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Title: URANS vs. Experiments of Flow Induced Motions of Multiple Circular Cylinders with Passive Turbulence Control

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Abstract: Two-dimensional Unsteady Reynolds-Average Navier-Stokes equations with the Spalart-Allmaras turbulence model are used to simulate the flow induced motions of multiple circular cylinders with passive turbulence control (PTC) in steady uniform flow. Four configurations with 1, 2, 3, and 4 cylinders in tandem are simulated and studied at a series of Reynolds numbers in the range of $30,000 < Re < 120,000$. Simulation results are verified by experimental data measured in the Marine Renewable Energy Laboratory. Good agreement was observed between the values of vorticity, amplitude ratio, and frequency ratio predicted by numerical simulations and experimental measurements. The amplitude and frequency response show the initial and upper branches in vortex induced vibration (VIV), transition from VIV to galloping, and galloping branch for all PTC-cylinders. The maximum amplitude of 2.9 diameters for the first cylinder is achieved at $Re=104,356$ in the numerical results. Compared with the first cylinder, the VIV initial branch starts at higher Re for the downstream cylinders due to the presence of the upstream cylinder(s). 2P and 2P+2S vortex patterns are observed at $Re=62,049$ and $Re=90,254$ for the single PTC-cylinder. Furthermore, the shed vortices of the downstream cylinders are strongly disrupted and modified by the vortices shed from the upstream one in the cases of multiple PTC-cylinders.

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September 20, 2014

To: Editor of Journal of Fluids and Structures

SUBJECT: Revised paper submission

“URANS vs. Experiments of Flow Induced Motions of Multiple Circular Cylinders with Passive Turbulence Control”

by Lin Ding, Li Zhang, Eun Soo Kim, Michael M. Bernitsas

Dear Editor,

We would like to submit our revised manuscript titled “URANS vs. Experiments of Flow Induced Motions of Multiple Circular Cylinders with Passive Turbulence Control”, for publication in the Journal of Fluids and Structures.

We revised the paper following the reviewer comments which we found very constructive.

I certify that this manuscript, or any part of it, has not been published and will not be submitted elsewhere for publication while being considered by the Journal of Fluids and Structures.

Thank you for your consideration.

A handwritten signature in black ink, appearing to be the name 'Lin Ding' in Chinese characters.

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RESPONSE TO THE REVIEWERS' COMMENTS

Manuscript No.: YJFLS-D-14-00255

Title: URANS vs. Experiments of Flow Induced Motions of Multiple Circular Cylinders
with Passive Turbulence Control

Article Type: Research paper

Corresponding Author: Lin Ding, PhD

All Authors: Lin Ding, PhD; Li Zhang, PhD; Eun Soo Kim, PhD; Michael M. Bernitsas,
PhD

The comments from the two reviewers are all constructive and valuable in revising and improving our paper. They are also helpful in guiding our research. We have checked our manuscript, including equations, tables, and figures, and revised the manuscript according to reviewers' detailed comments and suggestions.

Our detailed response follows.

REVIEWER #1

“In this article the authors present both experimental results and turbulent numerical simulations of the flow around one, two, three or four in-line cylinder which are free to move vertically. After a very thorough and well documented introduction the authors

present the configuration set-up, the numerical model and the results for one up to four cylinders.

The article is well written but in my opinion it is still incomplete and some conclusions are not proven at all. This article cannot be accepted in the current form, although the results are very interesting.

I will list some of the issue that have to be addressed in the revised version.”

Thank you for your constructive comments regarding our paper. We have revised our paper according to your comments. Our response is provided below.

Comment #1: p. 11 : can the authors give more details of the time integration method for the motion of the cylinder?

Response #1: Thank you for your suggestion. In our work, the classical mass-damper-spring oscillator model is used to describe the cylinder dynamics in FIM. The equation of motion is expressed as

$$m\ddot{y} + C\dot{y} + Ky = F_{fluid,y} \quad (1)$$

The solution to equation (1) can be obtained by solving the following system of two first-order differential equations (2) and (3):

$$m\dot{v} = F_{fluid,y} - C\dot{y} - Ky \quad (2)$$

$$\dot{y} = v \quad (3)$$

A mixed implicit-explicit method is adopted to solve the equation of motion together with the Navier-Stokes equations for velocity $\mathbf{u}$ and displacement $\mathbf{y}$ of the cylinder. Explicit time integration is used for velocity and implicit method is used for displacement using predicted velocity at a new time step:

$$\dot{\mathbf{u}}^n = \frac{\mathbf{u}^{n+1} - \mathbf{u}^n}{\Delta t} \quad (4)$$

$$\mathbf{u}^{n+1} = \mathbf{u}^n + \Delta t \left(\frac{\mathbf{F}_{fluid,y} - C\dot{\mathbf{y}} - K\mathbf{y}}{m} \right) \quad (5)$$

$$\mathbf{y}^{n+1} = \mathbf{y}^n + \Delta t \cdot \mathbf{u}^{n+1} \quad (6)$$

where Δt is the time step, the superscript $(n+1)$ represents values at the new time step $(n+1)$.

The manuscript has been revised accordingly to include this information in Subsection 3.1 “Governing Equations”.

Comment #2: What kind of scheme is used for the grid deformation?

Response #2: In the present study, the topological mesh change technique combined with General Grid Interfaces (GGI) is used for the mesh motion. This dynamic mesh technique is successfully used to simulate two cylinders in FIM in our earlier work (Ding L., Bernitsas M.M., Kim E.S., 2013. 2-D URANS vs. experiments of flow induced motions of two circular cylinders in tandem with passive turbulence control for $30,000 < \text{Re} < 105,000$. Ocean Eng. 72, 429-440). The detailed explanation of “topological mesh change” and “GGI” can be found in the paper by Beaudoin and Jasak (Beaudoin M, Jasak H.

Development of a generalized grid interface for turbomachinery simulations with OpenFOAM. In: Open Source CFD International Conference, Berlin, Germany; 2008). The reference has been added to the paper. Also we have clarified the scheme for the grid deformation in the last paragraph of Section 3 in the revised paper.

Comment #3: Is the near wall distance conserved during the re-meshing? Do the upper and lower walls have the same resolution? This kind of grid is non-orthogonal in the wake of each cylinder and it doesn't seem to be taken into account in the PISO scheme, did the authors checked if it matters or not?

Response #3: Yes, the near wall distance is conserved. The upper and the lower walls have the same resolution. In the computational domain, the subdomain in the vicinity of each cylinder is a $2D \times 2D$ square where the grid density for the near-wall region is enhanced (see in Fig. 4). When the cylinder is in FIM, the $2D \times 2D$ square is moving up and down with the cylinder. Therefore, the grid around the cylinder does not change and the near wall distance is conserved during the re-meshing. Only the grid of the top and bottom domains of the $2D \times 2D$ square will re-generate when the cylinder is in FIM. The grid upstream and downstream remains unchanged. The grid in the wake of each cylinder is orthogonal and remains unchanged as well. The PISO algorithm is appropriate for solving the momentum and continuity equations. The paper has been revised to reflect the question and answer

above.

Comment #4: p. 13 : what is the nature of the wall-function for roughness used? The authors claimed that they studied three different grids and chose the middle one, on which criteria? Did they perform this grid sensitivity on each configuration?

Response #4: In the grid for rough cylinder, the Passive Turbulence Control (PTC) is modeled as part of the cylinder surface, and treated as wall boundary. The roughness effects are modeled by wall functions. For our cases, wall function is used to connect the boundary layer with the outer flow, we just need to keep the first cell in the log-law flow region. Therefore, the first grid spacing y^+ is about 30-70 depending on the Reynolds number.

In order to determine the overall grid resolution to achieve a convergent and accurate solution in reasonable computational-time, three different grid densities were considered for each configuration in our work. For the sub-domain of the $2D \times 2D$ square around each cylinder, the three grid densities tried are: Coarse (circumferential $\times$ radial=180 $\times$ 40), Medium (240 $\times$ 70), and Fine (360 $\times$ 100). In the results of the grid sensitivity study for each configuration, the three grids produce similar results on drag coefficient, lift coefficient, and Strouhal number. Thus, the medium grid resolution was selected in the present work. In order to avoid confusion, we have clarified this problem in the last paragraph of Section 3 in the revised paper.

