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Interaction of free oscillating flat plate and VIV of a circular cylinder in laminar flow

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Abstract

1 The Flow-Induced vibration around bluff bodies is always of great interest in terms of control
2 and suppressing the motion or amplifying and harvesting energy and the flat plates are known
3 as useful means to make proper changes in flow structure. The objective of this paper is to
4 investigate the effect of a downstream flat plate with one degree of freedom on the wake
5 structure and FIV of an upstream circular cylinder in laminar flow. Numerical simulation is
6 used to simulate the two-dimensional incompressible flow around two rigid objects with
7 identical characteristic lengths and a variable horizontal spacing in a range of $0.5D \leq G \leq 4D$ to
8 focus on the near wake and interaction of two objects. The results show that a free to vibrate
9 flat plate located in the wake of a rigid cylinder not necessarily plays the role of a splitter in
10 the near wake and starts to vibrate in an even small spacing such as $G=1.5D$. The Recirculation
11 zone structure and its length are found to be the main reasons for induced vibration in such
12 close configurations. In the case of simultaneous free oscillation, reducing the horizontal gap
13 amplifies the amplitude of both objects. The interaction of the oscillating plate and the shear
14 layers results in vortices with more strength that affect both objects. This amplification in
15 oscillation amplitude shows that the present concept has a considerable potential to be
16 utilized in the design of an improved energy harvesting system by converting fluid dynamic
17 energy into electric energy from independent objects in a small area.

Keywords

18 Flow-Induced Vibration, Vortex Shedding, Circular Cylinder, Flat Plate, Low Reynolds Number

Introduction

19 The flow around bluff bodies has widespread applications in different industries and
20 understanding its characteristics has been always of great interest. In some cases vortex
21 formation and shedding, by changing the local pressure distribution around the body
22 periodically, results in some types of flow-induced motions (FIMs). This periodic change
23 comes with an asymmetric flow field and applies an oscillatory force which may induce a
24 continuous motion to the objects located in the flow field. This oscillatory force can be either
25 by the upcoming vortices that result in a wake-induced motion or buffeting or induced to the
26 object by its own vortex shedding, which causes Vortex-induced Vibration (VIV) [1].

27 Lots of studies and reviews (such as Sarpkaya [2],[3]; Bearman [4]; Williamson and Govardhan
28 [5]) have been performed to develop the knowledge and gain a better insight into different
29 aspects of FIMs. The studies on FIMs can be divided into three main categories: understanding
30 the phenomena, control and suppression, or enhance and amplifying the motion. Because of
31 their destructive effects on structures, primitive studies were conducted to understand the
32 phenomena and finding a way to suppress the vortex shedding and consequently the induced
33 motions. On the other hand, global warming and other environmental problems have drawn
34 attention to renewable energy sources in the past few decades. Harvesting energy from wind
35 and current has been developed and a great deal of attention has been given to research in
36 the field of FIMs as a clean and renewable energy source.

37 Different approaches are implemented to either suppress the FIMs to prevent structural
38 damages or enhance them for harvest energy. An arrangement of a cylinder as a bluff body
39 and an attached or detached flat plate has been extensively studied in the past decades.

40 Although the primary studies were mainly focused on control induced motions, several

41 attempts have been conducted to enhance them and to improve energy harvesting in recent
42 years.

43 Early studies were devoted to find out the effect of a flat plate attached to the rear side of a
44 fixed cylinder. In 1954 Roshko showed that the flat plate works as a splitter, stabilizes the
45 wake, and delays the vortex shedding at a subcritical Reynolds number [6]. Change in vortex
46 formation length and a lower Strouhal number are other results of this configuration found
47 by Gerrard [7]. Apelt [8] found higher base pressure, lower drag and Strouhal number, and a
48 narrower wake by stabilizing the separation point on the circular cylinder. The effect of a
49 relatively short splitter plate ($G < 3D$) in supercritical Reynolds numbers examined by Adachi
50 [9] and a noticeable vortex suppression was observed. Kwon and Choi [10] studied the effect
51 of the plate length in the laminar flow and concluded that attached splitters longer than a
52 specific value completely suppress the vortex shedding. They also found that the Strouhal
53 number rapidly decreases with increasing the length as long as $G < 1D$.

54 Detached flat plates are also used to control the vortex shedding of cylinders. Ozono [11]
55 studied the effect of detached splitter plate in subcritical Re numbers and found it useful to
56 make a change in base pressure, Strouhal number, and even vortex shedding suppression. Its
57 effectiveness is also reported to be connected to its both horizontal and vertical location.
58 Hwang [12] observed similar behavior in laminar flow and introduced a critical horizontal
59 spacing ($G = 2.6D$) for maximum control efficiency. Akili [13] found that placing the splitter
60 beyond $2D$ has no considerable effect on the cylinder. Dehkordi and Jafari [14] In an attempt
61 to achieve maximum suppression showed that the critical horizontal spacing and the plate
62 vertical location are significantly related. They also found that parallel splitters in proper
63 location result in a diminution of 20% in the drag force.

64 Based on the studies mentioned above, it can be concluded that while both of the attached
65 and detached plates can make a delay in vortex shedding and even a complete vortex
66 suppression, the detached plates effectiveness are strongly more dependent on their length
67 and location.

68 In 1990, Kawai studied the free vibration of a circular cylinder with an attached splitter plate.
69 While this configuration prevents the shear layers to interact, the interaction between the
70 flapping of the shear layers and the plate tip results in galloping with a high amplitude and
71 low-frequency oscillation [15]. A bare circular cylinder can only be vortex-excited, but it can
72 gallop in presence of a long stationary splitter plate with adequate length and this finding can
73 be applied to any bluff body whether its cross section is sharp-edged or smooth [16].
74 Stapenbelt showed a transition from VIV to galloping with an increase in the plate length. The
75 transition occurs in lower reduced velocities as the plate length increases [17]. Galloping with
76 high oscillation amplitude is also reported by Assi and Bearman for a circular cylinder with a
77 simple and slotted splitter[18]. On the contrary, Wu et al. found that a hinged plate could
78 effectively suppress the VIV of a circular cylinder especially in low reduced velocities[19].

79 Liang et al. divided the cylinder response, based on the splitter length, into four groups: VIV,
80 the interaction between galloping and VIV, a combination of the velocity-restricted excitation
81 and interaction of VIV and galloping [20]. Sun et al. in an attempt to find the reason for the
82 transition between VIV and galloping found that the lift components generated from the
83 splitter and the cylinder behave, respectively, as the driving force and the suppressing force
84 of galloping. Therefore it can be said that the transition is a result of competition between
85 these forces [21]. A single downstream splitter alleviates VIV in low reduced velocities while
86 exciting the galloping in higher velocities. On the other hand, an upstream splitter delays

87 vortex shedding and makes the wake narrower. Zhu et al. also stated that using simultaneous
88 oscillation can also suppress the galloping completely [22].

89 It can be concluded that using splitter plates to control the vortex shedding and flow-induced
90 motions, whether attached or not, have different and even sometimes contradictory effects
91 on the flow field and the cylinder behavior. Their length, rigidity, location, and type of
92 connection to the cylinder are the main effective parameters.

93 In the past few years, some studies were focused on harvesting energy from different
94 configurations of a cylinder and a plate. Installing an attached piezoelectric plate to the rear
95 side of a cylinder has been investigated in different studies. An et al [23] proposed a novel
96 vortex-induced piezoelectric energy converter (VIPEC) in which vortex-induced pressure
97 difference acts on the plate and drives the plate to squeeze piezo patches to convert fluid
98 dynamic energy into electric energy. They found that Different lengths of the plate change in
99 shear layers interaction and delays vortex shedding. Wang et al. investigated the VIV of a
100 piezoelectric cantilever plate located in the wake of a circular cylinder. They divided the gap
101 between two bodies into two sections: suppression, vortex shedding and showed that this
102 gap is related to the plate's length [24]. Armandei and Fernandes [1] in 2016 tried to extract
103 marine current energy through buffeting. They elastically installed a flat plate with rotational
104 freedom in the wake of a modified D shape cylinder. They found buffeting response
105 independent from the distance and the efficiency is dependent on the elastic axis position
106 and the spring rate. These results show that while the wake of any bluff body can be
107 considered as a source to harvest energy, utilizing a proper configuration is essential for
108 higher efficiency.

109 From the studies reviewed above, it is known that while the flat plates are widely used to
110 control and suppress the FIMs, they are also noticed as means to amplify the motion of the
111 cylinder and even harvesting energy directly. On the other hand, small changes in the length,
112 location, and degree of freedom may change the system response significantly. It seems there
113 are still unknown aspects in the interaction of a plate and a circular cylinder.

