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THE EFFECT OF PLATE LENGTH ON THE BEHAVIOUR OF FREE-TO-ROTATE VIV SUPPRESSORS WITH PARALLEL PLATES

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ABSTRACT

Experiments have been carried out on free-to-rotate parallel plates fitted to a rigid section of circular cylinder to investigate the effect of plate length on the stability of this type of VIV (vortex-induced vibration) suppressor. Measurements of the dynamic response and trajectories are presented for models with low mass and damping which are free to respond in the cross-flow and streamwise directions. It is shown that, depending on a combination of geometric and strucutral parameters, parallel plates might not be able to completely suppress VIV for the whole range of reduced velocities investigated. Plates with length between 1.0 and 2.0 diameters showed instabilities and induced high-amplitude vibrations for some specific reduced velocities. Rotational friction was increased for a second run and all plates stabilised and suppressed VIV for the whole range of reduced velocities tested. An undesirable steady lateral force was also observed to occur for all configurations. Experiments with a plain cylinder in the Reynolds number range from 1,000 to 20,000 have been performed to serve as reference.

Keywords: *VIV suppression, free-to-rotate parallel plates, stability of VIV suppressors, plate length.*

NOMENCLATURE

D Cylinder external diameter
L Length of parallel plates
m^* Mass ratio
ζ Structural damping ratio
f_0 Natural frequency in air
U Flow speed
U/Df_0 Reduced velocity
$\hat{x}$ Streamwise harmonic amplitude of vibration
$\hat{y}$ Cross-flow harmonic amplitude of vibration
Re Reynolds number
St Strouhal number

INTRODUCTION

The development of new suppressors for flow-induced vibrations (FIV) of offshore structures is a topic that became frequent in the literature in the past years. As previously discussed in Assi et al. [1–3], with the advancement of offshore oil exploration research on FIV suppressors was pushed to a new level. "The industry demands suppressors that are not only efficient for low mass-damping systems but also that could be installed under harsh environmental conditions; such is the case for offshore risers" [3].

The present work contributes to the understanding of the mechanism behind a type of free-to-rotate device known as the parallel-plate "fairing". Suppressors of this nature are already available as viable commercial solutions [4, 5] being more and more employed on offshore drilling risers. Drilling risers are not in operation for as long as production risers, therefore fatigue damage is not as important a concern as the loads caused by

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strong currents. Therefore, besides suppressing FIV, suppressors must contribute to reduce drag, consequently reducing pipe bend during drilling operation.

It is known that free-to-rotate suppressors may experience hydrodynamic instabilities that will cause a substantial increase in drag but also prevent it from suppressing vibrations [1]. Actually, an unstable free-to-rotate suppressor may induce the structure into more vigorous vibrations excited by a type of flutter mechanism. Assi et al. [1, 3] have shown that the instability of free-to-rotate suppressors is directly related to the level of rotational resistance encountered in the system as well as geometric parameters such as plate length.

Assi et al. [1, 3] performed experiments in laboratory scale and showed that a free-to-rotate suppressor formed by a single splitter plate may need a minimum rotational resistance (or be above a critical rotational friction) to enable a stable configuration with effective suppression. The same was verified for a free-to-rotate suppressor composed of two parallel plates with 1 pipe diameter in length. They have shown that a minimum rotational resistance is necessary to stabilise the devices. However, they have not investigated the behaviour of the parallel plates against plate length.

Assi et al. [1, 2] have already shown that $1D$-long parallel plates can be very efficient in suppressing both VIV (vortex-induced vibration) and WIV (wake-induced vibration, that occurs when the downstream body of a set is excited by the unsteady wake generated from another body placed upstream [6]). In the present work we set out to investigate if a free-to-rotate parallel-plates device is able to find stable configurations and suppress VIV for various plate lengths.