Comment #5: p. 14 : a graph with the different branches of the VIV would be nice. On which variable is computed each FFT?

Response #5: Thank you for your suggestion. For each configuration in our work, typical FIM branches can be observed in the results of amplitude and frequency response. In the revised manuscript, we can observe four branches for the oscillation of the single PTC-cylinder in Fig.5: the initial branch of VIV, the upper branch of VIV, transition from VIV to galloping, and galloping. We can obtain the different regions from Fig.10 for the two PTC-cylinders, Figs. 15-16 for the three PTC-cylinders, and Figs. 21-22 for the four PTC-cylinders as well. In addition, we have added the graphs of displacement time history for some typical cases. FFT is based on the time history of the cylinder displacement. This point has been clarified in the revised manuscript.

Comment #6: p. 15 : can the author explain the precision of the critical Reynolds number of each branch?

Response #6: The VIV response was either two branches or three branches depending on the product of mass ratio and damping ratio. In our work, the PTC triggers galloping at lower reduced velocity (U_{water}^*) leading to transition from VIV to galloping before the lower branch is reached. The Reynolds number of each branch is identified based on

the amplitude and frequency responses in the present study. It should be noted that the hysteresis response is often found between initial branch and upper branch of VIV in the experiment (Williamson and Govardhan 2004; Raghavan 2007). During the transition between upper branch and lower branch of VIV, intermittent switching could be observed instead (Raghavan 2007). Strictly, the upper and lower limits of the Re range for each branch are not the critical Re of the branch in our paper. We have clarified this point in the 1st paragraph of Subsection 4.1 in the revised manuscript.

Comment #7: p. 16 : the 2P+4S is not always obvious on figure 7. It would be nice to have each vortex identified on the figure (this remark is more important for multiple cylinders). The phase of the movement is hard to follow, especially for multiple cylinder and a graph of the displacement versus time with this figure (and the next ones) would be very helpful.

Response #7: Thank you for your suggestion. The vortex structure is 2P+2S in Figure 7 (Fig.8 in the revised manuscript). The vortices are tagged in the Figures in the revised manuscript. We have added the graphs of displacement time history for some typical cases, the figures are:

Figure 7. Displacement history of single PTC-cylinder at $Re=62,049$

Figure 9. Displacement history of single PTC-cylinder at $Re=90,254$

Figure 12. Displacement history of two PTC-cylinders at $Re=62,049$

Figure 14. Displacement history of two PTC-cylinders at $Re=90,254$

Figure 18. Displacement history of three PTC-cylinders at $Re=62,049$

Figure 20. Displacement history of three PTC-cylinders at $Re=90,254$

Figure 24. Displacement history of four PTC-cylinders at $Re=62,049$

Figure 26. Displacement history of four PTC-cylinders at $Re=90,254$

Comment #8: p. 17 : I don't see any proof for this statement : "Instead, the galloping mechanism based on instability and synchronization of FIM with the shear layer motion drives the cylinder motion. This observation is in agreement with the perfectly periodic motion of the cylinder."

Response #8: Thank you for your detailed reviewing. In order to avoid the confusion identified by the reviewer, Figure 9 which shows the displacement time history for the case of single PTC-cylinder at $Re=90,254$ was added in the revised manuscript. As shown in Figure 9, the cylinder presents a stable periodic motion at $Re=90,254$. However, the vortex structure is not stable, in the sense that the number and the structure of the vortices shed during each cycle are not exactly the same throughout the time history. Therefore, the wake pattern frequency does not synchronize with the cylinder motion any longer. The vortex patterns do not drive the FIM in galloping. The galloping mechanism is based on instability and synchronization of FIM with the shear layer motion. This phenomenon was observed in

our earlier work as well (Ding L., Bernitsas M.M., Kim E.S., 2013. 2-D URANS vs. experiments of flow induced motions of two circular cylinders in tandem with passive turbulence control for $30,000 < Re < 105,000$. Ocean Eng. 72, 429-440). The manuscript has been revised to clarify this issue in the 2nd paragraph of Subsection 4.1.

Comment #9: p. 18 : "the beating phenomenon resulting in low displacement in some cycles" is quite impossible to follow just by looking at fig. 9.

Response #9: We have added the graph of displacement time history for the case of two PTC-cylinders at $Re=62,049$. The motion of the 2nd cylinder exhibits response resembling a beating phenomenon resulting in low displacements in some cycles as shown in Figure 12(b) in the revised paper. This issue has been clarified in the second paragraph of Subsection 4.2.

Comment #10: p. 20 : any idea why the frequencies of the three cylinders are so close?

Response #10: For the case of the numerical study of the three PTC-cylinders at $Re=62,094$, the oscillatory frequencies of the three cylinders are close to each other. We have checked the motion of the cylinders during the whole simulated time. The vortex pattern 2P is observed for the first cylinder. The vortex pair sheds from the 1st cylinder when the maximum displacement is reached. The 2nd cylinder oscillates in the wake of the

1st cylinder. Due to the specific spacing between the 1st and 2nd cylinder, the vortex pair from the 1st cylinder doesn't collide into the body of the 2nd one. Instead, the vortex pair from the first cylinder triggers the vortex shedding of the 2nd cylinder and can change the direction of movement. Therefore, the oscillatory frequency of the 2nd cylinder is approximately equal to the wake pattern (2P) frequency of the 1st cylinder. And the 3rd cylinder follows the 2nd one. Thus, the frequencies of the three cylinders are close to each other. A few lines were added to the second paragraph of Subsection 4.3 in the revised paper to clarify this issue.

Comment #11: p. 21 : where can we see the phase lag between cylinder 3 and 4?

Response #11: Thank you for your detailed reviewing. The graph of displacement time history for the case of four PTC-cylinders at $Re=62,049$ was added in the revised paper. As shown in Fig. 24(c) and Fig. 24(d) in the revised manuscript, there is a nearly 180° phase lag between the 3rd and 4th cylinders.

Comment #12: Globally some conclusion are based on results (or result exploitation) which are not presented. Some of the numerical results are very far from experimental data and still all the flow physics are deduced from these simulations. Any ideas why such differences?

Response #12: We have added and modified some figures to prove the conclusions in the revised paper. In addition, we have checked all the results and found that the main difference between numerical simulations and experiments is in the amplitude response of a downstream cylinder at high Reynolds number. Specifically:

- (a) For the cases of three PTC-cylinders as shown in Figure 15, the amplitude of the 3rd cylinder is smaller in the numerical results than in the experiments.
- (b) This phenomenon can also be observed in the amplitude response of the 2nd and 3rd cylinders in the four PTC-cylinder case in Figure 21.

The vortex development of the downstream cylinder is strongly influenced by the vortices shed from the upstream cylinder. Depending on the spacing between cylinders, the motion of the downstream cylinder may be enhanced or weakened. In numerical animations, we have observed that depending on amplitude and speed the upstream vortices may coalesce with the downstream cylinder vortices and yank them away from the downstream cylinder thus depriving the downstream cylinder from forming and shedding complete vortices that induce high lift. On other occasions we have observed increased amplitude of the downstream cylinder due to a double effect: * the cylinders are out of phase thus each maintains its own vortices and * at the same time a gap flow forms forcing the downstream cylinder to high amplitudes.

The numerical simulation of FIM of multiple cylinders is a very complex

fluid-structure interaction phenomenon. Changes in spacing may result in different motion of the cylinders at high Reynolds number. In addition to the complexity of the phenomenon there are of course imperfections in our CFD model and its ability to model exactly the experiments. Those are: (i) The free surface is modeled as a wall in the current numerical study. The free surface boundary condition limits FIM in experiments. (ii) The FIM of multiple cylinders is simplified as a two-dimensional problem in the numerical study. (iii) In the numerical model, only the linear viscous damping is modeled while the physical model exhibits a very complex viscous model as we have explained in previous work (Ding et al. 2013).

