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ABSTRACT

As one of the fundamental problems in fluid mechanics, the flow-induced vibration of a flexible cylinder helps shed light on various complex fluid–structure interaction phenomena, such as the coupling effect of the cross-flow and in-line motions in the relationship between external fluid forces and vortical wake patterns. This paper devised a non-uniform in-flow condition (partially uniform flow and partially still water) for flexible cylinders in experimental and numerical conditions. Consistently, a new phenomenon is observed in different scale experiments and simulations where secondary traveling wave vibration of the flexible cylinder is excited from the still water part to the uniform inflow part due to the positive external fluid energy input in the still water. Furthermore, the detailed flow visualization on the vortical wake patterns around the vibrating flexible cylinders reveals that the external fluid force sources in the still water are due to the existence of the attached vortex pair with an uneven strength, which has been observed before in the still rigid cylinder in the oscillatory flow or the rigid cylinder oscillating in the still water at Keulegan–Carpenter number from 4 to 7.

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I. INTRODUCTION

Due to the unstable shedding vortices, long slender structures in ocean current, such as risers, cables, and hawsers, are subject to vibrations. It is referred to as vortex-induced vibrations (VIVs) that may cause the structures unpredictable fatigue damage. Therefore, it has attracted a significant amount of research attention over the past decades.1–6

Flexible cylinders in the non-uniform current exhibit a more complicated spatial distribution of structural vibrations and vortical wake patterns compared to rigid cylinder VIVs.7 For example, the flexible cylinder response in the highly sheared flow is more likely to show multi-mode, non-“lock-in” behavior than that in mildly sheared conditions.8–10 Standing waves dominate the VIV response of a rigid cylinder in uniform current, while significant traveling wave responses11–13 could be observed on the long flexible cylinder in large scale field experiments.14

Over the past decades, various numerical methods have been proposed to simulate this nonlinear VIVs problem.1 Among them, the high-fidelity CFD method shows a great advantage in revealing the vortical wake topology,15 even though the computational cost is too high to be applied to industrial applications. To this end, empirical prediction tools still dominate the offshore structure’s VIV design.16–18

In oceans, especially of the water column over 1000 m deep, the uniform current is not realistic and seldom met.19 Typically, the higher
velocity current is found close to the surface, and it decreases to a lower or even a zero value in the significant part of the water column.\(^{20}\) Therefore, the non-uniform current could result in a VIV scenario that the top part of the flexible cylinder vibrates in the current of higher velocity, while the other part of the cylinder vibrates in the current of lower speed or even in the still water.

The vibrating cylinder in still water or cylinder in oscillatory flow has attracted far less attention. Notably, the experiment on rigid cylinder in oscillatory flow performed by Sarpkaya\(^{123}\) showed that the drag (\(C_d\)) and added mass coefficient (\(C_m\)) are in tune with the theory: for smooth cylinders, under the critical \(KC = 2\pi f d \) number, the flow transitioned into unstable state depending on different \(\beta = \frac{d}{L}\), where \(A\) and \(f\) is vibration amplitude and frequency, \(d\) is the cylinder diameter, and \(\nu\) is the fluid kinematic viscosity. In addition, Williamson\(^{122}\) studied lift forces (perpendicular to the oscillation direction) on oscillating a circular cylinder in still fluid, which revealed the fact that the appearance of the different harmonics of the oscillation frequency of the lift force depended on \(KC\) too. With increasing \(KC\), the flow pattern changed from a pair of attached vortices around the cylinder to multiple-pair shedding vortices. A more comprehensive flow visualization study was carried out by Tatsuno and Bearman\(^{23}\) for a broad range of \(KC\) from 1.6 to 15 and various \(\beta\) between 5 and 160, and a comprehensive flow regimes map was established, where eight different flow regimes were identified (A*, A, B, C, D, E, F, and G). However, minimal work has been done on the flexible cylinder oscillating in the still water. It was reported by Fu and Wang\(^{24,25}\) that the response of flexible cylinder in oscillatory flow at large \(KC\) features an intermittent VIV response with amplitude modulation including three phases: build-up, lock-in, and dry-out.

In the current work, we will first carry out experimental measurements on the response of a flexible cylinder in stepped current, in other words, partially in the uniform flow and partially in the still water, which is achieved by hiding part of the flexible cylinder in particular, designed bucket to block incoming velocity when towing the model. Subsequently, we will perform the high-fidelity simulation to visualize the surrounding vortical wake pattern and get a deep insight into the corresponding VIV.

