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2010

# Flow-induced vibration mitigation using attached splitter-plates

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RESEARCH ONLINE

## Flow-induced vibration mitigation using attached splitter-plates

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### Abstract:

Prior stationary cylinder studies have continually demonstrated the effectiveness of attached splitter-plates in reducing drag and lift coefficients of cylinders in steady uniform flow. The use of attached splitter-plates has even been reported to be able to completely eliminate vortex shedding in fixed cylinder investigations. In the present study, the proposed utility of attached splitter-plate wake-stabilisation as a passive control mechanism for vortex-induced vibration (VIV) mitigation was investigated. A range of splitter-plate ratios from  $l/D=0$  to 4 were examined over a reduced velocity interval of  $U_r=3$  to 60. The addition of splitter-plates resulted in the desired effect of decreasing the maximum drag and lift coefficient values experienced by the bare cylinder. The amplitude response however was markedly increased at low splitter-plate lengths. A galloping type response was observed with the addition of even small splitter-plates to the cylinder. The response of the cylinders at low splitter-plate ratios appeared to be strongly influenced by vortex shedding. Key characteristics of the response such as the abrupt decrease in oscillation amplitude at higher reduced velocity aligned well with the bare cylinder vortex-induced vibration response. With increasing splitter plate ratio, there appears to be a smooth transition from pure vortex-induced vibration to a galloping type response strongly influenced by the vortex shedding at low reduced velocity and a predominantly galloping response at higher reduced velocity. Vibration mitigation was only achieved at splitter-plate lengths of  $l/D \geq 2.8$  where no significant vortex-induced vibration or galloping type response was observed.

*Keywords: vortex-induced vibration, galloping, splitter plates, vibration mitigation, wake stabilisation*

### 1. INTRODUCTION

Flow past a bluff object such as a circular cylinder generates vorticity due to the shear present in the boundary layer. This vorticity in the flow field coalesces into regions of concentrated vorticity known as vortices on either side of the cylinder. Flow above a threshold Reynolds number allows perturbations in the flow upstream to cause one of the vortices to grow larger. This vortex, with higher flow velocities and accompanying lower pressures, draws the smaller vortex from the opposing side across the wake centreline. The opposite vorticity from this smaller vortex severs the vorticity supply of the larger vortex, allowing it to convect downstream (Sumer & Fredsoe, 2007). This process is repeated in the reverse sense leading to alternating vortex shedding from the cylinder.

When the cylinder is elastically restrained and natural frequencies are introduced a fluid-elastic instability known as vortex-induced vibration (VIV) results. The time varying non-uniform pressure distribution around the cylinder resulting from the vortex shedding causes structural vibrations both inline and transverse to the flow (Hatton 1999). Near the natural frequency of the structure, the vortex-shedding frequency synchronises with the natural frequency and the vibration frequency. One of the primary mechanisms responsible for this synchronisation is the change in hydrodynamic mass,

as demonstrated in the experiments of Vikestad (1998). The range of reduced velocity over which this synchronisation occurs is known as the lock-in range. Mostly, the ensuing vibrations are undesirable, resulting in increased fatigue loading and component design complexity to accommodate these motions. The transverse vibrations also result in higher dynamic relative to static drag coefficients.

In an attempt to mitigate these vortex-induced vibrations, the adoption of splitter plates has been proposed (Every, King & Weaver, 1982). A splitter plate is a rigid plate attached to a structure so that it splits the wake. The splitter plate falls within the category of devices termed wake stabilisers. The intention of these devices is to prevent the interaction of the shear developed in the boundary layer flow at either side of the cylinder (Sumer & Fredsoe, 2007). By limiting this interaction the vortex shedding process is interrupted.

A number of stationary cylinder studies have been conducted to investigate the effectiveness of splitter plates in reducing vortex shedding and the accompanying hydrodynamic forces (see for example Apelt & West (1975) and Anderson & Szewczyk (1997)). The study by Roshko (1954) reported that at splitter plate ratios (i.e. the ratio of splitter plate length to the cylinder diameter) greater than or equal to five, vortex shedding was completely eliminated. The sub-

critical stationary cylinder experiments by Anderson & Szewczyk (1997) with splitter plate ratios of 0.5 and 1 demonstrated significant reduction in vortex interaction and a delay in the formation of vortices by extension of the separated shear layers downstream of the trailing edge. This was also observed in the studies by Roshko (1954) and Bearman (1965).

Gerrard (1966) reported that increasing plate length produced a decrease in the Strouhal number. The investigations by Apelt, West & Szewczyk (1973) and Anderson & Szewczyk (1997) concur, demonstrating that the variation of Strouhal number with splitter plate ratio is non-linear. These studies all agreed that even at small splitter plate lengths (i.e.  $0.25 < l/D < 1$ ) the Strouhal number, drag coefficient and lift forces decreased. Kawai (1990) reported that a drag reduction of up to 36% was attained with the use of splitter plates on stationary cylinders. Studies of stationary cylinders with detached splitter plates have shown similar decreases in lift and drag forces (Hwang, Yang & Sun, 2003; Akilli, Sahin & Tumen, 2005).

There exist a limited number of studies where the effect of splitter plates is investigated when the cylinder is allowed to vibrate (i.e. the cylinder is elastically restrained). The recent studies by Assi, Bearman, and Kitney (2009) and Parssinen & Eloranta (2009) and the discrete vortex modelling of Kawai (1990) for example demonstrated a galloping type response of a cylinder with splitter plate which was

elastically restrained in the transverse direction. The study by Nakamura, Hirata & Kashima (1994) showed that this galloping type behaviour did not align well with accepted galloping theory. Den Hartog's criteria for galloping instability (i.e. the rate of change of the transverse force coefficient at zero angle of incidence) for example did not hold true for the case of a circular cylinder with an attached splitter plate. The present study contributes to the systematic experimental examination of the effect of splitter-plates on the dynamics and kinematics of a cylinder allowed to vibrate transverse to the flow.