In our research we work continually to improve the numerical model for FIM of multiple cylinders to reduce the error.

REVIEWER #2

“This is an interesting study for applications like vibration control and energy harvesting, which combines simulation and experimentation to analyse the flow Induced Motions of multiple cylinders. Except for some details to improve the introduction and the presentation of the results, this submission can be accepted for publication in the Journal of Fluids and Structures.”

Thank you for your positive comment and support. We really appreciate it that you

find our research useful and worth publishing. We have revised our paper according to your comments. Our response is provided below.

Comment #1: To delimit the domains associated with the different branches it is better, rather than to use the Reynolds number, to choose the reduced velocity which is a more natural parameter to describe the fluid-structure interaction.

Response #1: Thank you for your suggestion. The reduced velocity is an important parameter to describe FSI. In the revised paper, three x -axes, including Reynolds number (Re), reduced velocity (U_{water}^*), and flow velocity (U), are used in Fig. 5, Fig. 10, Fig. 15, Fig. 16, Fig. 21, and Fig. 22 to show the amplitude and frequency response of the cylinders. Meanwhile, both Re and reduced velocity are employed to describe FIM in the text. In calculating the reduced U_{water}^* , the natural frequency in water is used. Since the natural frequencies of the four cylinders are so close to each other (Table 1), the value of 1.114 representing the natural frequency of the first and fourth cylinders was used. This is a compromise since a different natural frequency would result in a different U_{water}^* for each cylinder at the same flow speed.

Comment #2: In the introduction (p.3) : I suggest to associate MIV (Movement Induced Vibrations) to VIV instead of galloping in the sentence « The two common forms of FIM...

» .

Response #2: Thanks for your suggestion. Flow-induced motions (FIM) of multiple cylinders are studied in this paper. We are not sure what the term MIV implies; we think that it implies forced movement of the cylinder, which we do not consider in this paper.

The cylinders are elastically mounted and vibrate freely. Vortex shedding results in fluctuating forces acting on the cylinder and further induces vibrations. Vortex-induced vibration (VIV), flutter, and galloping are common forms of FIM. The phenomena of VIV and galloping are observed in our simulation and experiment. Thus, we discussed only VIV and galloping in the introduction. If the reviewer feels that MIV is a better term for VIV and galloping, which are flow induced, we would appreciate additional explanation to understand if it would be appropriate to introduce the term MIV.

Comment #3: It's true that nonlinear effects are important in FIM but lock-in can be roughly described by a linear approach (see for example E. de Langre. Frequency lock-in is caused by coupled-mode flutter. *Journal of Fluids and Structures*, 22(6-7):783-791, 2006.)

Response #3: Thanks for reminding us of this question. Indeed, the motion of the cylinder in the range of lock-in can be roughly described by a linear model (eq. 1 in the paper). In general, the empirical models of flow-induced motion can be classified into two groups: one is the force decomposition model, and the other is the wake-oscillator model. In a

wake-oscillator model, the Van de Pol-type equation is employed as the governing equation of the lift force and the motion. Based on the wake-oscillator model, de Langre (2006) developed a linear approach to describe the cylinder motion in the range of lock-in. We assume that the motion of the cylinder in VIV or galloping is sinusoidal as modeled by equation (1). We have mentioned this issue in the first paragraph of Section 1.

Comment #4: « Smooth circular cylinders do not gallop ». You must specify that it is about an isolated cylinder.

Response #4: Thank you for your suggestion. We agree with you, proximity of another cylinder may also induce galloping to a smooth circular cylinders. In order to avoid the confusion, this sentence has been revised as: “An isolated smooth circular cylinder does not gallop.”

General Response: Many grammatical and typographical errors have been corrected.

We acknowledge the reviewers’ constructive comments and suggestions. They have been very valuable in improving the quality of our manuscript.

HIGHLIGHTS

- URANS simulations of multi-cylinders with PTC in FIM agree well with experiments
- Simulations predict well FIM response as well as VIV branches and galloping
- Vortex patterns of multiple cylinders in VIV and galloping are predicted accurately

URANS VS. EXPERIMENTS OF FLOW INDUCED MOTIONS OF MULTIPLE CIRCULAR CYLINDERS WITH PASSIVE TURBULENCE CONTROL

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ABSTRACT

Two-dimensional Unsteady Reynolds-Average Navier-Stokes equations with the Spalart-Allmaras turbulence model are used to simulate the flow induced motions of multiple circular cylinders with passive turbulence control (PTC) in steady uniform flow. Four configurations with 1, 2, 3, and 4 cylinders in tandem are simulated and studied at a series of Reynolds numbers in the range of $30,000 < Re < 120,000$. Simulation results are verified by experimental data measured in the Marine Renewable Energy Laboratory. Good agreement was observed between the values of vorticity, amplitude ratio, and frequency ratio predicted by numerical simulations and experimental measurements. The amplitude and frequency response show the initial and upper branches in vortex induced vibration (VIV), transition from VIV to galloping, and galloping branch for all PTC-cylinders. The maximum amplitude of 2.9 diameters for the first cylinder is achieved at $Re=104,356$ in the numerical results. Compared with the first cylinder, the VIV initial branch starts at higher Re for the downstream cylinders due to the presence of the upstream cylinder(s). 2P and 2P+2S vortex patterns are observed at $Re=62,049$ and $Re=90,254$ for the single PTC-cylinder. Furthermore, the shed vortices of the downstream cylinders are strongly disrupted and modified by the vortices shed from the upstream one in the cases of multiple PTC-cylinders.

KEY WORDS

flow induced motion; multiple cylinders; vortex induced vibration; galloping; passive turbulence control.

1. INTRODUCTION

Flow induced motion (FIM) is a frequently observed fluid-structure interaction phenomenon of elongated, bluff, flexible bodies in a cross-flow. Vortex induced vibrations (VIV) and galloping are the two most common forms of FIM. VIV is caused by the excitation due to the vortices shed behind a bluff body. The shed vortices create an asymmetric oscillatory pressure distribution causing periodic lift forces to act on the bluff body. Oscillation of the bluff elongated body occurs when the body is elastically mounted or flexible transversely to the steady stream. FIM is inherently a nonlinear phenomenon because the motion can enhance the strength of vortices resulting in lock-in. The lock-in phenomenon means that the cylinder oscillates at or near the natural frequency of the system and vortex shedding is synchronized with the oscillation frequency. When the frequency of the vortex shedding locks onto the oscillating frequency of an elastically mounted cylinder, the cylinder would be excited to large amplitude of oscillation. The oscillatory lift forces on the cylinder lead to VIV. An isolated smooth circular cylinder does not gallop. Flow asymmetry caused either by incoming flow or geometric modification of the cylinder may induce galloping which is an instability phenomenon not driven by vortex shedding but motion of the shear layers on either side of the cylinder. Passive Turbulence Control (PTC) in the form of sand strips used in this work both in experiments and simulations result not rotationally symmetric body and, thus, galloping. The PTC-to-FIM was developed by Park, et al. (2012). The motion of the cylinder in the range of lock-in can

be roughly described by a linear model. In the present study, the motion of the cylinder in VIV or galloping is assumed to be sinusoidal. In general, the empirical models of FIM can be classified into the force decomposition model (Sarpkaya, 2004) and the wake-oscillator model (Hartlen and Currie, 1970; de Langre, 2006). In a wake-oscillator model, the Van de Pol-type equation was employed as the governing equation of the lift force and the motion. Based on the wake-oscillator model, de Langre (2006) developed a linear approach to describe the cylinder motion in the range of lock-in. More detailed information about flow induced motion for a single, isolated cylinder can be found in the reviews by Blackburn et al. (2001), Sarpkaya (2004), Williamson and Govardhan (2004), Bearman (2011), and Wu et al. (2012).