The rest of the paper is organized as follows: Sec. II presents the methods and models used in the experiment and simulation. Section III discusses the experimental and numerical results of the different flexible cylinders in stepped current. Section IV summarizes the main findings of the paper. In the Appendix, additional experimental results are given.

II. MATERIAL AND METHODS

A. Experimental method and models

The uniform flexible cylinder model used in current experiment has a diameter \(d = 1.2\) cm and a length \(L = 204\) cm (aspect ratio is 170). The model is constructed/molded via urethane rubber (density of 1.38 g/cm\(^3\)) with a mass ratio of \(m'/m = 4.0\) (the ratio between structural mass to displaced fluid mass). In addition, a fishing line is embedded in the center during the molding process to provide sufficient axial stiffness. In the experiment, the effect of the bending stiffness on the model’s natural frequency and modal shape is negligible compared to that of the tensions applied. Note that the structural damping of the urethane rubber material is higher than the traditional metal or ABS plastic cylinder model. Nonetheless, the damping ratio of the current flexible model is 8.2%, as obtained from the pluck test in the air. The model properties are shown in Table I labeled as Exp I.

Two foil-shaped holders are fixed on the supporting beam in the towing tank, with the model installed on the bottom of the holder that allows tension adjustment. A similar setup has been used in our previous research.\(^{27}\) To achieve the condition that only part of the flexible cylinder is exposed to the uniform inflow [length of \(l_1\) shown in Fig. 1(a)] and part of the flexible cylinder [length of \(l_2\) shown in Fig. 1(a)] responds in the still water, a specially designed bucket is constructed to block the incoming flow, shown in Fig. 1(b). The top and back of the bucket use the transparent acrylic plate that allows the model’s response to be directly observed by the cameras. In addition, a splitting plate is installed in the opening of the bucket to minimize the disturbance of the bucket to the model open-flow part during the towing experiment.

We applied our newly developed underwater optical methods using eight high-speed cameras to capture the Inline (IL) and Cross-flow (CF) vibrations for a total of 46 locations (staggered black and white markers) along the cylinder span. Compared to the traditional strain gauge and accelerometer measurement in flexible cylinder VIV experiments, the optical tracking system provides both temporally and spatially dense and direct measurements on the displacement response.\(^{29}\) Four cameras are installed over 80d downstream of the model to measure CF vibration, while six cameras are installed 50d above the model to measure In-Line (IL) vibration. In the meantime, four 1500-lumen underwater lightings are installed to provide enough camera background lighting. Corresponding image processing and

### Table I. Nomenclature.

| Case | Exp I | Exp II | Sim |
|------|------|-------|-----|
| Length (L) | 2.04 m | 13.12 m | 150 |
| Diameter (d) | 1.2 cm | 2.8 cm | 1 |
| Aspect ratio (L/d) | 170 | 469 | 150 |
| Exposure ratio (l1/L) | 0.5 | 0.45 | 0.47 |
| Mass ratio (m'/m) | 1.38 | 3.0 | 2.55 |

of the current flexible model is 8.2%, as obtained from the pluck test in the air. The model properties are shown in Table I labeled as Exp I.

![FIG. 1. Experimental setup of flexible cylinder half hiding in the still water: (a) sketch shows the axis origin and direction; (b) photo shows the flexible cylinder setup in the tow tank.](image-url)
motion tracking code have been successfully developed to capture and follow the trajectory of either white or black markers.\textsuperscript{26,29}

In this paper, the result of the larger-scale flexible cylinder experiment conducted by Chaplin et al. in the early 2000s\textsuperscript{30} is also discussed. Their experiment has been widely used as a benchmark for verification and validation for semi-empirical, and CFD software to predict flexible cylinder VIVs.\textsuperscript{31} The experiment was carried out at Delft Hydraulics in the Delta Flume, where the flexible cylinder (total length of 13.12 m) passed through the depth of water (6.5 m) in the flume and up to the top of a tank (the "vacuum tank"). The vacuum tank was filled with water by evacuating air during the experiments. When the carriage was moving, the riser thus experienced a stepped current consisting of uniform flow over its lower part and still water over its top part. The model properties are shown in Table I labeled as Exp II.