114 In this work, the FIMs of a combination of a circular cylinder and a flat plate with one degree
115 of freedom has been numerically studied in a laminar flow regime. The plate is located in the
116 cylinder wake, and the gap spacing is defined as the distance between the trailing point of the
117 cylinder and the leading edge of the plate. The objects are independent and each of them is
118 able to oscillate in crossflow direction freely. This study aims at exploring the effects of
119 horizontal spacing on the flow structure and system response when the plate oscillates
120 individually or both objects oscillate simultaneously.

Nomenclature

A	Plunging Amplitude
D	Circular Cylinder Diameter
L_P	Flat Plate Length
G	Non Dimensional Horizontal spacing
m^*	Mass Ratio
ζ	Damping Factor
CL	Lift Coefficient
CL_{rms}	Root Mean Square of Lift Coefficient
CD	Drag Coefficient
C_p	Pressure Coefficient
C_{p_b}	Base Pressure Coefficient
f	Plunging Oscillation Frequency
f_n	Transverse Natural Frequency
ρ	Fluid Density
m	Body Mass
m_A	Added Mass
K	Transverse Stiffness Factor
U_{in}	Free Stream Velocity
U_r	Reduced Velocity
t	Physical Time
T	Non Dimensional Time

Problem description

121 The FIV of a circular cylinder and a flat plate located in the wake is modeled by a mass-spring
122 system with one degree of freedom proposed by Facchinetti et al. [25]. Figure 1 presents the
123 schematic view of flow past the objects which are elastically mounted perpendicular to the
124 uniform flow. D and L_p represent the cylinder diameter and the plate's length respectively.
125 The streamwise gap between the rear end of the cylinder the leading edge of the plate is
126 shown by $G=L/D$. In all cases the length of the flat plate and the cylinder diameter are equal
127 and the plate thickness is set as $\delta = 0.03D$.

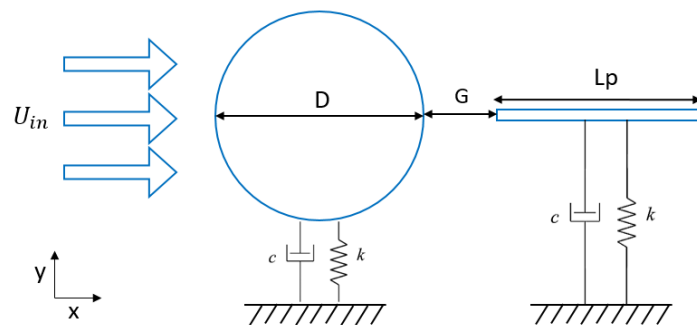


Figure 1 – schematic diagram of flow past a free to oscillate cylinder and flat plate in tandem arrangement

128 In FIV, the Reynolds number, the mass, and damping ratios play a significant role in the
129 dynamic response of the system. The mass ratio is defined as $m^* = m/m_A$ based on the mass
130 (m) of the body, and its added mass. Damping ratio can also be defined as $\zeta = c / (2\sqrt{mk})$
131 , where C and K are the damping and spring stiffness of the system. Williamson [26], [27]
132 reported that the wake transition of a circular cylinder occurs when Re is larger than 180,
133 which was confirmed by Jiang et al. [28]. The two-dimensional (2D) simulations were
134 performed at $Re = 100$ for all cases to avoid the flow three dimensionality effects. As the main
135 goal of the current study is to find out the effect of different horizontal spacing on the
136 behavior of the system, the mass ratio and Re number are kept constant at 10 and 100,
137 respectively. The structural damping is also set to zero ($\zeta = 0$) to encourage a high amplitude

138 of oscillation. The natural frequency of the system is also set based on the Strouhal number
 139 and vortex shedding frequency of a bare fixed circular cylinder. Table 1 summarizes the main
 140 parameters for simulation where U_r is the reduced velocity and can be defined as $U_{in}/f_n D$.

Parameter	symbol	value
Mass ratio	m^*	10
Damping ratio	ζ	0
Horizontal space	$G=L/D$	0.5-0.4
Reynolds number	Re	100
Reduced velocity	U_r	6

Table 1 – Main Simulation Parameters

Governing equations

141 Based on the above assumptions, the flow over the objects is governed by the 2D
 142 incompressible unsteady Navier–Stokes (NS) equations. The conservation of mass and
 143 momentum, therefore, is given as follows:

$$144 \quad \frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} = 0, \quad (1)$$

$$145 \quad \frac{\partial u^*}{\partial t^*} + u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} = -\frac{\partial p^*}{\partial x^*} + \frac{1}{Re} \left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} \right), \quad (2)$$

$$146 \quad \frac{\partial v^*}{\partial t^*} + u^* \frac{\partial v^*}{\partial x^*} + v^* \frac{\partial v^*}{\partial y^*} = -\frac{\partial p^*}{\partial y^*} + \frac{1}{Re} \left(\frac{\partial^2 v^*}{\partial x^{*2}} + \frac{\partial^2 v^*}{\partial y^{*2}} \right), \quad (3)$$

147 In equations (1), (2), and (3), the dimensionless variables are evaluated as follows:

$$148 \quad x^* = x/D \quad y^* = y/D \quad u^* = u/U_{in} \quad v^* = v/U_{in} \quad p^* = p/\rho U_{in}^2 \quad t^* = t/U_{in}D$$

149 In the above equations, u and v stand for the velocity components in x - and y -directions,
 150 respectively, t is the flow time, ρ is the fluid density, U_{in} is the flow inlet velocity, D is cylinder
 151 diameter and p represents the static pressure of the fluid.

152 The mechanical response of the mass-spring-damper model is governed by the corresponding
153 equation of motion as:

$$154 \quad m\ddot{Y} + 2m\zeta\omega_0\dot{Y} + m\omega_0^2Y = f_l(t) \quad (4)$$

155 Where Y , \dot{Y} and \ddot{Y} denotes the transverse displacement, velocity, and acceleration of the
156 structure respectively. $\omega_0 = 2\pi f_n$ is the natural circular frequency of the system and $f_l(t)$ is
157 the time dependent lift force acting on the subject.

Numerical method

158 The flow field in the current study is numerically simulated with reliable CFD software ANSYS
159 Fluent. In the case of FIV simulations, as the subject oscillates, a dynamic mesh is required to
160 change the grid in every time step. ANSYS Fluent employs an Arbitrary Lagrangian-Eulerian
161 (ALE) algorithm which includes three dynamic mesh schemes, namely, smoothing, layering,
162 and remeshing. A diffusion-based smoothing method is used in the current study and a User-
163 Defined Function (UDF) is also employed using the C Programming Language to solve the
164 system's response in FIV.

165 In numerical simulations, the size of the computational domain and especially the blockage
166 effect are of great importance and small domains may result in errors in the prediction of the
167 forces and system behavior. On the other hand, a vibrating cylinder, Compared to a stationary
168 one, is associated with a wider wake and consequently, the blockage is expected to play an
169 even more significant role.

170 Prasanth and Mittal showed that for a circular cylinder in laminar flow ($m^* = 10$, $Re = 100$)
171 blockage of more than 2.5% can lead to hysteresis in the response of the cylinder at the onset
172 of synchronization. It also leads to an error in the prediction of aerodynamic forces [29], [30].

173 They also studied the VIV of two circular cylinders in tandem and staggered arrangements
 174 and located the lateral boundaries in 25D from the upstream cylinder which results in a
 175 blockage of 2% [31].

176 The size of the flow domain, the geometry of the model, and the boundary conditions of the
 177 current study are presented in Figure 2. The center of the cylinder is the origin of the
 178 coordinate system and the size of the whole computational domain is 75D×50D. The
 179 upstream and lateral boundaries are located 25D from the center of the cylinder and the
 180 downstream boundary is located 50D from the origin to capture the wake accurately.

