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DYNAMIC ABSORBERS IN THE SUPPRESSION OF VORTEX INDUCED VIBRATIONS OF MARINE STRUCTURES

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Abstract. *The current work presents the effectiveness of dynamic absorbers acting as suppressors of vortex induced vibrations caused by the incidence of uniform flow around cylindrical bodies. The absorber device is a flexible beam with a lumped mass. The device works without direct interference with the flow, unlike most of the VIV suppressors already proposed as strakes and water jets. The main analysis consists of a comparison between a base case of a cylinder without any device and another one with the absorber, keeping the correspondence of the hydrodynamic and structural properties of the two cases. Parametrical analysis regarding to the values of natural frequency and maximum VIV amplitude related to variations on the supported mass, support length and cross section properties is also performed.*

Keywords: *dynamic absorber of vibration, VIV, VIV suppressor*

1. INTRODUCTION

In the past few years, the petroleum industry has tried extensively to solve the problem of vortex induced vibrations (VIV) on marine structures. Most of these structures are drilling and production risers, and mooring lines. For some of these structures, the solutions proposed up to date actuate directly on the control of the excitation caused by the vortex shedding, through the introduction of devices that alter the flow around these structures. This kind of control is also applied on large structures like Spar Buoy Platforms by means of strakes that reduces the lift coefficient but, on the other hand, cause an increase on the drag force. Unlike on the other structures previously mentioned, the Vortex Induced Motion (VIM) on Spar platforms can also be controlled directly on the response of the system. This is the focus of the current work that discusses the effectiveness of Dynamic Absorbers (DA) as a VIV controller in an experimental approach.

Initially, in the design of DA, the main objective was the synchronization of its first natural frequency with the vortex shedding frequency that causes the maximum response in terms of VIV amplitude. This frequency is equal to the damped natural frequency of the cylinder, which is rigid and contains only one degree of freedom. The cylinder system suffers the excitation caused by the vortex shedding (input) and gives an output in terms of VIV. The vortex induced vibration is then, transmitted as input for the DA that gives an output in terms of vibration of the suspended mass placed at the free extremity of the elastic beam. Figure 1 illustrates the dynamic absorber working properly.

The design of the DA was conducted in a prospective way, including the suspended mass and the device that support the mass defining its elasticity, geometry and material. The device is placed above the water level in order to investigate the concept only in terms of the frequencies involved, avoiding the introduction of the flow viscous damping as another parameter to be defined. It also avoids the evaluation of the added mass on the DA for the correct tuning of its first natural frequency with that of the cylinder.

The results are presented in terms of comparisons between the amplitude of VIV for the cylinder with and without the DA. The tests conducted for the last one are compared with classical experiments of VIV on elastically mounted cylinders in order to validate the base case of comparison.

