DESIGN OF ELASTIC BASES FOR VIV EXPERIMENTS

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Resumo. *An oscillating cylinder immersed in a current profile modifies vortex shedding. There are two different ways to investigate this phenomenon: employing forced oscillations or using cylinder mounted on an elastic base. Actually, this last technique is the most direct form to analyse vortex-induced vibrations (VIV). The advantage of using this method is that the responses can be measured directly, consequently being more realistic than forced-oscillation experiments. The practical interest related to VIV and the amplitude reached by elastically mounted cylinders are their relation to the life expectance of cylindrical elements of offshore platforms immersed in currents and waves. Although the freedom to oscillate in-line with the flow directions has some effect on the transverse response, the modes of vibration, or the vortex wake dynamics, two-dimensional elastic bases have been developed in the most important research centres of the world. In this present work, two projects of different conceptions of elastic bases are presented, which fulfil some design criteria, e.g. low mass and low damping ratios, extremely linear system dynamics and instantaneous and direct measurement of position. One of the elastic bases employ a mini-jet air bearing system, to achieve very low damping, and the other one makes use of thin and flexible steel blades, which behave like springs. Both models have already operated on the water channel of the "Núcleo de Dinâmica e Fluidos – NDF" , at " Escola Politécnica", University of São Paulo, while the projects of the other two bases are being analyzed.*

Keywords*: vortex shedding, vortex-induced vibration, bluff body flow*

1. Introduction

Vortex-induced vibrations (VIV) are self-excited and self-controlled oscillations, which can occur on structures in a fluid flow. These vibrations preoccupy not only for the high-amplitude attained (for a cylinder, of about one diameter), but also for their permanence and frequency, able to affect the integrity of the structure, causing a precocious damage by fatigue.

These oscillations, which happen on thin bodies with bluff cross-section and immersed in fluid flow, depend on the periodic variations of the field of pressures in its surroundings, originating the periodic vortex shedding around the body. It is already known that this vortex shedding precedes the oscillations of the body, since exactly bodies without freedom to oscillate present vortex wake shedding.

The vortex wake mode emitted in the flow by the body follows an oscillation standard, that is, the vortex is emitted in a constant and well determined frequency, which varies with the speed of the incident flow and with the characteristic dimension of the body (diameter, in the case of cylinders). Thus, it can occur that this emission frequency grows gradually (increasing speed, for example) until being near one of the natural frequencies of the body. At this moment a resonance phenomenon known as *lock-in* takes place, enhancing the vibration amplitude.

At resonance, the vortex shedding frequency is captured by the body's oscillation frequency. For reduced velocity values near of 5, the highest oscillation amplitudes take place. The range of reduced velocity in which occurs the synchronization represents an important point of investigations.

The VIV phenomenon is classified as an example of the classical mechanical problems without a definitive agreement, despite the intense effort spent in researches on the topic since the beginning of the XX century. Thus, this subject still reserve innumerable aspects without a consistent and decisive understanding. Associated to this condition, a reality in which the phenomenon reveals practical and increasing interest in several contemporary technological segments also contribute to keep it under constant focus of attention.

Offshore projects, responsible for the viability of oil production in deep waters, concentrate great part of the current inquiries concerning VIV problem. One of the reasons that take to this are the not homogeneous nature of exciting ambient condition (maritime rapids), that induces the coexistence of various natural modes of vibration, mainly those of the bigger order and minors flexion lengths, aggravating still more the fatigue.

The essential objective of the present article is to get an experimental easiness for the study of forces and answers associates with VIV in cylinders, have seen the importance of practical essays in the validation of the gotten numerical solutions. Many of the raised questions will be solved through simulations accomplished by the DFC group (dynamic of fluids computational), but these will only possible due to the results of gotten experimental procedures with models in reduced scale, mounted in elastic base free to oscillate.

2. Elastic Base

The case relative to the vibration of a cylinder mounted on an elastic base, happens exclusively due to the fluid force, is one of the most fundamental and revealing cases related to the subject of vortex-induced vibrations. The practical interest in getting these amplitudes is linked to the fact that this parameter is directly related to the estimate of life-span of cylindrical elements submitted to VIV.