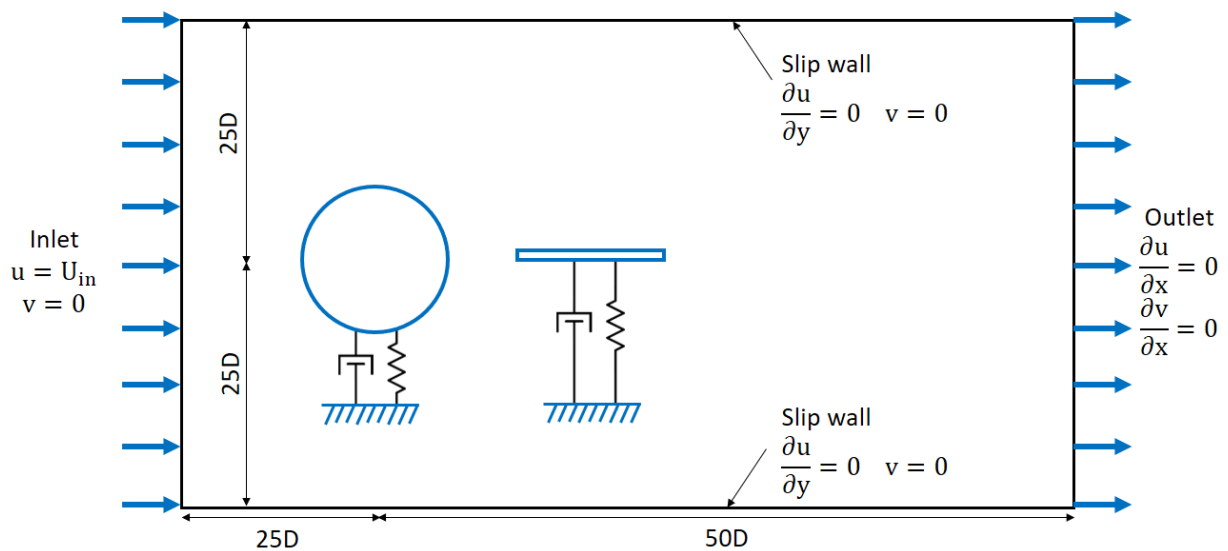


Figure 2 – The flow simulation domain, Geometry of the model and boundary Condition

181 The surface of the cylinder and the plate are assumed to be smooth, with a no-slip boundary
 182 condition ($u = 0, v = 0$). The inlet boundary is imposed with a uniform velocity boundary
 183 ($u = U_{in}, v = 0$). The top and bottom boundaries are prescribed with the normal component
 184 of the velocity and component of the stress vector along the boundary equal to zero ($\partial u /$
 185 $\partial y = 0, v = 0$). Finally, the zero normal gradient condition for velocity is assigned in the
 186 outlet and the pressure is specified as a reference value of zero ($\partial u / \partial x = 0, \partial v / \partial x = 0$
 187 $, p = 0$).

Computational domain and mesh dependency

188 In the first step, before conducting a detailed study on the main case, mesh independence
189 and temporal resolution validation are performed to achieve an independent solution
190 method.

191 Figure 3 shows the grid generated for the simulation of flow filed around a circular cylinder
192 and a downstream flat plate. In all of the cases the mesh around the cylinder's surface has
193 been divided into four sections: front, up and down and finally the rear side which the last
194 one is more important due to the presence of the plate and high flow gradients. The mesh is
195 also gradually coarsened to decrease the computational cost in regions far away from the
196 structures.

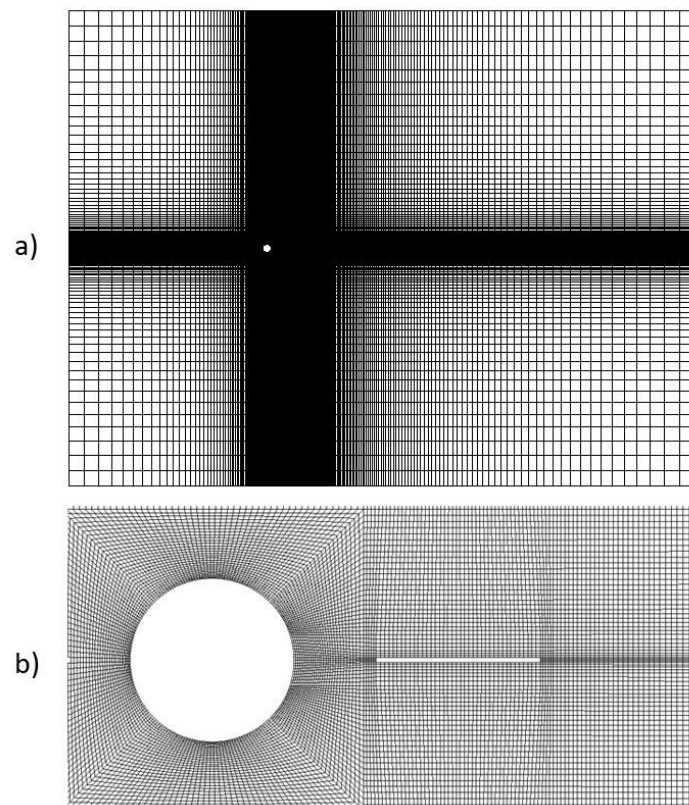


Figure 3 – Grid distribution: a) around the cylinder and plate, b) the whole grid system

197 The grid dependency study has been conducted on two configurations including the 2DOF VIV
198 of a bare circular cylinder and a fixed cylinder with a flat plate in the wake (G=3D). The grid's

199 structures are generally similar, but for the latter case, due to the presence of the plate, there
 200 are more cells in the wake.

201 For the first case, it can be seen that the variations of the normalized mean in-line and the
 202 maximum cross-flow amplitudes, the Strouhal number, and the mean drag coefficient
 203 converge in case G-1-2 with 200 nodes equally distributed on the cylinder surface (Table 2).

Grid	Nodes on the cylinder	X_{mean}	Y_{max}	St	CD_{mean}
G-1-1	4×40	0.116	0.656	0.1634	1.986
G-1-2	4×50	0.115	0.647	0.1635	2
G-1-3	4×60	0.1151	0.645	0.1635	1.997

Table 2 - Grid independency study for 2D VIV of circular cylinder ($Re=100$, $Ur=6.02$)

204 Table 3 presents the results of the second case. A grid with 40% more cells in rear side of the
 205 cylinder (G-2-3), assures that the results are independent from grid resolution.

Grid	Nodes on cylinder	Cylinder				Flat Plate			
		Cl_{max}	Cl_{rms}	St	CD_{mean}	Cl_{max}	Cl_{rms}	St	CD_{mean}
G-2-1	3×50+50	0.41	0.288	0.157	1.311	1.245	0.87	0.157	0.0782
G-2-2	3×50+60	0.405	0.288	0.157	1.310	1.254	0.875	0.157	0.0795
G-2-3	3×50+70	0.405	0.286	0.157	1.310	1.257	0.883	0.157	0.0799
G-2-4	3×50+80	0.405	0.285	0.157	1.309	1.258	0.884	0.157	0.0798

Table 3 - The grid independence test for a stationary cylinder and a flat plate 3D downstream at $Re=100$

206 According to the results, the grid G-2-3 is selected for further simulations on a fixed or
 207 oscillating combination of a cylinder and a flat plate. In the second test, for the temporal
 208 resolution analysis, the mesh G-2-3 is examined for several time steps and a non-dimensional
 209 time step of $\Delta t = 0.002$, is found to be sufficiently small to assure independency.

Model validation

210 This section aims to provide a validated computational approach to simulate the flow field
211 around stationary and oscillating subjects. Three cases including a stationary/oscillating
212 cylinder and a fixed cylinder with a flat plate in the wake are simulated and the results are
213 compared with previous studies. All the cases have a similar simulation domain which is
214 described before. In Table 4 the aerodynamic characteristics of the flow around the stationary
215 cylinder at $Re=100$ are compared with previous studies. The good agreement shows that the
216 current simulation approach is capable of predicting the case with acceptable accuracy.

Author	CD_{mean}	St	CL_{rms}	$-Cp_b$
Park et al. (1998) [32]	1.33	0.165	0.235	0.725
Kravchenko and Moin [33]	1.32	0.164	0.222	0.73
Shi et al. [34]	1.318	0.164	-	-
Mittal [35]	1.322	0.1644	0.226	-
Stalberg et al. [36]	1.32	0.166	0.233	-
Posdziech and Grundmann [37]	1.325	0.1644	0.228	0.709
Li et al. [38]	1.336	0.164	-	0.701
Qu et al. [39]	1.319	0.1648	0.225	0.709
Present study	1.34	0.165	0.232	0.71

Table 4 - Comparison of flow quantities with published results for flow around a stationary cylinder at $Re= 100$

217 For the second case, the variation of Strouhal number and Drag coefficient of the circular
218 cylinder and a wake mounted splitter plate is presented in Figure 4 and Figure 5.

