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Suppression of vortex-induced vibration of a circular cylinder
with fixed and rotating control cylinders

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**Suppression of vortex-induced vibration of a circular cylinder
with fixed and rotating control cylinders**

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To my family, for their
unconditional support.

“Militia est vita hominis super terram”

Job 7, 1

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RESUMO

A indústria *offshore* está envolvida no desenvolvimento de novas plataformas flutuantes como Spar, semi-submersível, TLP, FPSO e monocoluna para a exploração de águas profundas e ultra-profundas. Alguns destes sistemas flutuantes têm seções transversais circulares (ou de outras seções rombudas) sendo susceptíveis à vibrações induzidas por vórtices (VIV).

A esteira de vórtices desprendida de um corpo rombudo pode ser alterada ou suprimida ao longo de uma faixa de número de Reynolds. Várias técnicas de controle do escoamento foram sugeridas e testadas em geometrias simples, resultando na redução de forças de sustentação e arrasto. Um desses métodos é o controle de camada limite por superfícies móveis (CCLSM), no qual cilindrinhos rotativos de controle são colocados próximos ao corpo rombudo. Neste trabalho, este método foi abordado através de uma investigação experimental como um supressor de VIV para o escoamento omnidirecional.

Neste escopo três diferentes configurações foram montadas para comparar o efeito de supressão sobre um cilindro liso rodeado por dois, quatro e oito cilindros de controle, distribuídos simetricamente em torno dele. Foram realizados ensaios com o modelo estático, ensaios de VIV em um grau de liberdade com cilindros de controle fixos e rotativos. Foram medidos deslocamento e forças de sustentação e arrasto.

Os resultados mostraram que a posição dos cilindros de controle é um parâmetro importante para a supressão de VIV. A configuração com dois cilindros de controle aumentou as forças de sustentação e arrasto. Diferentemente, as configurações de quatro e oito cilindros de controle mostraram-se mais eficazes para suprimir VIV. Além disso, todos os casos da configuração de oito cilindros de controle fixos apresentaram redução nas amplitudes de vibração e nas forças de sustentação e arrasto, quando comparados com um cilindro liso. No entanto, quando os cilindros de controle foram acionados para rotacionar, mostrou-se um aumento na força de arrasto em relação aos cilindros de controle fixos.

Palavras-chave: *Escoamento ao redor de cilindros, Vibração induzida por vórtices, Supressores de VIV, Controle de camada limite, Cilindros de controle*

ABSTRACT

The offshore oil industry is engaged in the development of new floating platforms, such as Spar, semi-submersible, tension-leg, FPSO and monocolumn for the exploration of deep and ultra-deep waters. Some of these floating systems have circular cross sections (or cross sections of other bluff geometries) being susceptible to vortex-induced vibrations (VIV).

Vortex shedding behind a bluff body can be altered, suppressed or controlled over a limited range of Reynolds numbers. Various flow-control techniques, which result in the reduction of drag and unsteady forces, have been suggested and tested in simple geometries. One such method is the *moving-surface boundary layer control* (MSBC), in which smaller control rotating cylinders are placed close to the bluff body. This method is considered as an inspiration for the present experimental investigation of VIV suppression for omni-directional flows.

In this context, three different configurations have been assembled to compare the effect of suppression on a plain cylinder surrounded by two, four and eight control cylinders distributed symmetrically around it. Experiments were carried out with static models and models free to oscillate in one-degree-of-freedom with fixed and rotating control cylinders. Experiments with a plain cylinder were performed to serve as reference. Displacements, drag and lift forces were measured.

The position of the control cylinders proved to be an important parameter to VIV suppression. Configurations with two control cylinders increased lift and drag forces. In contrast, configurations of four and eight control cylinders showed to be more effective to suppress VIV. Furthermore the results for all the cases of the configuration of eight fixed control cylinders presented a reduction of displacement amplitude, lift and drag forces when compared to a plain cylinder. However, when the control cylinders were actuated, the two cases with rotating control cylinders increased drag force when compared to fixed control cylinders.

Keywords: *Flow around circular cylinders, Vortex-induced vibration, VIV Suppressors, Boundary layer control, Control cylinders*

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LIST OF ABBREVIATIONS

FFT	Fast Fourier Transform
FPSO	Floating Production Storage and Off-loading
MSBC	Moving Surface Boundary-layer Control
NOAA	National Oceanic and Atmospheric Administration (USA)
NDF	Dynamics and Fluids Research Group (USP)
PIV	Particle-image Velocimetry
RMS	Root Mean Square
USP	University of São Paulo
VIV	Vortex-induced Vibration

LIST OF SYMBOLS

A^*	Displacement of amplitude ratio: $A^* = \hat{y}/D$
c	Structural damping [Kg/s]
$C_D, \overline{C_D}, \hat{C}_D$	Total, steady and fluctuating drag coefficients
$C_L, \hat{C}_L$	Total and fluctuating lift coefficients
D	Diameter of the main cylinder [m]
d_c	Diameter of the control cylinders [m]
f	Frequency of oscillation [1/s]
f_0	Natural frequency in air [1/s]
f_n	Natural frequency in water [1/s]
f_s	Frequency of vortex shedding [1/s]
$F_D, \overline{F_D}, \hat{F}_D$	Total, steady and fluctuating drag force [N]
$F_L, \hat{F}_L$	Total and fluctuating lift force [N]
G	Gap between cylinder and control cylinder [m]
k	Structural stiffness [N/m]
L	Underwater cylinder length [m]
m	Structural mass [Kg]
m^*	Mass ratio: $m^* = 4m/\rho\pi D^2L$
Re	Reynolds number: $\text{Re} = UD/\nu$
S_G	Skopp-Griffin parameter
St	Strouhal number: $\text{St} = f_s D/U$
t	Time [s]
T	Period of oscillations [s]
U	Free stream velocity [m/s]
U_c	Tangential velocity on the control cylinders [m/s]

U_R	Reduced velocity: $U_R = U/Df_0$
$\hat{y}$	Harmonic amplitude of displacement [m]
$y, \dot{y}, \ddot{y}$	Cylinder displacement [m], velocity [m^2/s] and acceleration [m/s^2]
ϕ	Phase angle between y and C_L and displacement [degrees]
ν, μ	Kinematic [m^2/s] and dynamic [Kg/ms] viscosity of water
ρ	Specific mass of water [Kg/ m^3]
ζ	Damping ratio

1 INTRODUCTION

The last decades have been marked by a considerable development of offshore oil and gas activities. Because of an increasing energy demand and technological innovation, drilling activities extended and moved into deep (between 500 to 1499 meters) and ultra-deep (more than 1500 meters) water areas. As of today, almost a third of the oil and a quarter of the natural gas consumed in the world come from underwater areas and this rush to offshore oil and gas exploration and exploitation is not about to end (ROCHETTE et al., 2014). In this context the conventional platform, jacket structures are totally obsolete now. As illustrated in Figure 1.1. production facilities on any of these deepwater locations are expected to be a combination of subsea and floating systems such as spar, semi-submersible, tension-leg, FPSO (floating production, storage and offloading system) and monocolumn floating units.

The principal challenge goes to the operation of such a massive dynamic system under severe conditions of wind, sea currents and waves. Most of marine structures which have bluff cross section (risers, umbilicals, mooring lines and hulls) can be subjected to current loads sufficient to produce vortex-induced vibrations (VIV). Basically, as water flows around them it excites the structure. The cross-flow and stream-wise vibrations caused by the shedding of vortices will increase the level of stresses in the structure, contributing to the reduction of its operational life and, in some cases, resulting in fatigue failure.

Vortex-induced vibration is a self-excited, self-limited vibration governed by the hydroelastic interaction between the flow and the structure. VIV is a phenomenon which is well recognized, but for design purpose is not easily predicted and difficult to control in certain types of systems. The importance of scientific and technological applied studies of VIV requires the development of experimental and numerical modeling for prediction and better understanding of the hydrodynamic characteristics involved in this phenomenon. Attenuation and/or suppression of VIV is a problem of great appeal not only in the offshore industry but clearly in many other areas of engineering where the fluid-structure

Figure 1.1: Types of offshore platforms employed for oil exploration in different water depths. Adapted from NOAA Ocean Explorer: Expedition to the Deep Slope.

interaction of bluff bodies is found. A great deal of resources are currently been invested in the study and development of VIV suppressors (see Allen et al. (2004), for example).

Prevention of vortex-induced oscillations can be done by changing structural parameters, for example by increasing either the natural frequency or energy absorption. The former can be accomplished by increasing the stiffness or redistributing the mass, the latter by using various kinds of structural dampers. Regarding the hydrodynamics, the passive techniques are dependent on modifications of the bluff body geometry which affect the formation and shedding mechanism of vortices. Zdravkovich (1981) classified the techniques into three categories; surface protrusions, shrouds and near wake stabilizers.

Such passive methods require no external energy supply to the fluid structure system and have a simpler implementation. A good example of the first case (surface protrusions) is the helical strakes, one of the most common attenuator employed both in air and water flows. Strakes are particularly effective for omni-directional currents. Despite the proven efficiency of the strakes as VIV suppressors, they increase the mean drag (KORKISCHKO; MENEGHINI, 2011; ZDRAVKOVICH, 1981). The second (shrouds) type basically surrounds the structure with a perforated cylinder, axial rods or slats. The third type (near wake stabilizers) such as splitter plates, works close to the structure and are intended to change the vortex shedding mechanism in the near wake

Another method to suppress the regular vortex shedding and, consequently, to modify the cylinder wake is the addition of small control cylinders strategically located in the near wake, as investigated by Alam et al. (2003), Kuo et al. (2007), Dipankar

et al. (2007) and Korkischko & Meneghini (2012). This wake control method is known as *moving surface boundary-layer control* (MSBC). With few literature studies based on two-dimensional numerical simulations of stationary cylinders. Korkischko & Meneghini (2012) performed an experiment employing MSBC to suppress VIV of an isolated cylinder mounted on an elastic base with one degree of freedom in the transverse direction. Hence, MSBC has already been tested as a means to suppress vortex shedding of static cylinders as well as VIV of oscillating bodies. The smaller control rotating cylinders inject angular or linear momentum in the boundary layer of the main cylinder leading to vortex suppression and consequently drag decrease as well as reduction of the transverse velocity fluctuations in the wake.

1.1 Objectives

In this sense, the present work aims to conduct an experimental study about the VIV suppression motivated by the control rotating cylinders and the axial rods surrounding the structures, to attend the demands of offshore industry for large floating offshore structures. The experimental investigation will be performed with a one-degree-of-freedom cylinder fitted with suppression devices. In this context, the main goals are:

- To understand the physical aspects of the fluid-structure interaction with three different configurations of the main circular cylinder with diameter D , surrounded by two, four and eight small cylinders distributed symmetrically around it. For each arrangement the main geometrical parameters to be analyzed are the ratio between the diameter of the small cylinders and the diameter of the master cylinder (d/D) and the spacing (gap) between the wall of the control cylinders and the wall of the master cylinder (G/D), as illustrated in Figure 1.2
- To investigate the effect of eight small rotating cylinders as an omni-directional VIV suppressor.
- To characterize the VIV responses for each configuration by analyzing the results of displacements and frequencies of oscillation and fluid forces acting on the cylinder. Also, to understand the hydrodynamic mechanisms responsible for the attenuation of VIV with axial rods in the form of fixed and rotating control cylinders.

Figure 1.2: Geometrical parameters for the three configurations of control cylinders.

2 LITERATURE REVIEW

This chapter begins with a brief review of the phenomena of flow separation and vortex shedding of circular cylinders. Next it briefly discusses the VIV (vortex-induced vibration) mechanism and phenomenon, identifying the parameters that have traditionally been used to study VIV. Finally it describes studies on the suppression of VIV.

2.1 Flow around bluff bodies

The flow around a bluff body, such as a circular cylinder, is characterized by large zones of separation. An adverse pressure gradient in the direction of the flow, imposed by the geometry, causes the boundary layer to separate from the surface of the cylinder generating a recirculation in the flow. The wake of a cylinder is formed as a consequence. In this section there is a brief summary, of how the wake of a circular cylinder evolves as a function of different parameters. An excellent and extensive description of the flow around a cylinder can be found in Zdravkovich (1997b).

2.1.1 Regime of flow

The dimensionless hydrodynamic number that is useful in the description of the flow around a smooth circular cylinder is the Reynolds number (Re). By definition, the Reynolds number is the ratio of the inertial to viscous forces, formulated as

$$\text{Re} = \frac{DU}{\nu} = \frac{\rho DU}{\mu}, \quad (2.1)$$

in which D is the diameter of the cylinder, U is the flow velocity, ρ is the density, μ the dynamic viscosity of the fluid and ν is the kinematic viscosity.

Table 2.1: Flow regime around a smooth circular cylinder, adapted from Zdravkovich (1997b).