Assi et al. [1] have shown that suppression of cross-flow and in-line VIV of a circular cylinder, with resulting drag coefficients less than that for a fixed plain cylinder, has been achieved using two-dimensional control plates in low mass-damping systems. A free-to-rotate splitter plate was also found to suppress VIV but instead of remaining aligned with the flow on the centreline of the wake the plate adopted a stable but deflected position when it was released. VIV was suppressed, throughout the range of reduced velocity investigated, and drag reduced below that of a plain cylinder. Cimbala and Garg [7] had also observed this bi-stable behaviour for a free-to-rotate cylinder fitted with a splitter plate.

Particle Image Velocimetry (PIV) measurements showed that on the side to which the plate deflected the separating shear layer from the cylinder appeared to attach to the tip of the plate and this had the effect of stabilising the near wake flow [1]. Vortex shedding was visible downstream but this did not feed back to cause vibrations. An unwanted effect was that a steady transverse lift force developed on the cylinder towards the side to which the splitter plate had deflected. This steady lift could be eliminated by using a pair of splitter plates arranged so that the shear layers that sprung from both sides of the cylinder attached to the tips of the plates. When various geometries were tested, the maximum suppression and drag reduction occurred with a pair of free-to-rotate parallel plates installed on the sides of the cylinder.

Assi et al. [1] also found that the level of rotational friction between the free-to-rotate splitter plate and the cylinder played a fundamentally important role. Friction needed to be "high enough to hold the device in a stable position, while still allowing them to realign if the flow direction changes. Devices with rotational friction below a critical value oscillate themselves as the cylinder vibrates, sometimes increasing the amplitude of cylinder oscillation higher than that for a plain cylinder" [1]. All investigated devices with rotational friction above the critical value appeared to suppress VIV and reduce drag. However, if the rotational resistance was above a limiting threshold the suppressors could not rotate and an undesired galloping response was initiated.

Based on previous investigations [1] we believe that parallel plates and splitter plates are able to suppress VIV based on the same fluid-dynamic mechanism. Free-to-rotate parallel plates are not a "fairing" in the strict sense of the term, i.e., they do not make the cylinder a streamlined body. For this to occur the length of the fairing would have to be many times the diameter of the cylinder (as shown in [8], [9], [10]). In essence, we believe the parallel plates act in the near wake with fully separated flow avoiding the interaction between the shear layers and delaying vortex formation and shedding, therefore the same mechanism as splitter plates and short-tail (or teardrop) fairings [1,3].

EXPERIMENTAL ARRANGEMENT

Experiments have been carried out in the Circulating Water Channel of the NDF (Fluids and Dynamics Research Group) at the University of São Paulo, Brazil. The NDF-USP water channel has an open test section which is 0.7m wide, 0.9m deep and 7.5m long. Good quality flow can be achieved up to 1.0m/s with turbulence intensity less than 3%. This laboratory has been especially designed for experiments with flow-induced vibrations and more details about the facilities are described in Assi et al. [11].

Variations on the concept of double plates, some inspired by the early work of Grimminger [12] related to suppressing VIV of submarine periscopes, were the inspiration for the present study. In Grimminger's experiments the plates were fixed since the flow direction was known, but in our work the plates were free to rotate.

A rigid section of circular cylinder with an external diameter of $D = 50$mm was made of a perspex tube (please refer to Figure 1 for details). Four pairs of parallel plates were manufactured varying in length from $L/D = 0.5$ to 2.0. Starting at the $\pm 90^\circ$ points, a quarter of a diameter downstream of the centre of the cylinder, the plates trailed back from the back of the cylinder and were initially aligned to the flow. Although each

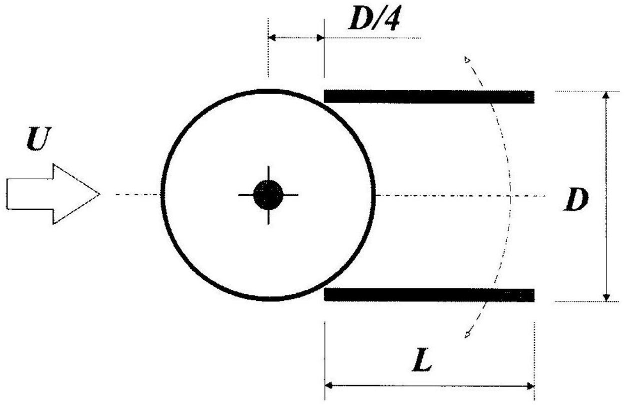


FIGURE 1. Free-to-rotate parallel plates installed on a cylinder. Total length of the suppressor is $D/4+L$ measured form centre of the cylinder to the tip of the plates.

plate varied in length (L), the starting point was kept at a constant distance from the centre, hence the total length of the suppressor measured from the centre of the cylinder was $D/4+L$.

