, and all of them conclude that its effectiveness is limited to the mitigation of low frequency vibration modes. In the case of the present work centrifugal pendulums are used, it will demonstrated that the addition of rotational velocity to slender structures as pendulums or blades, induces a raise in natural frequencies values allowing its use at higher frequencies. To do this, the first step is to derive the non-linear equation of motion of a Centrifugal Pendulum.

Consider the centrifugal pendulum model illustrated in Fig 2(a). A rigid link of length a allows to attach a pendulum bar with length l and mass M to a rotation axis which rotates at an angular velocity Ω. A mass m located at the bar tip experiences mainly two forces, the universal gravity g and the centrifugal force F_c product of the pendulum rotation. The mathematical modelling of the system can be obtain by simple application of Newton's second law making the sum of moments around point o', see Eq. 1.

$$\sum M_{o'} = J\ddot{\theta} \tag{1}$$

Where J is the total inertia moment of pendulum (bar+mass) system, that is, $J = J_M + J_m$, being $J_M = \frac{1}{3}Ml^2$ and $J_m = ml^2$. And θ the angle between the pendulum and a vertical reference. Then, making the sum of moments Eq 1 becomes Eq. 2.



(a) (b)

Figure 2. (a) CPDVA analytical model, (b) CPDVA numerical model

$$-\frac{Mglsin\theta}{2} - mglsin\theta + m\Omega^2 rlcos\theta = J\ddot{\theta} \quad (2)$$

The variable r represents the distance between the tip mass m and the rotation axis, it is given by the expression $r = a + lsin\theta$. Reorganizing Eq. 2 and defining the constante $b = \left(m + \frac{1}{3}M\right) l$leads to:

$$\ddot{\theta} + \frac{Mg}{2b}sin\theta + \frac{mg}{b}sin\theta - \frac{m\Omega^2(a + lsin\theta)}{b}cos\theta = 0 \quad (3)$$

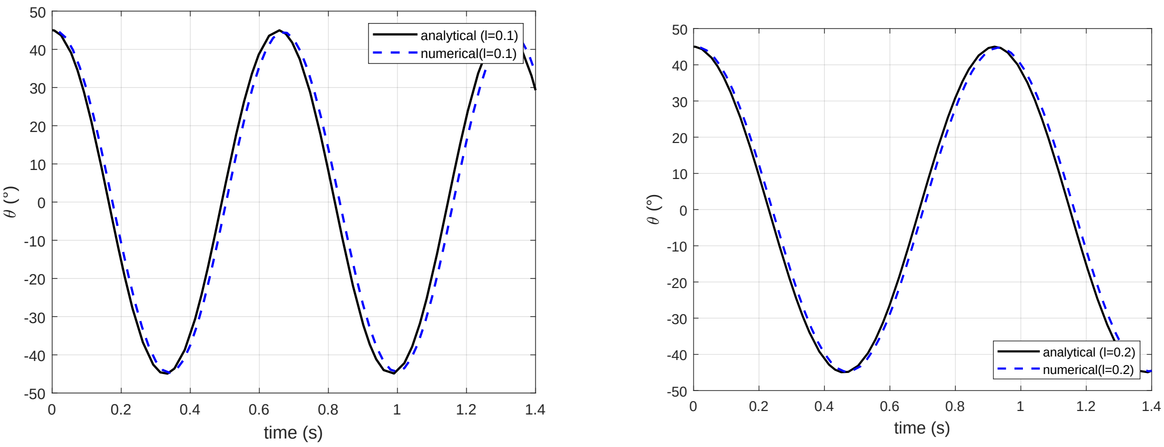
Equation 3 represents the non-linear differential equation that governs the behaviour of the centrifugal pendulum. This equation will be solved using "ode45" solver from Matlab/Simulink software.