STATE	Re RANGES	REGIME	
Laminar	$Re < 5$	No separation	
	$5 < Re < 47$	Steady separation	
	$47 < Re < 200$	Periodic laminar	
Transition in wake	$200 < Re < 250$	Far wake	
	$250 < Re < 400$	Near wake	
Transition in shear layers (Subcritical)	$400 < Re < 2 \times 10^3$	Lower	
	$2 \times 10^3 < Re < 4 \times 10^4$	Intermediate	
	$4 \times 10^4 < Re < 2 \times 10^5$	Upper	
Transition in boundary layers	$2 \times 10^5 < Re < 3.4 \times 10^5$	Precritical	
	$3.5 \times 10^5 < Re < 4 \times 10^5$	One-bubble (Critical)	
	$4 \times 10^5 < Re < 1 \times 10^6$	Two-bubble (Critical)	
	$1 \times 10^6 < Re < 3 \times 10^6$	Supercritical	
	$3 \times 10^6 < Re < \textit{Unknown}$	Postcritical	
Fully turbulent	$\textit{Unknown} < Re < \infty$	Invariable	

Different flow regimes appear as the Reynolds number changes, generating separated flow regions in the wake of the cylinder, thus producing vortices. This hydrodynamic mechanism of vortex shedding is illustrated in Table 2.1 as Re increases. At low values of Re ($Re < 5$), viscous effects are dominant in the flow dynamics near the body, resulting in a streamlined flow around the cylinder in which the vorticity produced at the boundary layer is totally dissipated in the vicinity of the body. The flow profile downstream of the cylinder body is again uniform and the flow pattern does not vary with time. For $Re = 5$, separation begins and a closed near wake is formed with a weak recirculation bubble consisting of two symmetric eddies. The shear layers meet at a point named stagnation point, closing to the near wake region. When the Reynolds number reaches 47 an instability in the near wake leads to an oscillation of the shear layers at the neutral point. A common feature for circular cylinder flows with Reynolds numbers greater than 47, is vortex shedding and the formation of a Kármán vortex street.

Vortex shedding is the result of an interaction between the shear layers in the near body wake involving the vorticity inherent to them. It occurs at a certain frequency, which is called the vortex shedding frequency (f_s). This frequency normalized with the flow velocity U and the cylinder diameter D results in the Strouhal number, define as

$$St = \frac{f_s D}{U} \quad (2.2)$$

Experiments from different researches have shown that the frequency of vortex shedding in the wake of a cylinder is a function of the Reynolds number, the roughness of the cylinder surface, the turbulence intensity in the incoming flow, the aspect ratio, etc. Figure 2.1 shows the variation of St with Re . The product between the shedding frequency and the flow speed divided by the diameter of the cylinder is practically constant in the subcritical regime, with a value of $St \approx 2$.

As Re is further increased, a transition from a laminar to turbulent flow regime will start to happen. First the wake further downstream of the cylinder becomes turbulent, but with an additional increase in Re the start of a turbulent regime travels upstream reaching the free shear layers and eventually affecting the boundary layers. Turbulent boundary layers have more kinetic energy close to the wall and are able to resist longer the effect of the adverse pressure gradient.

At the supercritical flow regime, the boundary layers on both sides of the cylinder are turbulent at the separation point. Consequently, the boundary layer separation is

Figure 2.1: Strouhal number versus Re for a smooth circular cylinder. Adapted from Norberg (2001)

delayed since the separation point moves downstream. At this point, the vortices are close to each other and shed at a faster rate than the rate in the subcritical regime, thereby leading to higher values of the Strouhal number. This results in a complex and not so well-defined evolution of the Strouhal number. However for postcritical Reynolds numbers above 3×10^6 a fully turbulent vortex wake reappears (as in the flow illustrated in Table 2.1).

2.1.2 Mechanism of vortex formation

The concept of boundary layer introduced by Prandtl (1904) states that an effect of friction was to cause the fluid immediately adjacent to the surface to stick to the surface, and that frictional effects were experienced only in a boundary layer. The boundary layer around a circular cylinder separates due to the adverse pressure gradient imposed on the flow by the geometry of the body, as can be seen in Figure 2.2. The angle of separation of the boundary layer, which is measured along the surface of the cylinder from the front stagnation point, changes depending on the transition from laminar to turbulent flow in the boundary layer, which depends on the Reynolds number.

A physical description of the mechanics of the vortex-formation region has been proposed by Gerrard (1966). He suggests that the mutual interaction between two separating shear layers is a key factor in the formation of a vortex street. Gerrard postulated that a vortex grows by gaining circulation from a connected separated shear layer. At

Figure 2.2: Characteristics of the boundary layer of a curved surface. Adapted from Fielding (2005).

a certain instant, the growing vortex is strong enough to draw the opposing shear layer across the near wake. The approach of oppositely signed vorticity, in sufficient concentration, cuts off further supply of circulation to the growing vortex, which is shed and convected downstream.

In more detail, Gerrard (1966) described the mechanisms of formation of vortices in the wake illustrated in Figure 2.3, where instantaneous filament lines are shown. He suggested that the fluid particles of the opposite shear layer drawn across the wake can follow one of the routes: a) can be entrained into the growing vortex thus reducing its strength, b) can find their way into the shear layer with vorticity with opposite sign, and c) can find their way into the near-wake region rolling to form the next vortex. The balance of the amount of fluid that follows these routes controls the shedding frequency, the strength of the vortices and the base pressure.

Figure 2.3: Scheme of the formation of the vortex wake of a circular cylinder. Adapted from Gerrard (1966).

2.1.3 Drag and Lift forces

As the result of the periodic change of the vortex shedding, the pressure distribution around the cylinder due to the flow will also change periodically, thereby generating a periodic variation in the force components acting on the cylinder. The hydrodynamic force can be divided into cross-flow and in-line components. The force of the cross-flow direction is commonly named as the lift force (F_L) while the latter is named as the drag force (F_D). The lift force appears when the vortex shedding starts to occur and it fluctuates at the vortex shedding frequency. Similarly, the drag force also has the oscillating part with twice the frequency due to the vortex shedding, but, in addition, it also has a non-zero mean force as a result of friction and pressure difference; this part is called the mean drag. Both lift and drag can be modeled as harmonic functions, such as

$$F_L = \hat{F}_L \sin(2\pi f_s t) \quad (2.3)$$

$$F_D = \overline{F_D} + \hat{F}_D \sin(2 \cdot 2\pi f_s t), \quad (2.4)$$

where $\hat{F}_L$ and $\hat{F}_D$ are the amplitudes of the oscillating lift and drag respectively and $\overline{F_D}$ is the mean drag.

C_D and C_L are the dimensionless coefficients for drag and lift respectively, and can be derived as:

$$\hat{C}_L = \frac{\hat{F}_L}{1/2\rho LDU^2} \quad (2.5)$$

$$\hat{C}_D = \frac{\hat{F}_D}{1/2\rho LDU^2} \quad (2.6)$$

$$\overline{C_D} = \frac{\overline{F_D}}{1/2\rho LDU^2} \quad (2.7)$$

where ρ , L , D and U are the fluid density, cylinder length, cylinder diameter and flow velocity, respectively.

According to Norberg (2001), Williamson (1996) and Zdravkovich (1997a) the value of $\hat{C}_L$ is defined as the RMS (root mean square) of force. Figure 2.4 shows the values of drag and lift coefficients for a wide range of Re, covering the transition from

laminar to turbulent regime in the wake, shear layers and boundary layers. $\overline{C_D}$ curve represents the time-average drag coefficient acting on the cylinder, which is divided into two terms according to their nature: friction drag $\overline{C_{Df}}$ and pressure drag $\overline{C_{Dp}}$. As Re increases, the frictional component tends to decrease in relation to the component form. In the subcritical regime $\overline{C_D}$ stabilizes close to 1. When Re is on the critical regime a drag crisis occurs resulting in a sudden drag reduction on the smooth cylinder. After the supercritical regime, $\overline{C_D}$ stabilizes at values less than 1, smaller than the subcritical regime, due to changes in separation point, as discussed in Section 2.1.1.

Figure 2.4: Variation of lift and drag coefficients with Reynolds number for a circular cylinder. $\overline{C_D}$ represent mean drag; $\hat{C}_L$ and $\hat{C}_D$ represent the magnitude of fluctuating lift and drag. Reproduced from Assi (2005), adapted from Zdravkovich (1997b).

Due to the symmetry of the flow, the mean lift coefficient $\overline{C_L}$ is zero, while $\hat{C}_L$ is significant from the beginning of the vortex shedding in the laminar regime. $\hat{C}_L$ is reduced in the transition from laminar to turbulent in the wake, and starts to increase until $\hat{C}_L \approx 0.6$ in the critical regime. During the critical regime, $\hat{C}_L$ decreases returning in values close to 0.2 in the supercritical regime. However, it is necessary to point out that there is a wide dispersion in $\hat{C}_L$ values found in the literature. Norberg (2001) experimentally measured this fluctuating lift on a circular cylinder and his results are shown in Figure 2.5, showing a considerable scatter of data.

Figure 2.5: Root-mean-square fluctuating lift coefficient, $\hat{C}_L$ for a stationary circular cylinder as a function of Reynolds number, Re . Reproduced from Norberg (2001).

2.2 Vortex Induced Vibration

For a fixed rigid body, vortex shedding is easily described. However, vibrations of flexible structures induced by transverse wind or ocean current forces are observed in numerous occasions. Therefore, a totally flexible body or a rigid body with flexible support should be considered. Vibrations of a body due to vortices lead to a continuous change in the boundary conditions of the flow and vortices, which may generate a more complex problem due to the interaction between dynamic flow and vibrating structure.

Vortex shedding in the wake of a stationary circular cylinder occurs for Reynolds number greater than 47. Under this situation, periodic shedding is formed downstream of the cylinder, creating a situation in which alternating fluid forces are generated on the cylinder and if it is flexibly mounted, vortex-induced vibrations (VIV) can occur. Once the oscillations starts, the flow is modified and the excitation affecting the moving cylinder changes, which again modifies the response and so on. This feedback process between the structure and the flow, leads to a very complex non-linear interaction. Extensively review works on VIV have been published by Sarpkaya (1979), Bearman (1984), Parkinson (1989), Williamson & Govardhan (2004).

Investigations on VIV problems are based on experimental, theoretical, numerical and a combined theoretical-experimental analysis. The VIV experimental study in cylinders can be addressed, according to the number of degrees of freedom, the cylinder type (rigid or flexible) and the type of vibration (forced or free). Each of these methods involves different physical mechanisms and requires different theoretical approaches. De-

scribing all of them is beyond the scope of this text. In this work the focus will be only on elastically mounted rigid cylinder with 1 degree of freedom in the transverse direction to the flow.

2.2.1 Elastically mounted cylinder

A free vibration of an elastically mounted cylinder is represented in Figure 2.6 by an idealized description of a vibrating structure with properties of mass (m), stiffness (k) and damping (c). Due to the cyclic nature of vortex shedding in the near wake the body experiences fluid force fluctuations that can excite the system into VIV. Allowing for displacements only in one degree of freedom (1-dof) in the y-axis (cross-flow direction), the equation of motion for a harmonic oscillator can be modeled by

$$m\ddot{y} + c\dot{y} + ky = F_L(t) \quad (2.8)$$

$$y(t) = \hat{y} \sin(2\pi ft) \quad (2.9)$$

$$F_L = \hat{F}_L \sin(2\pi ft + \phi) \quad (2.10)$$

where y , $\dot{y}$ and $\ddot{y}$ are respectively the displacement, velocity and acceleration of the body and $F_L(t)$ is the time-dependent fluid force in the cross-flow direction. $\hat{y}$ and $\hat{F}_L$ indicate the harmonic amplitudes of y and F_L , f is the oscillation frequency and ϕ is the phase angle between the displacement and the fluid force. For body excitation to occur, the phase angle between $y(t)$ and $F_L(t)$ must be between $\phi = 0^\circ$ and 180° . A phase angle equal either to 0° or 180° means that no energy is transferred from the fluid to the structure to excite any vibration. The inertial force is $m\ddot{y}$, the viscous damping force proportional to the velocity of the body and associated with the structure is $c\dot{y}$, the restoring force is ky .

The range of flow velocities over which high-amplitude oscillations occur, called the lock-in domain, depends on the mass of the cylinder, generally presented in its non-dimensional form m^* , defined as the ratio of the mass of the cylinder m to the mass of displaced fluid:

Figure 2.6: Sketch of a cylinder elastically mounted in 1-dof.

$$m^* = \frac{m}{\rho\pi\frac{D^2}{4}L} \quad (2.11)$$

The amplitude of oscillation in the lock-in range depends also on the damping of the system, appearing in the damping ratio ζ , defined as the ratio of the structural damping c to the critical damping of the system:

$$\zeta = \frac{c}{2\sqrt{k(m + m_a)}} \quad (2.12)$$

When a structure starts to vibrate due to VIV, the reduced velocity

$$U_r = \frac{U}{Df}, \quad (2.13)$$

where U is the free-stream velocity, D the cylinder diameter and f the natural frequencies for the structure f_0 for air or f_n to water, becomes a useful parameter. Vortex shedding can appear for all reduced velocities, often affected by the amplitude of the vibration. It has been seen that the vibration amplitude reaches a maximum at resonance but is sustained for a wider range of reduced velocity.

In order to evaluate the response the non-dimensional amplitude of vibration is defined as,

$$A^* = \frac{\hat{y}}{D} \quad (2.14)$$

where $\hat{y}$ is the amplitude of vibration and D the cylinder diameter. The non-dimensional frequency response of the system is

$$f^* = \frac{f}{f_n} \quad (2.15)$$

where f_n is the system natural frequency measured in water.

2.2.2 Typical VIV response

A classic experiment reviewing the behavior of a flexibly mounted cylinder was carried out by Feng (1968). Feng analyzed a system with a high mass ratio ($m^* \simeq 250$), immersed in air, by measuring the amplitude response of oscillation, shedding frequencies and surface pressures. Feng increased the speed of the flow in small steps to observe the response of the circular cylinder due to the current. In the response range, two separate branches could be observed, one achieving higher amplitudes than the other. The frequency of vibration was very close to the natural frequency of the system in air, in the whole range where non-zero response was found. Vortex shedding frequency followed the Strouhal relationship for a stationary cylinder, except in the range of reduced velocity that goes from 5 to 7, where the frequency of oscillations and the frequency of vortex shedding were equal and near the natural frequency of the system. This phenomenon, where oscillation frequency and vortex shedding frequency were roughly equal, was called lock-in, and it was where the higher amplitude responses were observed. The lock-in is also known as resonance, synchronization or wake capture. The data is shown in Figure 2.7 as appeared in Parkinson (1989)

Recent studies by several authors exploring low mass and damping cylinders, Khalak & Williamson (1996) and Brankovic (2004), have confirmed how bodies can oscillate with large amplitudes even though the oscillation frequency is not close to the natural frequency of the cylinder, changing the classic definition of lock-in, adopted by Sarpkaya (1996) in which the frequency of vortex shedding has to be equal to that of the oscillation of the cylinder (f). Under this situation, the excitation lift force is in phase with the motion, and it fluctuates at the same rate as the structure ($f \approx f_s \approx f_n$).