The parallel plates were mounted on ball bearings at the extremity of the cylinder and were always parallel to each other, freely rotating as one body around the centre of the cylinder. A negligible gap between the plates and the cylinder was left in order to avoid contact and increase friction. Rotational friction was not measured in this study, instead it was simply verified if the actual level of rotational friction in the bearings was high enough to stabilise the $1D$-long plates around the expected VIV peak of response. It was later verified that the current level of friction was on the edge of the critical friction required to stabilise the device, since it did not remain stable for the whole range of reduced velocity tested (as will be discussed later).

Models were mounted on a very low damping rig that allowed the cylinder to freely respond in both cross-flow and streamwise directions (Figure 2). The cylinder model was mounted at the lower end of a long titanium tube forming the arm of a rigid pendulum. The top end of the arm was connected to a universal joint fixed at the ceiling of the laboratory so that the cylinder model was free to oscillate in any direction in a pendulum motion. The water channel was filled up to 650mm, resulting in a submerged cylinder length to diameter ratio of 13. The design and construction of the pendular elastic rig was made by Freire et al. [13] based on a previous idea employed by Assi et al. [1, 2] for experiments with VIV suppressors. The present apparatus has been validated for VIV experiments by Freire at al. [14, 15].

Two independent optical sensors were employed to measure displacements in the x and y directions at the mid-length of the model. It should be noted that for a displacement equal to

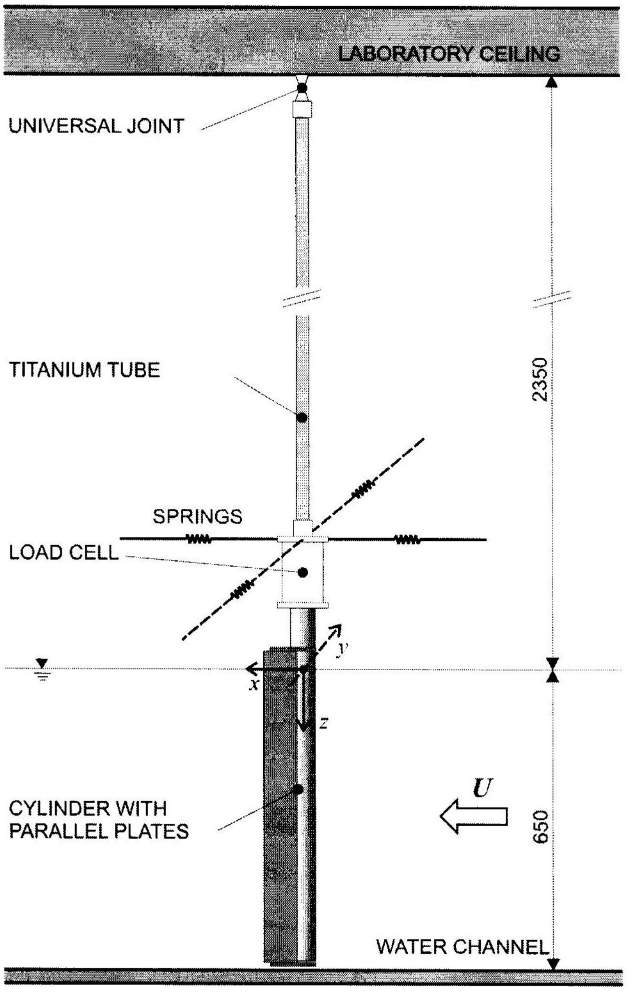


FIGURE 2. Experimental setup: cylinder with parallel plates mounted on the two-degree-of-freedom rig in the test section of the NDF-USP water channel.