2.2 Numerical model

The governing equation of motion of the CPDVA is developed using finite element method (FEM), Ansys APDL CAD/CAE software was utilized to this end. The pendulum bar was modelled with a standard two-node beam type element, after the application of the boundary conditions, only three degrees of freedom (DOF) were allowed, translations along x and y axes (see Fig. 2(b)) and rotation of node located at point o'. It was not necessary to model the rigid link, the location of end nodes of pendulum bar was enough to guarantee the effect of r variable in centrifugal force $F_c = m\Omega^2 r$. The tip mass was modelled using a discrete/point mass element. The non-linear pendulum deflection θ is captured by means of a transient analysis, that is, the nodal displacement (x and y) at discrete mass node is transformed in angular displacement using the function $atan2(y, x)$.

The parameters used at analytical and numerical models implementation were: standard gravity $g = 9.81$, offset $a = 0.2m$, pendulum length $l = 0.2m$, pendulum tip mass $m = 0.3kg$, quadrangular section pendulum bar with mass $M = (0.01 \times 0.01 \times l)\rho$ (aluminium: $\rho = 2700kg/m^3$, $E = 70GPa$) and rotational velocity $\Omega = 40rad/s$. In order to validate the FEM model, parametric case studies were performed and the results are compared with the analytical model response. In case 1 it was simulated the natural oscillation of a pendulum without centrifugal force ($\Omega = 0$) with two different pendulum bar lengths $l = 0.1m$ (Fig. 3(a)) and $l = 0.2m$ (Fig. 3(b)). At all cases the initial pendulum angle is $\theta_0 = 45°$. In this case, the natural frequency of the pendulum is approximately done by the expression $\omega_n \approx \sqrt{g/l}$ especially when the bar mass M and inertia J_M are negligible. That explains why the frequency at Fig. 3(b) is lower than in Fig. 3(a).

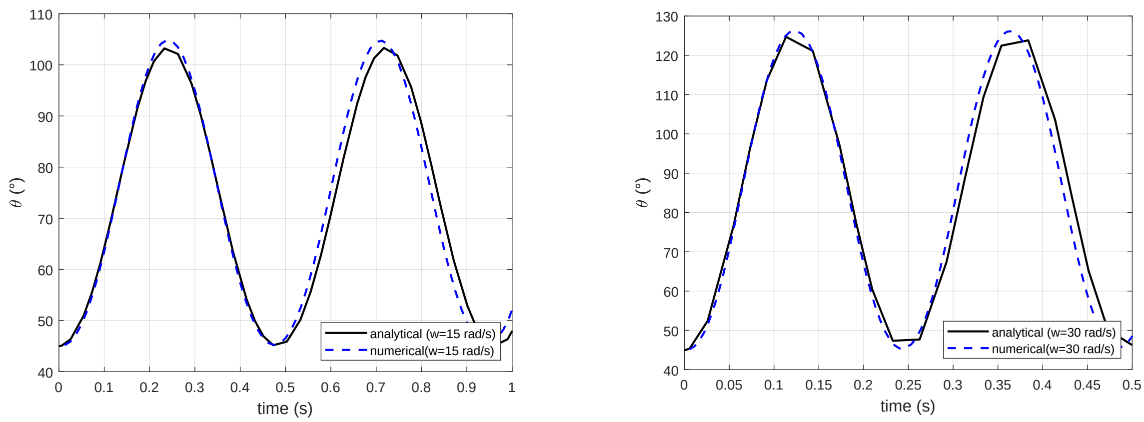
Case 2 considers the influence of centrifugal force by adding $\Omega \neq 0$. The pendulum bar legth was fixed in $l = 0.2m$ and two rotational velocities were tested: $\Omega = 15rad/s$ (Fig. 4(a)) and $\Omega = 30rad/s$ (Fig. 4(b)). It is worth mentioning that the centrifugal force makes the pendulum to oscillate near the rotor plane, at $\Omega = 15rad/s$ its equilibrium angle is approximately $75°$, in the case of $\Omega = 30rad/s$ the pendulum achieves its equilibrium inside the rotor plane $\theta \approx 90°$, this phenomenon can be observe in the video showed at `https://www.youtube.com/watch?v=Pu48f7s5Ru8`. In some works, the CPDVAs are called out-of-plane pendulum absorbers due to the continuous oscillation of the pendulum up and down the rotor plane as illustrated in



(a) (b)

Figure 3. (a) Pendulum angle ($\Omega = 0rad/s$, $l = 0.1m$), (b) Pendulum angle ($\Omega = 0rad/s$, $l = 0.2m$)

Fig. 4(b).