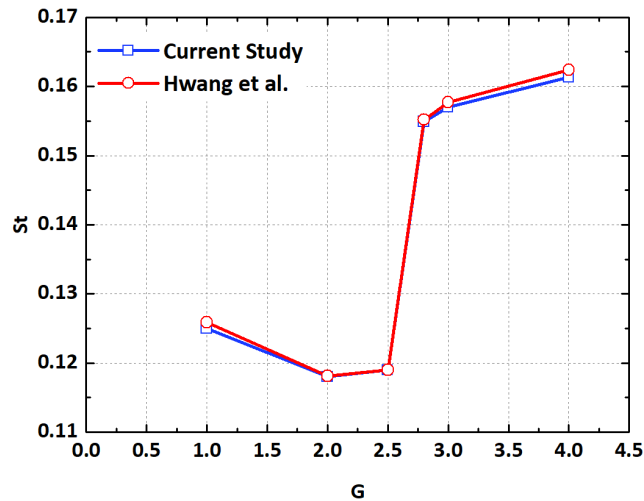


Figure 4 - Variation of Strouhal number for different spacing in Re=100 [12]

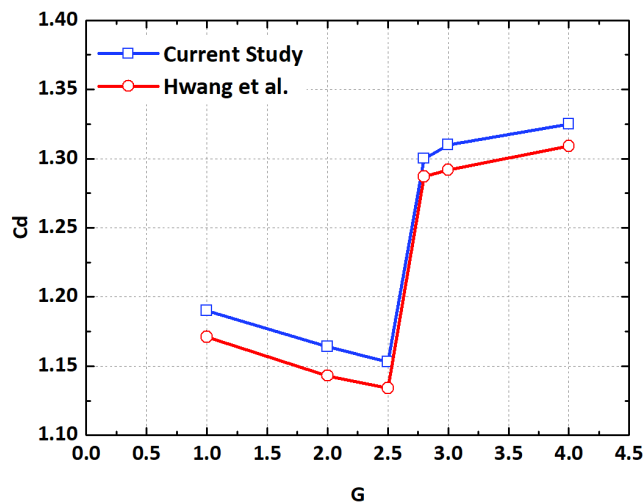


Figure 5 - Variation of drag coefficient for different spacing in Re=100 [12]

219 The results of the current simulation are in agreement with those of Hwang et al [12]. The
 220 plate changes the wake structure for $L/D \leq 2.5$ and decreases the drag coefficient and Strouhal
 221 number significantly. For $L/D \geq 2.8$, a regular vortex shedding appears and both of the
 222 parameters jump to higher values. To examine the accuracy of the numerical method to
 223 simulate the 2DOF VIV, the results of the current simulation are compared with some earlier
 224 studies in Table 5.

Author	Y_{max}	X_{mean}	St	CL_{max}	CD_{mean}	CD_{rms}
Prasanth and Mittal (2008) [30]	0.53	0.1115	0.1643	0.1929	1.90	0.2486
He et al. (2012) [40]	0.503	0.1082	0.1652	0.1985	1.81	0.2244
Tu et al.(2015) [41]	0.525	0.1307	0.1652	0.2220	1.88	0.2664
Present study	0.54	0.110	0.1640	0.235	1.88	0.2660

Table 5 - Comparison of the results of the current study with previous studies for 2D VIV of a circular cylinder at $Re=100$ and $Ur=6$

225 The good agreement confirms that the reliability of the numerical approach. Figure 6,
 226 presents the variation of maximum amplitude of the cylinder for a range of Re numbers. It
 227 can be seen that the results agree well with those of Prasanth and Mittal [30] (Decreasing
 228 case) and Tu Et al. [41].

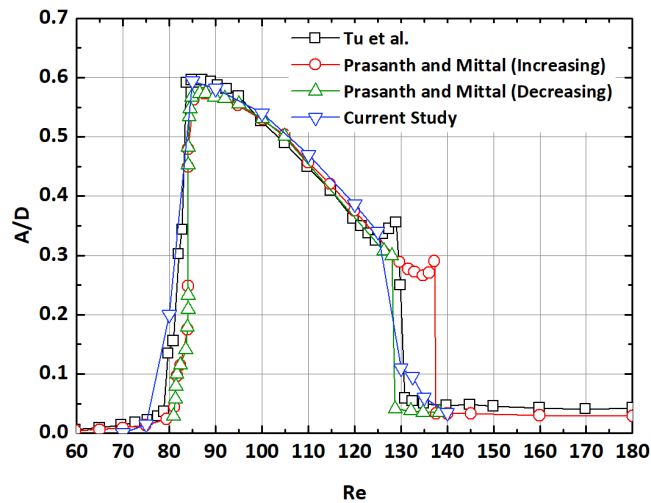


Figure 6 - Variation of statistical values of the cylinder responses with Re (Transverse oscillation amplitude)

229 Oscillation amplitude follows the trend; the maximum amplitude has been predicted well and
 230 there is only little differences in the beginning and end of synchronization regime.

231 Considering all the differences mentioned above, it can be concluded that the present
 232 numerical approach is able to predict the FIV of a cylinder and a downstream flat plate with
 233 adequate accuracy.

Results

234 The results of the current study are divided into two parts. In the first one, the flow field and
235 the response of a flat plate with one DOF in cross flow direction that is located in the wake of
236 a stationary circular cylinder are presented. The second part deals with simultaneous free
237 vibration of the circular cylinder and the plate and the interaction of two objects in small
238 horizontal spacing.

Stationary cylinder, Vibrating flat plate

239 According to the previous studies the location of the flat plate has a significant effect on the
240 vortex formation and the wake structure. Therefore 10 different configurations in a spacing
241 range of $0.5D < G < 4D$ have been numerically simulated. The cylinder diameter and the flat
242 plate's length are equal and all the simulations are conducted at a Re number of 100 which is
243 equivalent to a reduced velocity of 6.

244 A fixed flat plate located in the wake of a stationary cylinder can perform as a splitter and
245 delays or even suppress the vortex shedding depending on its location [12]. This behavior is
246 a result of changing the recirculation zone (R-Z) on the rear side of the cylinder. Based on the
247 current simulations, the system response for a flat plate with a 1DOF in transverse direction
248 is different. The vibration amplitude and frequency ratio of the flat plate in different
249 configurations are presented in Figure 7:

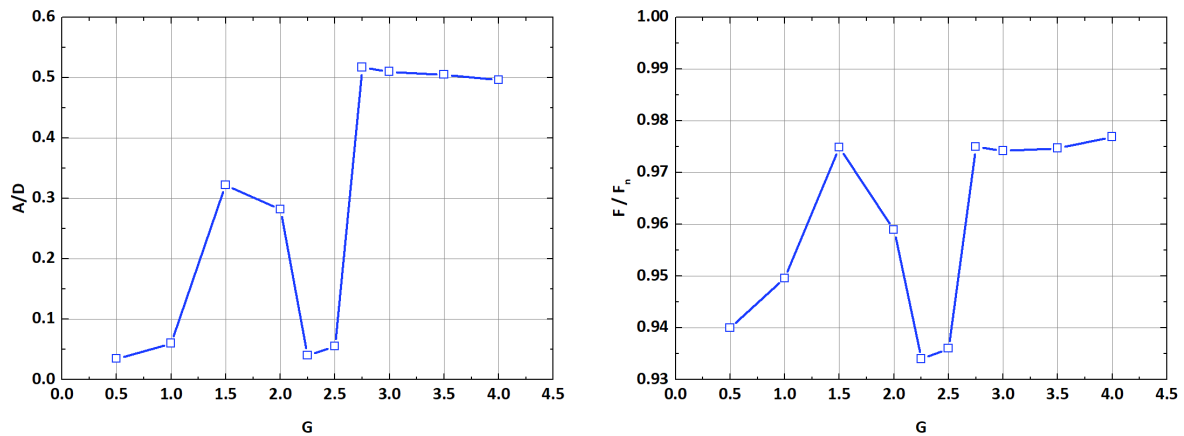


Figure 7 – Oscillation amplitude (Left) and frequency ratio (Right) of the flat plate located in the wake of a stationary cylinder

250 Flat plates which are located outside of a critical distance from the rear end of the cylinder,
 251 about 2.6D and introduced by Hwang [12], show an anticipated behavior and oscillates
 252 sinusoidally. It oscillates beyond this critical distance, because of adequate space for vortices
 253 to form and shed into the wake. Therefore the plate faces with a local asymmetric pressure
 254 field and starts to vibrate. In this condition, as it approaches the cylinder, due to facing
 255 stronger vortices, the oscillation amplitude slightly increases up to $A/D=0.517$ for $G=2.75D$.

256 The system response is just similar to that of a fixed plate for $G=2.25D$ and $G=2.5D$ and the
 257 vibration almost vanishes. While the frequency ratio follows the natural frequency of the fixed
 258 cylinder for $G \geq 2.75$, it gets far from it in this region.

259 Smaller spacing interestingly results in higher vibration amplitude, although it is about 50%
 260 less than those of $G=2.75-4D$. When the spacing sets as $G=0.5-1D$ the flat plate rolls as a
 261 splitter and both of the amplitude and frequency fall again.

262 Generally, the vibration amplitude in the crossflow direction and the variation of lift force are
 263 relevant. In Figure 8 the variation of $C_{l_{rms}}$ of the flat plate for different configurations are
 264 presented and compared with those of a fixed one.