## 2. METHODOLOGY

The experimental investigation conducted, examined the single (transverse motion only) degree of freedom VIV response of an elastically mounted rigid cylinder with various splitter plate ratios under steady, uniform current conditions. The apparatus utilised consisted of a towing carriage and a parallel linkage mechanism capable of translation motion in the transverse direction only (the mechanism is illustrated in Figure 1). Inline motion was restricted through the use of diagonal cross-braces between linkages. The splitter-plate length range (see the definition sketch provided in Figure 2) covered was 0 to  $4D$ . The apparatus for the experiment was lightly damped at around 0.3% of the critical damping. Experimental parameter values adopted in the present investigation are listed in Table 1.

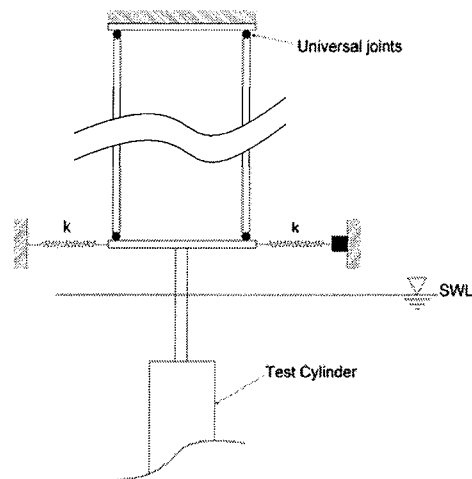


Figure 1 Elevation view of the experimental apparatus

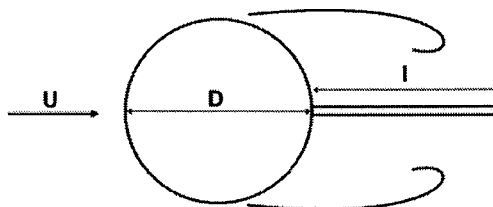


Figure 2 Attached splitter-plate parameter definition sketch

Throughout the present study, the flow velocities are presented normalised (i.e. as reduced velocities,  $U_r$ ) by the product of the cylinder diameter and the natural

frequency of the bare cylinder in still-water. Maximum amplitudes in a time series were defined as the mean of the top 10% half peak to peak values.

Table 1 Experimental parameters

Parameter	Symbol	Value	Units
Cylinder diameter	D	0.0554	m
Cylinder length	L	0.4432	m
Cylinder aspect ratio	-	8.0	-
Cylinder mass ratio	$m^*$	2.36	-
System stiffness	k	53.0	N/m
Damping ratio	$\zeta$	0.003	-
Natural frequency in air <sup>1</sup>	$\omega_v$	9.05	rad/s
Still-water natural frequency <sup>1</sup>	$\omega_{\omega v}$	3.95	rad/s
Splitter plate ratio range	l/D	0 – 4.0	-

<sup>1</sup>Cylinder without splitter plates attached.

To simulate steady, uniform current conditions the structure was towed through initially still water. The ratio of cylinder diameter to channel width was 1:25 (representing around 1% of the channel area). No significant variation in local current velocity due to blockage effects was therefore experienced. Experimentation was conducted in the Reynolds number range  $1.26 \times 10^4$ – $8.40 \times 10^4$  which is within the stable subcritical Reynolds number range. Only smooth test section surfaces were considered.

The cylinder aspect ratio adopted in the present study brought the test section close to commonly employed cylindrical floating structure ratios of around 3 to 5. The experiment did not use test section end-plates. The influence of the end-conditions on the cylinder forcing at this small aspect ratio are likely therefore to have resulted in lowered excitation frequencies. Strouhal number and vortex shedding frequency data are traditionally collected with very large aspect ratios (i.e.  $L/D \gg 100$ ) and commonly, using the walls of the tank effectively as very large end-plates (maintaining two-dimensional flow at the cylinder ends). A number of studies have shown a decrease in the shedding frequency with decreasing aspect ratio. Examples include the low Reynolds number study of Lee & Budwig (1991) and the higher Reynolds number study of Gowda (1975).

Experimentation and analysis of previous work by Gerich & Eckelman (1982) demonstrated regions near the cylinder ends where the shedding frequency was significantly lower (up to a reported 17% below the Strouhal frequency). Below an aspect ratio of approximately 15, the shedding frequency is dominated by these regions (i.e. the shedding is controlled by the end conditions). Further examination of the effect of the end-conditions (with Reynolds numbers up to  $10^4$ ) on vortex shedding by Norberg (1994), also demonstrated a decreased shedding frequency with reduced end-plate size. In the limit of this trend is of course the present case of a cylinder without end-plates.

In the present study, the cylinder was non surface-piercing and the top cylinder end was at a distance  $4D$  below the free surface. No compensating corrections were applied for the surface proximity or to account for the small hydrodynamic forces acting on the rectangular cylinder support structure.

### 3. RESULTS AND DISCUSSION

The amplitude and frequency response curves for the bare cylinder case are illustrated in Figures 3 and 4 respectively.