Williamson & Govardhan (2004) identified the different vortex structures in the

Figure 2.7: Transverse response of a freely vibrating cylinder in air from Feng's experiment. Reproduced Parkinson (1989).

wake of an oscillating cylinder and they associated different patterns to each response branch found in amplitude against reduced velocity diagrams. What they named the 2S (2 single vortices per cycle) was initially found in free vibration of the cylinder. A second pattern called 2P (2 pairs of vortices per cycle) appeared in free and forced vibrations and a third pattern, the P+S (1 single + 1 pair of vortices per cycle), appeared initially only in forced vibrations. They defined a vortex mode map, shown in Figure 2.8, in which the vortex structures in the wake of a circular cylinder appear as a function of the reduced velocity and the response amplitude.

Figure 2.8: Vortex patterns in the wake of circular cylinder, from Williamson & Govardhan (2004).

They compared the response of high mass ratio systems, using Feng's data, against their low mass-damping data. Figure 2.9, shows how in the low mass ratio system, the synchronization range was 4 times larger, and the oscillation frequency did not match the natural frequency, it was 1.4 times higher in the lower branch (for a $m^* = 2.4$). They classified three different branches of response for low mass and damping cylinders: an initial branch in which the oscillation starts to build up; an upper branch of maximum amplitude around the resonance peak; and a lower branch that persists until the oscillations eventually die out. There was a hysteretic behavior between the initial and the upper branch, but between the upper and the lower branch there was a intermittent change. The 2S mode was found in the initial branch and the 2P in the upper and lower branches. The jumps between the three branches were associated with changes in the phase angle between the motion and the excitation force.

Khalak & Williamson (1999) concluded that the maximum vibration amplitude is related to the product $m^*\zeta$, while the width of the range U_r where synchronization occurs is directly related to the value of m^* for cases of the same $m^*\zeta$, as can be observed in Figure 2.10. The increase in the reduced mass causes a decrease in synchronization width, so that for high values of m^* there are only two branches response. This justifies the difference between the results presented in Figure 2.9, because Feng's experiment

(1968) was conducted with m^* approximately 25 times higher than Khalak and Williamson (1999).

Figure 2.9: Comparison of low (Williamson) and high (Feng) mass ratio data from Williamson & Govardhan (2004).

Figure 2.10: The two distinct types of amplitude response are shown here schematically. (Vertical axes represent A^* and horizontal axes represent U_r), from Khalak & Williamson (1999)

The maximum amplitude achievable by a certain system is something that has captured the interest of researches working on VIV. Griffin & Ramberg (1982) published a work where the relationship between the maximum amplitude recorded for a system was plotted against a parameter called reduced damping which was based on the mass and the damping ratio of the system.

$$S_G = 8\pi^2 \text{St}^2 m^* \zeta \quad (2.16)$$

The amplitude of the response decreases with increasing S_G until motion disappears. An extensive compilation of non-dimensional maximum amplitude response data from several authors is presented in Williamson & Govardhan (2004). In Figure 2.11 data obtained from cylinders in air and water is presented.

Figure 2.11: Griffin plot in function of S_G . Reproduced from Williamson & Govardhan (2004).

2.3 Suppression of VIV

Vortex shedding could cause serious structural vibrations, acoustic noise and resonance, enhanced mixing, and significant increases in the mean drag and lift fluctuations. Therefore, the effective control of vortex shedding is important in engineering applications.

Blevins (1990) suggests alternatives to prevent vortex induced oscillations by increasing $m^*\zeta$, avoid resonance and add suppression device. For the first one, suppression can be accomplished by increasing the stiffness or redistributing the mass or employing materials with high internal damping or installing external dampers. The last one becomes an option when the structure cannot be modified accordingly and the installation of add-on devices to try to suppress the vibrations is advisable. According to Choi et al. (2008), these flow control mechanisms for bluff bodies can be classified as passive, active open-loop and closed-loop.

2.3.1 Passive control

Basically, the suppression of VIV starts with the disruption of vortex shedding from the body. Zdravkovich (1981) presents a discussion of several of the passive control devices presented in Figure 2.12. Solutions were classified into three categories according to the way they affect the shedding mechanism: (i) Surface protrusions which affect separation lines and/or separated shear layers: they involve helical strakes, wires, fins, studs, or

spheres, among others. (ii) Shrouds affect the entrainment layers around the body. The perforated shroud which has the effect of influencing the base pressure and the axial rods are two examples. (iii) Near-wake stabilizers affect the switch of the confluence point. Splitter plate, which prevents communication between the opposing shear layers of the wake, is a common example. These passive methods require no external energy supply to the fluid–structure system and have a simpler implementation.

Figure 2.12: Add-on devices for suppression of vortex-induced vibration of cylinders: (a) Helical strake; (b) Shroud; (c) Axial slats; (d) Streamlined fairing; (e) Splitter; (f) Ribbed cable; (g) Pivoted guiding vane; (h) Spoiler plate. Blevins (1990).

Among the various solutions for passive VIV suppression, the helical strakes are one of the most commonly used in air and water flows. Extensive experimental studies conducted by Bearman & Brankovic (2004), Trim et al. (2005) and Korkischko & Meneghini (2010) investigated the effects of the main geometric features (pitch and height) in VIV attenuation/suppression. Despite the proven efficiency of the strakes as VIV suppressors, they increase the mean drag (KORKISCHKO; MENEGHINI, 2011; ZDRAVKOVICH, 1981), which is undesirable in a great number of applications. This fact leads to the development of low-drag solution, such as a wake splitter (GROVE et al., 1964; PATNAIK et al., 1996) to improve base suction.

Using an upstream control rod is also a well-established strategy for drag reduction (LEE et al., 2004). Strykowski & Sreenivasan (1990) proved that if the small control

cylinder is placed within a defined region in the near-wake of the main cylinder, the wake behind the main cylinder could be effectively suppressed at a Reynolds number of 80. Suppression of the vortex street is associated with damping the instability in the near-wake region. They also showed that wake suppression is the most efficient when the small control cylinder is placed near the center of this contoured region, as, shown in Figure 2.13.

Figure 2.13: Regions of suppression for four values of D/d . These regions are symmetrical about the plane $y = 0$. $Re = 80$. Strykowski & Sreenivasan (1990).

Their experimental and numerical results also indicate that the domain of this region strongly depends on the Reynolds number as well as the diameter ratio. For the Reynolds number range from 80 to 300, Kuo et al. (2007) showed the detailed flow structures and revealed the primary mechanism that led to significant lift and drag reduction without completely suppressing the shedding of a vortex street.

Zdravkovich (1981) presented results of an axial-rod shroud, following the concept that the shroud should break-up the flow into a large number of small vortices. Axial-rod shroud were fitted about the circular cylinder. Several parameters were varied in order to find an optimum configuration; the porosity varied the numbers of rods (218 to 4), the shroud size D_s/D (1.08, 1.165, 1.25 and 1.50) and the circumferential distribution. The tests were performed in a wind tunnel for $Re = 4.3 \times 10^3$. For each case different values of amplitude were recorded. The most effective shroud geometry was $D_s/D = 1.25$ with a 63% porosity, this configuration was chosen to verify the circumferential distribution, where rods were pulled out. Figure 2.14 presents the effect of the orientation of the incomplete axial-rods shrouds in relation to the free stream on the effectiveness in reducing

the amplitude oscillation, the highest reduction of the amplitude was for the unshrouded portion faced the near-wake.

Figure 2.14: Effectiveness of incomplete axial-rods shrouds. Reproduced from Zdravkovich (1981).

2.3.2 Active control

Active control strategies, as compared to passive, do require power input to achieved the desired suppression. Active strategies are particularly useful in applications where flow conditions change dramatically and rapidly.

Active open-loop control methods have been further and newly developed because they are easier to implement than the active feedback control method, which requires sensing as well as actuation. Nevertheless, many attempts have been also made to improve control efficiency and effectiveness using active feedback control methods with the advent of the micro-electromechanical system, the development of control theory, and the fast growth of computer power.

Active mechanisms generally take the form of suction, jets, or moving surfaces. Wehrmann (1965) and Schumm et al. (1994) have used vibrating cylinders to study the wake vortices. The latter have employed lift force as a feedback signal to control vortex shedding behind a circular cylinder. Glezer & Amitay (2002) employed a control device based on a synthetic jet, which generates vortices at the edge of an orifice by the time-periodic motion of a flexible diaphragm in a sealed cavity. The net mass flux of the synthetic jet is zero, but nonzero linear momentum is transferred to the flow. Besides when a time-periodic open-loop forcing is applied, vortex shedding in the wake is in general locked in phase to the forcing (Blevins 1990); consequently, the forcing strengthens vortex

shedding and increases the mean drag and lift fluctuations.

There are a few successful active open-loop controls that attenuate vortex shedding and reduce drag. One example is the high-frequency rotation of the circular cylinder by Tokumaru & Dimotakis (1991) and another is base bleed (BEARMAN, 1967; WOOD, 1967). Although these controls are effective in reducing drag, their efficiencies are not that high. The moving surface on boundary layers produces significant drag reduction. The next section will describe this type of control, which is employed in this work.

2.3.3 Control of separated flow with rotating cylinder

An active separation control strategy employs the moving surface of a rotating circular cylinder. It is an active open-loop mechanism because the rotation of the cylinders requires power input. The level of control effort is measured by the rotation parameter, which is defined as the ratio between tangential velocity of the moving surface and the flow velocity. (U_c/U).

This method is known as *moving surface boundary-layer control* (MSBC) and relies on the injection of momentum in the boundary layer of the body by rotating elements, which usually are small circular cylinders. The momentum injection in the boundary layer postpones the effects of the adverse pressure gradient generated by the bluff body geometry, moving the separation points of the boundary layer to a more advanced position. As a result, the wake becomes narrower and the recirculation region behind the body is suppressed or drastically reduced (MODI, 1997).

Modi (1997) discusses the various arrangements in which rotating cylinders have been used. His review focused on experiments and simulations that used rotating cylinders in thin airfoils and at the sharp corners of angled bluff bodies at Reynolds numbers up to 4×10^4 . In many cases, cylinders arranged around in larger bodies and partially exposed to the flow while rotating with $U_c/U > 1$ provide increased lift and decreased drag. Some arrangements of cylinder applied to an airfoil are shown in Figure 2.15.

In arrangements in which the small control cylinders are completely external to the primary body, the presence of the control cylinders by itself can significantly alter the wake even without rotation, as presented in Sec. 2.3.1 on the study of Strykowski & Sreenivasan (1990). Kuo et al. (2007) found that locating two small cylinders in the wake of a large main circular cylinder can passively reduce mean drag as well as fluctuating lift and drag.

Figure 2.15: Rotating cylinder applied to airfoils with $U_c/U = 4$, plot to assess relative influence on the lift. Reproduced from Modi (1997).

Mittal (2001) employed numerical simulations to investigate the control of flow past a circular cylinder with two small rotating control cylinders external to the main cylinder, shown in Figure 2.16. Simulations were carried out for two values of Reynolds number ($Re = 100$ and 10×10^4) and gap between the main and control cylinders (gap = $0.01D$ and $0.075D$). He observed that $U_c/U = 5$ resulted in a steady flow at low Reynolds numbers. At $Re = 10 \times 10^4$ the flow remained unsteady, besides the wake was highly organized compared to the one without control. In all the cases, a significant reduction in the overall drag coefficient and the fluctuation lift force acting on the body was observed.

Mittal (2003) investigated the effect of the gap on drag reduction and found that there was a critical gap beyond which greater rotation achieves negligible drag reduction. Muddada & Patnaik (2010) performed further simulations developing this control strategy, which was employed to a circular cylinder fitted with two simple rotary mechanical

Figure 2.16: Diagram of the relative location of the main cylinder and the external rotating control cylinders. Adapted from Mittal (2001).

actuators located at different angles (the ratio of the main cylinder and the control cylinder $D/d_c = 10$ and the gap between them $g = 0.01D$). They found that the optimal angle was located at 120° in relation to the incident flow and presented the best effectiveness in reducing drag, as Figure 2.17 shows.

Figure 2.17: Schematic diagram of the main cylinder with two small rotating control cylinders located at 120° . Adapted from Muddada & Patnaik (2010).

Korkischko & Meneghini (2012) performed an experiment employing MSBC to suppress VIV of an isolated cylinder mounted on an elastic base with one degree of freedom in the transverse direction. The velocity ratio range was $U_c/U = 5$ to 10 , in the range of $Re = 1600$ to 7500 . In all the cases, a significant reduction in the overall drag coefficient and the unsteady hydrodynamic forces acting on the main cylinder were observed. Other experiment performed by Schulmeister (2012) investigated the active flow control with

rotating cylinders at intermediate sub-critical $Re = 5.2 \times 10^4$. The main cylinder was fixed and the rotating cylinders were positioned at 120° similar at the simulation study of Muddada & Patnaik (2010). When the rotation parameter increased for U_c/U greater than 1.0, he found that rotation suppressed vortex shedding from the model and significant drag reduction was realized.

Moving on from these research projects of MSBC method as a suppressor of VIV were essential to determinate all the geometrical parameters to be analyzed as;

- The number of control cylinder and their position around the model.
- The ratio, d_c/D between the diameter of the control cylinders and the diameter of the main cylinder.
- The spacing, G/D between cylinders.
- The velocity ratio, U_c/U .