**B. Numerical simulation**

Numerical simulation of the vortex-induced vibration of a flexible cylinder in stepped current is rather difficult. To the best of the authors’ knowledge, it has scarcely ever been achieved. In this paper, the incompressible flow is solved by employing the mixed spectral-element/Fourier method, i.e., spectral-element discretization on the \((x, y)\) plane and Fourier expansion along the cylinder axial direction \((z)\).\textsuperscript{32,33} In particular, the moving boundary due to the vibration is handled by a coordinate transformation method proposed by Ref. \textsuperscript{34}. Here, the benefit of using the Fourier method is twofold, as explained in\textsuperscript{35} first, the three-dimensional flow problem is transformed into a series of two-dimensional computations, which can significantly reduce the computing time; second, the well-developed FFTW library can be employed to achieve more efficiency. Moreover, the entropy-viscosity method\textsuperscript{36,37} based large-eddy simulation (LES) subgrid model is used to take into account the turbulence effect. The computational domain in \((x, y)\) plane is enclosed between two circles, namely, \(\sqrt{x^2+y^2} = 35d\) and \(\sqrt{x^2+y^2} = 0.5d\), where \(d\) is the diameter of the flexible cylinder. As shown in Fig. 2, the inflow boundary is placed on arc \(ABCDE\), where velocity is given by \(u = U, v = 0, w = 0\). Here, \(u, v, w\) are the three components of the velocity vector \(\mathbf{u}\). The outflow boundary is on arc \(EFGHA\), where \(\frac{\partial u}{\partial n} = 0\) is prescribed. Note that \(p\) is the pressure, and \(\mathbf{n}\) is the normal vector. Figure 3 shows the prescribed inflow velocity profile along the cylinder span. It could be seen that in section \(0 \leq z \leq 70d, U = 1\), while in \(70d \leq z \leq 150d, U = 0\). The hyperbolic tangential function \(\frac{\tan h(\eta)}{2} + 1\) \(U\) is used to smooth the stepped inflow velocity from high to low, where \(\eta\) is a constant, whose value is four times of the mesh size in \(z\) direction. It is worth noting that although the length of the flexible cylinder is \(l_1 + l_2 = 150d\), both the inflow velocity profile and the computational domain are expanded by an additional 16.67\(d\) to maintain the periodicity in \(z\) direction, where the aforementioned hyperbolic tangent function is again used to connect the low velocity \((U = 0\) and high velocity \((U = 1)\) section. The expanded section is named as "buffer region" in Ref. \textsuperscript{38}, which proves that the buffer region has negligible impact on the flow and structure vibrations of the rest of the riser, as long as the size of the buffer region is small compared to the cylinder length. Furthermore, the domain on the \((x, y)\) plane is partitioned into 2320 quadrilateral elements clustered around the cylinder to resolve the boundary layer. Specifically, on the radial direction, the size of the first layer element around the cylinder is \(0.01D\), which gives rise to \(y^+ < 1\) in all the simulations of this paper, and along the azimuthal direction, the element edge length is \(\frac{4d}{\pi}\).

In this paper, the reduced velocity is defined as

\[
U_r = \frac{U}{J_{n_1}d},
\]

(1)

where \(f_{n_1} = \frac{1}{2\pi} \sqrt{\frac{T}{m^*C_{m^*}}}\) is the first modal natural frequency, \(m^*\) is the cylinder mass ratio, \(T\) is the static tension applied on the cylinder, \(\rho_f\) is the fluid density, and \(C_{m^*} = 1.0\) is the assumed added mass coefficient. The flexible cylinder motion is governed by the following equation:

\[
\frac{\partial^2 \xi_j}{\partial t^2} + 2\xi_j\frac{\partial \xi_j}{\partial t} + \frac{EI}{\mu} \frac{\partial^4 \xi_j}{\partial x^4} + \frac{C_T \partial^2 \xi_j}{\partial z^2} = \frac{C_j}{2\mu},
\]

(2)

where \(\xi_j\) is the displacement along the \(J\)-direction \((J = x\) or \(J = y)\), and \(\mu = m^* \frac{\partial^2 d}{\partial \xi_j^2}\) is the cylinder mass per unit length, \(\xi\) is the damping coefficient with \(\omega_n = 2\pi \frac{\omega_{n_1}}{f_{n_1}}\), and \(EL\) is the bending stiffness. Note that in the simulation, \(T\) is a constant along the cylinder span, and \(EL < 10^{-4}\) to ensure that the riser is tension dominated, see Ref. \textsuperscript{39}. \(C_T\) is the J-component of the hydrodynamic force coefficient exerted on the cylinder surface. Equation (2) is constrained by the pinned boundary condition \((\xi_j = 0\) and \(\frac{\partial^2 \xi_j}{\partial z^2} = 0\) at \(z/d = 0\) and \(z/d = 150\). In the
simulation, \( m^* = 2.55, \ U_t = 15, \ \zeta = 2.0\% \), and the flow Reynolds number is set as \( Re = U_{\infty} d/\nu = 1000 \), where \( \nu \) is the kinematic viscosity. Note that the parameters used in simulation is placed in Table I labeled as Sim.