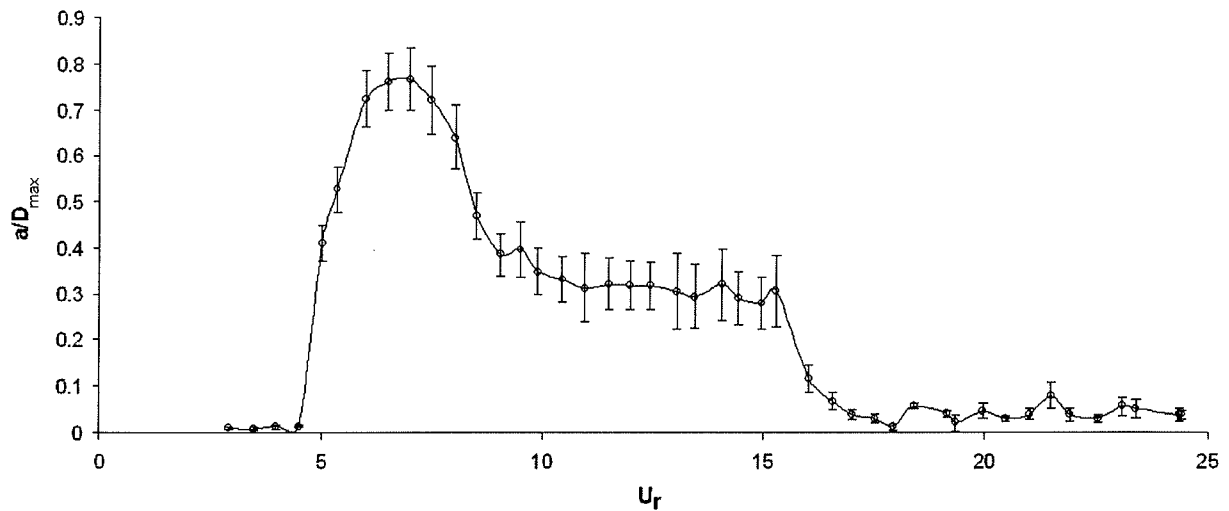


Figure 3 Amplitude response; bare cylinder case  $l/D=0$ .

The initial, upper and lower response branches are visible as is the de-coherence region. Lock-in (where the forcing and vibration frequencies synchronise with the system natural frequency) falls between reduced velocities of approximately 5 and 15. A continual rise in the system natural frequency during lock-in (as a result of the changing hydrodynamic mass) is observed as expected for a low mass ratio system undergoing vortex-induced vibrations (Vikestad, 1998). Since

galloping motion cannot be excited with an axisymmetric shape such as a circular cylinder, the transverse forcing frequency can reasonably be assumed to be the vortex shedding frequency. The Strouhal number from Figure 4 is then approximately  $St=0.13$  (with a fit statistic of  $R^2=0.995$ ). This value is consistent with former low mass and aspect ratio experiments by the author (Stappenbelt & Lalji, 2008).

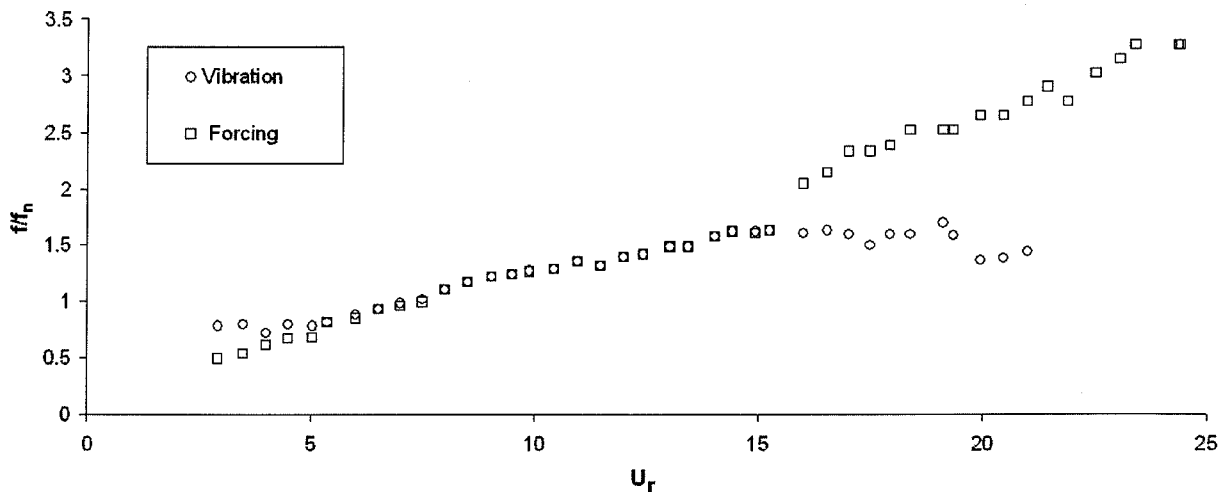


Figure 4 Frequency response; bare cylinder case  $l/D=0$ .

The mean drag coefficient results for the bare cylinder case are presented in Figure 5. The dynamic drag values are observed to increase as a function of response amplitude as described in Det Norske Veritas

(2000). The static drag coefficient of the cylinder (estimated from the data outside the lock-in range) is approximately  $C_d=0.92$  which agrees well with the experimental results reported by Achenbach (1971).

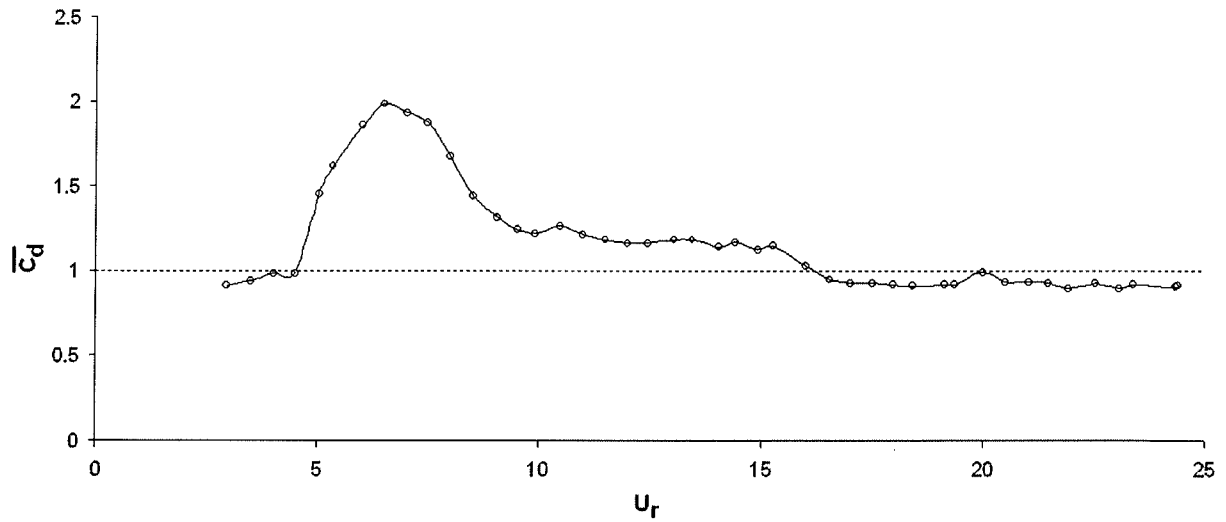


Figure 5 Mean drag coefficient; bare cylinder case  $l/D=0$ .