The next chapter will describe in detail each parameter and the description of the experimental set up of this work.

3 EXPERIMENTAL METHOD

In this chapter details are given of the experimental setup employed to investigate the suppressor's behavior. Experiments were carried out at the Dynamics and Fluids Research Group laboratory (NDF) at the University of São Paulo (USP).

3.1 Experimental setup

A rigid section of circular cylinder was mounted on a one-degree-of-freedom rig fixed to the water channel presented in Figure 3.1. Two, four or eight identical control cylinders were axisymmetrically supported around the main cylinder, as seen in Figure 1.2. Rotating of the case of eight control cylinders was achieved by a servo motor fixed on the channel.

Figure 3.1: Photograph of the experimental setup in apparatus.

Figure 1.2 presented three configurations of models varying the number of control cylinders (also repeated in Table 3.1). Two, four, or eight smaller cylinders with diameter d_c were fitted about the main cylinder with diameter D . The gap between the main cylinder and the control cylinders was set to $G/D = 0.05$, 0.10 and 0.15 , measured from

wall to wall. Table 3.1 summarizes the geometrical parameters considered in the present investigation.

Table 3.1: Geometrical parameters.

Number of control cylinders			
Diameter of control cylinders, d_c/D	0.04, 0.06, 0.08		
Gap between cylinders, G/D	0.05, 0.10, 0.15		
Velocity ratio, U_c/U	-	-	0 - 3

3.1.1 Flow Facility

The test section of the circulating water channel shown in Figure 3.2 has the following dimensions: 0.7m wide, 0.9m deep and 7.5m long. The maximum flow rate in the circuit is $0.4m^3/s$. An axial pump controlled by a frequency inverter ensures the required flow in the circuit with the possibility of varying the flow rate over a wide range. An electromagnetic flow meter provided a reading of the instantaneous flow rate with uncertainty 0.5% of the instantaneous flow. The speed is measured by the ratio between the flow rate and the section area. In this work, flow speed varies from 0.05 to $0.5m/s$. The channel operates at a low turbulence level less than 3%. Sidewalls and bottom of the section were made of glass mounted on a steel frame, allowing access through the free surface and a complete view of the models for flow visualization purposes. Assi (2005) investigated the velocity profile along the height of the channel to verify the influence of the free surface, finding a maximum variation of 5% for the average value of the velocity profile to $0.4m/s$. Hence the free surface does not affect significantly the results due to the long span of the cylinders. Additional information about the recirculating water channel NDF-USP can be found in Assi (2005).

Figure 3.2: Recirculating water channel of the NDF-USP. Reproduced from Assi (2005)

3.1.2 Elastic base

Models were mounted from its upper end to an one-degree-of-freedom elastic mounting, named the 1-dof rig, developed by Assi (2009) for experiments with low mass ratio and damping. Figure 3.3 shows the 1-dof rig installed on the water channel. The cylinder is free to oscillate in the transverse direction of the flow. Two shafts made out of a carbon fiber tube are coupled to four brass aerostatic bearings, which result in low structural damping. They were connected at mid-length by an aluminum support, from which a rigid load cell and models were firmly attached. The low friction is guaranteed by the formation of a film of air through which the shaft slides without direct contact with the bearings. Two springs are responsible for the restoration of the system. The springs must be chosen to ensure that the natural frequency of the system is adequate for VIV resonance to occur within the flow speed range, giving rise to the phenomenon or lock-in. The 1-dof rig mass is 2.8Kg, which allows testing models with low mass ratio m^* . Further details can be found in Assi (2009).

Figure 3.3: 1-dof rig mounted on top of the water channel.

3.1.3 Models

The circular cylinder model is made of rigid acrylic with a smooth surface. The main cylinder external diameter is $D = 100mm$ with an immersed length of $L = 700mm$; the resulting aspect ratio of the model was $L/D = 7$. Two perforated acrylic discs with a diameter of $170mm$ and thickness of $10mm$ were attached to the ends of the cylinder to support the control cylinders, as shown in Figure 3.4. The clearance between the acrylic disc to the bottom of the channel is $3mm$. The main cylinder without any control cylinder, as see in Figure 3.4 is called plain cylinder in the following results.

Figure 3.4: Plain cylinder model.

The control cylinders are made of brass ($d_c = 4mm$) and acrylic ($d_c = 6$ and $8mm$). For those made in acrylic two brass rods of $4mm$ of diameter and $30mm$ long were fixed at the ends, in order to be installed on the acrylic discs of the main cylinder. Two acrylic support discs were installed on the plain cylinder at $L/3$ distances to prevent the control cylinders to vibrate on the first mode. A Figure 3.5 illustrates the three different configuration, each with distinct control cylinder with the supports.

(a) Two control cylinders, $d_c = 8mm$ (b) Four control cylinders, $d_c = 6mm$ (c) Eight control cylinders, $d_c = 4mm$

Figure 3.5: The three model configurations tested of control cylinders.

A pulley system was employed to transmit the rotation from the motor to each of the control cylinders. All pulleys were controlled by one principal double pulley located in a base plate outside the acrylic disc. A flexible shaft connected the double pulley with the servo motor, as seen in Figure 3.6.

The maximum projected area of the model would result in a blockage ratio of 13.8%. Brankovic (2004) performed VIV tests on a single cylinder in a similar water channel with three ratios of blockage: 11.3%, 13.6% and 17%. She concluded that in VIV

Figure 3.6: The pulley system and the connection with the flexible shaft installed in the 1-dof rig.

tests, although the maximum amplitude of oscillation decreased slightly for higher blockage ratios the results remained qualitatively the same, indicating that the hydrodynamic mechanisms have not changed. Thus, the present results were not corrected for blockage effect.

3.1.4 Instrumentation

A load cell was attached between the model and the base to measure instantaneous and time-averaged hydrodynamic forces acting on the cylinder. The cell was designed to measure forces in the stream-wise and cross-flow directions. It was instrumented with strain gauges as Figure 3.7 illustrates. Each set of strain gauges was wired to form a complete Wheatstone bridge, which was connected to a power supply and signal amplifier. This configuration permitted each individual cell to be insensitive to moments but responsive only to the force component acting in its direction. The load cell was designed to measure a load of up to $50N$ with an uncertainty of 1% in each direction. Additional information about the design and construction of the cell can be found Assi (2009)

The transverse displacement of the model is measured by an optical distance sensor, fixed on the 1-dof rig. The sensor resolution is 0.1 mm and the absolute accuracy of measurement is $\pm 2\%$. The operating principle of this sensor is based on triangulation reflection, where a laser diode emits light falling on the surface of the object whose distance is to be measured and the reflection of this beam passes through a set of lenses, focusing on receptors that assess their intensity.

Figure 3.7: Load cell showing a pair of strain gauges installed. Reproduced from Assi (2009)

The tangential velocity of the control cylinders U_c is controlled by a servo motor, illustrated in Figure 3.8. The characteristics are; $750W$, $3000rpm$ and torque of $2.39N \cdot m$ (without break). It was mounted on top of the water channel. The servo control device provides internal parameter function to define rotation and speed.

Figure 3.8: The AC servo motor coupled whit the flexible shaft.

The signal acquisition was performed by a National Instruments system consisting of acquisition boards and LabVIEW code developed at NDF-USP. The program stores all data in a single text file with five columns, relating to time, velocity, displacement and hydrodynamic forces, respectively. It also controls the flow channel and allows the uses to program a series of runs with different speeds without user interaction. The frequency of acquisition was 100 Hz.

3.2 Experiments

Experiments in the present study were performed through 64 runs producing an extensive set of data. Preliminary tests were executed with a plain cylinder to serve as reference for comparison. A description of the methodology used in decay tests for determining the natural frequency and damping coefficient, is also presented. The Figure 3.9 shows a general classification of all the tests performed.

Figure 3.9: Experimental campaign

3.2.1 Fixed Models

The first tests were performed with the models fixed in a static position as the arrangement shown in Figure 3.10, in order to evaluate the hydrodynamic loads on the body, the air bearings of the rig were switched off and the cylinder was not allowed to respond with vibration in any direction. Measurements of time, speed, lift and drag forces were made, in the range of flow velocity $U = 0.05$ to $0.5m/s$, which alters the Reynolds number between 5,000 and 50,000. Data were processed by a routine developed in MatLab program to calculate $\overline{C}_D$ and $\hat{C}_L$. Additional information on the test are shown in Table 3.2.

Figure 3.10: Cross-sectional view of the water channel and 1-dof rig used in the static tests.

Table 3.2: Test parameters of static model

Range of flow speed, m/s	0.05 - 0.5
Re range	$5 \times 10^3 - 5 \times 10^4$
D	100mm
L/D	7
Number of control cylinders	2, 4, 8
d_c/D	0.04, 0.06, 0.08
G/D	0.05, 0.10, 0.15
Acquisition time for each data set	120s

3.2.2 Decay test

Decay tests were performed to determine the natural frequency and the structural damping. They were carried out to all configurations with models free to oscillate. The decay is performed by imposing an initial displacement to the model and monitoring its trajectory over time.

A pair of springs was installed between the moving table and the rigid supports in order to provide the structural restoration force of the oscillator. The spring stiffness (k) combined with the mass of all oscillating parts (m) results in the undamped natural frequency of oscillation in air f_0 . Springs provided the system with a natural frequency of oscillation of $f_0 = 0.25Hz$, determined by performing a series of free decay tests in air. It was possible to vary reduced velocity to a maximum of $U_R = 20$ when U was increased up to $0.5m/s$. The minimum flow was $U = 0.05m/s$, resulting in the lower limit of $U_R = 2$. All moving parts of the elastic base contributed to the effective mass oscillating along with the cylinder. Therefore, m^* could be altered by adding extra mass on top of the rig, but not to the cylinder itself. This way the mass ratio of the system was always kept constant, it was found to be $m^* = 1.09$.

The structural damping of the system was mainly caused by friction between the sliding guides and the air bearings. The air bearings proved to be an efficient way to reduce damping without compromising the stiffness of the structure. By carrying out free decay tests in air it was possible to estimate the structural damping of the system, it was $\zeta = .61\%$. Therefore, yielding the product $m^*\zeta = 0.066$ for the experiments.

3.2.3 VIV in one-degree-of-freedom with fixed control cylinders

This section refers to the VIV tests performed in one-degree-of-freedom with fixed control cylinder. The flow velocity U (and range Reynold number) were the same as in the test with the static model. The pair of springs was installed according to the experimental arrangement presented in Figure 3.11. Other parameters are shown in Table 3.3.

The response of the cylinder to VIV is analyzed by means of amplitude of displacement and frequency of vibration. Frequencies were obtained by the fast Fourier transform of the displacement signal. The values of the mean drag and fluctuating lift coefficients were calculated with the Eq. 2.7 and Eq. 2.5. The measurement of the amplitude of displacement $\hat{y}$ can be realized in different ways as shown by Assi (2005). In this work, the amplitude $\hat{y}$ was calculated by measuring the RMS of displacement and

multiplying the result by $\sqrt{2}$, yielding the harmonic amplitude of the signal. Nevertheless, several authors measured $\hat{y}$ by the average of the 10% maximum peaks, which generates a value slightly higher than that obtained here. The difference between methods should be taken into account when other experiments are compared.

Figure 3.11: Cross-sectional view of the water channel and the 1-dof rig used in the VIV tests.

Table 3.3: Test parameters of VIV

Number of control cylinders	2 , 4, 8
d_c/D	0.04, 0.06, 0.08
G/D	0.05, 0.10, 0.15
Range of U_R	2 - 20
f_0 (Hz)	0.25
m^* (%)	1.09
ζ (%)	0.61
Acquisition time for each data set	240s

3.2.4 VIV in one-degree-of-freedom with eight rotating control cylinders

The configuration with eight control cylinders is approached to work as a suppressor for omni-directional flow. Since the focus of this test is to evaluate the amplitudes of displacement in the upper branch when rotation is applied to the control cylinders, only two cases of the tests of one-degree-of-freedom with fixed cylinders were chosen. The servo motor with the pulley system controlled the rotation, as illustrated in Figure 3.12.

The maximum peak of response of the plain cylinder was identified. Keeping that reduced velocity U_R of each case, the velocity ratio U_c/U was modified in a range of 1 to 10, where the velocity flow U was constant and only the tangential velocity U_c was altered. Once the amplitude was calculated and identified the lowest value, three distinct velocity ratios (U_c/U) were selected whose parameters are presented in Table 3.4.

To maintain U_c/U constant depends on the flow speed and the tangential velocity of the control cylinders, as U_c is controlled by the servo motor, the maximum tangential velocity correspond to the maximum motor speed at 3000 RPM. One test for each case were made with U_c constant, the parameters are presented in Table 3.5.

Figure 3.12: Cross-sectional view of the water channel and 1-dof rig used with eight rotating control cylinders.

In the following chapters the results are presented; in Chapter 4 for a static cylinder, Chapter 5 for VIV with fixed control cylinders and finally in Chapter 6 for VIV with rotating control cylinders.

Table 3.4: Test parameters of VIV with eight rotating control cylinders

Models	Case A	Case B
d_c/D	0.06	0.08
G/D	0.10	
Range of U_R	2 - 12	
U_c/U	2, 2.5, 3	
f_0 (Hz)	0.25	
m^* (%)	1.09	
ζ (%)	0.61	
Acquisition time for each data set	240s	

Table 3.5: Test parameters of VIV with eight rotating control cylinders with a constant tangential velocity (U_c)

Models	Case A	Case B
d_c/D	0.06	0.08
G/D	0.10	
Range of U_R	2 - 20	
U_c/U	0.5 - 4.7	
f_0 (Hz)	0.25	
m^* (%)	1.09	
ζ (%)	0.61	
Acquisition time for each data set	240s	

4 RESULTS FOR A STATIC CYLINDER

This chapter will present and discuss the results of the first series of experiments (campaign I of Figure 3.9), carried out with a static model. Measurements of lift and drag forces were made for each configuration. Initially a validation of the results obtained for the plain cylinder with data available in the literature is presented. Subsequently, the results of suppressor models are presented and discussed, followed by the conclusion of this chapter.