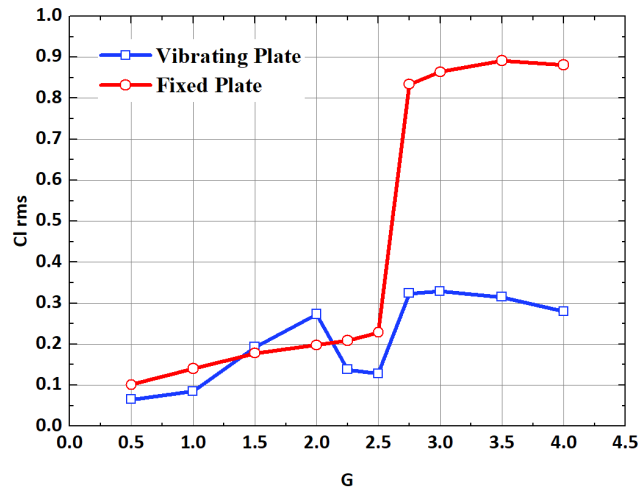


Figure 8 – Variation of CL_{rms} of the flat plat in different configurations

265 For large horizontal spacing ($G \geq 2.75$), the oscillation of the plate and change in its position
 266 results in considerably lower CL_{rms} than that of a fixed one. The lift force falls in $G=2.5D$ for
 267 both of the fixed and free plates. As expected, this coincides with the vanishing of the
 268 vibration. While the force decreases gradually for the fixed plate, there is a temporary growth
 269 for the cases with $G=1.5, 2D$ which once again has similar behavior to the vibration amplitude.
 270 The CL_{rms} falls again for smaller spacings.

271 This system response is related to the near wake flow field and as mentioned before, the R-Z
 272 has a significant effect on its structure and vortex shedding pattern. Figure 9 presents the
 273 variation of this region in different configurations:

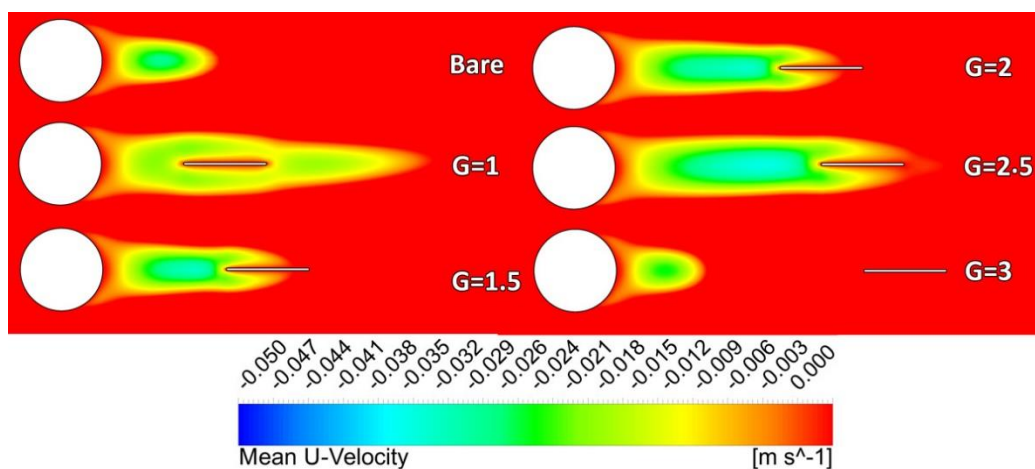


Figure 9 –Recirculation zone of a bare cylinder and the effect of free to oscillate flat plate in different configurations

274 The presence of the flat plate with 1DOF changes the R-Z in a way which is different to that
 275 of a fixed one presented by Hwang [12]. For $G=1D$, the recirculation length grows
 276 significantly, and the plate is entirely confined to this region. When the plate is placed slightly
 277 further ($G=1.5, 2D$), a shorter R-Z appears and a small part of the plate is outside of it. In the
 278 case of $G=2.5D$, a larger R-Z appears that is stretched up to the trailing edge of the flat plate.
 279 Larger horizontal spacing ($G \geq 3D$) results in a short R-Z which is similar to that of a bare one.
 280 Comparing the R-Z, the plate location and vibration amplitude shows that the plate starts to
 281 vibrate with a considerable amplitude only in cases that the plate is not entirely surrounded
 282 by this region.

283 In the case of $G=3D$, the plate which is located completely outside of this region faces
 284 shedding vortices and consequently has a considerable vibration amplitude. The vibration
 285 mechanism for $G=1.5, 2D$, in which the R-Z is larger and no regular vortex shedding has been
 286 formed before the plate, is different. To achieve a better understanding of this mechanism, a
 287 closer look into the wake structure for different configurations might be beneficial. Figure 10
 288 shows the streamlines in near wake of the cylinder for an identical moment ($C_{l_{cylinder}}=0$). A
 289 regular vortex shedding pattern of a bare cylinder is also presented for a better comparison.

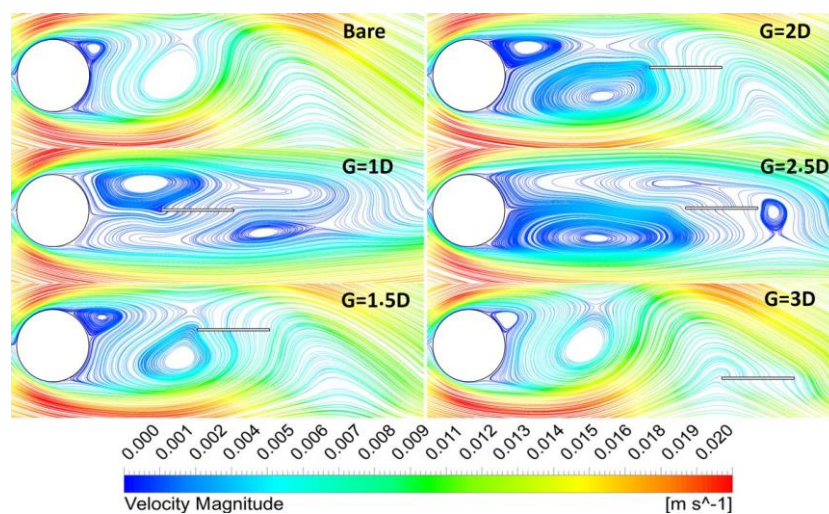


Figure 10 – Comparison of the wake structure in different configurations

290 When the plate is placed in $G=1D$, unlike the bare cylinder, Large and stretched vortices are
 291 generated that surround both the upper and lower sides of the plate. This flow structure
 292 confirms the large R-Z and may explain the reason for low vibration amplitude due to low
 293 pressure difference. For $G=1.5, 2D$, the wake structure is somehow similar to the case of a
 294 bare cylinder and the vortices form and move downstream periodically. As the plate goes
 295 further ($G=2.5D$), once again large and stretched vortices appear that are even larger than
 296 those of $G=1D$. These large structures are matched with the growth of R-Z. Placing the plate
 297 in further locations ($G \geq 3D$) results in a near wake similar to a bare cylinder. A regular vortex
 298 shedding pattern appears and the vibration is based on the interaction of the plate and the
 299 vortices which are completely formed and move downstream periodically. Instantaneous
 300 streamline contours at equal intervals are presented in Figure 11 during one cycle of plate
 301 vibration for three configurations.

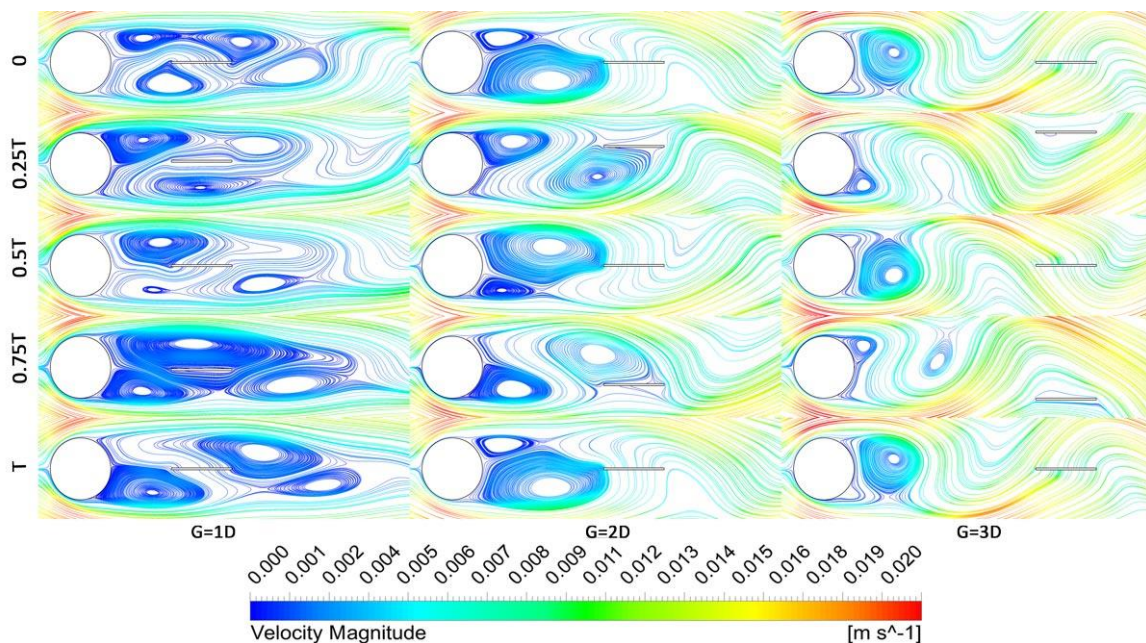


Figure 11 – Wake structure during one cycle of plate oscillation for cases $G=1D$, $G=2D$, $G=3D$

302 When the plate is located at $G=1D$, the lack of interaction between two shear layers, changes
 303 the R-Z structure and formation of each vortex is based on the same side shear layer and
 304 results in large and stretched vortices. As the plate goes further ($G=2D$), a familiar wake

305 structure can be observed with a periodic vortex formation. In each side, the vortex grows up
 306 to the plate's leading edge, separates from the cylinder by the shear layer of the opposite side
 307 and finally moves downstream which creates an asymmetric flow field around the plate . This
 308 mechanism is almost similar to that of a bare cylinder or a cylinder and a plate which is located
 309 far enough ($G=3D$). The main difference between the cases $G=2, 3D$ is in the larger size of the
 310 vortices for the first one.