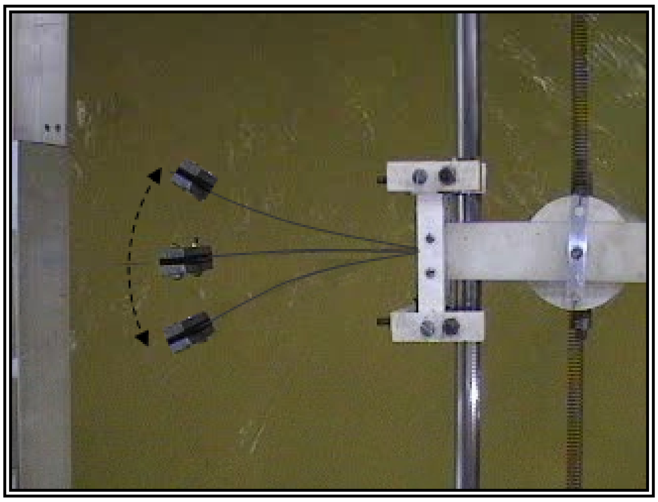


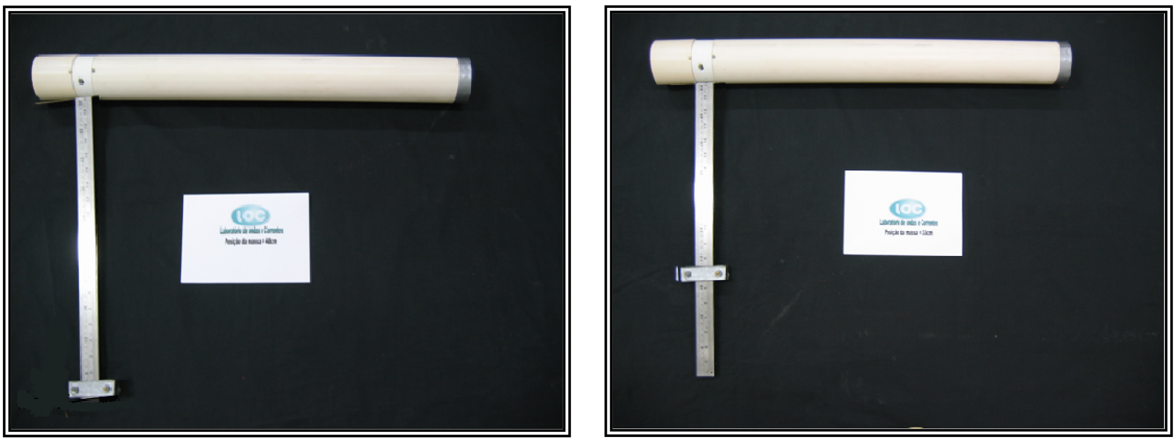
Figure 1 – Dynamic Absorber of Vibration placed at a rigid cylinder.

2. MODELS AND EXPERIMENTAL ARRANGEMENT

The proceedings adopted for the tests are described in sequence. The elastic base that supports the cylinder can be clearly observed on Figure 1. The base restricts the in-line motion (in the current direction) and allows motion on the transverse direction. The bottom of the cylinder is free. The distance between the cylinder and the current channel bottom was small enough to assure two-dimensional flow around this point.

The data acquisition was performed by an image acquisition system (Siqueira, 2000) that works on the contrast between targets positioned on the top of the system and a black background provided by the experimental setup.

The experimental comparisons were conducted for two different models. Model 1 was designed in a way that the first natural frequency of the DA (beam-mass) was equal to the theoretical vortex shedding frequency that causes the higher cylinder response in terms of non-dimensional amplitude of VIV (*A/D*); where *D* is the cylinder external diameter. During the progress of the tests, it also was sensed the need to put the DA’s first natural frequency on a value equal to a vortex shedding frequency that causes lower values of *A/D* after the resonance region (usually called lock-in region for VIV). Model 2 presents a DA with this value as first natural frequency. The first natural frequencies of models 1 and 2, presented on figures 2 (a) and (b), contain values of 0.47 and 0.8 Hz.



(a) (b)

Figure 2 – (a) Model 1: first natural frequency equal to 0.47 Hz. (b) Model 2: first natural frequency equal to 0.8 Hz.

For the definition of these two frequencies, it was performed a parametrical study based on the classical analytical solution for the problem described here. The dimensions of the DA evaluated on the parametrical study are represented on Figure 3.

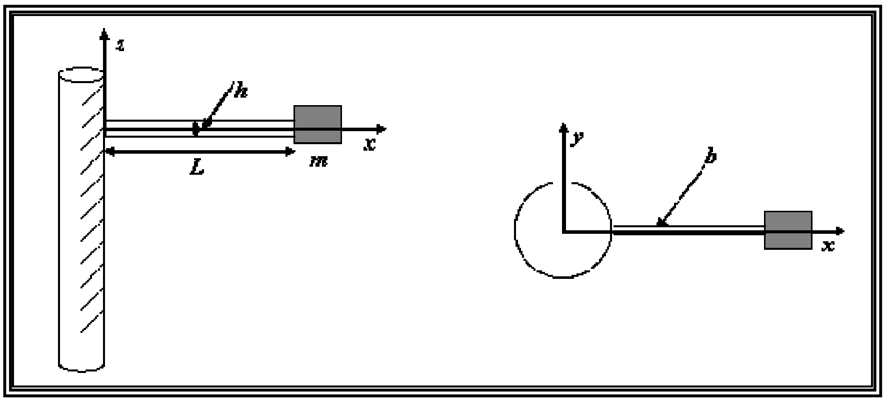


Figure 3 – Representation of DA dimensions.