4.1 Validation of a plain cylinder

Experiments with a plain cylinder in the Re range from 5,000 to 50,000 have been performed to serve as reference. The range corresponds to the subcritical regime, where the transition to turbulence occurs in the shear layers, as seen in Chapter 2. The mean drag ($\overline{C_D}$) and the RMS of the lift ($\hat{C}_L$) coefficient were calculated by the force measurements.

Zdravkovich (1997b) summarizes results for $\overline{C_D}$ obtained from various authors as shown in Figure 4.1 compared with the results obtained in the present work. Note that $\overline{C_D}$ of the plain cylinder remains stable at $\overline{C_D} \approx 1.35$ on the range of Re 1.2×10^4 to 4×10^4 , not too far from the reference value.

The comparison of $\hat{C}_L$ with the literature can not be done directly as in the previous case, because the magnitude is too sensitive to boundary conditions. Apart from that, there are important differences in the way the forces are measured. In this study, the overall force acting on the cylinder was measured with a load cell positioned on its top. Due to the three-dimensional flow effects, the coefficient obtained by this total magnitude ($\hat{C}_L$) differs from that calculated for a 2D cylinder section. So it is necessary to distinguish the two coefficients, where $\hat{C}_L$ is the sectional RMS lift coefficient. Norberg (2003) analyzed several experimental and numerical results obtained under different conditions. In that paper, he proposes an expression for $\hat{c}_L$ defined by:

Figure 4.1: Mean drag coefficient for smooth circular cylinder.

$$\hat{c}_L = 0.52 - 0.06x^{(-2.6)} \quad (4.1)$$

$$x = \log \frac{\text{Re}}{1 \times 10^3} \quad (4.2)$$

Further on, Norberg (2003) proposes a regression to transform the total lift value in a sectional value $\hat{c}_L$.

$$\gamma_L = \frac{\hat{C}_L}{\hat{c}_L} \quad (4.3)$$

$$\gamma_L = \frac{\sqrt{2}}{a} \sqrt{e^{-a} + a - 1} \quad (4.4)$$

Where $a = \frac{l_c}{\Lambda_1}$, l_c is the length of the cylinder and Λ_1 is the correlation length, determined by the following regression:

$$\frac{\Lambda_1}{D} = 2.6 \left(\frac{\text{Re}}{2.4 \times 10^5} \right)^{(-0.20)} \quad (4.5)$$

The value of $\hat{C}_L$, its correction $\hat{c}_L$ and the regression proposed by Norberg (2003) are shown in Figure 4.2. The regressions employed are empirical and must be utilized carefully, especially in this range examined as the article shows the values of $\hat{c}_L$ has a great dispersion. Nevertheless this most of the corrected values of the RMS sectional coefficient match with the proposed regression.

Figure 4.2: RMS of lift coefficient for a smooth circular cylinder.

Based on the data presented and discussed, it is estimated that the results for the plain cylinder are in good agreement with data in the literature and will be a good reference for the results obtained with suppressors models, described in subsequent sections.

4.2 Static cylinder with fixed control cylinders

Tests with fixed (not rotating) control cylinders with different diameter (d_c), were performed with the same Re range and acquisition time as the plain cylinder. For each configuration the drag and lift forces were measured for 36 equally spaced values of reduced velocities in the range between $Re = 5,000$ and $50,000$. In order to evaluate the force coefficients, the cylinder diameter (D), was employed as a standard dimension for all configurations. However, it is known that the effective diameter for each configuration is different depending on the presence of the control cylinders and the gap analyzed.

4.2.1 Control cylinder with $d_c/D = 0.04$

Mean drag coefficient $\overline{C_D}$ is shown in Figure 4.3. At $Re = 5 \times 10^3$ all configurations presented $\overline{C_D}$ lower than that for the plain cylinder. In the range of $5 \times 10^3 < Re < 5 \times 10^4$ for all cases with two control cylinders, $\overline{C_D}$ is slightly higher than that for the plain cylinder. For the case of four control cylinder with the highest gap $G/D = 0.15$, the value of $\overline{C_D}$ remains very close to that for the plain cylinder. For the other two cases of four control cylinder, $\overline{C_D}$ is always below that for the reference model, especially $G/D = 0.05$

in the range of $2.5 \times 10^4 < Re < 5 \times 10^4$, which has the lowest values of $\overline{C_D} \approx 0.8$. Likewise in the case with eight control cylinders, $\overline{C_D}$ is lower than that for the isolate cylinder; for $\overline{C_D} \approx 1.2$ $G/D = 0.10$ for all the Re range.

Figure 4.3: Mean drag coefficient versus Re for sets of control cylinders with $d_c/D = 0.04$ and varying gap. (the maximum limit of the vertical axis was set to 3.5 to allow for direct qualitative comparison whit the next figures)

Figure 4.4 shows the fluctuating lift coefficient. Like the previous graph it is clear that for the configuration of two control cylinders $\hat{C}_L$ increases over that for a plain cylinder. For all four control cylinders configurations from $Re = 1.8 \times 10^4$, the values of $\hat{C}_L$ are lower than that for the reference cylinder. Similarly the case of eight control cylinders present lower values along the entire Re range, except the case of $G/D = 0.15$. The case of eight control cylinders with $G/D = 0.05$ presents the lowest and constant value $\hat{C}_L \approx 0.1$ from $Re = 1.5 \times 10^4$.

4.2.2 Control cylinders with $d_c/D = 0.06$

Figure 4.5 shows the mean drag coefficient of this section. Firstly, one can note that the addition of static control cylinders increases the mean drag coefficient, compared to that of the plain cylinder, for all cases with two control cylinders from $Re = 8 \times 10^3$ to

Figure 4.4: RMS of lift coefficient versus Re for sets of control cylinders with $d_c/D = 0.04$ and varying gap. (the maximum limit of the vertical axis was set to 1.5 to allow for direct qualitative comparison with the next figures)

5×10^4 and for two cases with eight control cylinders ($G/D = 0.05$ and $G/D = 0.15$) until $Re = 1.5 \times 10^4$. Besides, at $Re = 5 \times 10^3$ to 5×10^4 for the configuration of four control cylinders at different gaps, the mean drag coefficient is lower than that observed for an isolated cylinder; the lowest value being $\overline{C_D} = 0.75$ for the gap $G/D = 0.05$. Likewise, for each configuration with eight control cylinders, the mean drag coefficient slightly decreases from $Re = 1.5 \times 10^4$ to 5×10^4

The measured values of $\hat{C}_L$ are shown in Figure 4.6 for the various gaps for each configuration of two, four and eight control cylinders. The RMS of the lift coefficient obtained at $Re = 5 \times 10^3$ in most of the cases is higher than that observed for a plain cylinder. Values of $\hat{C}_L$ for four and eight control cylinders at different gaps reduce for increasing Re when compared to that of the isolated cylinder. Unlike the $\overline{C_D}$ plots, the lowest value of $\hat{C}_L = 0.2$ is for the configuration with eight control cylinders with $G/D = 0.15$. The configuration of two control cylinders at different gaps results in an increase of $\hat{C}_L$.

Figure 4.5: Mean drag coefficient versus Re for sets of control cylinders with $d_c/D = 0.06$ and varying gap. (the maximum limit of the vertical axis was set to 3.5 to allow for direct qualitative comparison with the next figures)

Figure 4.6: RMS of lift coefficient versus Re for sets of control cylinders with $d_c/D = 0.06$ and varying gap.

4.2.3 Control cylinders with $d_c/D = 0.08$

$\overline{C_D}$ is presented in Figure 4.7 for all configurations with $d_c/D = 0.08$. Four cases present an increase of $\overline{C_D}$: For two control cylinders the increase is around 40% when compared with that of the plain cylinder. For the other two cases of eight control cylinders with $G/D = 0.10$ and 0.15 , $\overline{C_D}$ remains almost equal to that of the isolated cylinder. For the configuration four control cylinders $\overline{C_D}$ slightly decreases (in the range of $5 \times 10^3 < Re < 2 \times 10^4$), but remains constant at $\overline{C_D} \approx 0.9$ for $Re > 2 \times 10^4$.

Figure 4.7: Mean drag coefficient versus Re for sets of control cylinders with $d_c/D = 0.08$ and varying gap.

The measured values of $\hat{C}_L$ are shown in Figure 4.8. The configuration of two control cylinders at the different gaps results in an increase of $\hat{C}_L$. The RMS of lift coefficient obtained at $Re < 1 \times 10^4$, in most of the cases, is higher than that observed for a plain cylinder. In the range $1 \times 10^4 < Re < 5 \times 10^4$ values of $\hat{C}_L$ for four and eight control cylinders at different gaps are reduced when compared to that of the plain cylinder. The lowest values that remain constant at $\hat{C}_L \approx 0.1$ are for the configurations with eight and four control cylinders, the first one with $G/D = 0.10$ and the other with $G/D = 0.05$.

Figure 4.8: RMS of lift coefficient versus Re for sets of control cylinders with $d_c/D = 0.08$ and varying gap. (the maximum limit of the vertical axis was set to 1.5 to allow for direct qualitative comparison with the previous figures)

4.3 Discussion

The most important observation for the analysis of the results is the influence of the position (in terms of angular location and gap) of the control cylinders employed. For three different values of d_c/D the response according to their position was almost the same. The addition of two control cylinders caused an increase in $\overline{C_D}$ and $\hat{C}_L$ for all cases.

A similar behavior was perceived by Mittal (2001), on his numerical simulation of flow past a circular cylinder at $Re = 10^4$, where the geometrical parameters were $d_c/D = 0.05$ and $G/D = 0.075$. For $d_c/D = 0.06$ and 0.08 the configuration of four control cylinders has the most reduction of the mean drag coefficient.

Strykowski & Sreenivasan (1990) performed one method of controlling vortex shedding with a small secondary cylinder located in the near wake. Experiments were carried on a wind tunnel for low Re . They assumed that the vortex shedding is connected to a global instability and proposed an instructive representation of the influence of the control cylinder for different Reynolds number as showed in Figure 2.13 on Chapter 2.

Recently, motivated by the work of Strykowski & Sreenivasan (1990), Patino

et al. (2015) studied the effects of control cylinders located around the main cylinder $d_c/D = 0.06$, $G/D = 0.07$ for $Re = 47$. They found that certain cylinders located at $0 - 50^\circ$, $135^\circ - 225^\circ$ and $310^\circ - 360^\circ$, stabilize the system. In spite of the system always presented instabilities (the eigenvalues are always non-zero and positive) those cylinders created regions where the most unstable eigenvalue of the system are lower than the baseline given by the unstable eigenvalue of the principal cylinder (continued line). The Figure 4.9 shows the distribution the unstable eigenvalue as a function of the angle. According whit this distribution the configuration of two control cylinder are located in zone unstable (90° and 270°) the opposite with the configuration of four and eight control cylinders, which are located below the continued line.

Figure 4.9: Distribution of unstable eigenvalue as a function of the angle. Reproduced from Patino et al. (2015)

Another noticeable fact is related to the diameter of the control cylinders. For $d_c/D = 0.06$ most of the values of mean drag and fluctuation lift coefficients were lower in comparison to those of the other two ratios. For the smallest diameter analyzed of $d_c/D = 0.04$, the values of $\overline{C_D}$ and $\hat{C}_L$ were not quite as remarkable the influence of the position for four and eight control cylinder as $d_c/D = 0.06$ and $d_c/D = 0.08$.

Finally the gap parameter has the lowest influence when compared to the position and diameter of the control cylinders. The analysis does not show a constant response to conclude which is the optimal value of G/D that minimizes drag or lift.

4.4 Conclusion

In this Chapter, the effect of the configuration of two, four and eight control cylinders of different diameters with varying gap has been investigated. All cases have been compared with the hydrodynamic forces of a plain cylinder.

The suppression effectiveness and drag efficiency may be directly related to the configuration and diameter of the control cylinders. The configuration of four cylinders with $G/D = 0.05$ and $d_c/D = 0.06$ produced less drag when compared to all other configurations: $\overline{C_D} \approx 0.75$, approximately 50% lower than that of the isolated cylinder. For the configuration with eight control cylinders with $d_c/D = 0.04$, all G/D ratios showed an average coefficient of $\overline{C_D} \approx 1$, which corresponds to a decrease of 33%. Only the configurations of two control cylinders presented an increase in the drag coefficient of $\overline{C_D} \approx 1.66$, an increase of 10% in average.

It was found that the configuration is an important parameter for the static suppressor models. The configuration of 4 cylinders has been found to be the best solution for all tested cases, presenting the lowest values of $\overline{C_D}$ and $\hat{C}_L$. Despite the configuration of 8 control cylinders aims to be used in omni-directional flow their position was not as effective as the four control cylinders when $\overline{C_D}$ and $\hat{C}_L$ are concerned.

A cylinder with low $\hat{C}_L$ is not guaranteed to be unaffected by VIV. However, a low $\hat{C}_L$ is a good indication that a similar cylinder free to respond to the flow excitation might present a low VIV response, if not suppressed altogether. This will lead us to investigate the same models now free to respond in VIV.

5 RESULTS: VIV WITH FIXED CONTROL CYLINDERS

This chapter will present the results of the second series of experiments carried out with one-degree-of-freedom VIV, with fixed control cylinders, in the transverse direction to the flow. Initially a validation of the results obtained for a plain cylinder is presented, to serve as a reference for comparison. Then the results for suppressors are presented and discussed.