1 diameter the inclination angle of the cylinder was only just over 1 degree from the vertical. Two pairs of springs were installed in the x and y axes to set the natural frequencies in both directions of motion. The springs were chosen to provide the same natural frequency (f_0) measured in air in both the cross-flow and streamwise directions.

By using two pairs of springs perpendicular to each other, the assembly has nonlinear spring constants. Movement in the transverse direction will cause a lateral spring deflection in the in-line direction and vice versa. This nonlinearity is minimised by making the springs as long as possible, hence the in-line springs were installed at the end of 1.5m-long wires.

TABLE 1. Structural properties.

	m^*	ζ	$m^*\zeta$
Plain cylinder	2.0	0.02%	0,0040
Cylinder with parallel plates	2.2	0.02%	0,0044

A especially built load cell was attached between the cylinder and the support system to deduce the instantaneous and time-averaged hydrodynamic forces on the cylinder model. In order to obtain the dynamic forces acting, the inertia force (cylinder structural mass times acceleration) was subtracted from the forces recorded by the load cell. (Hydrodynamic forces will not be discussed in the present paper.)

Decay tests have been performed in air in order to determine the natural frequencies of the system in both direction as well as the level of structural damping. The apparatus with one universal joint and four springs turned out to present a very low structural damping of $\zeta = 0.20\%$, measured as a fraction of the critical damping. The total oscillating mass of the system was measured in air, resulting in a non-dimensional mass parameter m^* around 2.0, defined as the ratio between the total mass and the mass of displaced fluid. Consequently, the mass-damping parameter $m^*\zeta$ of the system was kept to the lowest possible value in order to amplify the amplitude of response.

Preliminary tests have been performed with a plain cylinder to serve as reference for comparison. Table 1 presents a summary of the structural parameter for both the plain cylinder and the cylinder fitted with parallel plates.

Measurements were made using a fixed set of springs and the reduced velocity range covered was up to 12, where reduced velocity (U/Df_0) is defined using the cylinder natural frequency of oscillation in the cross-flow direction measured in air. The only flow variable changed during the course of the experiments was the flow velocity U, which, as for full-scale risers, alters both the reduced velocity and the Reynolds number between 1,000 and 20,000.

Throughout the present study, cylinder displacement amplitudes nondimensionalised by the plain cylinder diameter ($\hat{x}/D$ for streamwise and $\hat{y}/D$ for cross-flow) were found by measuring the root mean square value of response and multiplying by the square root of 2 (the so called equivalent harmonic amplitude). This is likely to give an underestimation of maximum response but was judged to be perfectly acceptable for assessing the effectiveness of VIV suppression devices.

PRELIMINARY RESULTS FOR A STRAIGHT CYLINDER

A preliminary VIV experiment was performed with a straight cylinder in order to validate the set-up and methodology. The same pendulum rig was employed, only replacing the model with parallel plates by a plain cylinder with the same diameter.

Figure3 compares the reference cross-flow and streamwise responses obtained for the plain cylinder with those obtained for each suppression device. The observed peak amplitude of $\hat{y}/D = 1.2$ between $U/Df_0 = 5.0$ and 6.0 for the plain cylinder is in good agreement with other results presented in the literature [1, 17]. The general behaviour of the cross-flow response confirms the typical response for two-degree-of-freedom VIV of a system with the same natural frequency in both directions.

The recorded streamwise response presented a peculiar feature. Around reduced velocity 6.0, corresponding to the transition from the upper to the lower branch in the cross-flow response, we observed high-amplitude vibration above $\hat{x}/D = 0.8$ in the streamwise direction. At first sight one might conclude that such a distinct peak could be related to a local resonance between the streamwise excitation and a higher harmonic in that direction. However, this idea was discarded once the time series for the displacement signal was analysed. In fact, it occurred that the cylinder experienced an unstable transition from the upper to the lower branch in the cross-flow oscillations, jumping back and forth from one mode to the other. This alternation between two different levels of amplitude had an effect on the streamwise response due to fluctuations on the mean drag induced by the cross-flow vibrations. As a result, the response appeared as if the cylinder were oscillating with $\hat{x}/D > 0.8$ around a mean position, but in fact it was alternating between two branches of vibration as long as the transition was not completed.