The amplitude response data for all splitter plate ratios covered is shown in Figure 6. Evident from these plots are the very large vibration amplitudes observed in the splitter plate cylinder results between  $l/D=0.34$  and  $2.4$ . The near linear response curve of the cylinder with

splitter plates agrees well with the galloping type response described in Blevins (2001). As reported by Bearman and Luo (1988) the galloping response onset is at the VIV onset point or higher reduced velocity.

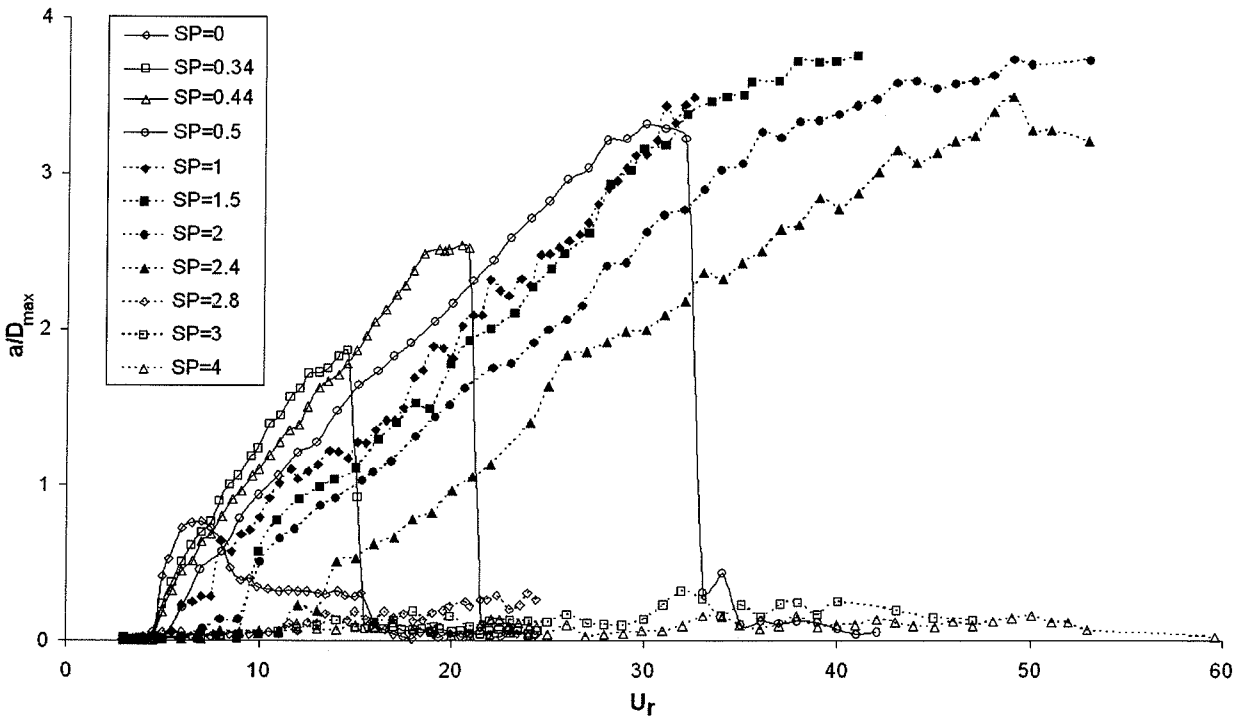


Figure 6 Amplitude response plots; splitter plate ratios  $l/D = 0 - 4$ .

In the splitter plate ratio range of  $l/D=0.34$  to  $0.5$ , the galloping type response halts abruptly at higher reduced velocity. When the responses of the various splitter plate ratio cases are normalised by the true natural frequency (rather than the bare cylinder natural

frequency), it is evident that the location of this transition is at the end of the VIV lock-in range. This behaviour may be explained by a system that is only just stable with respect to galloping motions. A system close to a galloping instability may be excited readily

by vortex shedding forces and the changing relative free stream velocity due to the induced motions (Nakamura, Hirata & Kashima, 1994). For splitter plate ratios of  $l/D=2.8$  to 4, no significant vortex-induced or galloping type vibrations were observed.

For the bare cylinder case during VIV lock-in it may be seen, that the standard deviation (SD) of the amplitude response (see Figure 7) is low. This is especially evident throughout the upper response branch vortex shedding mode. The variation in response amplitudes of the  $l/D=0.34$  and 0.44 cases are interesting as they display a similar low standard deviation which continues until the reduced velocity at which the amplitude response drops abruptly. In contrast to this, the higher splitter plate ratios cases (as shown in Figure 7 with splitter-plate ratios of  $l/D=2$  and 3) show far greater amplitude response variation. All standard deviation plots in Figure 7 converge at higher reduced

velocity and share a common value prior to lock-in or galloping onset.

The mean drag data presented in Figure 8 shows the dynamic drag amplification typical of VIV (i.e. as described earlier with reference to Fig. 5) at low splitter plate ratio ( $l/D < 0.5$ ). This dynamic drag amplification is absent at higher splitter plate ratios. At  $l/D \geq 1$  the drag coefficients measured are equivalent to the static drag coefficient values. These values are again consistent with the drag coefficient data from for example Achenbach (1971). At higher splitter plate ratio, the mean drag coefficient continues to drop with increasing splitter plate length. This is expected as the form drag is further decreased by improving the streamlining of the structure. The maximum drag coefficients at each splitter-plate ratio do not exceed the maxima of the bare cylinder case.