On the other hand, multiple circular cylinders in cross-flow are employed in many engineering disciplines and applications, such as heat exchangers, cooling systems for nuclear power plants, offshore structures, chimneys, and cables. In addition, contrary to the traditional engineering challenge of reducing FIM to prevent structural damage, the VIVACE (Vortex Induced Vibration for Aquatic Clean Energy) team at the University of Michigan works towards increasing the oscillatory response of a cylinder in FIM (Bernitsas et al., 2008, 2009; Bernitsas and Raghavan, 2009, 2011; Chang et al., 2011; Ding et al., 2013; Kim et al., 2013; Lee and Bernitsas, 2011; Park et al., 2012, 2013). Their purpose is to harness marine hydrokinetic energy from ocean/river currents and maximize the power density of VIVACE by using multiple cylinders in synergistic FIM (Kim et al., 2013).

Despite the widely spread applications, research on flow around multiple cylinders started rather late in comparison with research on a single cylinder. Analyses and scientific debate of FIM of single cylinder in cross-flow is on-going due to the complexity of its dynamics. FIM of multiple cylinders is much more complex resulting in new phenomena of synergistic motion and is expected to continue for long time (Kim et al., 2013).

Flow interference in multiple-cylinder configurations is a common subject due to its practical interest in engineering fields. Two stationary parallel circular cylinders positioned perpendicular to a uniform, steady, free-stream is being investigated the most as it presents the next level in complexity of FIM of bluff elastic bodies. Most extensive is the work by Zdravkovich (1985, 1987, 2003). Zdravkovich's classification has been widely referenced by other researchers. The interference effects strongly depend on the arrangement of the two cylinders and their orientation to the free stream (Zdravkovich, 2003). So far though, few studies focus on interference between oscillating cylinders. For two cylinders positioned in tandem, there are three possible configurations regarding their elastic support: (a) elastic-fixed, (b) fixed-elastic, and (c) elastic-elastic. The configuration (b) is always called wake oscillator because it can experience wake-induced vibration (WIV) due to the wake of the upstream stationary cylinder. Tandida et al. (1973) investigated the stability of the wake oscillator at low Reynolds numbers (Re). They concluded that synchronization occurred only when the downstream cylinder oscillated inside the vortex-formation region of the upstream one. Furthermore, Bokaian and Geoola (1984) observed wake induced

galloping in the experiments and a quasi-steady theory was developed to predict the galloping behavior. Brika and Laneville (1999) conducted the experiment to study the dynamic response of a long flexible cylinder in the wake of a stationary one. They found that the synchronization onset of the downstream cylinder depends on the reduced velocity and the in-flow spacing. Assi et al. (2006, 2010) investigated the mechanism of WIV by experiments. They found that WIV is different from a typical VIV response and suggested that WIV of the downstream cylinder is excited by the unsteady vortex structure interactions between the body and the upstream wake. In contrast, few investigations have studied the upstream oscillator, which means an elastically mounted cylinder upstream of a stationary one. To and Lam (2007) conducted experiments to investigate the response of a small cylinder placed upstream of a twice larger cylinder and showed that the motion of the upstream cylinder is affected by the stationary downstream cylinder.

Compared with the wake oscillator and upstream oscillator models, the setup of two oscillating cylinders is the most complex case. The spacing between cylinders, defined as the center-to-center distance between them, has significant effect on the response. There is a critical spacing at which the fluctuation lift coefficient suddenly jumps, and it is dependent on the conditions of the two oscillating cylinders system and the free stream. The critical spacing is usually observed between 2.7 and 4 diameters. Laneville and Brika (1999) examined the response of two flexible circular cylinders with spacing between 7 and 25 diameters. They found that the upstream cylinder behaves as an isolated one and the

response of the downstream cylinder depends on the spacing. This behavior of the upstream cylinder was also observed in the numerical work by Mittal and Kumar (2001, 2004) when the spacing was 5.5 diameters in the simulations at $Re=1,000$. Kim et al. (2009) investigated the effect of the spacing between the two cylinders on FIM and concluded that the upstream cylinder vibrates like an isolated cylinder when the spacing exceeds a critical spacing of 2.7 diameters in the range of $4,365 < Re < 74,200$. Lin et al. (2013) found that the dynamic response of both cylinders is strongly influenced by the spacing in laminar flow ($Re=100$). As the spacing between the two cylinders increases, the interaction of the cylinders first enhances and then reduces. Zhao (2013) conducted simulations for the two cylinders in tandem with different spacing at $Re=150$ and found the lock-in regime of the reduced velocity depends on the spacing as well. Ding et al. (2013) studied FIM of two rigid circular cylinders in tandem for $30,000 < Re < 105,000$, which were free to oscillate transversely to the flow by simulation. The spacing was set at 2 diameters. The simulation results were in excellent agreement with experimental data and oscillatory amplitude of 3.5 diameters was reached in galloping when $Re=105,000$. In addition, only a limited amount of papers study the interference between three or four cylinders (Hanenkamp and Hammer, 1981; Liu et al., 2008; Wang et al., 2002) and all the cylinders are stationary even when three or four cylinders were considered. Kim et al. (2013) have studied the problem of multiple cylinders in FIM experimentally.

CFD can be useful in understanding phenomena that are difficult to measure experimentally such as VIV to galloping transition, vortex structures, motion of separation point, and pressure distribution for multiple cylinders. In this paper, four configurations with one, two, three, and four cylinders are investigated numerically and results are compared to experiments to validate the developed CFD code. The physical and numerical models are summarized in Sections 2 and 3. In Section 4, the results are compared with experiments conducted in the Low Turbulence Free Surface Water (LTFSW) channel of the Marine Renewable Energy Laboratory (MRELab) at the University of Michigan for a single isolated cylinder (Chang, 2010) and for multiple cylinders (Kim et al., 2013). Amplitude ratio, frequency ratio, and vortex patterns match well with experimental observations. Conclusions are presented at the end.

2. PHYSICAL MODEL

A simple schematic of an elastically mounted circular cylinder is shown in Figure 1(a). The elements of this oscillator are: (a) a rigid circular cylinder of diameter D and length L , (b) two end-supporting linear springs of stiffness K , and (c) damping of the transmission mechanism C . The cylinder is placed perpendicular to the flow velocity U , which is in the x -direction. The cylinder oscillates in the y -direction, which is perpendicular to its axis and the flow velocity in x . A picture of four single-cylinder oscillators mounted on the LTFSW Channel is presented in Figure 1(b). In the experiments, each cylinder is mounted on a rail

system allowing adjustment of the longitudinal spacing between the four cylinders. The spacing between the centers of the neighboring cylinders is set at $2.5D$. For the case of two or three cylinders in tandem, the experimental setup is the same as the four-cylinder case shown in Figure 1(b).

In the present study, simulations are verified by comparisons to experimental measurements on the flow-induced motions of multiple circular PTC-cylinders in tandem. The system parameters in the numerical simulations are the same as those used in the corresponding experiments. Particulars of the isolated cylinder oscillator and the four single-cylinder oscillators are listed in Table 1. The stiffness of the springs and the system damping are measured by a series of free decay tests in air.

To enhance FIM of a circular cylinder and extract more hydrokinetic energy from fluid flows, passive turbulence control (PTC) was introduced and studied extensively in the MRELab. Park et al. (2012) thoroughly investigated the effects of PTC on VIV and galloping of a circular cylinder, and produced the PTC-to-FIM Map based on the amplitude response of the cylinder. All modeling parameters of PTC are defined in Figure 2. Two roughness strips parallel to the cylinder axis are attached running along the entire length of the cylinder. Strip thickness is about equal to the thickness of the boundary layer and affects FIM profoundly. PTC is used in this paper to induce synergetic dynamics of multiple cylinders with large amplitude galloping resulting in dramatic expansion of the synchronization range. Thus, the system with multiple oscillators works like a

multi-cylinder reciprocating engine. Table 2 shows the details of the roughness strip used in the present study. It should be noted that the entry angle of the PTC (α_{PTC}) for each oscillating system is different: (a) for the single cylinder, $\alpha_{\text{PTC}}=\pm 20^\circ$; (b) for the two-cylinder configuration, $\alpha_{\text{PTC}}=\pm 20^\circ$ for the 1st cylinder and $\alpha_{\text{PTC}}=\pm 30^\circ$ for the 2nd cylinder; (c) for the three-cylinder configuration, $\alpha_{\text{PTC}}=\pm 30^\circ$ for all three cylinders; (d) for the four-cylinder configuration, $\alpha_{\text{PTC}}=\pm 30^\circ$ for all four cylinders.