With this type of apparatus, oscillations only happen for ranges of reduced velocity in which the energy transferred from the fluid to the body is positive and the vortex shedding frequency is near to the natural frequency of the base or one of its multiples or submultiples.

The advantage to simulate the fluid flow around of a model mounted on an elastic base comes from the fact of that, in this case, we measure oscillation amplitudes directly. In forced oscillations, the effect of the coupling can also occur for negative energy ranges, oscillations that never would take place if the model were mounted on an elastic base. In this aspect, free oscillation experiments are more realistic than those in which the oscillations are imposed. However, the price that we pay for is that, in the free oscillation experiments, the number of parameters is significantly higher than those with forced oscillations.

Among the parameters to be considered in this paper, the relation between Strouhal number (St) and spring stiffness plays a special role. It is already known that for certain values of Re, more specifically between 200 and $5x10^5$, the Strouhal number is practically constant and around 0.2. The vortex shedding frequency can be evaluated, for a give current speed, and it can be compared to the natural frequency of the spring.

The total mass of the system (base + model) must be low (close to displaced fluid mass), in order to the system oscillate with the possible highest amplitude. Therefore, it must be carried out a detailed study concerning the form, dimension and materials to be used in the process of designing the base, in order to minimize its mass effect on the experiment. The base damping must be also small in order to make the oscillation amplitude as high as possible. These are the main concerns during design. Here, our main goal was to manufacture a base with damping below 1%.

In the development of the base, the fixed parameters of the water channel where it will tested, are also considered, specially the dimensions of the tests section and the flow velocity. Additionally, its flexibility must be taken into account, in order to be applied in different types of experiments. For this, it is fundamental that the system allows to the alteration of spring stiffness and the coupling of diverse types and model configurations.

3. Analyzed parameters

In this section, we first analyse the structural parameters of the elastic base and their relation to the hydrodynamic forces. In order to obtain the equation that characterizes the problem and the influence of its parameters the reasoning applied by Khalak and Williansom (1996) and Meneghini (2002) will be employed.

The basic equation of the problem is the one of an oscillator:

$$m\ddot{y} + c\dot{y} + ky = F \tag{1}$$

where m is the structural mass, c is the structural damping, k is the spring stiffness and F, the fluid force in the transverse direction. For the case of oscillations in the current direction, the fluid force in the transverse direction is substituted by the drag force.

The mass ratio is given by the ratio of the structural mass to the displaced fluid mass, i.e.

$$m^* = \frac{m}{m_d} = \frac{4m}{\pi D^2 L\rho} \tag{2}$$

where m is the structural mass, m_d is the displaced fluid mass, D is the cylinder diameter, L is the cylinder length and ρ is the fluid density.

It must be said that, for the estimate of structural mass, all mobile parts of the base are taken in consideration, such as the shaft, carriage and springs, in the case of the air bearing base, and the flexible blades base, the inferior base and the screws, in the case of the fletor base, beyond the cylinder being tested.

The damping of the system is given by the ratio of the structural damping to the fluid damping

$$\zeta_a = \frac{c}{2\sqrt{k \cdot m}} \tag{3}$$

where m_a represents the additional mass and C_a the additional mass potential coefficient.

The mass-damping parameter must be the lowest possible. Variations of this parameter influence directly in the experimental data obtained. Its importance is essential to define the type of structure global behaviour inside the lock-in range.

Frequency ratio is given by the ratio of the oscillator frequency to the natural frequency of the system

$$f_a^* = \frac{f_{osc}}{f_{na}} \tag{4}$$

The system natural frequency and the period are given by

$$T_{na} = \frac{1}{f_{na}} = \frac{2\pi}{\omega_{na}} = 2\pi \Big/ \sqrt{\frac{k}{m}} \tag{5}$$

The reduced velocity is given by the ratio of the fluid velocity to the cylinder oscillation velocity:

$$V_r = \frac{U}{f_n D} \tag{6}$$

where U is the fluid velocity, f_n is the natural frequency of the system and D is the cylinder diameter.