The first natural frequency of the DA can be determined by Equation 1 [Blevins, 1979] for the lateral vibration (*xy* plane) of a beam fixed on an extremity and free on the other, but with a mass lumped at the free boundary. The beam utilized for the tests has geometry form of a ruler, properly chosen in a way of preventing torsion on the *x* axis, proportioning only bending on the plane *xy*.

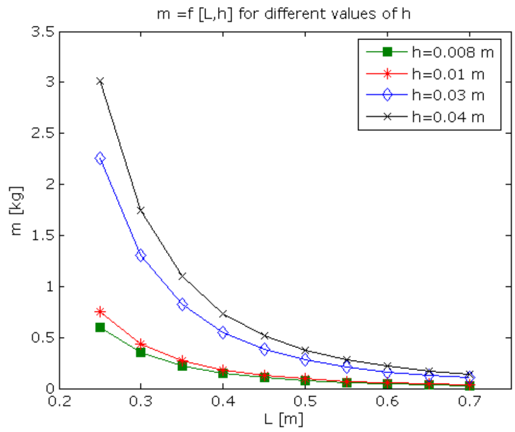
$$f_1 = \frac{\lambda^2}{2\pi}\sqrt{\frac{EI_{zz}}{mL^3}} \tag{1}$$

Where λ = 1.8751, *E* is the Young Modulus of the beam material, *m* is the mass lumped at the free boundary of the beam summed to the beam mass, *L* is the beam length and *I* is the inertia moment of a beam with rectangular section related to the *z* axis (Figure 3) given by Equation 2.

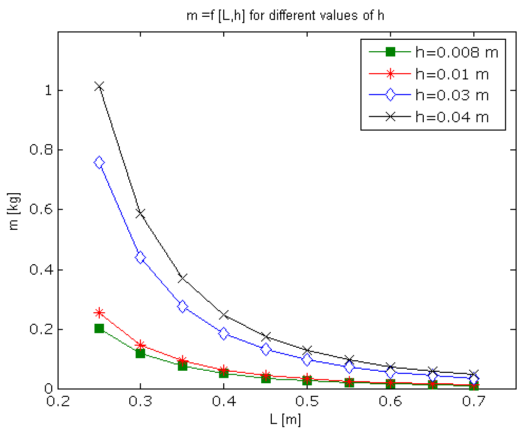
$$I_{zz} = \frac{hb^3}{12} \tag{2}$$

Where *h* is the high and *b* is the width of the beam.

As previously mentioned, a parametrical study was performed in order to map the values of *m*, *h, b* and *L* that ensure the first natural frequency desired for the DA. In order, to prevent torsion, many rulers were tested, and for the two models of DA presented here, it was utilized a ruler of Steel 302 with width equal to 0.6 mm. The adjustment can be easier obtained through variations on the high and length of the rulers in contrast with manufacturing widths. Then, the problem was reduced to the determination of lumped mass, length and height of the ruler in a combination that avoids torsion on the *x* axis. Figures 4 (a) and (b) presents the parametrical relations that assure first natural frequencies for the DA of 0.47 and 0.8 Hz, respectively.



(a)



(b)

Figure 4 – Parametrical relations that assure first natural frequencies of the DA around 0.47 Hz (a) and 0.8 Hz (b).

Due to practical reasons, a high of 0.03 m was utilized. In follow, a combination of mass and beam's length was chosen. Tables 1 (a) and (b) present, respectively, these values in order to ensure the values of 0.47 and 0.8 Hz as first natural frequencies of the DA. The chosen values are in the ranges defined by the bolded values on the tables.