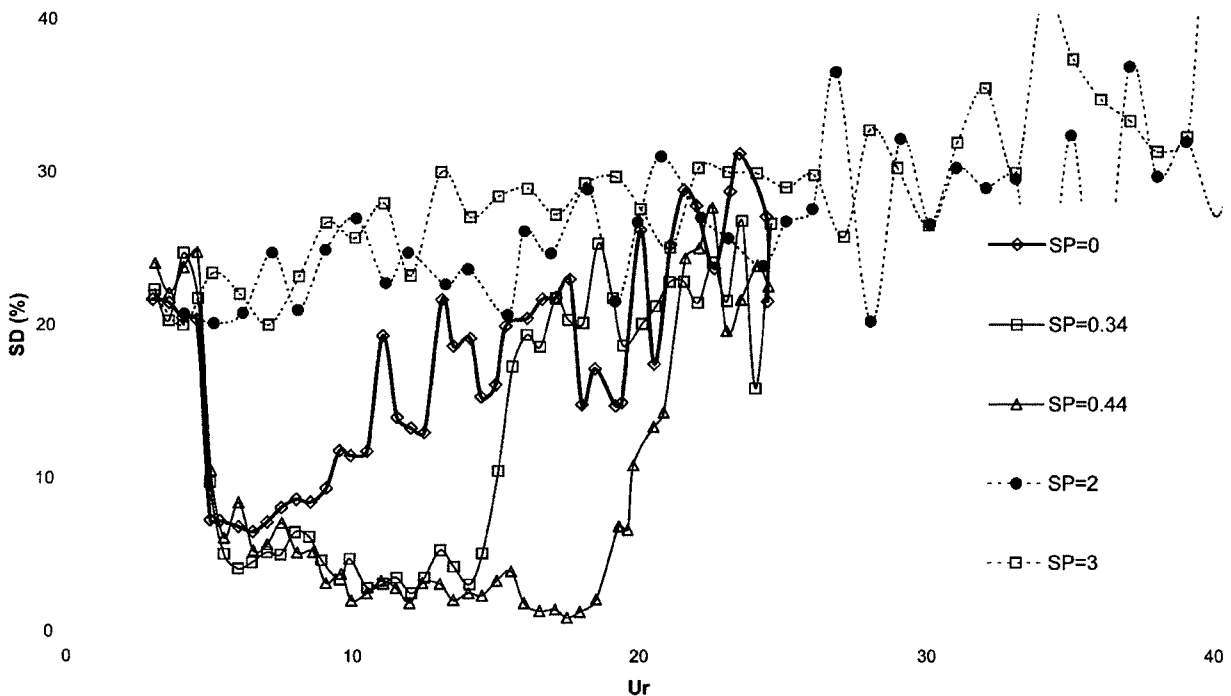


Figure 7 Amplitude response standard deviation; splitter plate ratios  $l/D=0-3$ .



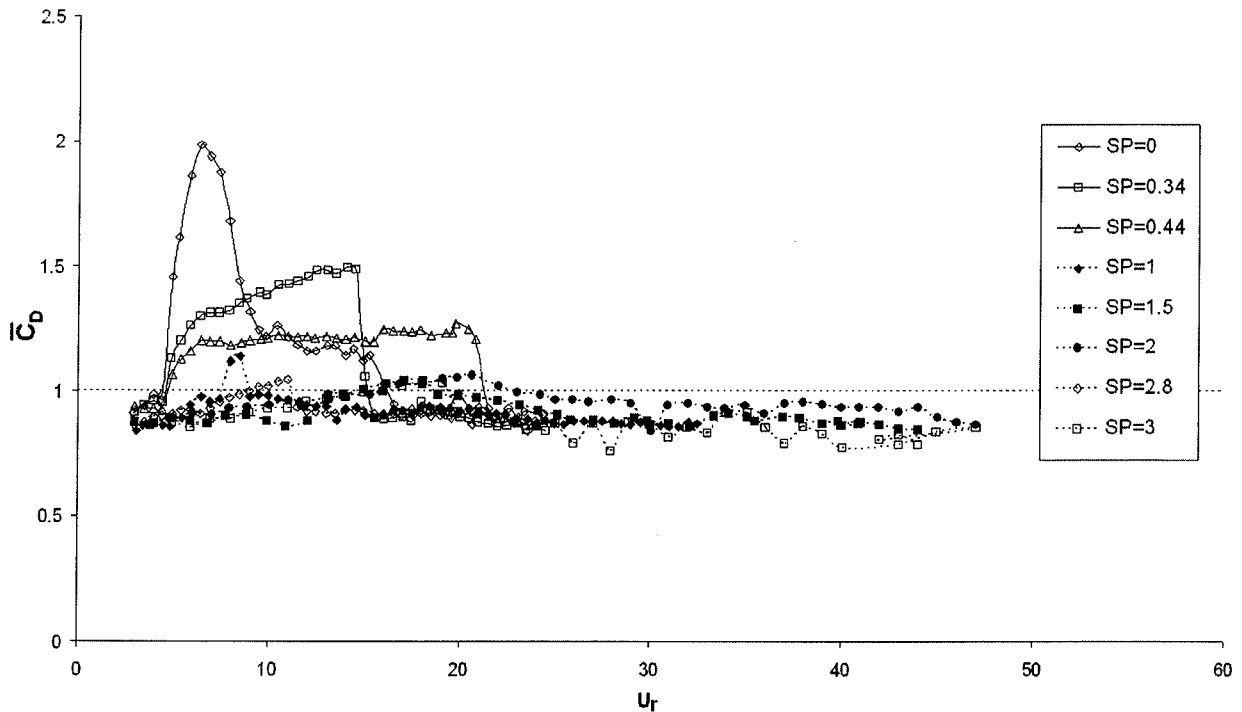


Figure 8 Mean drag coefficients; splitter plate ratios  $l/D = 0 - 3$ .