The amplitude ratio given by the ratio of the oscillation amplitude to the cylinder diameter:

$$A^* = \frac{A}{D} \tag{7}$$

where A is the cylinder oscillation amplitude.

4. Elastic base with air bearing system

Elastic base with mini-jet air bearing system was the first alternative chosen in order to get, not only low damping, but also low rigidity and mass parameter, and an extremely linear instrument. The original model in which this base was inspired was developed at Imperial College, by PhD student Maša Brankovic (2004), and employed by her to investigate VIV attenuation devices on cylinders with low damping and mass.

This base consists of a carriage that supports vertically the cylinder immersed in the fluid. Two springs had been used in an way that the oscillation frequency of the mass-spring system is near the vortex formation frequency for a given current speed. In this in case, the cylinder is free to oscillate in the transversal direction of the water channel, that is, perpendicularly to the flow direction. The design of the elastic base with mini-jet air bearing system is shown in Fig. 1.

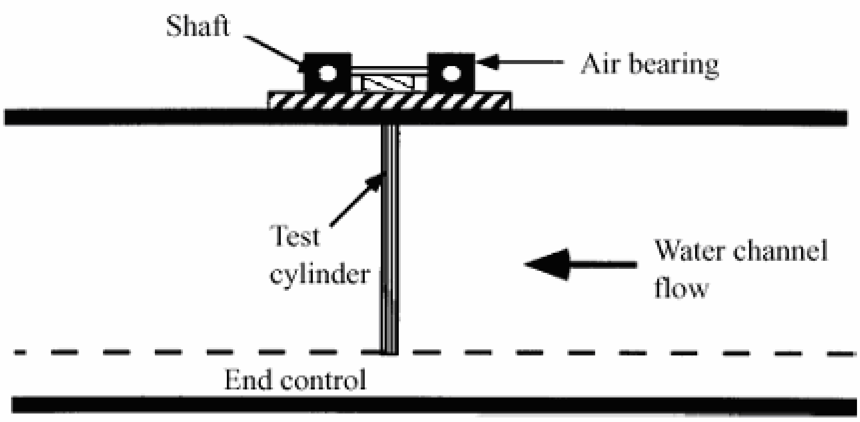

Figure 1. Lateral view of the elastic base with air bearing system, reproduced by Khalak and Williansom (1996)

The project of the elastic base construction was subdivided in accordance with the integrant parts of the system, in the case carriage, bearings and springs. Bearing was designed to supply a stiff and lightweight system. So that such objective was reached, an acrylic plate of 15 mm thickness was used, where the tested cylinder is fixed. The carriage project depend on two cylindrical shafts of 600 mm length and 30 mm diameter, confectioned in aluminium of 1 mm wall, that are connected the plate and pass freely through each pair of air bearing.

The cylinder model is constructed in aluminium and it is assembled in the water channel with a gap to the floor with less than 5mm. The cylinder diameter can vary in accordance with the experiment to be carried and Reynolds number chosen. Therefore, a coupling part to the base have been developed guaranteeing that cylinders with different diameters could be essayed. The mass ratio obtained was 2.66 with a cylinder of 32 mm diameter. The cylinder was opened on its bottom extremity up to water level. Consequently the mass of water inside the model was considered to calculate the mass ratio.

The system stiffness is controlled using two springs of approximately 100 mm length, placed between the carriage and a plate settled perpendicularly to each pair of air bearing. The springs had been dimensioned in accordance with the Eq. (6), considering the reduced velocity range required for the experiments, Vr from 2 to 18, and optimizing the system natural frequency so that such range is obtained. Variations in the velocity of essay, the Reynolds number and the cylinder diameter were also considered when a preliminary table of essays was confectioned.

The low structural damping required in the base project was achieved through the construction of four air bearings. In the air bearing, air deriving from a pneumatic system passes through a restrictor orifice followed for one determined pressure fall and then it flows axially in direction to the atmosphere between the shaft and bearing, causing a second fall of pressure. Thus, an air film is formed where the shaft slides without direct contact with the bearings. This arrangement gives a damping ratio, that is, the ratio of the damping in air to the critical damping, below 1%.