311 Figure 12 presents the velocity magnitude and the spanwise vorticity contours of the flow
 312 field for cases $G=1, 2, 2.5, 3D$ in an identical moment ($Y_{plate}=0$).

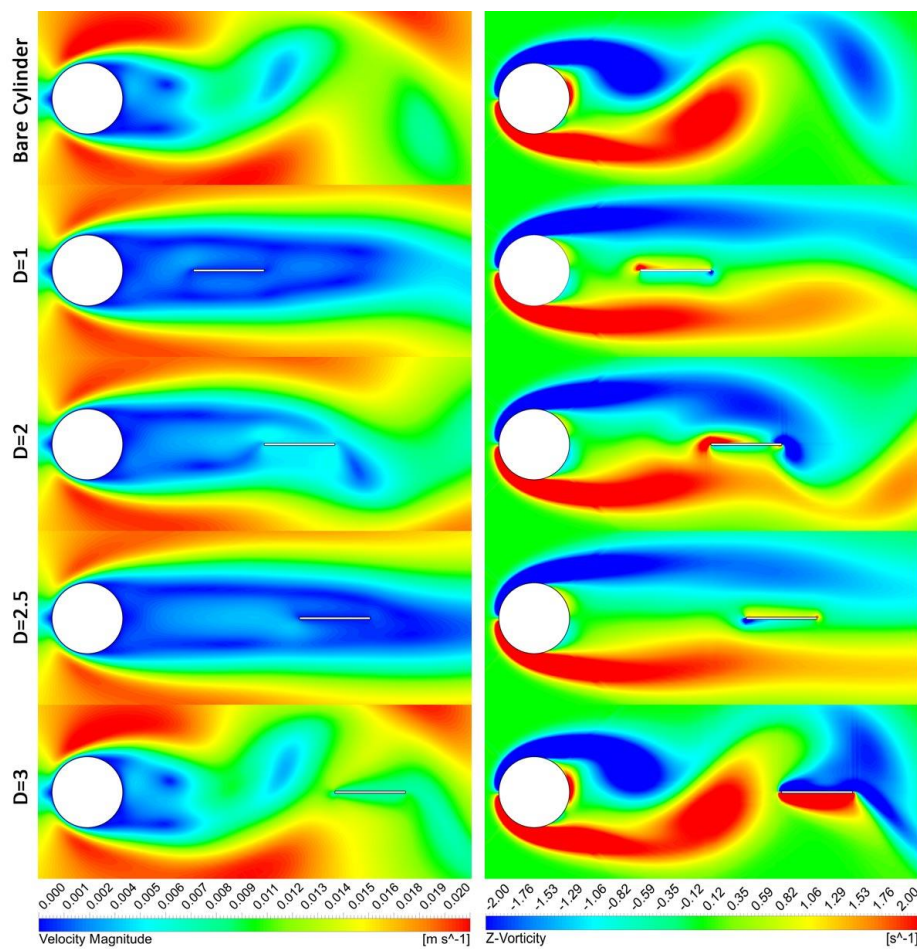


Figure 12 – Velocity magnitude and spanwise vorticity contours in different configurations ($Y_{plate}=0$)

313 For the case of $G=1D$, the near wake includes two separated shear layers and the plate is
 314 entirely covered by the R-Z with a low velocity magnitude. In this case, the cylinder and the

315 plate can be considered as a whole body. A periodic vortex formation occurs just before the
316 plate with a larger gap in case $G=2D$. This mechanism results in an asymmetric flow field
317 around the plate surface and consequently, it starts to vibrate. Due to the presence of the
318 plate, the vortex shedding occurs further downstream when they just have passed the trailing
319 edge of the plate. In the case of $G=2.5D$, the objects just behave as a whole body, and a similar
320 pattern to the case $G=1D$ can be seen. This Larger gap also delays the start of vortex shedding
321 even further downstream. In the last case, there is enough space for a fully formation and
322 shedding of the vortices. The flow pattern and vibration mechanism are similar to that of a
323 bare cylinder.

324 **Simultaneous FIV of the cylinder and the flat plate**

325 The results of simultaneous FIV of a cylinder and a flat plate in tandem arrangement are
326 presented in this part. All of the configurations and simulation parameters are similar to the
327 last part. It is necessary to mention that the term (bare oscillation/vibration) deals with the
328 FIV of a single and bare object in the flow field and the term (individual oscillation/vibration)
329 indicates the FIV of one object while the other object is stationary which indicates the results
330 of last section.

331 The oscillation amplitude and frequency ratio of the cylinder are presented and compared to
332 those of a bare cylinder in Figure 13:

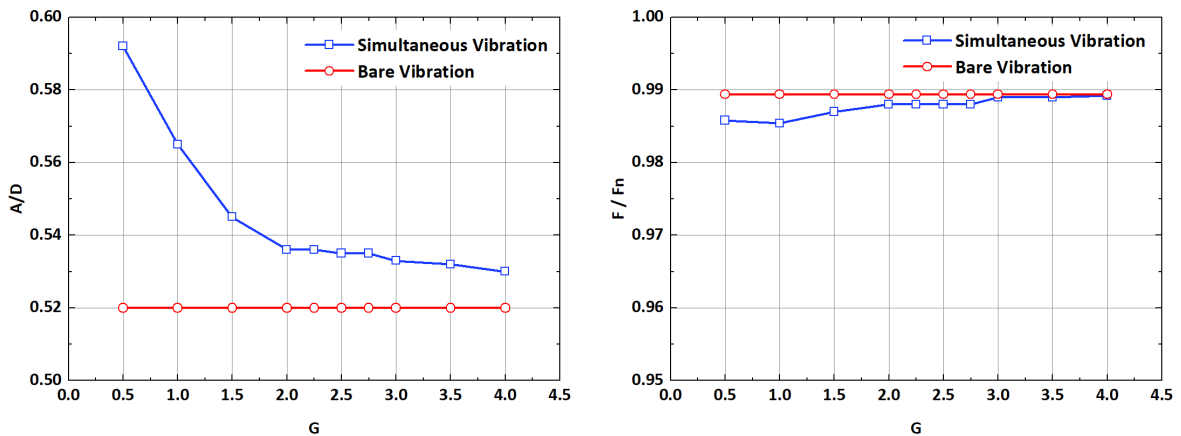


Figure 13 – Oscillation amplitude (left) and frequency ratio (right) of the bare cylinder and the cylinder in simultaneous vibration

333 The oscillation amplitude is slightly higher in simultaneous vibration than that of the bare
334 cylinder for large gaps. As the spacing decreases, the amplitude grows gradually and reaches
335 up to $A/D=0.592$ for the case of $G=0.5D$ which is about 14% higher than the bare cylinder. The
336 frequency ratio is almost constant and follows system's natural frequency for all cases.