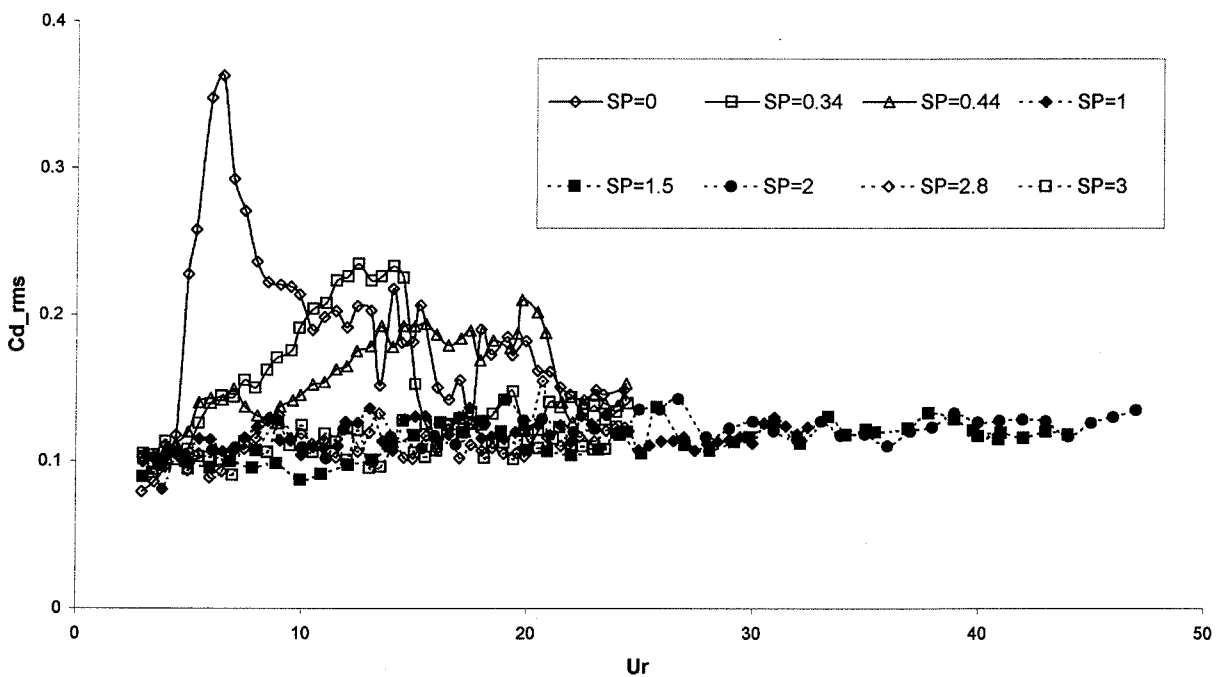


Figure 9 Root mean square drag coefficients; splitter plate ratios  $l/D = 0 - 3$ .

As for the mean drag, distinct VIV response branches are visible in the oscillatory drag coefficient data (Figure 9) of the bare cylinder case. This is again accompanied by drag amplification during the peak

transverse amplitude response (i.e. particularly evident during the upper response branch oscillations). The galloping type response curve can be seen in the cases with splitter plates attached (i.e.  $l/D \geq 0.34$ ). This

response ends abruptly at a reduced velocity consistent with that observed for the amplitude response. Above  $l/D=0.44$ , the oscillatory drag characteristics of the cylinders with various splitter plate ratios become indistinguishable and the dynamic drag amplification typical of VIV is again not observed. All oscillatory drag data in Figure 9 tends to a root-mean-square drag coefficient value of around  $C_{dRMS}=0.1$ , increasing slightly with reduced velocity.

The lift coefficient data for all cases is presented in Figure 10. Note that the lift forces were normalised by the projected area of the cylinder and plate in the transverse direction. Consistent with the amplitude response plots of Figure 6 and the drag coefficient data

of Figures 8 and 9, the lift coefficients show the typical VIV response for the bare cylinder and a galloping type response at low splitter plate ratios. The maximum lift coefficients at each splitter-plate ratio do not exceed the maxima of the bare cylinder case.

The splitter plate ratios of  $l/D=0.34$ ,  $0.44$  and  $1$  demonstrate remarkably similar initial lift coefficient values, decreasing monotonically at different rates. Improved alignment in the rate of decrease of the lift coefficients with increasing reduced velocity may be realised by normalising the flow velocity by the true natural frequency for each splitter-plate arrangement. Above  $l/D=1$ , the oscillatory lift coefficients deviate little with increasing splitter-plate length.