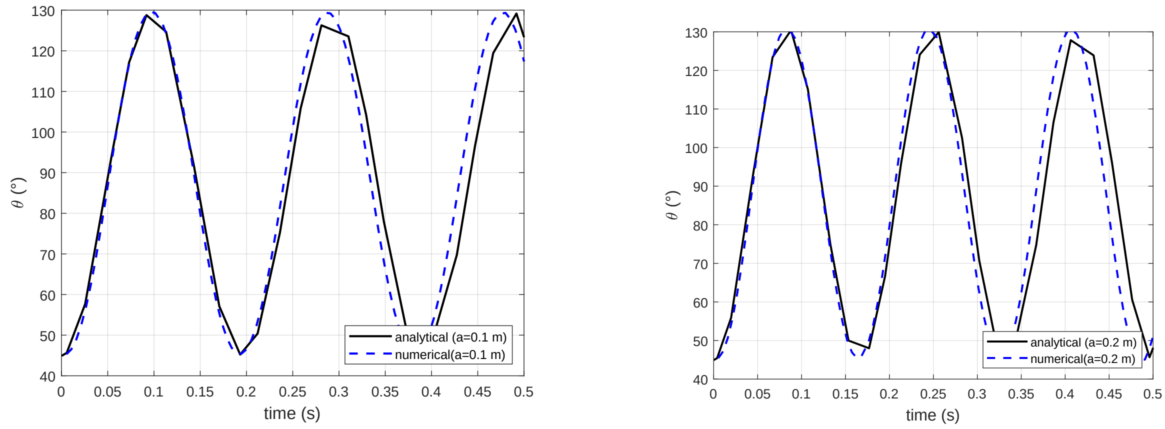


(a) (b)

Figure 4. (a) Pendulum angle θ at $\Omega = 15rad/s$, (b) Pendulum angle θ at $\Omega = 30rad/s$

Finally, case 3 illustrates the influence of the offset value a in the natural frequency of the pendulum. The rotational velocity was fixed at $\Omega = 30rad/s$ and the offset effect was observed with $a = 0.1m$ (Fig. 5(a)) and $a = 0.2m$ (Fig. 5(b)). It easy to see that the higher the offset value a the higher the pendulum natural frequency is.

As can be seen from the results (Figures 3, 4 and 5), there is a good agreement between the numerical and analytical results, then, the pendulum model is validated. As concluded by [7] there is a relationship between the pendulum parameters studied in this section, the relation is given by the equation $w_n \approx \Omega\sqrt{1 + a/l}$, and only valid for high rotational rotor speeds. The partial validating results showed at this section also agree with this expression. The natural frequency w_n of the pendulum response in the time domain is simply computed using $w_n = 2\pi f$ and $f = 1/T$ being T the period of the θ oscillation.



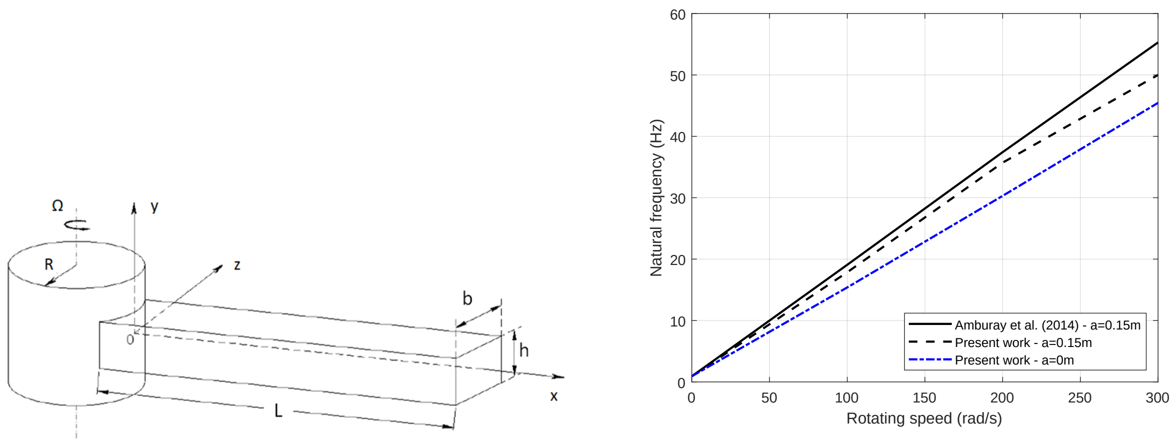
(a) (b)