Equation (2) is discretized by the second-order central-difference scheme in space and the Runge-Kutta method in time. Three spectral-element modes in each element on the \((x, y)\) plane, and 512 Fourier planes along the axis \((z)\) direction are employed in the simulation. Note that in order to minimize the aliasing error, we employ over integration, i.e., we use five-points Gauss–Lobatto quadrature in each element and the 3/2 de-aliasing rule in the Fourier direction. The total computational time \( \frac{\Delta t}{D_s} \geq 400 \) with a time step \( \Delta t \Delta z = 1.0 \times 10^{-3} \), which results in the CFL number less than 0.3.

III. RESULT AND DISCUSSION

A. Structural response in the experiments

We measured the structural response of the flexible cylinder in the range of towing velocities \((U)\) from 0.2 to 0.6 m/s. The results of the two cases of \( U = 0.5 \) and \( U = 0.35 \) m/s are presented in this section and in Appendix. Figure 4 shows the structural response the displacement \( A \) is defined as \( A_u \) for the IL direction and \( A_v \) for the CF direction along the flexible cylinder at \( U = 0.5 \) m/s of Exp I: the CF frequency response, the IL frequency response, the displacement response in both directions and phase response of the CF first, IL first and the second harmonics along the cylinder span.

As shown above, the interesting phenomenon that strong first IL harmonic vibration travels from the cylinder section in still water to that in open flow, there might be an external disturbance source that can transfer energy from fluid to structure in the still water in the IL direction. Note that the hypothesis of an external fluid disturbance source in the still water inducing the first IL vibration harmonics in the IL direction as well, and Fig. 5(d) shows that the first IL vibration harmonics travels a different direction from still water part to open flow part, opposite from the first CF vibration and the second IL vibration harmonics.

It should be noted that the hypothesis of an external fluid disturbance source in the still water inducing the first IL vibration
maybe not be correct. This IL vibration response could be due to the structural coupling effect between flexible cylinder CF and IL response in the experiment, as the flexible model is twisted in the installation.
B. Vortical wake visualization in simulation

To determine whether an external fluid disturbance source in the still water exists, LES is performed on an IL and CF coupled flexible cylinder in stepped current to visualize the shedding vortices.

Here, the response of the flexible cylinder with 47% of its length in the uniform flow at $U_r = 15$ and $Re = 1000$ is given in Figs. 6(a)–6(d). The result is similar to those found in Exp I and Exp II that there is a strong first harmonic vibration component in the IL direction traveling from the part of the cylinder part in still water to that in open flow, in the opposite direction to the first harmonic vibration component in the CF direction and the second harmonic vibration component in the IL direction. The simulation result further demonstrates that the induced strong first harmonic vibration component in the IL direction is a result of external fluid source; it is not due to the IL and CF structural coupling effect because of the imperfect experimental installation.

A vortical flow pattern analysis is performed to understand better the mechanisms of interaction between the fluid and the structure. The snapshots of the 3D wake visualization obtained from the LES of the flexible cylinder at $U_r = 15$ and $Re = 1000$ at five-time instants are plotted in Fig. 7. In the lower part of the cylinder that is subject to uniform inflow, it could be observed that the IL nodes can separate the vortex tube formation behind the cylinder into different cells alternatively distributed along the spanwise direction. A similar phenomenon has been identified and explained in detail in our previous experimental and simulation work on the flexible cylinder in the uniform flow. Differently, in the upper still water region, it could be seen that the vortex tube is attached to the vibrating flexible cylinder with an alternating sign of vorticity.

FIG. 7. 3D vortices from the simulation of a flexible cylinder at $U_r = 15$ and $Re = 1000$. From left to right, the time snapshots are $t = \frac{T}{4}, 3.75, 4.875, 6.125, 7.375$, and $8.625$, corresponding to black dashed line in the sub-figure (d) of Figs. 6–11. Here, vortices are represented by iso-surfaces of $Q = 0.2$ and colored by $x_z$.