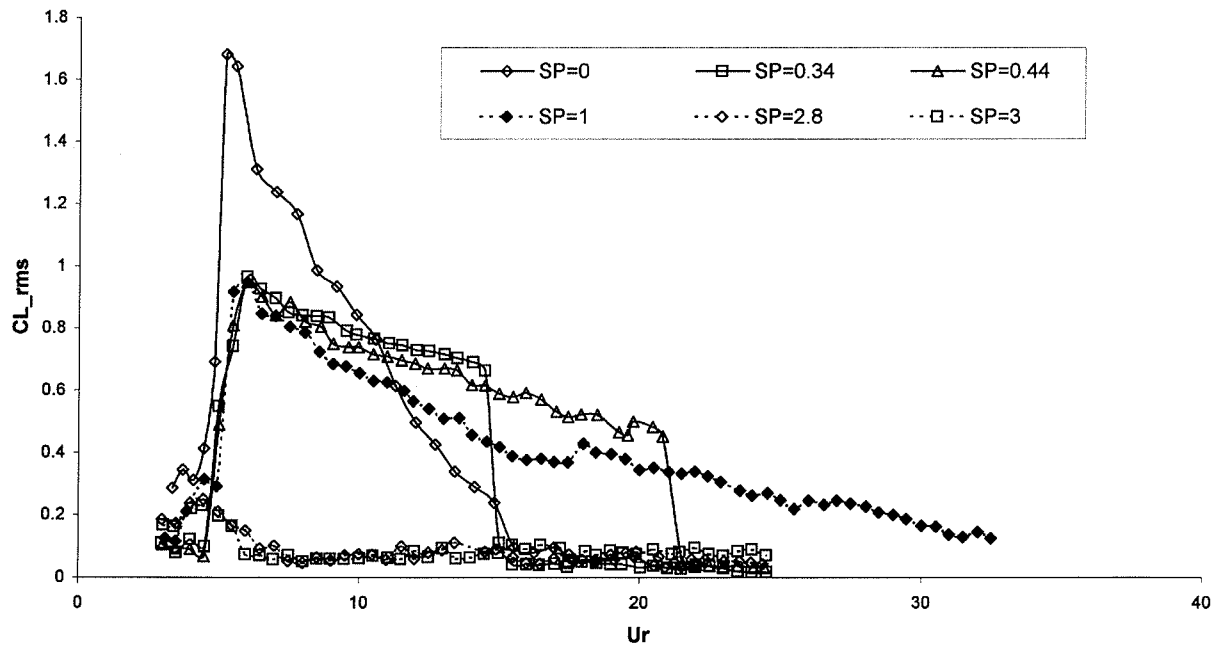


Figure 10 Root mean square lift coefficients; splitter plate ratios  $l/D = 0 - 3$ .

The frequency response of the  $l/D=0.34$  and  $0.44$  cases (Figures 11 and 12 respectively), demonstrate the dual presence of galloping and vortex-shedding forces. Within the resonance region, the response is typical of galloping, with the forcing and vibration occurring at the system natural frequency. Outside this region, where there is no appreciable motion and hence no significant galloping forces, the forcing measured is the

vortex-shedding force. The Strouhal number for  $l/D=0.34$  and  $0.44$  may therefore be determined as  $St=0.087$  ( $R^2=0.9807$ ) and  $St=0.067$  ( $R^2=0.9872$ ) respectively. The observed decrease in Strouhal number with increasing splitter plate length is consistent with previously reported results (Anderson & Szewczyk, 1997).

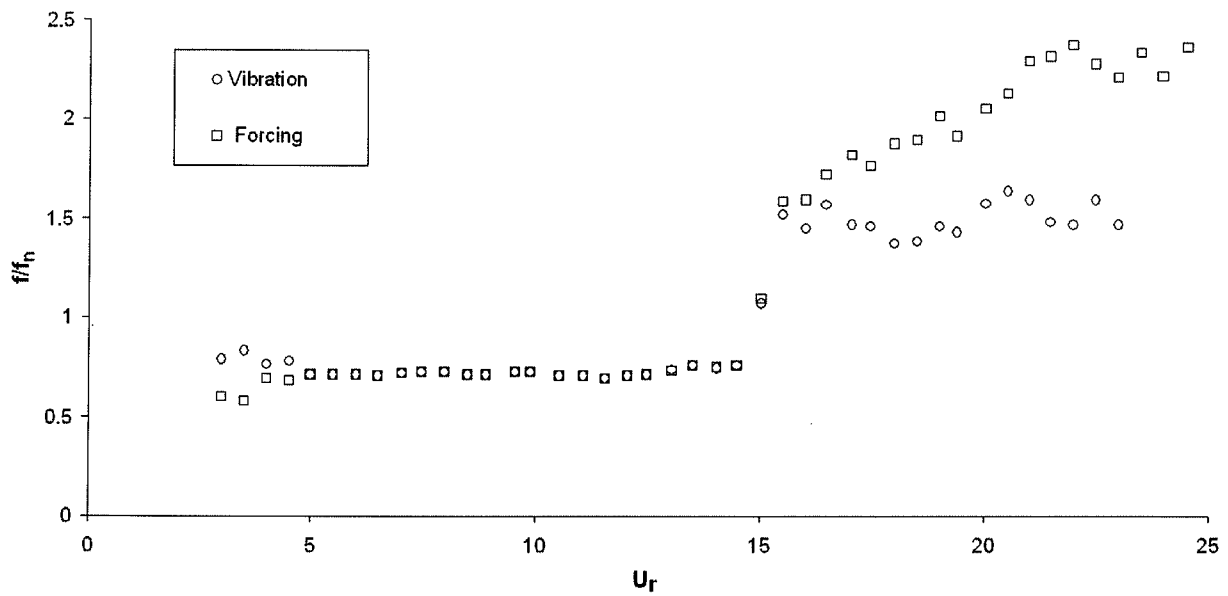


Figure 11 Frequency response;  $l/D=0.34$ .