3. COMPUTATIONAL METHOD

3.1. Governing Equations

In the present study, flow simulations are performed by a code developed in the MRELab based on open source CFD tool OpenFOAM, which is composed of C++ libraries solving continuum mechanics problems with a finite-volume discretization method. In addition, code was developed (a) for the dynamics of the multiple oscillators and (b) for dynamic adjustment of the grid near a cylinder as it approaches a boundary. The time-dependent viscous flow solutions are obtained by numerical approximation of the incompressible URANS equations in conjunction with the one-equation Spalart-Allmaras turbulence model (Spalart and Allmaras, 1994). The two-dimensional URANS governing equations are:

$$\frac{\partial U_i}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} (2\nu S_{ji} - \overline{u'_j u'_i}) \quad (2)$$

where ν is the molecular kinematic viscosity and S_{ij} is the mean strain-rate tensor

$$S_{ij} = \frac{1}{2} \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \quad (3)$$

where U_i is the mean flow velocity vector. The quantity $\tau_{ij} = -\overline{u'_i u'_j}$ is known as the Reynolds stress tensor, which is symmetric. In the Spalart-Allmaras model, a common method employs the Boussinesq approximation to relate the Reynolds stresses to the mean velocity gradient as

$$\tau_{ij} = 2\nu_T S_{ij} \quad (4)$$

where ν_T is the kinetic eddy viscosity and the related defining equations are as follows:

$$\nu_T = \tilde{\nu} f_{v1}, \quad f_{v1} = \frac{\chi^3}{\chi^3 + c_{v1}^3}, \quad \chi = \frac{\tilde{\nu}}{\nu}$$

$\tilde{\nu}$ is an intermediate working variable of the turbulence model and obeys the following transport equation

$$\frac{\partial \tilde{\nu}}{\partial t} + u_j \frac{\partial \tilde{\nu}}{\partial x_j} = c_{b1} \tilde{S} \tilde{\nu} - c_{w1} f_w \left(\frac{\tilde{\nu}}{d} \right)^2 + \frac{1}{\sigma} \left\{ \frac{\partial}{\partial x_j} \left[(\nu + \tilde{\nu}) \frac{\partial \tilde{\nu}}{\partial x_j} \right] + c_{b2} \frac{\partial \tilde{\nu}}{\partial x_i} \frac{\partial \tilde{\nu}}{\partial x_i} \right\} \quad (5)$$

Additional definitions of functions and constants can be found in the work by Spalart and Allmaras (1994). In this study, the trip terms f_{t1} and f_{t2} are turned off, and at the same time, the “trip-less” initial condition for $\tilde{\nu}$ is used following Travin et al. (2000).

The classical mass-damper-spring oscillator model is used to describe the cylinder dynamics in FIM. That is, the cylinder response in VIV is assumed to be sinusoidal. The one-degree of freedom equation of motion can be expressed as

$$m\ddot{y} + C\dot{y} + Ky = F_{fluid,y} \quad (6)$$

where m is the total oscillating mass of the cylinder and attachments including 1/3 of the spring mass and $F_{fluid,y}$ is the fluid force acting on the cylinder in the y -direction. The solution to equation (6) can be obtained by solving the following system of two first-order differential equations:

$$m\dot{v} = F_{fluid,y} - C\dot{y} - Ky \quad (7)$$

$$\dot{y} = u \quad (8)$$

where u is the velocity of the cylinder. A mixed implicit-explicit method is adopted to solve the equation of motion together with the Navier-Stokes equations for velocity u and displacement y of the cylinder. Explicit time integration is used for velocity and implicit method is used for displacement using predicted velocity at a new time step (Wu et al., 2011):

$$\dot{u}^n = \frac{u^{n+1} - u^n}{\Delta t} \quad (9)$$

$$u^{n+1} = u^n + \Delta t \left(\frac{F_{fluid,y} - C\dot{y} - Ky}{m} \right) \quad (10)$$

$$y^{n+1} = y^n + \Delta t \bullet u^{n+1} \quad (11)$$

where Δt is the time step, the superscript $(n+1)$ represents values at the new time step $(n+1)$.

In the numerical simulations in this work, a second-order Gauss integration scheme with a linear interpolation for the face-centered value of the unknown is used for the divergence, gradient, and Laplacian terms in the governing equations. The second-order backward Euler method is adopted for time integration. Thus, the numerical discretization scheme gives second order accuracy in space and time. A pressure implicit with splitting of operators (PISO) algorithm is employed to solve the momentum and continuity equations together in a segregated way.

3.2. Computational Domain and Grid Generation

The computational domain size is different in the stream direction depending on the number of cylinders in the flow field. The domain of $50D \times 9D$ is used for the single-cylinder simulation, $52.5D \times 9D$ for the case of two cylinders, $55D \times 9D$ for three cylinders, and $57.5D \times 9D$ for four cylinders. Figure 3 shows the computational domain in the four PTC-cylinders case. The entire domain includes five boundaries: inflow, outflow, top, bottom, and the cylinder walls. The distance between the inlet boundary and the center of the 1st cylinder, l_{up} , is set at $25D$. The downstream length of the domain, l_{down} , is also set at $25D$. l_{up} and l_{down} are the same regardless of the number of cylinders placed in tandem in the flow. The horizontal spacing d between two cylinders is fixed at $2.5D$. The inflow velocity is considered as uniform and constant. At the outflow boundary, a zero gradient condition is specified for velocity as well as v_t and $\tilde{v}$. The bottom condition is defined as a wall-boundary to match the experimental conditions. In the present numerical study, the

free-surface is simplified by modeling it as a wall. A moving wall boundary condition is applied for the cylinder surface when the cylinders are in flow induced motion. In addition, similar to the “trip-less” initial condition for the one-cylinder simulation (Wu et al., 2011), the fluid domain is divided into two regions: (a) From the upstream inlet to the center of the 1st cylinder, where a zero value is applied for the eddy viscosity. (b) From the downstream-half of the 1st cylinder through the downstream cylinders to the outlet of the flow domain where a nonzero value is used for the eddy viscosity. The nonzero value is set equal to the molecular eddy viscosity for all simulations. In summary, the body and channel boundary conditions in the numerical model match the physical model conditions as described in Section 2 with the exception of the free surface, which is modeled by a wall. In addition, the safety stop is considered in the simulations. When the displacement of a cylinder reaches $2.9D$, the body hits the safety stop in the simulation to match the condition of the experiments.

Two-dimensional, structured, computational grids were generated for all cases using the Gambit grid-generating software. The topological mesh change technique combined with General Grid Interfaces (Beaudoin and Jasak, 2008) is used for dynamic mesh motion. This dynamic mesh technique is successfully used to simulate two cylinders in FIM in earlier work by Ding et al. (2013). In the computational domain, the upper and the lower walls have the same resolution. The sub-domain in the vicinity of each cylinder is a $2D \times 2D$ square where the grid density for the near-wall region is enhanced to solve for high

resolution in flow properties. When the cylinder is in FIM, the $2D \times 2D$ square is moving up and down with the cylinder. Therefore, the grid around the cylinder does not change and the near wall distance is conserved during the re-meshing. The grid in the wake of each cylinder is orthogonal and remains unchanged. Thus, PISO algorithm is appropriate for solving the momentum and continuity equations. For the cylinder with PTC, due to the specifically modified surface geometry, a wall function type boundary condition is used to account for the effect of surface roughness. Due to the nature of the wall-function for the roughness model used in this study, the near-wall grid spacing was selected to produce a y^+ between 30-70, depending on the Reynolds number. In order to determine the overall grid resolution to achieve a convergent and accurate solution in reasonable computational-time, three different grid densities were considered for the grid sensitivity study for each configuration. For the sub-domain of the $2D \times 2D$ square around each cylinder, the three grid densities tried are: coarse (circumferential $\times$ radial=180 $\times$ 40), medium (240 $\times$ 70), and fine (360 $\times$ 100). In the results of the grid sensitivity study for each configuration, the three grids produce similar results on drag coefficient, lift coefficient, and Strouhal number. Thus, the medium grid resolution was selected in the present work. A close-up of the medium grid, total about 117,800 cells, for the case of four PTC-cylinders is shown in Figure 4.