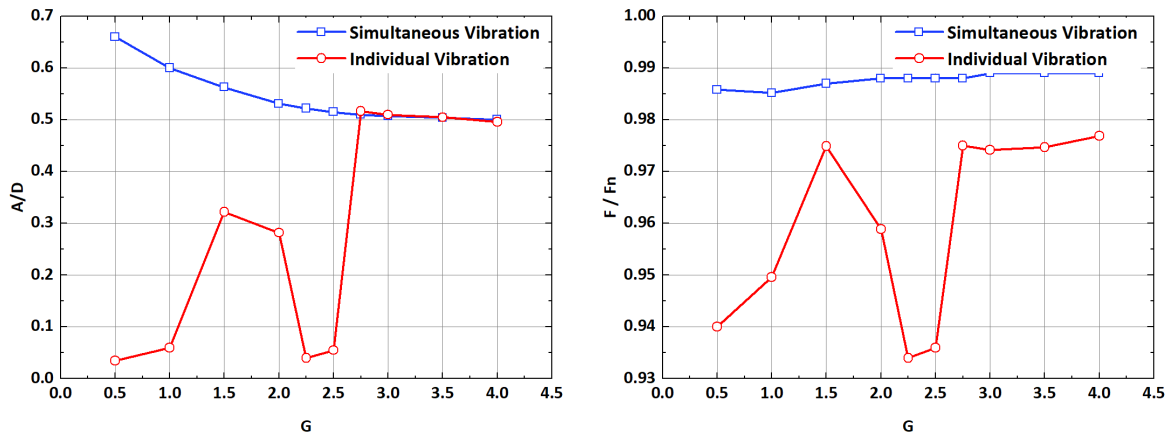


Figure 14 –Oscillation amplitude (left) and frequency ratio (right) of the plate in individual and simultaneous oscillation

337 Although the flat plate only vibrates for a specific range of G in the last part, it keeps vibrating
 338 for every configuration in simultaneous vibration, and its amplitude also increases gradually
 339 with the reduction of G . As shown in Figure 14, the maximum amplitude occurs for $G=0.5D$
 340 and reaches up to $A/D=0.66$ while it almost disappeared in the similar case of the last part.

341 The frequency ratio behaves similar to the cylinder and follows the natural frequency.

342 The higher amplitude of both of the objects in simultaneous oscillation, especially in small
 343 horizontal spacing, shows that the present concept has a considerable potential to be utilized
 344 in the design of an improved energy harvesting system by converting fluid dynamic energy
 345 into electric energy from independent objects in a small area.

346 The phase difference between the motions of the two objects varies linearly and is a function
 347 of horizontal spacing. The maximum difference occurs for $G=1D$ and in the case of $G=3D$ two
 348 objects vibrate almost with the same phase (Figure 15).

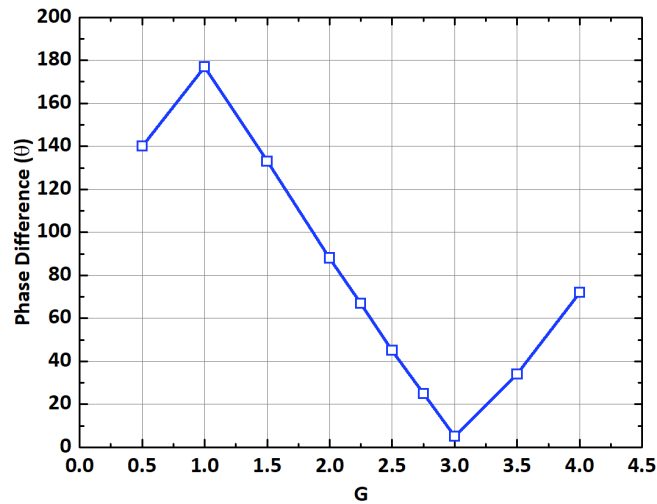


Figure 15 – The phase difference in oscillation of the circular cylinder and the flat plate in simultaneous oscillation

349 The CL_{rms} for both of the cylinder and the plate growth gradually as the plate gets closer to
 350 the cylinder (Figure 16).

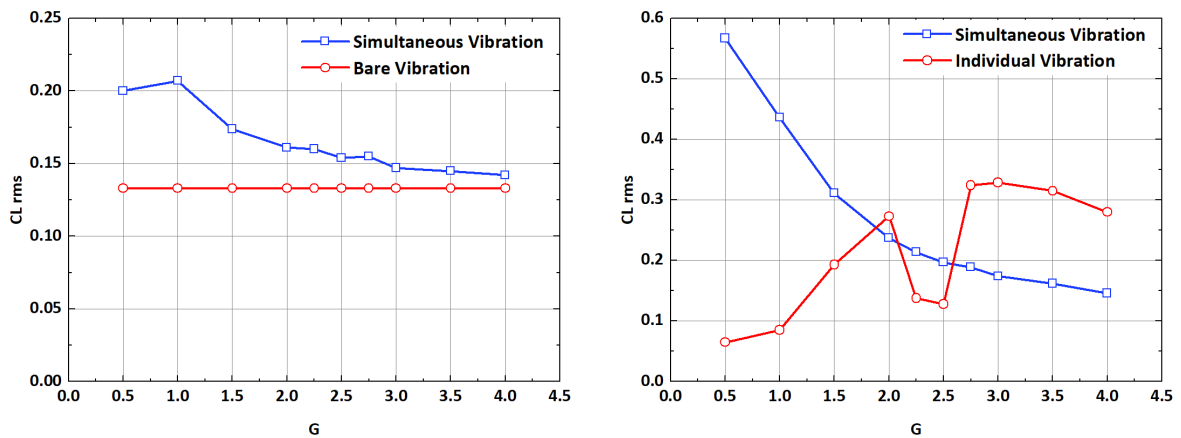


Figure 16 – CL_{RMS} of the circular cylinder (Left) and the flat plate (right) in simultaneous oscillation and bare or individual vibration

351 The higher value of CL_{rms} in small spacing is a result of change in lift force during one cycle of
 352 oscillation. As shown in Figure 17, not only the maximum lift coefficient rises about 20% for
 353 $G=0.5D$, but a secondary peak also appears in CL diagram just after the first one as the cylinder
 354 moves up or down (about 40%, 90% of a vibration cycle).

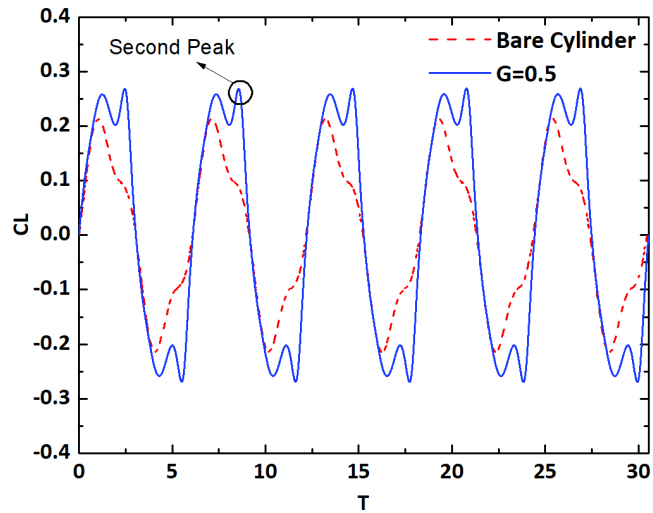


Figure 17 - Variation of Lift coefficient for the bare cylinder and in case $G=0.5D$

355 The Pressure Coefficient distribution also confirms the higher lift coefficient. The pressure
 356 difference on both of the front and rear side of the cylinder are slightly higher than that of a
 357 bare oscillating one (Figure 18).

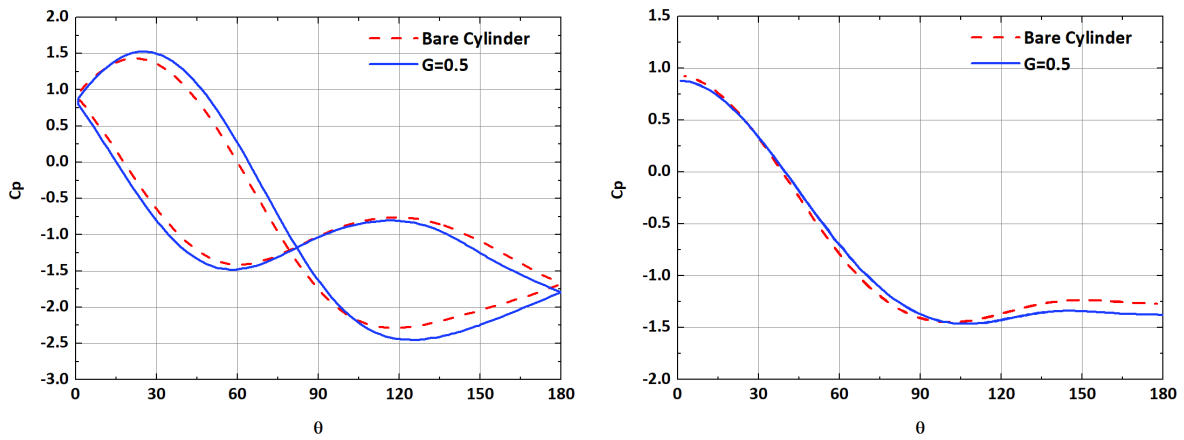


Figure 18 – Left) pressure coefficient on the circular cylinder for the bare one and in case $G=0.5$ at $t=0.4T$. Right) Time averaged pressure coefficient of the circular cylinder for the bare one and in case $G=0.5D$

358 Due to the change in shear layers and vortex formation, the time averaged pressure
 359 coefficient also shows a slightly higher pressure difference specially on the rear side of the
 360 cylinder in case $G=0.5D$. This change results in higher lift forces which is related to higher
 361 oscillation amplitude of this case (Figure 18).