Figure 5. (a) Pendulum angle θ at $a = 0.1m$, (b) Pendulum angle θ at $a = 0.2m$

3 CPDVA as Vibration Absorber

3.1 FEM Blade validation

In order to demonstrate the effectiveness of the centrifugal pendulums to reduce the vibration amplitude in rotating structures, the helicopter blade is modelled in the present work as a rotating beam with rectangular section, in future approaches a more realistic blade geometry will be used. The main objective of this study is to demonstrate the capacity of CPDVAs to reduce vibration and contribute to explain how this devices work. The ”blade” modelled using FEM is the same used at the work of [8], its properties are described as follows, it is a aluminium cantilever rotating beam (as a rigid helicopter rotor blade), with rectangular section ($b \times h = 0.15 \times 0.01m$) and length $L = 3m$. The beam is attached to a rotor hub of radius $R = 0.15m$ and subjected to a rotational velocity Ω (see Fig. 6(a)).



(a) (b)

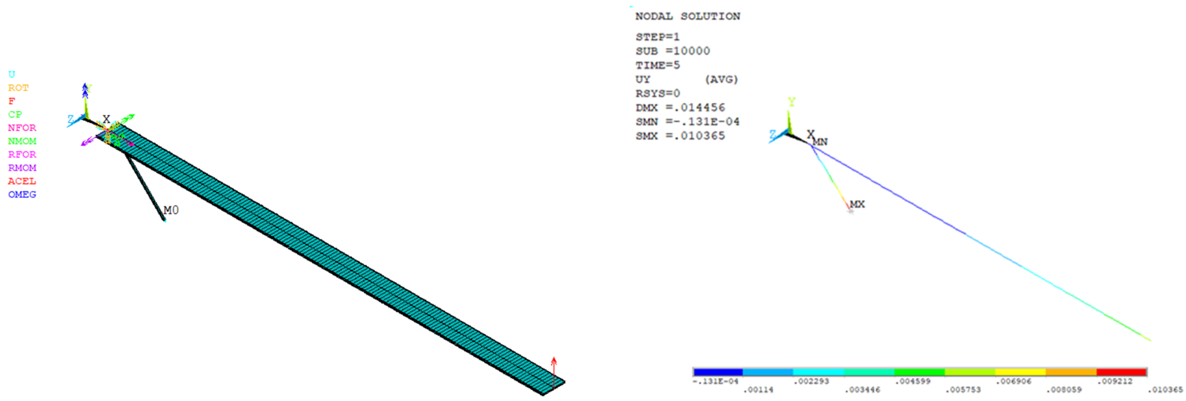
Figure 6. (a) Blade model. Adapted from [1], (b) blade natural frequency for several rotational velocities

The numerical model was developed using beam elements with six DOF at each of its two nodes, 100 elements were sufficient to the convergence of the mesh. For the validation of the blade model, the first natural frequency of the beam was captured varying the rotational velocity. Results can be seen at Fig. 6(b), it is possible to note that for rotational velocities lower than $\Omega = 150rad/s$, there is a good agreement in terms of the natural frequency of the rotational blade/beam between the present work (continuos line) and the developed by [8] (dash line). Blue dotted line represents the change in blade natural frequency with rotational speed when the rotor hub radius R

is zero, then, as expected, it is possible to see that the "stiffness" of the blade increases as the offset a increases too, the same phenomenon was observed in the pendulum. In the case of the blade the offset a is the same hub radius R.