4. RESULTS AND DISCUSSION

A series of simulations were carried out for validating the ability of the code to predict hydrodynamic loads and response of multiple cylinders undergoing flow induced motions using 2D-URANS. Comparisons are made with experiments conducted in the LTFSW Channel of the MRELab. The Reynolds numbers are in the high-lift TrSL3 regime in all the tests, where TrSL stands for Transition in Shear Layer and “3” indicates the third region (Zdravkovich, 1997). Five Reynolds numbers are chosen and each represents a different oscillating branch in the FIM response range. Typical VIV branches are (a) the initial branch, (b) the upper branch, (c) the lower branch, (d) the desynchronization range. The relative widths of the branches are very different in the TrSL2 regime ($Re < 10,000$, Williamson and Govardhan 2004) and the TrSL3 regime ($24,000 < Re < 300,000$, Bernitsas et al. 2009). The cases studied in this work are in the TrSL3 regime. Beyond VIV, galloping may appear by introducing a geometric asymmetry in the structure or flow. The PTC roughness strips, though symmetrically placed on the cylinder, introduce asymmetry due to the motion of the cylinder resulting in an angle of attack leading to galloping. With PTC implemented in this work, the gap between VIV and galloping is closed by triggering galloping earlier resulting in back-to-back VIV and galloping. Thus, the lower branch of VIV and the desynchronization range in VIV have been not observed in the results. Instead, typically two branches past the end of VIV appear: (e) transition from VIV to galloping, (f) fully developed galloping. In (e), both hydrodynamic mechanisms of VIV and galloping coexist.

In all multiple-cylinder cases, cylinders are placed in tandem. Only cylinders with PTC are considered for two reasons: (a) PTC promotes high-amplitude vibration and expands the range of synchronization (Chang et al. 2011, Park et al. 2013), which is the goal of the MRELab for harnessing marine hydrokinetic energy. (b) CFD tests typically fail to predict well flow around smooth circular cylinders for $Re > 1 \times 10^4 - 1.2 \times 10^4$ because it is difficult to capture the location of the separation points, which oscillate significantly by about $5^\circ - 10^\circ$. By placing PTC on a cylinder, the separation point is specified both in experiments and in CFD resulting in very good agreement between experiments and CFD simulations (Wu et al. 2011, Ding et al. 2013).

The simulations and experiments have been conducted with one-cylinder, two-cylinder, three-cylinder, and four-cylinder configurations. The amplitude and frequency response of the cylinders are discussed in this section. Vortex patterns for each arrangement of multiple cylinders are analyzed as well. In both the experiments and the simulations, the amplitude, A_{peak} , of each cylinder is calculated by averaging the absolute values of the 60 highest positive and negative peaks. The frequency of oscillation for each cylinder is processed using Fast Fourier Transform of the time history of the cylinder displacement. The dominant frequency of oscillation is found and non-dimensionalized by the corresponding system natural frequency in water, $f_{n,water}$. It should be noted that $f_{n,water}$ has different value for each one of the four oscillators as shown in Table 1. In the presentation of the results, three x -axes (all proportional to the flow velocity), including Reynolds number (Re),

reduced velocity (U_{water}^*), and flow velocity (U), are used in the figures to show the amplitude and frequency response of the cylinders. In the text, both Re and reduced velocity (U_{water}^*) are employed to describe FIM. In calculating the reduced velocity ($U_{water}^* = U / f_{n,water} D$), the natural frequency in water is used. Since the natural frequencies of the four cylinders are so close to each other (Table 1), the value of 1.114 representing the natural frequency of the first and fourth cylinders was used. This is a compromise since a different natural frequency would result in a different U_{water}^* for each cylinder at the same flow speed.

4.1. FIM of Single PTC-Cylinder

The amplitude and frequency ratios for the numerical study and experiments for the single PTC-cylinder are plotted in Figure 5. In the experiments, there are clearly four branches of the amplitude ratio curve, which are the VIV initial branch, VIV upper branch, region of transition from VIV to galloping, and the galloping branch. When $Re < 52,178$ ($U_{water}^* < 4.79$), the cylinder is in the VIV initial branch. The range from $Re = 52,178$ ($U_{water}^* = 4.79$) to $Re = 81,793$ ($U_{water}^* = 7.51$) is the VIV upper branch. The next region is the transition from VIV to galloping ranging from $Re = 81,793$ ($U_{water}^* = 7.51$) to $Re = 90,254$ ($U_{water}^* = 8.28$). As shown in Figure 5(a), the transition region initiates with a jump of the amplitude at approximately $Re \approx 81,793$ ($U_{water}^* \approx 7.51$) and again the amplitude increases at a relatively high rate until Re reaches 90,254 ($U_{water}^* = 8.28$), where galloping starts. Meanwhile, the frequency encounters a drop-off when switching from VIV upper branch to

galloping in Figure 5(b). The Reynolds number range of each branch is identified based on the amplitude and frequency responses in the present study. It should be noted that the hysteresis response is often found between initial branch and upper branch of VIV in the experiment (Williamson and Govardhan 2004; Raghavan 2007). During the transition between upper branch and lower branch of VIV, intermittent switching could be observed instead (Raghavan 2007). Strictly, the upper and lower limits of the Re range for each branch are not the critical Re of the branch in this paper. The five CFD cases simulated in this work are $Re=47,947$ ($U_{water}^*=4.40$), $Re=62,049$ ($U_{water}^*=5.70$), $Re=76,152$ ($U_{water}^*=6.99$), $Re=90,254$ ($U_{water}^*=8.28$), and $Re=104,356$ ($U_{water}^*=9.58$), which correspond to the typical branches of FIM as identified by the experimental data. The CFD results follow the basic trend of the experiments and match the experimental data very well in the VIV upper branch. Specifically, the amplitude of $2.9D$ is achieved at $Re=104,356$ ($U_{water}^*=9.58$), which means the amplitude in the galloping branch reaches the limit in the simulations and is higher than that in the experiments.

The amplitude and frequency are the important measurements that can quantify VIV of the cylinder. The vortex structure in the near wake is another useful property in understanding the fluid structure interaction. The typical results of vortex structures and time histories of displacement ratio of the single PTC-cylinder at $Re=62,049$ ($U_{water}^*=5.70$) and $Re=90,254$ ($U_{water}^*=8.28$) are presented in Figure 6-9. It should be noted that the reason for the successful numerical prediction of experimental results lies in the application of the

surface roughness PTC (Wu et al. 2011, Ding et al. 2013). As shown in Figures 6 and 7, the cylinder presents a stable periodic motion in its VIV upper branch at $Re=62,049$ ($U_{water}^*=5.70$). The 2P vortex pattern is observed in one oscillating cycle, where P stands for “pair”. The vortex pattern is always stable when the cylinder is in a certain branch of VIV. Transition between branches is accompanied by vortex pattern change. When $Re=90,254$ ($U_{water}^*=8.28$), the vortex structure P+S+S+P is observed in one cycle in Figure 8, where S stands for “single”. Actually, the vortex structure is not stable that means the number and the structure of the vortices shed during each cycle are not exactly the same throughout the time history. This is the case at the end of the transition from VIV to galloping and also at the start of the galloping branch, where the frequency of vortex shedding is faster than that of the motion. As shown in Figure 9, the cylinder presents a stable periodic motion at $Re=90,254$ ($U_{water}^*=8.28$). Therefore, the wake pattern frequency does not synchronize with the cylinder motion any longer. When galloping is fully developed, the vortex patterns do not drive the FIM. Instead, the galloping mechanism is based on instability and synchronization of FIM with the shear layer motion. This phenomenon was observed in earlier work by Ding et al. (2013) as well.