Although the cylinder was initially aligned in the vertical position, in flowing water the mean drag displaces the cylinder from its original location reaching a slightly inclined configuration from the vertical. This was judged not to be detrimental to the experiment, hence the inclination of the cylinder was not corrected between each step of increasing flow speed. The same procedure was adopted for the cylinders fitted with parallel plates.

RESULTS FOR CYLINDERS FITTED WITH FREE-TO-ROTATE PARALLEL PLATES

Figure 3 also presents two sets of data with cross-flow and streamwise response curves for all four suppressors investigated. The first run, with low rotational friction, is marked with black symbols, while the second run, with high rotational friction, is marked with red symbols. We shall discuss the low-friction run (black symbols) first.

The general behaviour of all free-to-rotate parallel plates shows a remarkable reduction in vibration in both direction for most of the reduced velocity range investigated. Nonetheless, not all parallel plates were found to be stable for the whole range of reduced velocities given the minimum level of rotational friction provided by the bare bearings.

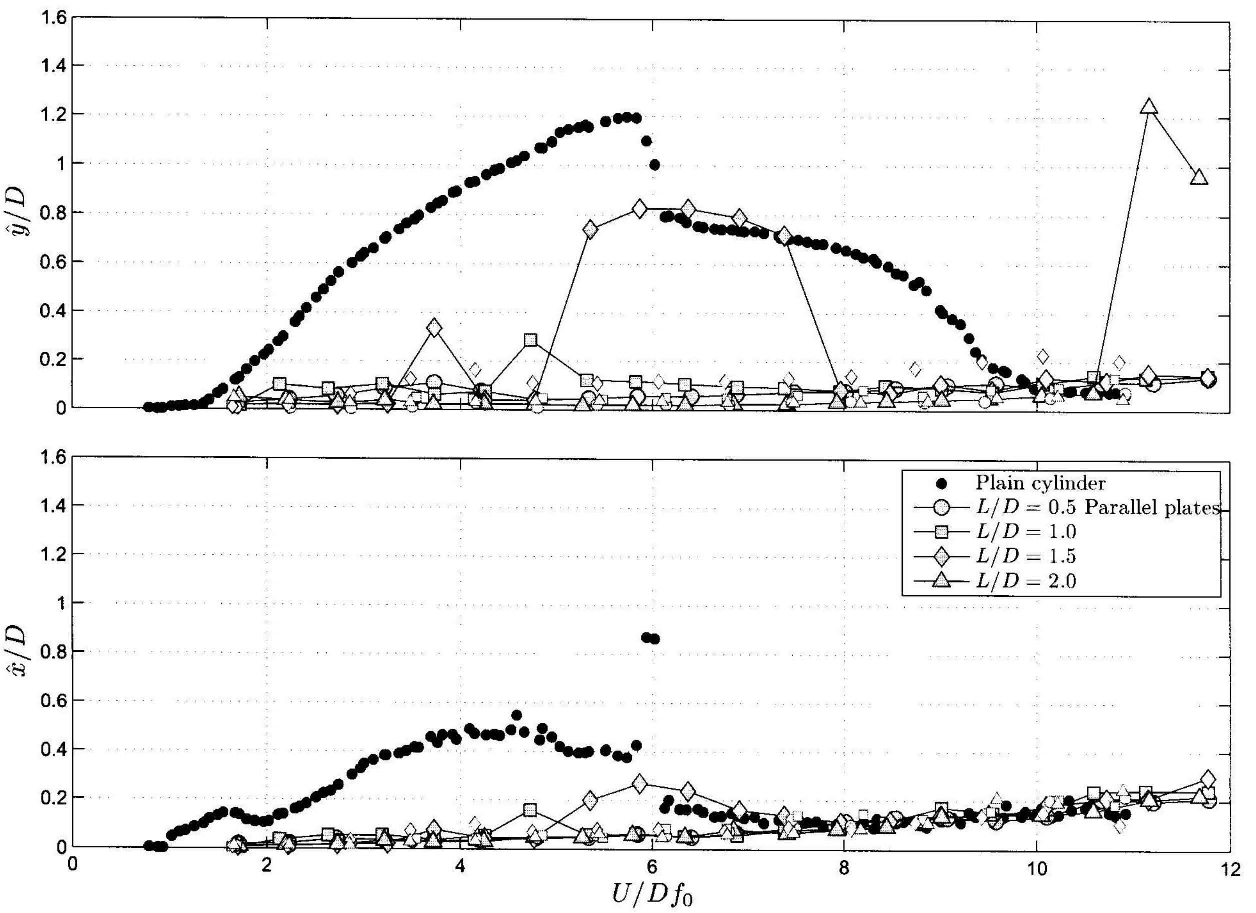


FIGURE 3. Cross-flow ($\hat{y}/D$) and streamwise ($\hat{x}/D$) amplitude of vibration versus reduced velocity for a plain cylinder compared to cylinders fitted with parallel plates of various lengths. Low-friction run in grey symbols; high-friction run in red symbols.

Parallel plates with $L/D = 0.5$ were able to practically mitigate vibrations below $\hat{y}/D = 0.2$ for the whole range of reduced velocities even during the cross-flow and streamwise resonances. Parallel plates with $L/D = 1.0$ showed a similar behaviour except for a single point of $\hat{y}/D \approx 0.3$ around $U/Df_0 = 4.5$. No considerable streamwise vibration was observed for either configuration.

Parallel plates with $L/D = 1.5$ showed the most distinct behaviour with a localised point of $\hat{y}/D \approx 0.35$ at $U/Df_0 = 3.5$, followed by a sudden reduction and followed by a higher branch of considerable $\hat{y}/D \approx 0.8$ between reduced velocities 5.0 and 7.5. It was observed that during this branch the plate was unable to find a stable position but oscillated vigorously exciting the system into severe movements.