Table 1 – *L* x *m,* for *b* = 0.0006 m and h=0.03 m, that ensure first natural frequencies of the DA around 0.47 Hz (a) and 0.8 Hz (b).

(a)

L (m)	m (kg)
0.08	68.93554
0.1	35.295
0.15	10.45778
0.2	4.411875
0.25	2.25888
0.3	1.307222
0.35	0.823207
0.4	0.551484
0.45	**0.387325**
0.5	**0.28236**
0.55	0.212141
0.6	0.163403
0.65	0.128521
0.7	0.102901

(b)

L (m)	m (kg)
0.08	23.20966543
0.1	11.8833487
0.15	3.520992207
0.2	1.485418587
0.25	0.760534317
0.3	**0.440124026**
0.35	**0.277162652**
0.4	0.185677323
0.45	0.130407119
0.5	0.09506679
0.55	0.071425086
0.6	0.055015503
0.65	0.043271183
0.7	0.034645331

Summarizing, in order to obtain a first natural frequency for the DA equal to 0.47 Hz, it was utilized a ruler with 0.0006 m of width, 0.03 of high, 0.46 m of length and a mass that in bounds with the ruler characterize the DA with a mass equal to 0.36 kg. Similarly, in order to obtain the first natural frequency equal to 0.8, a ruler with the same width and high, but with 0.32 m of length was utilized. The same DA mass was obtained for this last one.

3. EXPERIMENTAL RESULTS

The tests were conducted for two different models of dynamic absorber. In general, for each test, around twenty measurements with different current values were performed. The Reynolds Number obtained for the tests ranges from 1 x 10^4 to 4 x 10^4 characterizing laminar flow, once the experiments deal with external flow around a cylindrical structure. For each model, comparisons that involve the effectiveness of the DA are presented in terms of non-dimensional amplitude of VIV on the cylinder versus Reduced Velocity that is given by Equation 3.

$$V_r = \frac{U}{f_n D} \tag{3}$$

Where *U* is the current velocity, f_n is the damped natural frequency of the cylinder and *D* is its diameter. In order to determine the damped natural frequency of the cylinder, deployment tests were carried out for the cylinder without the DA. The natural frequency obtained for the cylinder is equal to 0.47 Hz.

Firstly, in order to validate the tests performed here, the base case (defined in the sequence of the paper) was compared with classical results from literature (King, 1974) and (Williamson, 2004). Figures 5 (a) and (b) are, respectively, recovered from the mentioned references, with red points indicating the placement of the case base tests performed here. These comparisons were performed in terms of the maximum non-dimensional amplitude of vibration (*A/D* for lock-in) as a function of the Skop-Griffin ($S_G = 2\pi^3 S^2 m^* \zeta$) and the Stability parameters ($K_S = 2m^* 2\pi\zeta$) of the tests (*St* refers to the Strouhal Number). As it can be seen, the results for the case base present good agreement with cited results.