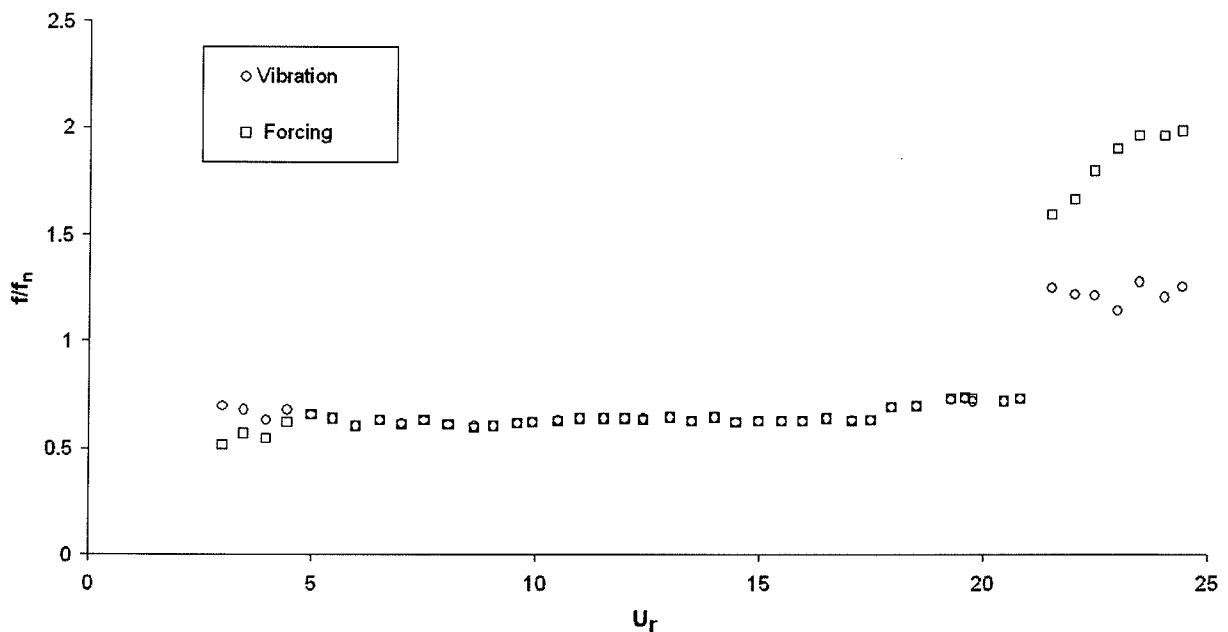


Figure 12 Frequency response;  $l/D=0.44$ .

Figure 13 contains the vibration frequency response data for selected splitter plate ratios where the galloping type response is evident. The natural frequencies of each case correspond well with the observed galloping frequencies. It may be noted from

this figure that there exists a decrease in the system natural frequency with increasing splitter plate length. This is due to the increased hydrodynamic damping (with the transverse motions) with longer splitter-plates.

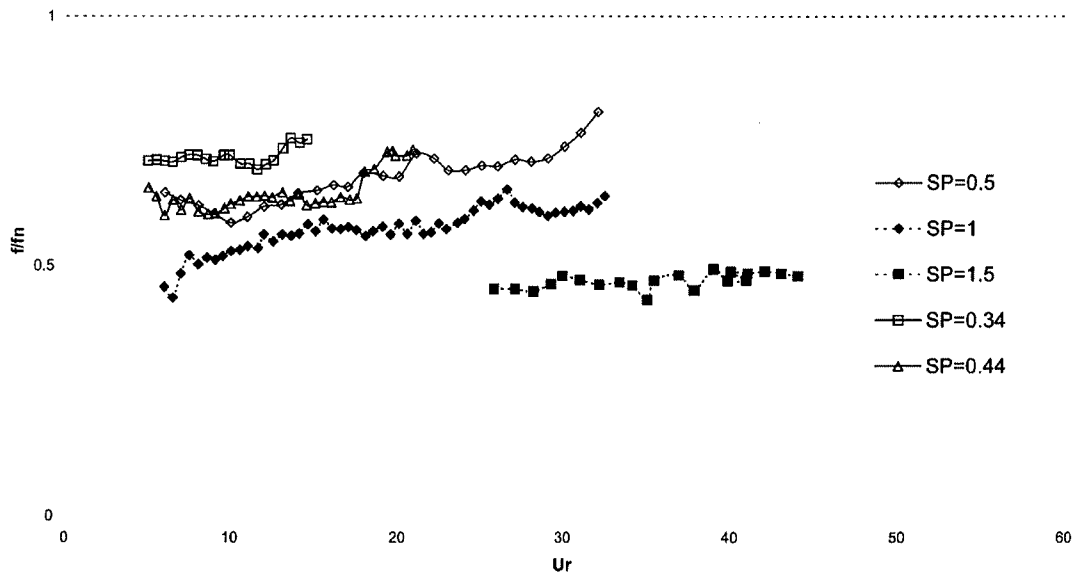


Figure 13 Vibration frequency response;  $l/D = 0.34 - 1.5$ .

#### 4. CONCLUSIONS

The introduction of rigid attached splitter plates to an elastically restrained cylinder, free to vibrate transverse to the flow, introduces the potential for a galloping type response. As a result, no vibration mitigation was achieved at splitter plate ratios less than  $l/D=2.8$ . Due to the galloping type response generated, significantly higher maximum amplitudes were experienced relative to the bare cylinder case. This of course has implications for the passive control of flow induced vibration using splitter-plates.

The galloping type response for small splitter-plate ratios ( $l/D \leq 0.5$ ) terminates abruptly at the VIV lock-out boundary. These cases also share the low response amplitude variance feature of the bare cylinder undergoing vortex-induced vibration. The mean drag for splitter-plate ratios  $l/D \leq 0.5$  has a form similar to that expected for a cylinder undergoing VIV. In addition, it is evident from the frequency response that both vortex shedding and galloping forces are acting on the structure.

It appears therefore that there is a transition, with increasing splitter plate ratio, from pure VIV (i.e.  $l/D=0$ ) to a galloping type response heavily influenced

by the vortex shedding at low reduced velocity and a predominantly galloping response at high reduced velocity. This transition, between the region where vortex shedding strongly influences the vibrations and where galloping is dominant, is smooth. This is consistent with the observations made in the square-section galloping experiments by Bearman and Luo (1988).

In all cases, a reduction in maximum drag and lift coefficients was realised with the introduction of splitter-plates. This is consistent with prior stationary cylinder investigations (e.g. Apelt & West (1975) and Anderson & Szewczyk (1997)).

For splitter plate ratios of  $l/D=2.8$  to 4, no significant vortex-induced or galloping type vibrations were observed for the low aspect ratio cylinder utilised in the present study. It is only within this region that vibration mitigation was realised.

#### 5. ACKNOWLEDGEMENTS

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