The air bearing was made out of brass blocks. Each bearing presents one 35mm hole drilled along its length and one 19mm hole at the top. An internal ring, with 12 rows of 3 circumferential holes drilled on each, was soldered centrally in the brass block. Each circumferential hole is 0.5 mm and transmits air for the aluminium shaft. The working pressure was estimated in 40-60 psi.

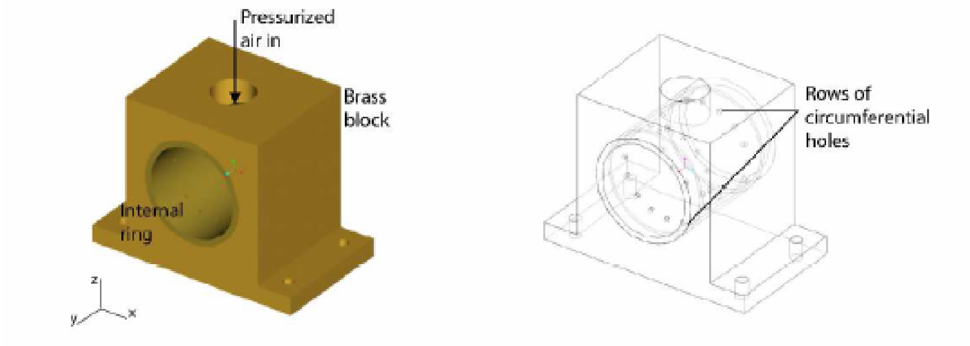

Figure 2. Detailed view of one of the air bearing

The bearing stays fixed on an aluminium plate by four screws and its alignment is criticizes, having to be guaranteed. This plate, fixed on the top of the water channel, present a central hole, through where the cylinder is able to oscillate freely inside the water. Screw-threaded legs placed at the four corners of the plate allow the adjustment of its height and inclination, and the calibration is made using a manometer levelling of high precision.

The reduced velocity (*Vr*) versus amplitude curve was obtained varying the current velocity. The reduce velocity can be calculated with the water velocity (m/s) and with the natural frequency: $Vr=U/fnD$. The natural frequency in air has been chosen for the evaluation of *Vr*, the reason for choosing its value in air is discussed in Meneghini (2002). The measured natural frequency in air for this base was found to be 1.725 Hz. The critical damping in air was equal to 0.006933. In Figure 3, this curve is shown. The maximum amplitude observed was slightly above one diameter and for a reduced velocity around 3.

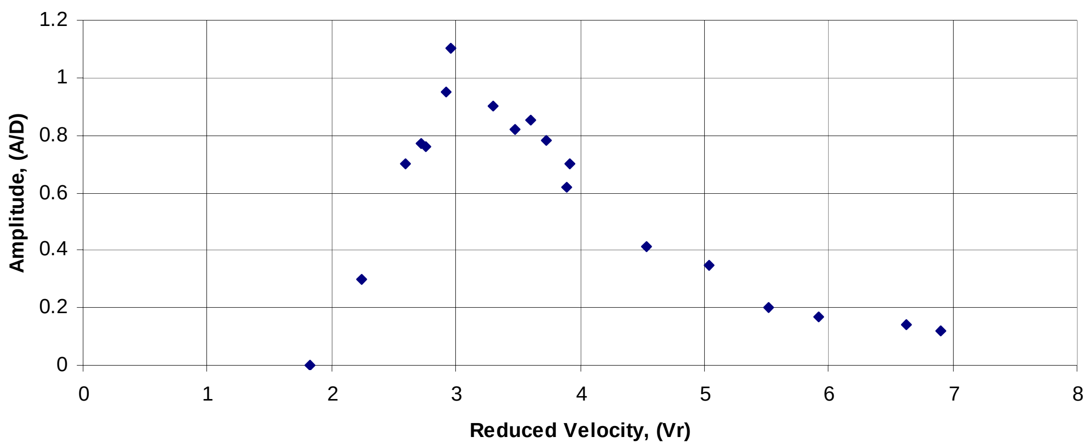

Figure 3. Response amplitude versus reduced velocity for cylinder mounted in elastic base with air bearing system

5. Flexible Blade Base

The mechanism of flexible blades employs springs in the form of thick blades that allow deformations in the elastic regime. One of the great advantages of this mechanism is the absence of slipping, and this characteristic allows a linear behaviour with low structural damping. Another advantage is related to its constructive simplicity, marked for the small number of mechanical components involved.