FIG. 8. Snapshots of two-dimensional vortex at location of the cylinder $z/L = 0.867$ at $t = 0$ (a), $t = \frac{T}{4}$ (b), $t = \frac{T}{2}$ (c), $t = \frac{3T}{4}$ (d), and $t = T$ (e), as well as the corresponding motion and force (f).
To visualize the relative motion between the local shedding vortex and oscillating cylinder, the snapshots of two-dimensional vortex at four locations of the cylinder, namely, $z/L = 0.867$, $z/L = 0.533$, $z/L = 0.333$, and $z/L = 0.167$, as well as the corresponding motion and force are plotted in Figs. 8–11. It could be observed that in the still water region, see Figs. 8 and 9(a)–9(e), these cylinder sections oscillate in the $Y$ direction (corresponding to the CF direction in the uniform open flow region) with an attached vortex pair of unequal strength. These asymmetric vortex pairs do not form simultaneously, upon flow reversal, which gives rise to an oscillatory force perpendicular to the main oscillation direction, at the main oscillation frequency, as shown in Figs. 8(f) and 9(f). Note that a similar phenomenon was first reported by Williamson, where the rigid cylinder in the oscillatory flow in the region of $K_C = 2 \pi \beta / \alpha < 7$, which is in the attached vortex pair regime, a vortex pair of an unequal strength will give rise to the "a lift force of low amplitude fluctuating at the oscillation frequency."
Therefore, the mechanism of the observed phenomenon in both the experiment and simulation can be explained as follows: the continuously alternating vortex shedding from the flexible cylinder in the uniform open flow results in a steady vibration of first harmonics in the CF direction with an amplitude close to 0.5–1 diameter and the second harmonics in the IL direction with a small amplitude. These vibration components hence travel from the cylinder section in open flow to that in the still water. If the oscillation in the still water is in $K_C^2$ ($A/d = 0.5, 1$), it gives rise to oscillatory force in the IL direction with the same first harmonic frequency, which in turn results in a first harmonic vibration component in the IL direction traveling back from still water to the open flow.

### IV. CONCLUSION

In this paper, we reported for the first time, both experimentally and numerically, a new phenomenon of secondary traveling wave vibration in the IL direction of the flexible cylinder when the flexible cylinder is placed partially in the uniform inflow and partially in the still water. It is a universal phenomenon that could be found in different flexible cylinder experiments and simulations of various scales. The phenomenon is characterized by (1) the same as that of the traditional flexible cylinder, and VIV of first CF harmonics and the second IL harmonics vibration were excited in the uniform inflow region, but (2) the unconventional external disturbance (positive energy-in) in the still water region generates the strong first IL harmonic vibration traveling from the still water region to uniform inflow region.

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**AUTHOR DECLARATIONS**

**Conflict of Interest**

The authors have no conflicts of interest to disclose.

**Author Contributions**

Z.W., A.L., and B.W. contributed equally to this work.

**DATA AVAILABILITY**

The data that support the findings of this study are available within the article.

**APPENDIX: ADDITIONAL EXPERIMENTAL CASES**

Two additional experimental cases of $U = 0.35$ m/s in Exp I and $U = 0.95$ m/s (case 21A018) in Exp II are shown in Figs. 12 and 13.
FIG. 12. Structural response along the flexible cylinder at $U = 0.35 \text{ m/s}$ in Exp I: (a) the CF frequency response along the model span, normalized by the peak CF vibration frequency $f_v = 5.91$; (b) the IL frequency response along the model span, normalized by the peak CF vibration frequency $f_v = 5.91$; (c) the standard deviation of the CF and the IL displacement response along the model span; and (d) the phase response of the CF first, IL first, and second harmonics along the model span.

FIG. 13. Structural response along the flexible cylinder at $U = 0.95 \text{ m/s}$ (case 21A018) in Exp II: (a) the CF frequency response along the model span, normalized by the peak CF vibration frequency $f_v = 5.78$ Hz; (b) the IL frequency response along the model span, normalized by the peak CF vibration frequency $f_v = 5.78$ Hz; (c) the standard deviation of the CF and the IL displacement response along the model span; and (d) the phase response of the CF first, IL first, and second harmonics along the model span.
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