4.2. FIM of Two PTC-Cylinders in Tandem

Two cylinders with PTC arranged in tandem are free to oscillate transversely to the free stream. The center-to-center distance between the two cylinders is $2.5D$. PTC is applied with its leading edge at $\alpha_{PTC}=\pm 20^\circ$ for the 1st cylinder and $\alpha_{PTC}=\pm 30^\circ$ for the 2nd cylinder.

The width of the PTC strips is 1.26cm corresponding to 16° coverage. The experiments are conducted at $31,025 \leq Re \leq 101,536$ in the LTFSW channel. The numerical simulations are carried out at the Reynolds numbers used in the CFD cases of the single PTC-cylinder. Figure 10 shows the amplitude and frequency response of the two PTC-cylinders. For the 1st cylinder, within the test Reynolds number range, four regions are observed in the amplitude and frequency ratio curves: VIV initial branch ($30,000 < Re < 40,000$, $3.88 < U_{water}^* < 5.17$), VIV upper branch ($40,000 < Re < 80,000$, $5.17 < U_{water}^* < 10.34$), transition from VIV to galloping ($80,000 < Re < 90,000$, $10.34 < U_{water}^* < 11.64$), and galloping ($Re > 90,000$, $U_{water}^* > 11.64$). Compared with Figure 5, the curves of amplitude and frequency ratios of the 1st cylinder are similar to the results of the single PTC-cylinder case. When $Re = 104,356$ ($U_{water}^* = 13.50$), the channel stops (both top and bottom) are hit at $2.9D$ by the 1st cylinder and the maximum amplitude reaches $2.9D$. For the 2nd cylinder, the response trends are quite different from those of the single PTC-cylinder and the 1st cylinder. As shown in Figure 10(a), an amplitude bump appears before the occurrence of the upshot of the initial branch at $Re \approx 55,000$ ($U_{water}^* \approx 7.11$). The frequency trends also show a typical response around $Re = 55,000$ ($U_{water}^* = 7.11$) in Figure 10(b). A similar phenomenon of amplitude response is also observed in the transition region from VIV upper branch to galloping at $Re \approx 95,000$ ($U_{water}^* \approx 12.29$). This behavior is due to the fact that the 2nd cylinder is driven by two frequencies, one is due to the wake frequency of the 1st cylinder and the

other due to its the own vortex shedding. The dominant frequency in the displacement time history is plotted in Figure 10(b).

The wake vortex-structure and displacement history of two PTC-cylinders in tandem at $Re=62,049$ ($U_{water}^*=8.03$) and $Re=90,254$ ($U_{water}^*=11.67$) are illustrated in Figure 11-14. When $Re=62,049$ ($U_{water}^*=8.03$), both cylinders generate 2P vortex patterns as shown in Figure 11. The upstream vortices directly and closely interact with the downstream cylinder. The motion of the 1st cylinder shows a periodic pattern in Figure 12(a). The amplitude ratio of the 2nd cylinder is close to that of the 1st cylinder at $Re=62,049$ ($U_{water}^*=8.03$), but the motion of the 2nd cylinder exhibits response resembling a beating phenomenon resulting in low displacements in some cycles as shown in Figure 12(b). Deeper examination of the animation of the cylinder response and wake vorticity reveals the reason for the small amplitude in those cycles. The specific spacing between the cylinders results in the following scenario: The vortex development of the 2nd cylinder is weakened by the shed vortices from the 1st cylinder. Specifically, when a vortex that sheds from the 1st cylinder reaches the 2nd cylinder, it coalesces with the partially developed vortex of the same rotation sign - attached to the same side of the 2nd cylinder - and takes it away. Thus, the 2nd cylinder is deprived of the circulation of its own vortex and the induced lift. This results in quick reversal of the cylinder motion and suppressed amplitude of the 2nd cylinder.

For $Re=90,254$ ($U_{water}^*=11.67$), galloping is initiated for the 1st cylinder. Both cylinders shed vortices in 2P+2S mode as presented in Figure 13. Occasionally, in the 1st cylinder

shedding, one more vortex is shed in the upward travel; that is, there is cycle-to-cycle variation in vortex shedding. For the 2nd cylinder, the vortices are disrupted and modified by the vortices shed from the upstream cylinder. This makes a smaller displacement of the 2nd cylinder (see in Figure 14), with average value of about $1.77D$.

4.3. FIM of Three PTC-Cylinders in Tandem

Besides the studies with one-cylinder and two-cylinder configurations, simulations and experiments have been carried out with three PTC-cylinders in tandem. The in-flow center-to-center distances between cylinders are kept equal to $d=2.5D$. The angles of the PTC are $\alpha_{PTC}=\pm 30^\circ$ for all three cylinders. Results of three PTC-cylinders are shown in Figures 15 and 16. The different oscillating branches can be clearly observed in the curves of amplitude and frequency ratios. The dynamic response of the cylinders placed downstream of the 1st cylinder is affected by the 1st one. Comparing the results of the single PTC-cylinder and two PTC-cylinders, we observe that the downstream cylinders have an influence on the FIM for the 1st cylinder as well. As shown in Figure 15, the amplitude ratios of the 1st cylinder rise from the VIV upper branch to galloping by an easy gradient instead a sharp jump. For the 2nd and 3rd cylinders, similar to the behavior of the downstream cylinder of the 2-PTC-cylinder case, the results show drops in the amplitude and frequency response around $Re=50,000$ ($U_{water}^*=6.47$). When galloping is fully developed ($Re>95,000$, $U_{water}^*>12.29$), the response trends of the two downstream cylinders are profoundly affected by the upstream cylinder. Further, the amplitude ratio for

the 2nd and 3rd cylinders is lower than that of the 1st cylinder. Notably, $2.9D$ is obtained for the 1st cylinder in the simulation results.

For the three-cylinder cases, the motion of the downstream cylinders is strongly influenced by the vortices shed from the 1st cylinder. When $Re=62,094$ ($U_{water}^*=8.03$), as presented in Figure 17, the vortex pattern of the 1st cylinder is 2P and it is the same as that of the 1st cylinder for the two-cylinder case. The vortices developed by the 2nd and 3rd cylinders are taken away and absorbed by the upstream vortices. It is difficult to identify the vortex pattern for the 2nd and 3rd cylinders in animations due to the presence of the wake of the 1st cylinder. Moreover, the amplitude is higher for the upstream cylinder as shown in Figure 18. Due to the specific spacing between the 1st and 2nd cylinder, the vortex pair from the 1st cylinder doesn't collide into the body of the 2nd one. Instead, the vortex pair from the first cylinder triggers the vortex shedding of the 2nd cylinder and can change the direction of movement. Therefore, the oscillatory frequency of the 2nd cylinder is approximately equal to the wake pattern (2P) frequency of the 1st cylinder. The 3rd cylinder follows the 2nd one. Thus, the frequencies of the three cylinders are close to each other.

In Figure 19, the vortex pattern of three PTC-cylinders for $Re=90,254$ ($U_{water}^*=11.67$) is plotted. Eight vortices are shed by the 1st cylinder in a 2P+4S vortex-structure. The vortex patterns are not clear for the 2nd and 3rd cylinders due to the presence of the 1st cylinder wake. It should be noted that the 2nd cylinder follows the 1st cylinder and its vortex shedding is disturbed by upstream vortices. Thus, the displacement of the 2nd cylinder is

influenced and small amplitude is obtained, as shown in Figure 20. Amplitude ratio of $A_{peaks}/D=2.49$ is reached for the 1st cylinder.