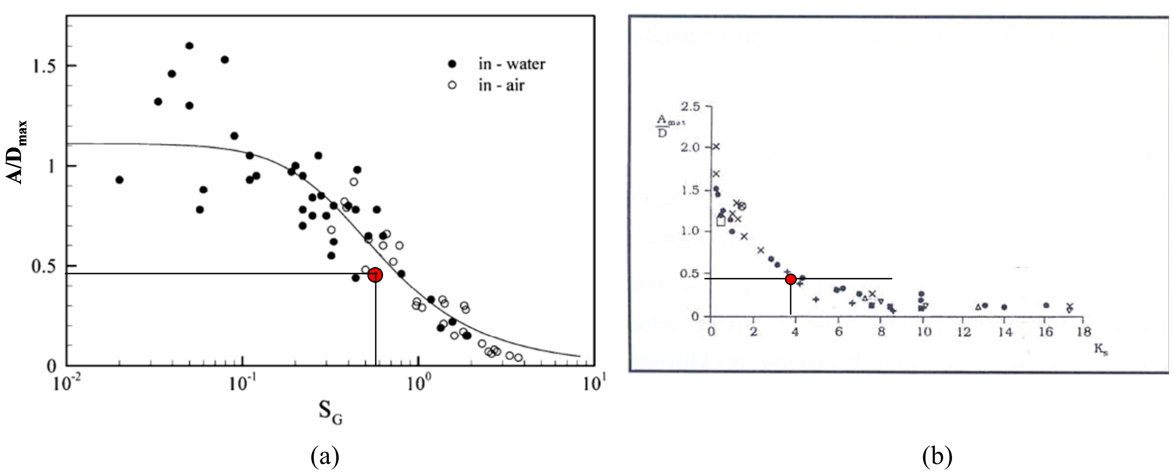


Figure 5 – Comparison between the base case and classical results from literature.

Once the objective of the study is to notice the effect of synchronization between the first natural frequency of the dynamic absorber with frequencies related to the excitation of the cylinder (vortex shedding frequencies), the system "cylinder plus DA" must be compared with the cylinder utilized on the system without the effect of the dynamic absorber. However, it is necessary to consider that the mass ratio ($m*$) – parameter that relates mass of the system and mass of displaced fluid – of the cylinder without the absorber device will be different from the system with DA. The effect of this difference can cause a changing on vibration amplitude that is not related to the real effect of the dynamic absorber. The cylinder without DA has a mass ratio of 0.89 and the system with DA has a mass ratio equal to 0.98.

In order to certify that the mentioned difference between these two mass ratios are not significant, on the point to interfere on the results, a test was conducted with a cylinder without DA, but containing a mass ratio equal to 0.98. This cylinder was filled internally with distributed mass along its length. The Figure 6 presents the results of the comparison between these two cylinders and it can be noted that the difference in terms of VIV non-dimensional amplitude is not significant.

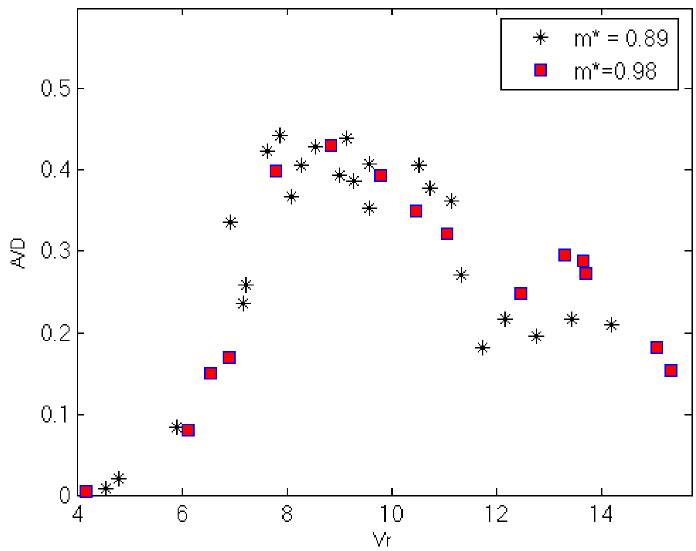


Figure 6 – Comparison of the VIV response between cylinders with different mass ratios.

Based on the previous result, the comparisons that follow take as base case the cylinder with mass ratio equal to 0.89.

Figure 7 presents comparison between the cylinder with and without the DA with first natural frequency of 0.47 Hz. As it can be observed from the results, the dynamic absorber works properly for VIV suppression until some point on the curve. The point where the dynamic absorber starts increasing the VIV amplitude corresponds to the beginning of curve's falling after the first mode resonance of the DA. It indicates that the dynamic absorber presented a negative tendency from the point where the excitation frequency went over its first natural frequency that for this case is 0.47 Hz and coincide with the cylinder natural frequency. From this point, possibly, the second natural frequency of the dynamic absorber has become the most significant one.