5.1 Validation of a plain cylinder

The fundamental natural frequency (f_0) in air was found to be $0.25Hz$. Based on that natural frequency the reduced velocity (U_R) achieved the range of 2 to 20 during the experiments. Experimental results for the dynamic response in terms of displacement amplitude and frequency of oscillation, as well as mean drag and fluctuating lift coefficients were measured for a plain cylinder and the other suppressor models. The validation of the plain cylinder was based on the papers of Khalak & Williamson (1977,1999).

Figure 5.1 presents the displacement amplitudes and frequencies of oscillation of a plain cylinder compared with the results obtained by Khalak & Williamson (1997). This paper had similarity between the parameters: the reduced mass of the experiments are $m^* = 3.3$ for the study of Khalak & Williamson (1997) and $m^* = 1.09$ in this work. In both an elastic base with compressed air bearings was employed. In the $\hat{y}$ curve (top graph) it is possible to identify the typical three branches of response discussed by Williamson & Govardhan (2004) for low-mass ratio and damping models. An initial branch for the plain cylinder shows a build up of response at lower reduced velocities $2 < U_R < 4$, and the frequency of oscillation follows the Strouhal relationship. On the upper branch for $4 < U_R < 8$ a characterized peak of $\hat{y}/D = 1$ is on reduced velocity of 4. At this point the frequency of oscillation stabilizes close to the natural frequency, starting the synchronization range.

Figure 5.1: VIV response of a single cylinder free to oscillate in the cross- flow direction. Top: displacement; bottom: frequency of oscillation

A lower branch with predominant amplitude of about $\hat{y}/D = 0.45$ smoothly follows the upper branch after a reduced velocity of 9, when the frequency of oscillation rises above the natural frequency in air ($f/f_0 = 1.0$). The frequency of oscillation remains at a roughly constant level until the end of this synchronization range, up to a reduced velocity of 13, when the amplitude of oscillation drops to levels below $\hat{y}/D = 0.2$. Small differences between the curves may be attributed, besides the difference in m^* , to different

values in the aspect ratio ($L/D = 10$) in Khalak and Williamson experiment and $L/D = 7$ in the present study.

Figure 5.2 shows $\overline{C_D}$ as a function of U_R . An increase of the mean drag coefficient at $U_R \approx 5$ is observed. After this point as the reduced velocity increases, $\overline{C_D}$ decreases until it remains around values similar to those for a static cylinder ($\overline{C_D} \approx 1$). The differences between curves may be attributed to the different values of aspect ratio or other experimental variations.

Figure 5.2: Mean drag coefficient versus reduced velocity for a plain cylinder.

The fluctuating lift coefficient ($\hat{C}_L$) as a function of U_R is shown in Figure 5.3. One may note that there is a maximum value where the highest oscillation occurs, and out of synchronization range the $\hat{C}_L$ is almost null. Comparing the results obtained with the literature, the curves show the same tendency with a small difference in the maximum value found in this work. This difference may be explained by the different values of m^* between experiments. Furthermore the boundary conditions can influence on the results, $\hat{C}_L$ is very sensitive to them and there is a wide dispersion of data in this Re range, as described in Chapter 4.

As far as validation is concerned, these first results presented in this section are in good agreement with the literature and provide a good reference for other experiments.

Figure 5.3: Fluctuating lift coefficient versus reduced velocity for a plain cylinder.

5.2 VIV response with fixed control cylinders

This section presents the responses obtained with the configuration of two, four and eight fixed control cylinders as suppressors. All the configurations were performed with in the same range $2 \leq U_R \leq 20$ and acquisition time as the plain cylinder. In this range the values are in the synchronization region and it is possible to observe the classical response which includes the initial, upper and lower branches as well as the desynchronization region. For each configuration, different diameters of the control cylinder (d_c) and gaps between the main cylinder and the control cylinders (G/D) were tested.

5.2.1 Response with two control cylinders

Figure 5.4 illustrates the displacement amplitude and frequencies of oscillation of the configuration with two control cylinders compared with the plain cylinder. The plain cylinder presents the typical VIV responses already discussed in the previous section, the maximum amplitude is $\hat{y}/D \approx 1.00$ at $U_R \approx 4$. For the first group of control cylinders with $d_c/D = 0.04$ there is a decrease in the amplitudes in the upper branch region if compare with the plain cylinder, the maximum amplitude is $\hat{y}/D \approx 0.7$ at $U_R \approx 4$.

On the lower branch when $U_R > 9$, for the case of $d_c/D = 0.04$ with $G/D = 0.15$ the amplitude remains constant at $\hat{y}/D \approx 0.5$. The case of $d_c/D = 0.04$ and $G/D = 0.10$ the amplitude is slightly higher than that of the plain cylinder, even in the desynchronization range. The synchronization can also be seen in the frequency graph, when the oscillation frequency approaches the natural frequency. The case which presents a higher variation on amplitude is that with $d_c/D = 0.04$ and $G/D = 0.05$.

Furthermore two different types of responses are characterized, one is for the cases $d_c/D = 0.06$ with $G/D = 0.05$ and 0.10 and $d_c/D = 0.08$, with $G/D = 0.05$ where the amplitude increased to $\hat{y}/D \approx 0.65$ until $U_R = 5$ and then remained constant in all the range of U_R , no lower branch or desynchronization range were verified. The oscillation frequency remains close to the frequency of vortices (or Strouhal frequency) throughout the analyzed interval without a common characterization of synchronization as for the plain cylinder. The other response is for the cases of $d_c/D = 0.08$ with $G/D = 0.10$ and 0.15 and $d_c/D = 0.06$ with $G/D = 0.10$ present initial and upper branches, however at $U_R > 9$ the amplitude increases almost linearly as a function of the reduced velocity.

The mean drag coefficient ($\overline{C_D}$) as a function of U_R is shown in Figure 5.5. For almost all the cases at the initial branch the mean drag coefficient is slightly higher than that of a plain cylinder. However the maximum value of $\overline{C_D} \approx 3.8$ at $U_R \approx 3$ it is lower than that of a plain cylinder. In the range of $3 < U_R < 8$, $\overline{C_D}$ decreases as a function of U_R . In the region of the lower branch, for $U_R > 8$, the mean drag coefficient remains constant at $\overline{C_D} \approx 1.8$ for the cases of $d_c/D = 0.08$ with $G/D = 0.10$ and 0.15 and $d_c/D = 0.06$ with $G/D = 0.05, 0.10$ and 0.15 . For the cases of $d_c/D = 0.04$ with $G/D = 0.05$ and 0.10 they present the same behavior on the desynchronization region with $\overline{C_D}$ slightly higher than that of the plain cylinder about $\overline{C_D} \approx 1.4$. Nonetheless all the cases at $U_R > 8$ the mean drag is higher when compared with that of the plain cylinder.

Figure 5.4: VIV response with two control cylinders for $m^* = 1.09$. Top: displacement; bottom: frequency of oscillation

Figure 5.5: Mean drag coefficient versus reduced velocity for the cylinder with two control cylinders and $m^* = 1.09$.

Figure 5.6 illustrates the fluctuating lift coefficient ($\hat{C}_L$) as a function of U_R . The plain cylinder in the initial branch grows until it reaches a maximum coefficient of $\hat{C}_L \approx 2.5$ around $U_R \approx 4$, linearly decreasing to values close $\hat{C}_L \approx 0.5$ in the lower branch region and then declining to almost zero at the end of the synchronization. The case $d_c/D = 0.08$ with $G/D = 0.10$ presents the highest value of $\hat{C}_L \approx 3.8$ on the initial branch and then decreases linearly as a function of U_R . Until $U_R = 8$ the fluctuation lift coefficient remains constant at $\hat{C}_L \approx 0.2$, the rest of the cases present low values of the initial branch, reaches the maximum value ($\hat{C}_L \approx 3.2$) in the upper branch region and decreases linearly until it reaches $\hat{C}_L \approx 0.2$ at $U_R \approx 7$, for high values of U_R its value does not tend to zero remains constant at the end of the range.

Figure 5.6: Fluctuating lift coefficient versus reduced velocity for the cylinder with two control cylinders and $m^* = 1.09$.

5.2.2 Response with four control cylinders

The displacement amplitude and frequencies of oscillation of the configuration with four fixed control cylinders are shown in Figure 5.7. For the group of control cylinders with $d_c/D = 0.04$, the upper branch presents different values of the maximum amplitude (for each gap $G/D = 0.05, 0.10$ and 0.15 the values are $\hat{y}/D \approx 0.6, 0.8$ and 1.1). For the cases with $G/D = 0.05$ and 0.10 after the highest peak of amplitude, they begin to decrease to $\hat{y}/D \approx 0.45$ and 0.6 . From $U_R \approx 5$ they remain constant until $U_R \approx 12$, where both drop in desynchronization to the same values of the plain cylinder. For the case with $G/D = 0.15$, after the lower branch, the amplitude increases to $\hat{y}/D \approx 0.8$ at $U_R \approx 11$, but for $U_R > 12$ it drops in desynchronization until the end of the U_R range. For the case with $G/D = 0.05$ the frequency of oscillation remains close to the frequency of the plain cylinder.

Figure 5.7: VIV response of plain cylinder with four control cylinder for $m^* = 1.09$. Top: displacement; bottom: frequency of oscillation

As in the previous section, there are two typical responses in addition to those presented for the case of $d_c/D = 0.04$. One of them, the amplitude of the upper branch clearly decreases in respect to the plain cylinder, reaching $\hat{y}/D \approx 0.45$ at $U_R \approx 3$. The lower branch occurs until $U_R \approx 9$; after this point desynchronization is started which have lower values than the plain cylinder. In contrast with the previous response, the cases $d_c/D = 0.06$ with $G/D = 0.15$ and $d_c/D = 0.08$ with $G/D = 0.15$ show a peaks and drops of amplitude. For $d_c/D = 0.06$ with $G/D = 0.15$ the first peak of $\hat{y}/D \approx 1.1$ occurs at $U_R \approx 5$ followed by an amplitude drop until $\hat{y}/D \approx 0.6$ and then by a linear increases at $U_R \approx 9$ when another drop in amplitude occurs ($\hat{y}/D \approx 0.4$) followed by linear increase as a function of the reduce velocity.

Figure 5.8 presents the mean drag coefficient ($\overline{C_D}$) as a function of U_R . The only case $d_c/D = 0.06$ with $G/D = 0.15$ presents a slightly higher value of $\overline{C_D}$ on the upper branch ($\overline{C_D} \approx 4.1$) and in the desynchronization region. Other cases present lower values of $\overline{C_D}$ in comparison to the plain cylinder. The lowest value of $\overline{C_D} \approx 1.2$ on the upper range corresponds to the case $d_c/D = 0.06$ with $G/D = 0.10$ on the desynchronization region $\overline{C_D} \approx 0.08$. The case $d_c/D = 0.04$ with $G/D = 0.15$ presents a similar response to that of the plain cylinder, except in the range that corresponds to the lower branch, where the values of $\overline{C_D}$ are lower.

The fluctuating lift coefficient ($\hat{C}_L$) as a function of U_R for the four control cylinder configuration is shown in Figure 5.9. Similar to the graph of the mean drag coefficient, the case $d_c/D = 0.06$ with $G/D = 0.15$ presents $\hat{C}_L$ values slightly higher on the upper branch and desynchronization region compared with those of the plain cylinder. For the other cases, in the upper branch region of $3 < U_R < 5$, the values of $\hat{C}_L$ are varied and lower than that of the plain cylinder. In the desynchronization region all values tend to zero. Differently from the $\overline{C_D}$ graph the lowest fluctuating lift coefficient in the region of the upper branch is $\hat{C}_L \approx 0.6$ for the case $d_c/D = 0.08$ with $G/D = 0.05$.

Figure 5.8: Mean drag coefficient versus reduced velocity for the cylinder with four control cylinders and $m^* = 1.09$.

Figure 5.9: Fluctuating lift coefficient versus reduced velocity for the cylinder with four control cylinders and $m^* = 1.09$.

5.2.3 Response with eight control cylinders

For the configuration with eight fixed control cylinders, the amplitude and frequencies of oscillations are presented in Figure 5.10. The case of $d_c/D = 0.08$ with $G/D = 0.10$ presents the lowest amplitude in the upper branch ($\hat{y}/D \approx 0.01$ at $U_R \approx 6$). The amplitude slightly increases at $\hat{y}/D \approx 0.05$ but remains constant to the end of the range. The highest amplitude of $\hat{y}/D \approx 0.7$ is found for the case of $d_c/D = 0.04$ with $G/D = 0.15$, but is still lower in comparison with the plain cylinder. When $U_R > 12$ the amplitude does not present a desynchronization, but remains at $\hat{y}/D \approx 0.5$. The other cases present different ranges of the branch regions in comparison with the plain cylinder: the initial corresponds to $2 \lesssim U_R \lesssim 3$; the upper, to $3 \lesssim U_R \lesssim 4$; the lower, to $4 \lesssim U_R \lesssim 8$; and finally the desynchronization starts at $U_R \gtrsim 8$ with $\hat{y}/D$ lower than that of the plain cylinder.

For the case with $d_c/D = 0.04$ and $G/D = 0.10$ the frequency of oscillations remains close to the Strouhal frequency through to the end of the reduced velocity range. Cases $d_c/D = 0.06$ with $G/D = 0.05$ and $d_c/D = 0.04$ with $G/D = 0.15$ remain close until $U_R \approx 10$. For the case of $d_c/D = 0.08$ with $G/D = 0.10$ the oscillation frequency does not present a synchronization (in the initial branch) with the Strouhal frequency, but remains constant in at $f/f_0 \approx 0.3$.