The base shown in this section was basically composed by two blades of small thickness, the under base and the superior base where they were fixed. The under base is where the cylinder was fixed. The cylinder was partially immersed in the water, the superior base was connected to a fixed table in the channel, as it can be seen in Figure 4. An elastic base with similar design was employed by Fujarra et al. (2001).

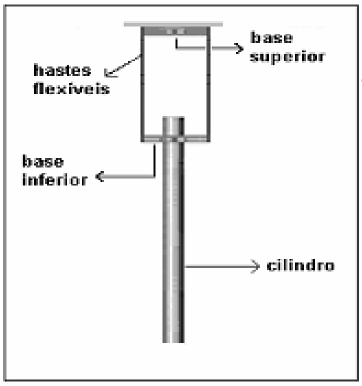

Figure 4. Cylinder fixed on the flexible blade

Considering the absence of mechanical components working for landslide in this base, the low damping requirement of the elastic base is inevitably achieved. With this configuration, innumerable assemblies had been considered according to the variation of dimensions and types of materials used in the mechanical components in order to attend its operating conditions. The choice of the best option happened through the evaluation aspects related with its structure. To determine the dimensions of the blade the theory of resistance of materials was considered to evaluate tensions and buckling.

From the data of the figure bellow it is possible to equate the problem:

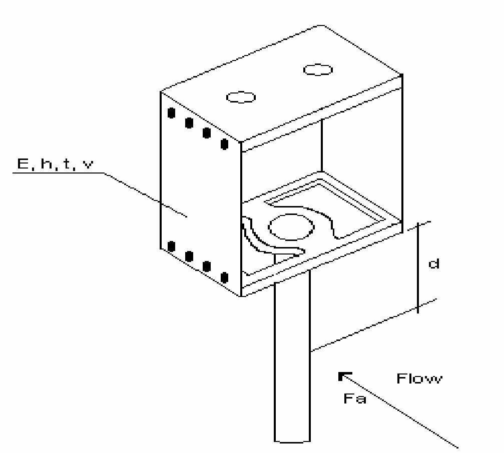

Figure 5. Parameters involved on the structural equation

The parameters shown in figure 5 allows us to develop the governing structural equation of this base.

The drag force is given by

$$F_a = \frac{C_d A\rho . V^2}{2} \quad (8)$$

Knowing the value of the drag force and admitting that it acts in the centre of the immersed part of the cylinder, the fletor moment in each blade can be calculated by

$$M_f = \frac{F_a . d}{2} \quad (9)$$

Since the moment is calculated, the value of the tension that corresponds to the maximum compression is calculated by:

$$\sigma_{max} = \frac{M_f b}{2I} \tag{10}$$

In order to prevent problems due the buckling of the plate, the values of width and length are chosen. The idea is to compare the tension of buckling with the tension caused by the fletor moment.

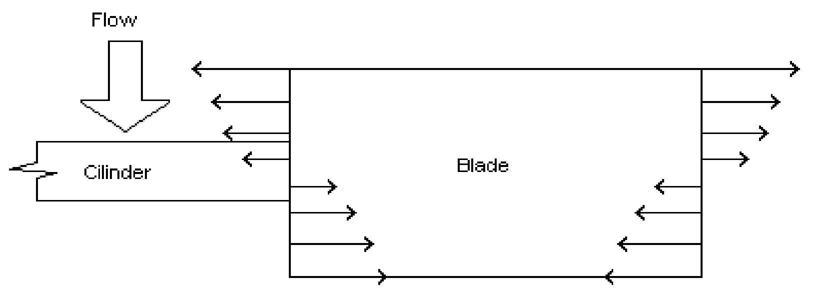

Figure 6. Buckling Illustration.