362 To achieve a better understanding of how the flow field and the response of the system
363 change, the vorticity contours for a bare cylinder and cases $G=0.5$, $3D$ are presented In Figure
364 19 for an identical moment ($Y_{cylinder}=0$).

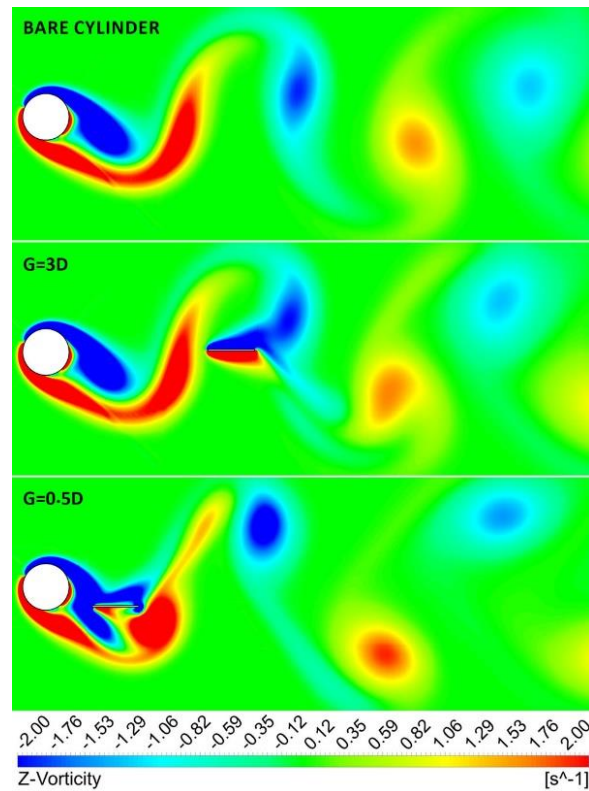


Figure 19 – Spanwise vorticity contours of the circular cylinder and the flat plate in simultaneous oscillation

365 Comparing the results shows a similar pattern for the case of $G=3D$ and the bare one. The
366 plate is far enough from the cylinder to form and shed the vortices, and has no significant
367 effect on the upstream flow structure. In cases with low spacing (Ex. $G=0.5D$), the interaction
368 of the flat plate and shear layers changes the vortex formation in such a way that increases
369 the vortices strength and not only amplifies the cylinder oscillation amplitude but also lets
370 the plate to oscillate with a considerable amplitude within such a short gap. Figure 20
371 presents the variation of the spanwise vorticity during one cycle of cylinder vibration at equal
372 intervals.

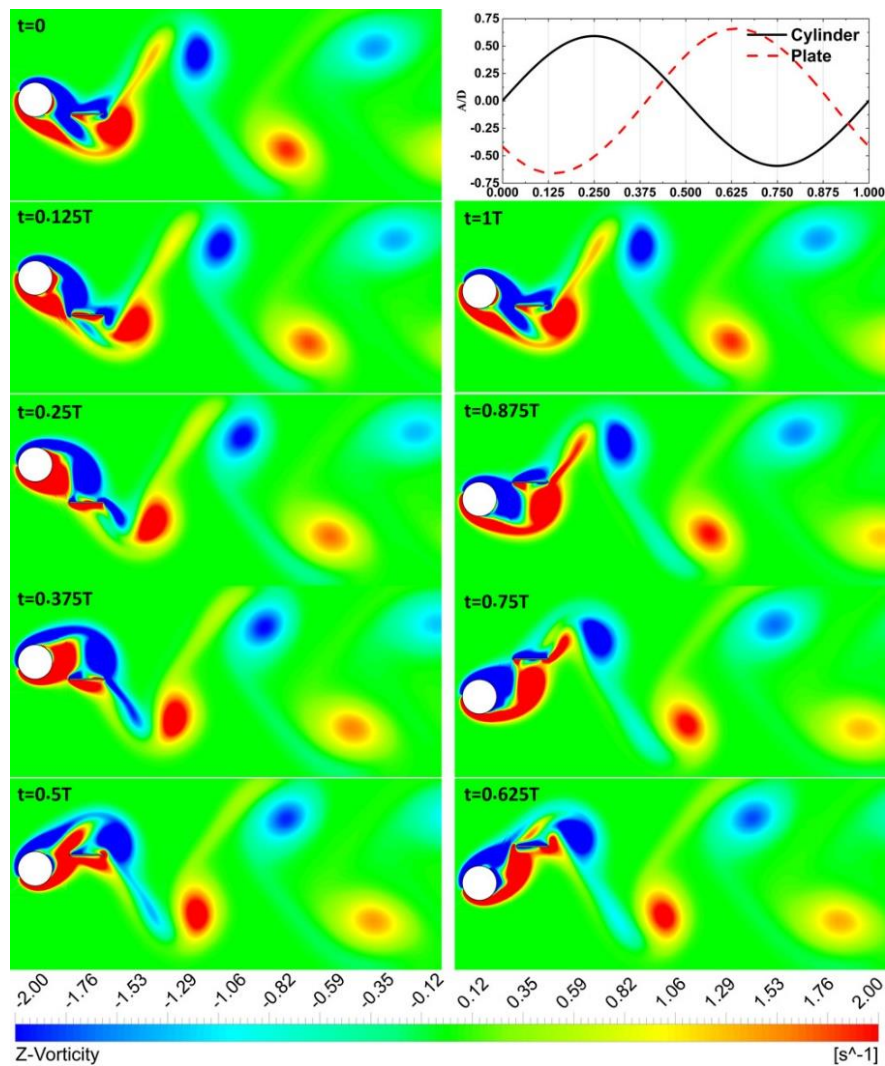


Figure 20 – Spanwise vorticity contours during one cycle of simultaneous oscillation for $G=0.5D$

373 The wake in this configuration is a bit different to that of a bare one. It is a bit wider, Vortices
 374 separate slightly sooner with a farther apart cores. They also have a long arm that is stretched
 375 toward the opposite side. The interaction between the plate, the shear layer, and forming
 376 vortex in each side of the plate, results in formation of a leading edge vortex (LEV) on the
 377 same side. The LEV extends and covers the entire plate surface and finally moves toward the
 378 trailing edge during one cycle of plate vibration. At this moment the second shedding vortex
 379 of the same side passes the plate, the TEV separates and joins the large vortex as its long arm.

Conclusion

380 Numerical simulations were conducted to investigate the effect of a free to vibrate flat plat
381 on the wake structure and vortex induced vibration of an upstream circular cylinder in
382 different horizontal spacing at a Re number of 100. The following major conclusions are
383 drawn:

384 1. When the flat plate with one DOF in crossflow direction is placed beyond $G=2.5D$, just like
385 a fixed one, has no considerable effect on the vortex formation and shedding of the cylinder.
386 Therefore a regular wake forms and the plate starts to vibrate due to facing with an
387 asymmetric flow domain.

388 2. Unlike the fixed one, placing the plate closer to the cylinder ($G<2.6D$), will not necessarily
389 be associated with a delay in vortex shedding or suppression. According to the results, for a
390 specific range of horizontal spacing ($G=1.5, 2D$), the plate starts to vibrate once again.
391 Although its amplitude is less than the previous cases.

392 3. The cause of this phenomenon is attributed to the size of recirculation zone and the plate
393 location. The flat plate starts to vibrate when it is completely, or at least a small part of it,
394 located outside of the R-Z.

395 4. The plate starts to oscillate as facing with the upstream shedding vortices for cases with
396 $G\geq 2.75D$. In those cases, as the shedding vortices move toward the plate, the asymmetric
397 flow domain changes the flow field around the plate and induces a free vibration. The
398 vibration mechanism is found different in cases of $G=1.5, 2D$. In these cases, although the
399 most part of the plate is inside the R-Z, the vortices inside this region are smaller than those
400 of $G=2.5D$ and periodically move downstream and pass the plate. This flow structure results
401 in a periodic pressure change, so the plate starts to vibrate.

402 5. In free simultaneous oscillation, the plate no longer plays the role of a splitter at any
403 horizontal spacing. In large spacing, the flow structure around the cylinder is similar to that
404 of a bare one and the cylinder and the plate start to vibrate due to VIV and WIV respectively.

405 6. As the plate gets closer, the oscillation amplitude for both of the objects increases
406 gradually. This increment occurs due to the interaction of the plate and the shear layers which
407 results in stronger vortices.

408 7. The higher amplitude in simultaneous oscillation, especially in small horizontal spacing,
409 shows that the present concept has a considerable potential to be utilized in the design of an
410 improved energy harvesting system from independent objects in a small area.

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