Figure 5.11 presents the mean drag coefficient ($\overline{C_D}$) as a function of U_R for the eight fixed control cylinders. For the case of the lowest amplitude $d_c/D = 0.08$ with $G/D = 0.10$ does not present the lowest value of $\overline{C_D}$. At the initial branch it has the highest $\overline{C_D} \approx 3$ and begins to decrease until $\overline{C_D} \approx 1.5$ at $U_R \approx 7$. After it remains constant to the end. Other cases present lower values of $\overline{C_D}$ in the upper branch region in comparison to the plain cylinder. The cases of $d_c/D = 0.06$ with $G/D = 0.05$ shows a constant and low value of $\overline{C_D} \approx 1.7$, followed by a decrease to $\overline{C_D} \approx 1.25$ just after $U_R \approx 9$.

The fluctuating lift coefficient ($\hat{C}_L$) as a function of U_R is presented in Figure 5.12. The case $d_c/D = 0.04$ with $G/D = 0.10$ is the only that presents $\hat{C}_L$ values slightly higher on the desynchronization region compared with those of the plain cylinder. For the cases with almost null amplitude, the $\hat{C}_L \approx 0.2$ is the lowest value in the upper range. For the other cases, $\hat{C}_L$ is lower than that of the isolated cylinder and higher for the case of $d_c/D = 0.08$ with $G/D = 0.10$.

Figure 5.10: VIV response of plain cylinder with eight control cylinder for $m^* = 1.09$.

Top: displacement; bottom: frequency of oscillation

Figure 5.11: Mean drag coefficient versus reduced velocity for the cylinder with eight control cylinders and $m^* = 1.09$.

Figure 5.12: Fluctuating lift coefficient versus reduced velocity for the cylinder with eight control cylinders and $m^* = 1.09$.

5.3 Discussion

In the synchronization range, most of the configurations reduced the amplitude of response. In the upper branch region of the plain cylinder, the configuration of eight control cylinders showed the largest decrease. In fact the case of $d_c/D = 0.08$ with $G/D = 0.10$ presented an amplitude of responses almost negligible, followed by the group of two control cylinders, the amplitude of response was $\hat{y}/D \approx 0.65$. The last configuration of four control cylinders presented the most diverse response, with the case of $d_c/D = 0.08$ with $G/D = 0.05$ presenting the second lowest amplitude of response. However the configuration $d_c/D = 0.06$ with $G/D = 0.15$ showed the highest amplitude responses $\hat{y}/D \approx 1.1$. In general, drag and lift forces were also reduced for almost all the cases in the three different configurations in the range of U_R that correspond to the upper branch. On the lower and upper branches, the configurations of two and four control cylinders presented behaviors that did not follow the same trend expected for the phenomenon of VIV for a plain cylinder.

For the configuration of two fixed control cylinders Korkischko & Meneghini (2012) reported that the amplitude of response was completely different when compared to that of a plain cylinder. Their response was similar to our case of $d_c/D = 0.04$ with $G/D = 0.15$, where the peak of amplitude was $\hat{y}/D \approx 0.7$. For Korkischko & Meneghini (2012), $\hat{y}/D \approx 0.6$ at $U_R = 4$ and slowly decreased along the U_R range tested. Their parameters were $d_c/D = 0.06$, $G/D = 0.07$ and $m^* = 1.8$.

Four cases; three for the two control cylinders ($d_c/D = 0.08$ with $G/D = 0.10$ and 0.15 and $d_c/D = 0.06$ with $G/D = 0.10$) and one for the four control cylinders configuration ($d_c/D = 0.06$ with $G/D = 0.15$) presented classical galloping instability (PARKINSON; DICKER, 1971). Korkischko & Meneghini (2012) suggested that the fact that the fixed control cylinders fix the boundary-layer separation points, the cylinder could be subjected to this phenomenon.

Zdravkovich (1981) presented results of an axial-rod shroud. Several parameters were varied in order to find an optimum configuration, the shroud size (d_c/D) and porosity. The most effective shroud geometry was $d_c/D = 1.25$ with a 63% porosity, this configuration was chosen to verify the circumferential distribution, where rods were pulled out. Figure 2.14 presents the effect of the orientation of the incomplete axial-rods shrouds. When the unshrouded portion with only 11 rods on each size, the amplitude of oscillation was higher than that found for the plain cylinder. The position of those 11

rods is similar where the two control cylinders are located. This article evidences that the position around the cylinder is an important parameter for the suppressor.

Three cases of two control cylinders configuration ($d_c/D = 0.06$ with $G/D = 0.05$ and 0.15 and $d_c/D = 0.08$ with $G/D = 0.05$) showed similar behavior with the response of Govardhan & Williamson (2002), which presented constant amplitude and frequency after the synchronization oscillation near vortex emission, for systems with $m^* < 0.54$, much smaller than that of this experiments ($m^* = 1.09$). The authors have called this behavior *resonance forever*, the frequency ratio tends to infinity and consequently the lower branch can never be reached. The upper branch of the VIV response continues indefinitely, resulting in an infinite resonance regime. The lower branch essentially ceases to exist below the critical mass ratio. In order for a system with $m^* = 1.09$ to maintain vibrations for high U_R values it is probable that the configuration is capable of absorbing more energy from the flow, sustaining higher lift forces. The curve of $\hat{C}_L$ does not show this behavior, however.

Another observation that is worth highlighting is two the cases where it seems to be a transition between VIV and possibly galloping. This is assumed for the changing of peaks generated on the amplitude responses. It is more evident in the configuration for four control cylinder for the case $d_c/D = 0.08$ with $G/D = 0.15$ and in the configuration of two control cylinder for $d_c/D = 0.04$ with $G/D = 0.05$.

5.4 Conclusion

Amplitude responses, frequencies of oscillations, mean drag and fluctuating lift coefficients were measured for the plain cylinder and showed good agreement to the results found in the literature. Therefore, the experimental method has been validated.

Most of the configurations presented resonant response and reduced the amplitude response when compared with that of a plain cylinder. In the upper branch region, the configuration of two control cylinders reduced about 35% of the maximum vibration amplitude, while four cases ($d_c/D = 0.08$ with $G/D = 0.05$ and 0.10 and $d_c/D = 0.06$ with $G/D = 0.05$ and 0.10) of the configuration of four control cylinders reduced about 60%. The best case for response reduction was the configuration of eight control cylinders ($d_c/D = 0.08$ with $G/D = 0.10$), which reduced about 99%.

$\overline{C_D}$ and $\hat{C}_L$ were also reduced for some cases in all the configurations from $4 < U_R < 10$. The best case was the configuration with eight control cylinders, which reduced

the mean drag coefficient in about 50% and the fluctuating lift coefficient in about 92% when compared to those of a plain cylinder.

For $U_R > 10$, different responses appeared in some cases, one of the responses is the linear increase of the amplitude along with U_R . In others the amplitude was almost constant until the end of the test. As a result, the mean drag force for that cases remained higher than that of a plain cylinder, while in other cases their amplitudes and mean drag coefficient were reduced.

The behavior of the cases of two control cylinders ($d_c/D = 0.08$ with $G/D = 0.10$ and 0.15 and $d_c/D = 0.06$ with $G/D = 0.10$) and the four control cylinders ($d_c/D = 0.06$ with $G/D = 0.15$) are the same as in the galloping phenomenon presented by Parkinson & Dicker (1971). The other distinct behavior ($d_c/D = 0.06$ with $G/D = 0.05$ and 0.15 and $d_c/D = 0.08$ with $G/D = 0.05$) is very similar to that seen by Govardhan and Williamson (2002) for very low mass ratio systems, called "resonance forever". It is true that the reduced mass of the system is not below the critical value of the mechanism ($m^* < 0.54$). However, it can be inferred that those configurations are able to capture energy from the flow as the system oscillates.

Finally, one may conclude that the presence of the control cylinders alone may interact with the flow in the near-wake, disrupt coherent vortex shedding and suppress vibration.

In the next chapter we will allow the control cylinders to rotate to enhance flow interference in the vortex formation region.

6 RESULTS: VIV WITH ROTATING CONTROL CYLINDERS

The helical strakes are one of the most commonly VIV suppressor used in air and water flows. An advantage, it may be used for an omni-directional flow, despite the increase the mean drag. The moving surface boundary-layer control (MSBC) is usually employed as a suppressor with the objective of reducing the drag of bluff bodies, previously discussed in Chapter 2.

In order to investigate the VIV suppression efficiency and drag reduction of the this mechanisms to be employed in an omni-directional fluid, the third experimental campaign was conducted with one-degree-of-freedom VIV in the transverse direction to the flow with rotating control cylinders. Two cases of the configuration of eight control cylinders of the Chapter 5 were selected.

In this chapter results for a plain cylinder will serve as a reference for comparison; the validation was presented in the Chapter 5. Initially we will present the results for the two cases: with varying velocity ratio (U_c/U) and with constant the tangential velocity of the control cylinders. Finally a discussion and conclusion will be given.

6.1 Response varying the velocity ratio (U_c/U)

This section presents the results of VIV regarding the rotating of the control cylinders around the circular cylinder as Figure 6.1 shows. The control parameter is the ratio between the tangential velocity of the control cylinder and the free stream velocity, as the rotation parameter, U_c/U . The control cylinders are actuated by a servo motor previously described in Chapter 3. Due to the characteristic of the servo motor there one restriction in the maximum range of U_c/U that can be effectively investigated with this experimental apparatus. Therefore, we will concentrate our attention on the effect of rotation over the response already obtained with the fixed control cylinders.

Figure 6.1: Schematic diagram of the main cylinder with eight rotating control cylinders.

Two cases will be tested: the first is based on the case with fixed cylinders that showed the highest reduction in amplitude response; the parameters are $d_c/D = 0.08$ with $G/D = 0.10$. The second is based on the case with fixed cylinders in which the reduction of the response was in average value; the parameters are $d_c/D = 0.06$ with $G/D = 0.10$.

6.1.1 Response of case 1: eight control cylinder, $d_c/D = 0.08$ with $G/D = 0.10$

The reduced velocity was kept constant at $U_R = 4.2$, which corresponds to the point of maximum amplitude for the plain cylinder. The U_c/U ratio was increased from 1 to 10. Figure 6.2 shows that the decrease of amplitudes of response for each ratio, is not monotonic. The lowest value of amplitude was $\hat{y}/D = 0.18$ at $U_c/U = 2.5$. Based on that result, U_c/U immediately higher and lower were selected to be analyzed, which correspond to $U_c/U = 2$ and 3.

Figure 6.2: Amplitude response of a cylinder with rotating control cylinders at the reduced velocity $U_R = 4.2$

Figure 6.3 shows the amplitude of response and frequency of oscillation of the configuration of eight control cylinders varying U_c/U . When the control cylinders are actuated, one can verify the reduction of the amplitude with respect to the plain cylinder on the upper branch region. However the response is still higher than that for the case with fixed control cylinders ($U_c/U = 0$). For $U_c/U = 2.5$ there is an increase of the amplitude to $\hat{y}/D \approx 0.2$ until $U_R \approx 4$ followed by a slight decrease and another increase between $4 < U_R < 7$, reaching $\hat{y}/D \approx 0.25$ and remaining constant until the end of the test at $U_R = 12$. $U_R = 12$ is the last value for which the rotation parameter can be controlled due to limitations in the servo motor. For $U_c/U = 2$ the maximum value of amplitude is $\hat{y}/D \approx 0.3$ and for $U_R > 4$ it remains constant. In the lower branch ($8 < U_R < 12$) the amplitude decreases to $\hat{y}/D \approx 0.15$ only for the case with $U_c/U = 3$.

The frequency of response has two different behaviors with respect to the plain cylinder, one for $U_c/U = 0$ and the other for $U_c/U = 2$ and 2.5 configurations. For the $U_c/U = 2$ and 2.5 cases, the frequency f/f_0 coincides with the Strouhal frequency in all the interval of the test. Contrary to the case $U_c/U = 0$, the other do not present a synchronization regime, but remain constant at a frequency $f/f_0 = 0.3$.

Figure 6.3: Amplitude and frequency responses of a cylinder with rotating control cylinders ($d_c/D = 0.08$ and $G/D = 0.10$).

Figure 6.4 shows the mean drag coefficient. Differently from the plain cylinder in the initial branch, the mean drag coefficient begins with very high values, but as the reduced velocity increases $\overline{C_D}$ gradually decreases. In the upper branch ($3 < U_R < 5$) the lowest values of $\overline{C_D}$ is found for the fixed control cylinders ($U_c/U = 0$), the case of $U_c/U = 2.5$ also present a reduction to $\overline{C_D} \approx 3.2$, however the cases $U_c/U = 2$ and 3 present almost the same values of $\overline{C_D}$ as that for the plain cylinder. For $U_R > 6$ $\overline{C_D}$ for the cases with rotating control cylinders are higher than that for the plain cylinder and for the case with fixed control cylinders. One may note a reduction in $\overline{C_D}$ is directly related to a reduction in the amplitude of response.

Figure 6.4: Mean drag coefficient of a cylinder with rotating control cylinders ($d_c/D = 0.08$ and $G/D = 0.10$).

The fluctuating lift coefficient ($\hat{C}_L$) as a function of U_R is shown in Figure 6.5. In the range of $2 < U_R < 9$ all the cases present a decrease of $\hat{C}_L$ when compared to the plain cylinder. The lowest value $\hat{C}_L \approx 0.1$ is found at $U_R = 4$ for $U_c/U = 0$. Nevertheless when $U_R > 9$ the cases with $U_c/U = 2$ and 2.5 present $\hat{C}_L$ is slightly higher than that of the plain cylinder.

Figure 6.5: Fluctuating lift coefficient of a cylinder with rotating control cylinders ($d_c/D = 0.08$ and $G/D = 0.10$).