Such estimate is given by the following equation:

$$\sigma_{flamb} = \frac{K\pi^2 E.t}{12(1-\upsilon)b^2} \tag{11}$$

where K *is* the proportionality constant that depends on the ratio of width b to length a of the blades and can be found in the literature related to the subject. The term υ is the coefficient of Poisson, approximately 0.3 for steel carbon, E is the elasticity constant, generally 210Gpa for steel of high carbon percentage, and t is the thickness of the blades.

The equations above helped to create a spread sheet for diverse dimensions of the requested blade and for maximum readiness of the channel, observing itself if the buckling occurs. After the design and the construction of the base it was necessary to make a calibration to initiate the experiments. Finished its calibration the first set of experiments started. The essays consisted of finding an average of the maximum amplitude of oscillation (*A/D*) for different values of reduced velocities. With the value of the outflow and natural frequency in air (1,9531Hz), the reduced velocity (*Vr*) was evaluated for each value of current velocities. The critical damping in air was equal to 0.0075 and the mass ratio was equal to 1.91. Doing so, the graph depicted in figure 7 was obtained.

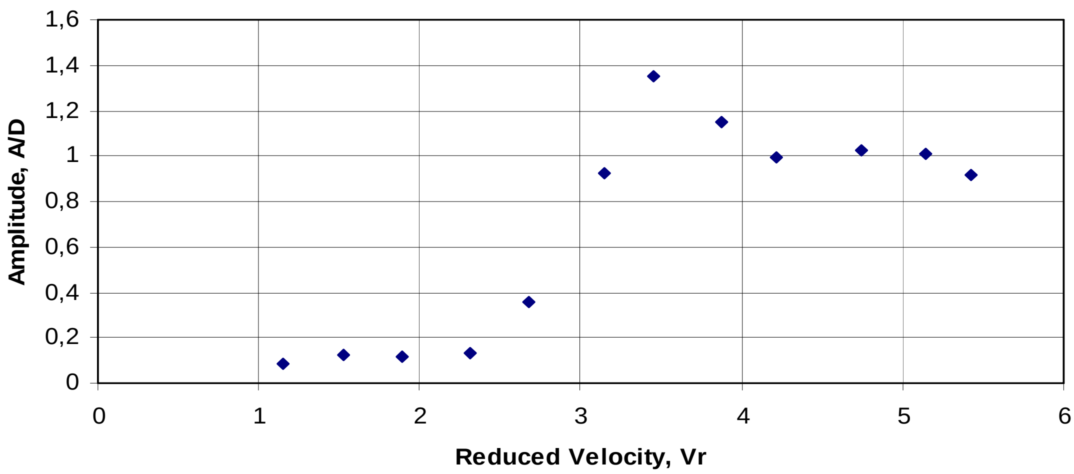

Figure 7. Response amplitude versus reduced velocity for cylinder mounted in flexible blade base. The maximum amplitude observed was slightly around 1.4 diameters and for a reduced velocity around 3.5.

6. Conclusions

The design process of two elastic bases had been presented in this paper together with the advantages and disadvantages of each one of them. The base formed for flexible blades presents the advantage of the low mass ratio. This kind of base, however, can suffer torsion if a high drag forces is acting of the cylinder. This base does not alow testing in high reduced velocity range. The air bearing base, in its turn, has a substantially superior mass ratio. Its main advantage is related to its high torsional stiffness and the easiness of measuring directly the oscillation amplitude through optical or inductive sensors. The two bases described in this article had already been assembled at the NDF (EPUSP) and are being deployed in VIV experiments.

7. Acknowledgements

The authors would like to acknowledge the support of CNPq – Conselho Nacional de Desenvolvimento Científico e Tecnológico, FAPESP – Fundação de Amparo à Pesquisa do Estado de São Paulo e FINEP - Financiadora de Estudos e Projetos, and Petrobras. The development of the research presented in this paper would not be possible without their support. The authors would also like to acknowledge the commentaries made by Professors Celso P. Pesce, André Fujarra, Clóvis A. Martins and José A. P. Aranha.

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