Curiously, parallel plates with $L/D = 2.0$ presented rather good suppression for most of the reduced velocity range, but a sudden point of $\hat{y}/D \approx 1.2$ appeared around reduced velocity 11. Again, the plate was not able to find a stable configuration and severe displacements resulted. Such a strong response past the synchronisation region for VIV is strong evidence that the physical mechanism might not be related to pure vortex excitation, but could be related to some other kind of fluidelastic instability such as flutter.

The same could be occurring for the $L/D = 1.5$ plate. Even though the highest displacements appeared around the region where a plain cylinder would be changing from the upper to the lower branch, the cylinder with parallel plates could have been disturbed by vortex shedding before being driven into a flutter-like form of excitation. This explanation might sound rather speculative at this point, but we believe PIV measurements of the wake around the cylinder and the plates could provide data to support or deny this hypothesis.

Now, turning to the high-friction run (red symbols), we observe that all four suppressors were able to find stable positions and suppress VIV below $\hat{y}/D \approx 0.2$ and $\hat{x}/D \approx 0.2$ for the whole range of reduced velocities investigated. The only difference between the two runs was that additional friction was added to the bearings, proving that given enough rotational friction the same suppressor might be able to stabilise and suppress vibrations. We shall return to this point when discussing the trajectories in the next section.

Results obtained for the $1D$-long parallel plates are in good agreement with previous experiments reported by Assi et al. [1, 16, 17]. Although their parallel plates had a slightly different geometry than the ones tested in this experiment (their plates started flush from the $\pm 90°$ points of the cylinder, leaving no $D/4$ gap down from the centre), the general behaviour agrees rather well.

Although not discussed in detail in this paper, it was also verified that all parallel plates were able to reduce drag below that of a fixed cylinder of the same diameter.