6.1.2 Response of case 2: eight control cylinder, $d_c/D = 0.06$ with $G/D = 0.10$

Following the same procedure as in the previous section, the reduced velocity was fixed and the rotation parameter was varied from 1 to 9. For this case the lowest amplitude was $\hat{y}/D = 0.21$ found at $U_c/U = 2.5$, as Figure 6.6 shows. The rotation parameters investigate in detail are $U_c/U = 2, 2.5$ and 3.

Figure 6.6: Amplitude response of a cylinder with rotating control cylinders at the reduced velocity $U_R = 4.2$

The amplitude of response and frequency of oscillation for this case is shown in Figure 6.7. When the control cylinders are actuated there is a substantial reduction in the oscillating amplitude for all the U_c/U ratios tested. As a general behavior for $2 < U_c/U < 3$, the amplitude begins to increase until $\hat{y}/D \approx 0.25$ at $U_R = 5$ and remains constant until $U_R = 12$ lower than that of the plain cylinder. Nonetheless in the range of $7 < U_R < 12$, the lowest value is found for $U_c/U = 0$ (fixed cylinders).

Between $2 < U_R < 7$ the frequency of response for all the cases coincide with the Strouhal frequency. Beyond this interval f/f_0 the case of $U_c/U = 0$, diverges from the Strouhal frequency with increasing reduced velocity and present a constant value of $f/f_0 = 0.3$. All other cases remain close to the Strouhal frequency.

Figure 6.7: Amplitude and frequency responses of a cylinder with rotating control cylinders ($d_c/D = 0.06$ and $G/D = 0.10$).

The mean drag coefficient for this case is shown in Figure 6.8. For the cases of rotating control cylinder the highest value of $\overline{C_D} \approx 5$ is found at $U_R = 2$, then it gradually decreases. In the range $3 < U_R < 8$ the cases $U_c/U = 0, 2.5$ and 3 show amplitude of response that are lower than that of the isolated cylinder. Beyond $U_R > 9$ only the case with fixed control cylinder ($U_c/U = 0$) shows amplitude as low as the values for the plain cylinder.

Figure 6.8: Mean drag coefficient of a cylinder with rotating control cylinders ($d_c/D = 0.06$ and $G/D = 0.10$).

Figure 6.9 presents the fluctuation lift coefficient ($\hat{C}_L$). For all the cases $\hat{C}_L$ is lower than that of the plain cylinder. At $U_R = 4$ the lowest value is $\hat{C}_L \approx 0.3$ for the case $U_c/U = 2$.

Figure 6.9: Fluctuating lift coefficient of a cylinder with rotating control cylinders ($d_c/D = 0.06$ and $G/D = 0.10$).

6.2 Response with a constant tangential velocity (U_c)

In the previous section it was clear that the characteristics of the servo motor imposes a rotation limitation so that it was not possible analyze the response for the maximum value of the reduced velocity $U_R = 20$. It is decided to run another case where the tangential velocity of the control cylinders remains constant over the entire range of reduced velocity from 2 to 20. The selection of the tangential velocity was based on the rotation ratio, (U_c/U), that presented the lowest amplitude of response at $U_R = 4.2$, which corresponds to the maximum amplitude observed for the plain cylinder. For both cases $U_c/U = 2.5$, according to the results presented in the previous section. The responses for each case are compared with that of the plain cylinder, the fixed control cylinders and the rotating control cylinders at $U_c/U = 2.5$.

6.2.1 Response of case 1: eight control cylinder, $d_c/D = 0.08$ with $G/D = 0.10$

Figure 6.10 presents the amplitude response and frequency of oscillation. For $U_c = 0.266m/s$ the amplitude response increases until $\hat{y}/D \approx 0.3$ at $U_R = 6$, but remains constant as the reduced velocity increased until $U_R = 9$; beyond this point the amplitude decreased until the end of $U_R = 20$. Compared to the plain cylinder one may note a substantial reduction in the oscillation amplitude, nevertheless is not as low as the case of the fixed control cylinder. The frequency coincide with the Strouhal frequency until $U_R = 6$, then f/f_0 presents a well-defined pattern of oscillation as observed for the plain cylinder in the range $6 < U_R < 14$.

The mean drag coefficient is presented in Figure 6.11. As well as the case of $U_c/U = 2.5$, when the control cylinders are actuated in a constant tangential velocity, the initial branch presents high values of $\overline{C_D}$ in comparison with that of the plain cylinder. For $U_R > 4$ to the case $U_c = 0.266m/s$ the mean drag coefficient gradually decreases until $U_R = 10$; these values are lower than that of the isolated cylinder. On the desynchronization region $\overline{C_D}$ remains constant at 1.2.

Figure 6.12 presents the fluctuating lift coefficient ($\hat{C}_L$). For the case where the tangential velocity remains constant $\hat{C}_L$ is lower than that of the plain cylinder. Nevertheless on the upper branch region, at $U_R = 4$, the lowest value is $\hat{C}_L \approx 0.1$ found for the case $U_c/U = 0$.

Figure 6.10: Amplitude and frequency responses of a cylinder with control cylinders with a constant tangential velocity ($d_c/D = 0.08$ and $G/D = 0.10$).

Figure 6.11: Mean drag coefficient of a cylinder with control cylinders with a constant tangential velocity. ($d_c/D = 0.08$ and $G/D = 0.10$).

Figure 6.12: Fluctuating lift coefficient of a cylinder with control cylinders with a constant tangential velocity ($d_c/D = 0.08$ and $G/D = 0.10$).

6.2.2 Response of case 2: eight control cylinder, $d_c/D = 0.06$ with $G/D = 0.10$

Figure 6.13 illustrates the displacement amplitude and frequencies of oscillation of the case where the tangential velocity remains constant compared with the plain cylinder and with control cylinders at $U_c/U = 0$ and $U_c/U = 2.5$. In the range of $2 < U_R < 7$ the amplitude response increases until $\hat{y}/D \approx 0.3$. Beyond this point it decreases. At $U_R = 10$ the amplitude drops to $\hat{y}/D \approx 0.1$ and remains constant to the end. Compared to the plain cylinder it is noted a substantial reduction in the oscillation amplitude. The frequency coincides with the Strouhal frequency until $U_R = 11$, beyond this point f/f_0 is intermittent of the oscillations.

Figure 6.14 illustrates the mean drag coefficient. As in the previous section, the behavior of $\overline{C_D}$ is almost the same for when the tangential velocity is constant. It starts with high values and, as the reduced velocity is increased, $\overline{C_D}$ decreases until $U_R = 10$. Beyond this point it remains constant at 1.2 until the end. On the range of $4 < U_R < 10$, $\overline{C_D}$ values are lower than that of the plain cylinder.

Figure 6.15 illustrates the fluctuating lift coefficient. The case of $U_c = 0.266m/s$, in the initial branch, reaches the maximum coefficient $\hat{C}_L \approx 0.6$ and begins to decrease to zero from $U_R = 10$. The values of $\hat{C}_L$ are lower than that of the plain cylinder.

Figure 6.13: Amplitude and frequency responses of a cylinder with control cylinders with a constant tangential velocity ($d_c/D = 0.06$ and $G/D = 0.10$).

Figure 6.14: Mean drag coefficient of a cylinder with control cylinders with a constant tangential velocity ($d_c/D = 0.06$ and $G/D = 0.10$).

Figure 6.15: Fluctuating lift coefficient of a cylinder with control cylinders with a constant tangential velocity ($d_c/D = 0.06$ and $G/D = 0.10$).

6.3 Discussions

In the higher vibration region of the plain cylinder, all the cases with rotating control cylinders reduced the amplitude of response and, consequently, the fluctuation lift coefficient. For the first configuration analyzed (eight control cylinders with $d_c/D = 0.08$ and $G/D = 0.10$), varying the rotation parameter does not produce the same result as that found by Korkischko & Meneghini (2012). They reported that when control cylinders are actuated the amplitude of response presented the lowest values when compared with those of the plain cylinder and the fixed control cylinders ($U_c/U = 0$). Although the parameters and the configuration are different (they presented results for two control cylinders with $d_c/D = 0.06$, $G/D = 0.07$ and $U_c/U = 5 - 25$), when the second configuration is analyzed (eight control cylinders with $d_c/D = 0.06$ and $G/D = 0.10$), the highest reduction of the amplitude corresponds to the cases where the control cylinders are actuated.

In general, the behavior of the response found when the control cylinders are rotating is almost the same for both configurations. The amplitude increases to reach a maximum value and remains constant there after. It seems that the behavior is similar to the results obtained by Govardhan & Williamson (2002), with a constant amplitude and frequency after the synchronization range systems with $m^* < 0.54$, which is much smaller than this experiment ($m^* = 1.09$). On the previous chapter some cases presented the same behavior, where the lower response branch essentially ceases to exist.

When the tangential velocity remains constant, the amplitude of response present a similar behavior of VIV with initial, upper and lower branches followed by a desynchronization region. However, with a substantial reduction of the amplitude of oscillation, frequency of oscillations is intermittent in the desynchronization region.

The $\overline{C_D}$ values for the cases when the control cylinders are actuated do not present a substantial reduction when compared to the fixed control cylinders. In both configurations, when $U_c/U > 9$, the mean drag coefficient remains at a constant value $\overline{C_D} \approx 3$. When the tangential velocity remains constant the $\overline{C_D}$ value remains close to the values of the plain cylinder.

6.4 Conclusion

Amplitude of response, frequency of oscillations, mean drag and fluctuating lift coefficients were measured for two configurations of eight rotating control cylinders. Result were

obtained for three different rotation parameters and finally with a constant tangential velocity.

Both configurations reduced the amplitude of response when compared with that of the plain cylinder. In the upper branch, for the two configurations when the rotating parameter is $U_c/U = 2.5$, it reduced about 80% of the maximum vibration amplitude. A similar reduction was found when the tangential velocity remained constant. Nevertheless the best case in terms of VIV suppression is the eight fixed control cylinders, ($d_c/D = 0.08$ with $G/D = 0.10$ and $U_c/U = 0$) which reduced about 99% of the peak amplitude.

In terms of efficiency, the tangential velocity remaining constant along the reduced velocity presented the best performance in comparison to a fixed rotating ratio (U_c/U). At $U_R = 4.2$ it presented the same reduction in the amplitude of response and lift coefficient. In addition, it does not present a restriction regarding the energy supplied to the servo motor as required to maintain the rotation parameter.

The measurements of mean drag demonstrate that these configurations do not reduce the mean drag coefficients as much as reported experimentally by Korkischko & Meneghini (2012) and with numerical simulation by Mittal (2001). A considerable reduction of the mean drag coefficient was achieved when the eight control cylinders are fixed ($U_c/U = 0$).

7 CONCLUSIONS

The suppression of vortex induced vibration of a circular cylinder with fixed and rotating control cylinders has been experimentally studied employing a model with an aspect ratio of $L/D = 7$ and a mass ratio of $m^* = 1.09$. Various geometric parameters, such as diameter ratio ($d_c/D = 0.04, 0.06$ and 0.08), cylinder gap ($G/D = 0.05, 0.10$ and 0.15) and the configuration of two, four and eight control cylinders, have been investigated.

For the first campaign of experiments with the static cylinder, the range of Reynolds number was $5000 < Re < 50000$ and the control cylinders were fixed. All the cases were compared to the hydrodynamic forces measured on a plain cylinder. Tests showed that the configuration of two control cylinders increased the mean drag in about 10% on average when compared with that of a plain cylinder. On the other hand, the configurations of four and eight control cylinders presented a reduction of 50% and 33%, respectively. Configurations of four and eight control cylinders also showed, on average, a fluctuating lift of almost zero. It was found that the number of control cylinders and their location around the cylinder is an important parameter for the static suppressor model.

The VIV tests with fixed control cylinders were performed for the same Re range and in the range of $2 < U_R < 20$. The results were presented according to their configurations. Most of the configurations reduced the amplitude response compared with that of the plain cylinder on the upper branch. Considering the maximum oscillation amplitude of a plain cylinder, the configurations of two, four and eight control cylinders achieved a reduction of 35%, 40% and 54%, respectively on average. Beyond $U_R > 8$ some cases with two and four fixed control cylinders presented galloping infinitely. Other responses appeared, in which vibration amplitude was almost constant until the end of the test and the oscillation frequency remained close to the frequency of vortex shedding. Some other cases reduced the amplitude of response and mean drag coefficient to the levels found for a plain cylinder. Furthermore it was found one case of eight control cylinders ($d_c/D = 0.08$ with $G/D = 0.10$) that reduced the amplitude of response in about 99%, the mean drag coefficient in about 50%, and the fluctuating lift coefficient in about 92%.

Finally, the last campaign of experiments for two configurations of eight rotating control cylinders employed three variations in the rotation parameter as well as a constant tangential velocity. All cases reduced the amplitude of response, on average, in about 80% in relation to the highest peak of the upper branch of the plain cylinder. A constant tangential velocity appeared to be a more efficient case than those in which the control cylinders rotate at a ratio (U_c/U). Nevertheless rotating the control cylinders did not reduce the mean drag coefficient as expected and reviewed in the literature of *moving surface boundary layer* methods.

The present study concludes that, for an omni-directional flow, employing eight fixed control cylinders suppressed VIV better than using rotating cylinders.

7.1 Future works

Future works in this line of research of VIV suppressor should propose;

To identify the emission modes of vortices through velocity fields obtained with a PIV (Particle-Image Velocimetry) system. As a suggestion, one may select the most efficient configurations of each experimental campaign in order to observe the influence of the control cylinders in the wake.

To investigate, for eight rotating control cylinders, the rotation parameter, considering that not all of the eight cylinders need to rotate.

To perform an experimental investigation related to the sensitivity study presented by Patino et al. (2015), where the control cylinders may be positioned in specific zones to stabilize the wake.

To investigate, for omni-directional flows if the suppression persist when eight fixed control cylinders are located around a main cylinder with lower aspect ratio (close to a floating offshore platform).

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