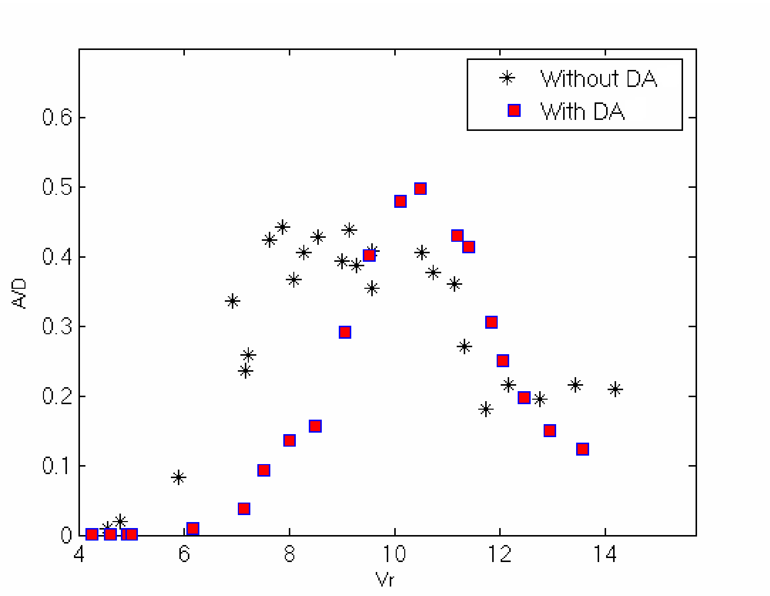


Figure 7 – Comparison between the cylinder with and without a DA of first natural frequency equal to 0.47 Hz.

In order to verify this tendency, another test was perform for the same cylinder, but now, with a dynamic absorber with 0.8 Hz of first natural frequency. Figure 8 presents the comparison with the cylinder without DA. As it can be observed from this result, the absorber becomes effective for a wider range of Reduced Velocities. It proves the justification given for the last result, once for this case, the first natural frequency of the DA is over passed by the theoretical vortex shedding frequency ($f_S=U*St/D$) only for high Reduced Velocities values.

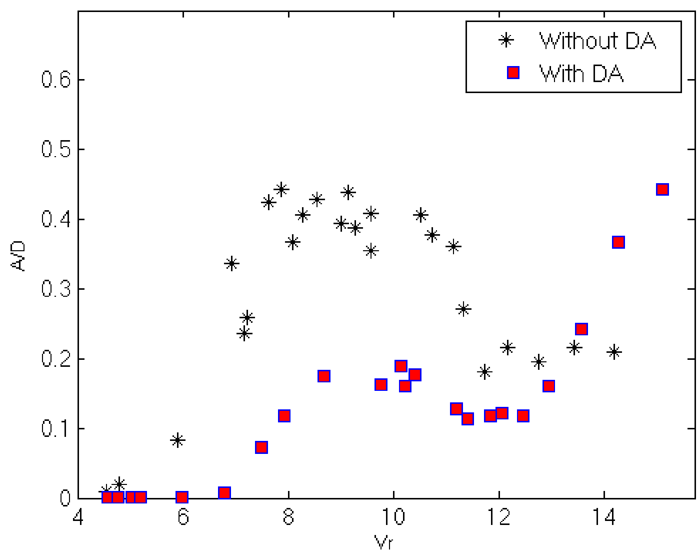


Figure 8 – Comparison between the cylinder with and without a DA of first natural frequency equal to 0.8 Hz.

4. CONCLUSIONS

In general terms, the experiments conducted for this work represented a starting point for the study of a concept of VIV suppressor not tested yet.

The concept of DA has shown to be effective as long as the design takes into account the requirement of the first natural frequency to be higher than the highest possible excitation frequency of the cylinder.

This tendency is positive, once a high first natural frequency is not difficult to be obtained, allowing indeed a low mass in bounds with a low length beam.

Future work will pursue a smaller device which probably will be placed in water. This was avoided in the present paper because the necessity of simpler problems in prospective analysis as this one.

5. ACKNOWLEDGMENTS

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