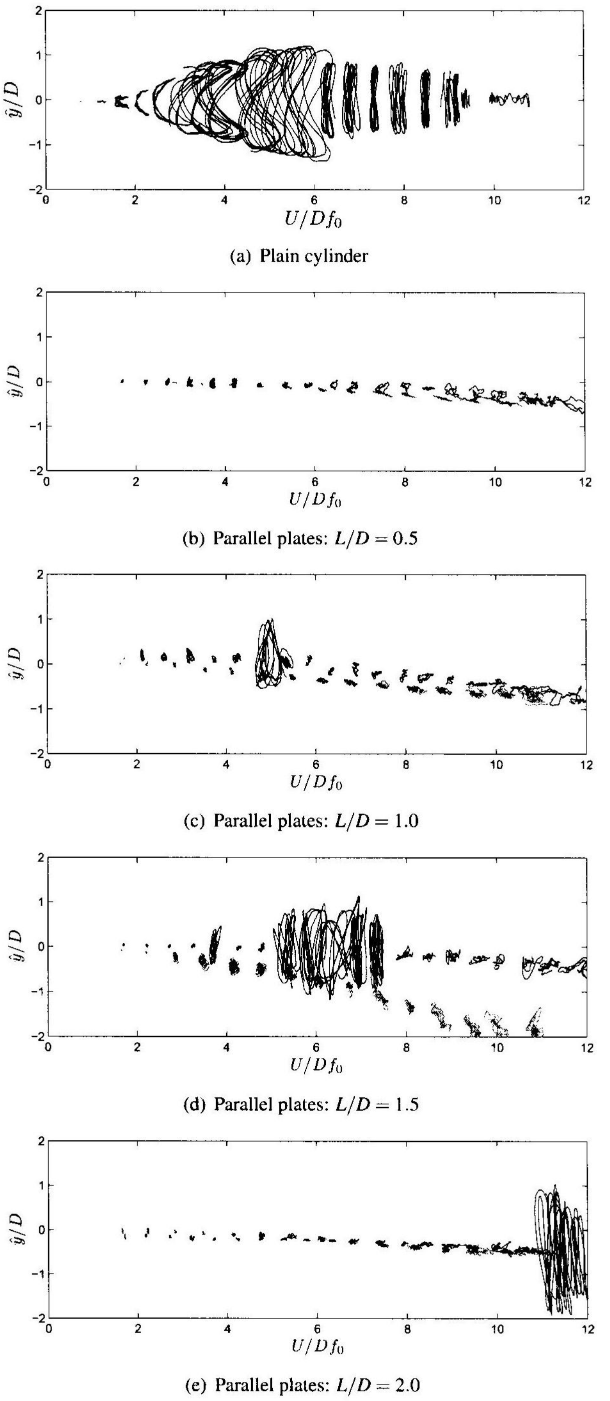
TRAJECTORIES AND LATERAL FORCE

Figure 4 compares samples of displacement trajectories obtained for the plain cylinder and the four suppressors for the low-friction run (in black) and high-friction run (in red). Trajectory figures are a simple and qualitative manner to analyse the response presented in Figure 3.

The plain cylinder response in Figure 4(a) presents characteristic figures typical of VIV in two degrees of freedom. A C-shaped trajectory at the initial branch progressively changes into an eight-shaped trajectory up to the peak amplitude at the upper branch. When the response changes from the upper to the lower branch, around reduced velocity 6.0, the trajectories immediately take a flatter shape with reduced displacement in the streamwise direction.

Trajectories for the four suppressors investigated are very different from the plain cylinder VIV of Figure 4(a). In general terms the cylinders fitted with free-to-rotate parallel plates present small movement around a mean position for most of the reduced velocity range except for a few points during the low-friction run. Sometimes the parallel plates were unable to stabilise and high-amplitude vibrations were registered for a few cycles, as illustrated in Figure 4(c) for $L/D = 1.0$ at $U/Df_0 \approx 5.0$. A similar sporadic behaviour is observed for the highest reduced velocities of $L/D = 2.0$ in Figure 4(e).

However, the most interesting behaviour is observed for the low-friction $L/D = 1.5$ case in Figure 4(d), a case which sustained high-amplitude vibrations for reduced velocities around the transition from the upper to the lower branch. For some reason still unexplained — but believed to be related to the geometric and structural parameters of the system — the plate remained unable to stabilise not only for a localised point but for



(a) Plain cylinder

(b) Parallel plates: $L/D = 0.5$

(c) Parallel plates: $L/D = 1.0$

(d) Parallel plates: $L/D = 1.5$

(e) Parallel plates: $L/D = 2.0$

FIGURE 4. Trajectories of motion for a plain cylinder compared to cylinders fitted with parallel plates of different lengths. Low-friction run in black; high-friction run in red.

a wider range of reduced velocities.