4.4. FIM of Four PTC-Cylinders in Tandem

Simulations and experiments are performed to investigate the flow-induced motions of four cylinders with PTC in tandem. In all cases, the in-flow center-to-center distance between consecutive cylinders is fixed at $2.5D$ and PTC is placed symmetrically on cylinders at $\pm 30^\circ$ from the front stagnation point of the corresponding ideal flow. The amplitude and frequency ratio are presented in Figures 21 and 22, respectively. The maximum amplitude of $2.9D$ is achieved for the 1st cylinder in galloping. Compared with the results of the three PTC-cylinder cases, the amplitude and frequency response are very similar for the three upstream cylinders. That means the 4th cylinder hardly affects the three upstream cylinders. But the FIM of the 4th cylinder is strongly disrupted by the vortices shed from the upstream cylinders. As shown in Figure 21, four oscillating branches can be clearly observed and the VIV upper branch starts at $Re \approx 40,000$ ($U_{water}^* \approx 5.17$) for the 1st cylinder. However, due to the presence of the upstream cylinders, the velocity corresponding to the start point of the VIV upper branch moves to higher velocities. Especially for the 4th cylinder, the VIV upper branch appears at $Re \approx 60,000$ ($U_{water}^* \approx 7.76$).

For $Re=62,049$ ($U_{water}^*=8.03$), the presence of the other three cylinders practically does not affect the oscillations of the 1st cylinder. In this case, the 1st cylinder experiences VIV and the 2P vortex pattern is observed in Figure 23. It is worthwhile to note that significant

phase difference exists between two adjacent cylinders. As shown in Figure 24, there is a nearly 180° phase lag between the 3rd and 4th cylinders. The shear layers of the downstream cylinders are significantly affected by the wake of the upstream cylinders. The oscillating frequencies are very close for all four cylinders as can be seen in Figure 22. The amplitude of the 1st cylinder is higher than the downstream cylinders and reaches $1.48D$.

As shown in Figure 25 for $Re=90,254$ ($U_{water}^*=11.67$), when the 1st cylinder moves down from the point of maximum displacement, a pair (P) of vortices sheds from the cylinder, followed closely by a single vortex (S). The same is observed in the upward journey of the cylinder. Thus, the vortex model is 2P+2S for the 1st cylinder. For the downstream cylinders, the vortex pattern is hard to identify as the shed vortices are affected and modified by the vortices from the upstream cylinder wakes. In addition, the motion of the 1st cylinder has a regular form, which can be observed in the displacement ratio curve in Figure 26. Amplitude ratio $A_{peaks}/D=2.49$ is achieved by the 1st cylinder which is much higher than those of the downstream cylinders.

5. CONCLUSIONS

The flow induced motions (FIM) of one, two, three, and four circular cylinders with passive turbulence control (PTC) in tandem were investigated using two-dimensional Unsteady Reynolds-Averaged Navier-Stokes equations with the Spalart-Allmaras turbulence model at Reynolds numbers in the high-lift TrSL3 regime. Results were verified

by comparison to experimental measurements derived in the LTFSW Channel of the MRELab in the University of Michigan. The amplitude and frequency response of all cylinders in a group were presented and compared to experiments. The vortex structures in the wake of each configuration at $Re=62,049$ and $Re=90,254$ were analyzed as well. PTC was used for all the cylinders in the study and was proven to be the key factor in achieving agreement between numerical and experimental results. The following conclusions can be drawn.

- (1) A 2D-URANS code based on OpenFOAM was developed to study numerically the FIM of multiple circular cylinders with passive turbulence control. Comparison with experiments shows that the developed CFD tool captures the salient features of the flow and, thus, can be useful in understanding the behavior of multiple cylinders in VIV and galloping.
- (2) Four configurations with one, two, three, and four cylinders were investigated and the simulation results are in good agreement with experimental data. Simulation points were selected at each important branch: the initial branch in VIV, the upper branch in VIV, transition from VIV to galloping, and fully developed galloping for all PTC-cylinders.
- (3) Displacement of 2.9 diameters is reached by the 1st cylinder at $Re=104,356$ in the numerical simulations. This means that the cylinder in galloping would hit the safety

stops in the LTFSW Channel, thus limiting further comparison between simulations and experiments.

- (4) In the cases of multiple PTC-cylinders in tandem, the amplitude of the downstream cylinders exhibit a hump before the upshot of the VIV initial branch. Compared with the first cylinder, the VIV initial branch for the downstream cylinders starts at a higher Reynolds number.
- (5) For the single PTC-cylinder, 2P and 2P+2S vortex patterns are observed at $Re=62,049$ and $Re=90,254$ corresponding to VIV upper branch and galloping, respectively. The shed vortices of the downstream cylinders are strongly disrupted and modified by the vortices shed by the upstream cylinder for the multiple PTC-cylinders.

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FIGURE CAPTIONS

Figure 1. (a) Schematic of single PTC-cylinder physical model; (b) Four cylinders with PTC in tandem in the LTFSW Channel

Figure 2. Configuration of sand strips (PTC) along the cylinder

Figure 3. Computational domain for 4 cylinders

Figure 4. Close-up of the medium grid for 4 PTC-cylinders

Figure 5. Comparison of experimental and numerical results for single PTC-cylinder

Figure 6. Vortex pattern for single PTC-cylinder at $Re=62,049$

Figure 7. Displacement history of single PTC-cylinder at $Re=62,049$

Figure 8. Vortex pattern for single PTC-cylinder at $Re=90,254$

Figure 9. Displacement history of single PTC-cylinder at $Re=90,254$

Figure 10. Amplitude and frequency ratios comparison for two PTC-cylinders

Figure 11. Vortex pattern for two PTC-cylinders at $Re=62,049$

Figure 12. Displacement history of two PTC-cylinders at $Re=62,049$

Figure 13. Vortex pattern for two PTC-cylinders at $Re=90,254$

Figure 14. Displacement history of two PTC-cylinders at $Re=90,254$

Figure 15. Amplitude ratios comparison for three PTC-cylinders

Figure 16. Frequency ratios comparison for three PTC-cylinders

Figure 17. Vortex pattern for three PTC-cylinders at $Re=62,049$

Figure 18. Displacement history of three PTC-cylinders at $Re=62,049$

Figure 19. Vortex pattern for three PTC-cylinders at $Re=90,254$

Figure 20. Displacement history of three PTC-cylinders at $Re=90,254$

Figure 21. Amplitude ratios comparison of four PTC-cylinders

Figure 22. Frequency ratios comparison of four PTC-cylinders

Figure 23. Vortex pattern for four PTC-cylinders at $Re=62,049$

Figure 24. Displacement history of four PTC-cylinders at $Re=62,049$

Figure 25. Vortex pattern for four cylinders with PTC at $Re=90,254$

Figure 26. Displacement history of four PTC-cylinders at $Re=90,254$

Table 1

Table 1. Particulars of the isolated cylinder and the four-cylinder oscillators

Parameters	Symbol	Isolated Cylinder	1 st Cylinder	2 nd Cylinder	3 rd Cylinder	4 th Cylinder
Diameter [m]	D	0.0889	0.0889	0.0889	0.0889	0.0889
Length [m]	l	0.9144	0.9144	0.9144	0.9144	0.9144
Oscillating mass [kg]	m	10.75	9.53	9.59	9.51	9.58
Spring stiffness[N/m]	K	1600	744.29	757.41	737.24	747.39
Damping ratio	ζ	0.0122	0.0206	0.0198	0.0172	0.0168
Damping [Ns/m]	C	3.2	3.464	3.382	2.882	2.834
Natural frequency[Hz]	$f_{n,water}$	1.57	1.114	1.121	1.109	1.114
Mass ratio	m^*	1.896	1.681	1.690	1.677	1.689
Added mass [kg]	m_a	5.671	5.671	5.671	5.671	5.671

Table 2. Details of roughness strip of PTC (P60)

Angular coverage of strip	θ [degrees]	16
Sand paper thickness	P [mm]	0.587
Average grit height	k [mm]	0.260
Total thickness	$T=P+k$ [mm]	0.847

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