3.2 CPDVA case study

Once validated the centrifugal pendulum and rotating beam separately. At this section, the Centrifugal Pendulum (CP) will be proved as vibration absorbers, and its ability on reducing the transverse vibration response of the beam is studied. To do this, it is considered the CPDVA attached to the blade subjected to a harmonic excitation force applied at the tip of the beam and comparing the transverse vibration response at the tip of the beam with and without the absorber. The FEM model of the set (blade + CPDVA) is shown in Fig. 7(a). The parameters configured to this case study are, blade: $L = 3m$, $b = 0.15m$, $h = 0.01m$ and $R = 0.15m$, CPDVA: $l = 0.55$, $m = 0.3kg$ and $a = 0.2m$. The rotational velocity was considered as $\Omega = 40rad/s$ (main rotor rotation of helicopter BK117 of Fig. 1).



(a) (b)

Figure 7. (a) Blade and CPDVA FEM model, (b) Blade and CPDVA vertical displacements

In Fig. 7(a) it is possible noting the boundary conditions and loads applied to the model, the U and ROT condition allows to define the clamped end of the blade at the hub ($Rot(z) = 0$), the CP legend indicates in Ansys, Clamped Free, that is, the application of a free hinge to the free rotation of the pendulum at a particular node without affecting the stiffness of the blade itself. ACEL and OMEG are related to the gravitation acceleration ($-g$ at y axis) and angular velocity ($Omega$ around y axis) respectively, the force F at the tip of the blade is an harmonic resonance force given by the function: $F = 1 * sin(\omega_n t)$. Figure 7(b) shows the vertical displacements of the system during transient analysis.

The l and a parameters of the CPDVA where chosen by a trial and error process, in order to make the CPDVA natural frequency match with the blade natural frequency ω_n such as described in [7]. According to [7] the suppression effect is improved when the pendulum is tuned to a frequency which is slightly lower than multiples of the rotational frequency due to the effect of non-linearities, then, there is not an exact expression to determine the its tuned parameters, this justify the use of a trial and error process. The vibration amplitude at the blade tip with and without the CPDVA under a resonance excitation force is showed in Fig. 8.

4 Conclusions

In the present study, the dynamic characteristics and transverse vibration suppression of helicopter rotor blades using CPVA are investigated and clarified. The governing differential equation of motion of a rotating centrifugal pendulum was developed and used to validated the FEM model. The validity of the developed finite element formulation for the blade was demonstrated by comparing the results of previous works in terms of natural frequencies and some physical facts about the behaviour of CPDVAs were clarified in relation to changes in its parameters. The following conclusions can be summarized: *i)* FEM pendulum behaviour coincides with the expected theoretical one, *ii)* a good agreement in terms of natural frequencies was observed between the FEM and previous works, *iii)* it is not strictly necessary to analytically develop the governing equations of motion of the

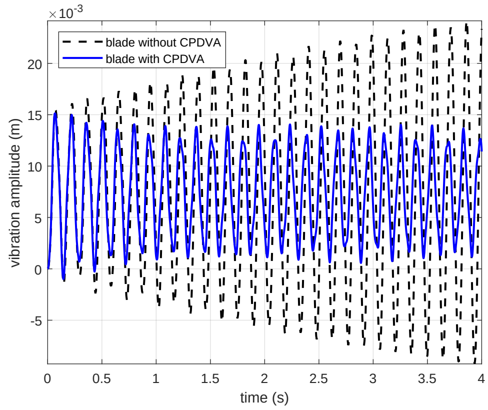


Figure 8. Blade tip amplitude with and without CPDVA

beam or the absorber, the use of FEM to design the system is more intuitive and easy to do, *iv)* centrifugal forces produced by rotational speeds increase the natural frequencies of centrifugal pendulums allowing its use in more rigid structures and not limited to low frequencies of tall buildings and *v)* the tuning process of a CPDVA is not an easy task, it will be necessary for future works the use of an optimization process in order to facilitate and enhance its design, the construction of a parametric FEM model such as the presented in this work can be helpful.

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