While Figure 3 plots the harmonic amplitude of vibration, (i.e, where the mean displacement has been removed) Figure 4 kept the information of the real cross-flow position of the cylinder through time. In Figure 4(a) we notice that the plain cylinder describes eight-shaped figures around the zero position in the y axis. On the other hand, plots from (b) to (e) reveal that when the parallel plates were able to suppress oscillations the cylinder presented a drift in the lateral direction that increased with flow speed.

Assi et al. [1,3] showed that such a drift is due to occur for a single splitter plate that finds a deflected but stable position to one of the sides while suppressing VIV. They showed that the system would experience a steady lift force towards the side to which the plates had deflected. However, they did not find this behaviour to occur for parallel plates, which, differently from splitter plates, did not need to find a deflected position to stabilise.

In the present work we found that this may not be the case for all plate lengths or rotational friction. From Figure 4 it is clear that all cylinders with stable parallel plates experienced a steady lift force to one side that happened to be the side the plate presented a very small defection. Such an small angle could not be determined in the present study but is evident in the trajectory plot.

Curiously, the $L/D = 1.5$ plate (Figure 4(d)) stabilised with a larger deflection angle in the high-friction run than in the low friction. As a result, a stronger lift force generated on the system increased the lateral drift as observed in the red trajectory curves. As mentioned above, rotational friction was not quantified for the low-friction and high-friction runs, but we can suppose that the high level of friction on the second run was enough to hold the plate in a slightly larger angle than that encountered in the low-friction run.

We may reason that the main hydrodynamic difference between models employed by Assi et al. [1] and those in the present study is exactly the gap of $D/4$ left between the shoulders of the cylinder and the leading edge of the plates. Because flow separation is so important to stabilise (or unstabilise) such systems [1], we can suggest that the leading edge of the plate might be playing an important role in the separation and reattachment mechanism encountered by be shear layers. We intend to employ PIV measurements to investigate the phenomenon in the near future.

Assi et al. [1] concluded that rotational inertia played no significant role in the stability of single splitter plates, however this might not be the case for the oscillator formed by two parallel plates. The dependency of the stability on this parameter is yet to be investigated.

CONCLUSIONS

In the present work we have investigated the stability of free-to-rotate parallel plates as far as plate length from $L/D = 0.5$ to 2.0 is concerned.

The results presented in this study raise the issue that even a symmetrical device as parallel plates may be susceptible to instabilities depending on geometrical and structural parameters such as plate length and rotational friction.

We conclude that:

(i) Even though we have not quantitatively investigated geometric or structural parameters necessary to stabilised the plates, we believe that the configuration set in the low-friction run of this experiment is on the edge of a combination that might provide plate stability. This conclusion may be drawn because we observed that sometimes the cylinder would oscillate for many cycles and suddenly vibration would be suppressed for other many cycles for the same flow speed. In addition, when rotational friction was increased for the second run vibration was suppressed for all flow speeds.

(ii) The suppression effectiveness and drag efficiency must be directly related to plate length, since a parallel plate of $L/D = 1.5$ showed less suppression than the others for the same reduced velocity and Reynolds number range. Because we could not find a continuous trend while varying L/D we have reasons to believe that there might be critical plate lengths able to interact with the separated flow (or the separation bubble) causing positive or negative effects on the stability of the system. This also points out that there might be an optimum plate length to increase suppression and reduce drag.

(iii) An undesirable lateral force appeared to act on the system causing the cylinder to drift to one side. We suppose this is being caused by a small deflection of the plates (although such a deflection angle was too small to be noticeable or measured in the present work) or asymmetrical flow separation around the cylinder.

Future work in this topic should focus on the understanding of the suppression mechanism and concentrate on performing flow measurements to analyse the interaction between the separated flow and the free-to-rotate parallel plates.

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