**Numerical Simulation and Experimental Validation for Energy Harvesting of
Single-Cylinder VIVACE Converter with Passive Turbulence Control**

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ABSTRACT

Flow Induced Motions (FIM) of a single, rigid, circular cylinder on end-springs are investigated for Reynolds number $30,000 < Re < 110,000$. Passive Turbulence Control (PTC) in the form of roughness strips is applied to enhance FIM and increase the efficiency of the VIVACE (Vortex Induced Vibration for Aquatic Clean Energy) converter in harnessing marine hydrokinetic energy. Numerical simulations are performed using a solver for two-dimensional Unsteady Reynolds-Averaged Navier-Stokes equations, which is developed based on OpenFOAM using a finite-volume discretization method. The results are in excellent agreement with experiments conducted in the Marine Renewable Energy Laboratory (MRELab) at the University of Michigan. Amplitude and frequency are predicted correctly in the initial and upper VIV branches, in VIV-to-galloping transition, and in galloping. The 2S (S=Single) vortex pattern is observed in the initial VIV branch and the 2P+2S (P=Pair) in the upper branch for $30,000 \leq Re \leq 80,000$. Transition from VIV to galloping initiates at $Re=90,000$ with a P+2S+P+S pattern. Maximum amplitude of $3.5D$ (D =Diameters) is achieved and more than seven vortices per cycle are observed in fully developed galloping. With PTC, the VIVACE converter can harness hydrokinetic energy from currents or tides over the entire range of FIM synchronization. Energy conversion efficiency reaches 37% in simulations and 28% in experiments.

KEY WORDS

hydrokinetic energy; flow induced motions; vortex induced vibrations; galloping; passive turbulence control; VIVACE converter

1. INTRODUCTION

Vortex shedding from a bluff body is a common physical phenomenon, which involves complex flow mechanisms. Vortex shedding results in fluctuating forces acting on the bluff body, which may induce vibrations when the bluff body is elastically mounted or deformable [1, 2]. This vibration further alters vortex formation and shedding. This interaction between fluid and structure, called Vortex Induced Vibration (VIV), is a form of Flow Induced Motions (FIM). Another form of FIM is galloping, which is an instability driven phenomenon, resulting in even more vigorous vibrations. With the rapid development of ocean engineering, wind engineering, aerospace, and nuclear engineering, FIM has attracted wide attention and has become one of the challenging subjects in fluid mechanics and its engineering applications. In several machines and engineering problems involving fluid-flow past a bluff body, FIM is typically treated as a destructive phenomenon because of the resulting fatigue damage.

With entirely opposite objective of those previous efforts, which are mainly focused on reducing FIM effects, the VIVACE (Vortex Induced Vibration for Aquatic Clean Energy) converter is designed to generate power by utilizing potentially destructive FIM; specifically, VIV and galloping. It is a hydrokinetic power-generating device invented by Bernitsas and Raghavan [3] and further developed by the Marine Renewable Energy Laboratory (MRELab) at the University of Michigan [4-8]. In its simplest form, the VIVACE converter consists of a rigid circular cylinder mounted on elastic springs and connected to a power take-off system via a transmission mechanism. It can harness hydrokinetic energy from ocean and river currents as

slow as 0.4m/s [6]. The VIVACE converter generates power with high power-to-weight density [9]. In order to enhance the cylinder FIM and extract more hydrokinetic energy from fluid flows, Passive Turbulence Control (PTC) utilizing roughness strips was developed and tested experimentally in the MRELab. The FIM of a rigid circular cylinder mounted on end linear-springs with PTC to attain higher power generation is studied in this paper.

VIV and galloping are the most commonly observed FIM phenomena. VIV is generally excited by three-dimensional separated flow and widely treated as a fluid-structure interaction problem. This inherently self-limiting highly nonlinear phenomenon has been receiving extensive attention in the academia and industry for several decades. A large number of researches on VIV can be found in the comprehensive reviews [10-16]. The frequency and amplitude of motion are the most important characteristics of FIM that can effectively describe and quantify VIV and galloping. They are influenced by several system parameters, including the mass ratio, spring stiffness, structural damping, and surface roughness. In VIV, large amplitude response occurs when the oscillation frequency of the cylinder is close to the vortex shedding frequency, which is called “synchronization” or “lock-in” [15]. Throughout the broad synchronization region, the oscillation frequency and vortex shedding frequency remain nearly equal. Two alternate explanations are still debated in the literature: (a) Lock-in of vortex shedding frequency to the frequency of oscillation [16]. (b) Variable fluid added mass altering the natural frequency of the oscillating body in water resulting in variable frequency resonance as the flow velocity increases [17]. Furthermore, Khalak and Williamson [14, 17, 18] divided the

regime of VIV into three branches based on the amplitude and frequency response; those are the VIV initial branch, upper branch, and lower branch. Hysteretic transition between the initial and upper branches and intermittency between the upper and lower branches were also observed by Khalak and Williamson [14]. These observations were made experimentally for Reynolds number around 3,800 and have been confirmed for Re up to 10,000. This range covers up to the TrSL2 (Transition Shear Layer 2) [19]. These branches are also present at higher Reynolds numbers $24,000 < Re < 130,000$ but with very different relative magnitude and shape [9, 20, 21].

Galloping is the other FIM phenomenon studied in this work since it is used by the VIVACE converter to convert hydrokinetic energy to mechanical. Galloping is enhanced here as a one-degree of freedom instability which is known to be a high-amplitude and low-frequency motion. The galloping generation-mechanism is the interruption of communication between the separated upper and lower flows. It is instigated by geometric asymmetry [22, 23]. As body and fluid oscillate with respect to each other, oscillatory fluid forces are exerted on the body. If the resulting fluid-structure interaction increases the body vibrations, the body motion is dynamically unstable. When the flow separates from the bluff body, the fluid force is a nonlinear function of the relative angle of attack, and the resulting FIM of the bluff body is referred to as galloping [22]. A long slender body with bluff cross-section may be subjected to large amplitude galloping oscillations only if it does not have a smooth circular cross-section. In recent decades, many researchers have investigated the galloping characteristics of bluff bodies with different cross sections. Nakamura et al. [23] found that a circular section with attachments may

experience galloping. Blevins [22] shows data for galloping of a variety of cross-sections. More recent experimental studies [6, 20, 21, 24-26] demonstrated as well that galloping is highly dependent on the cross-sectional geometry and the angle of incident flow. Furthermore, proximity of another cylinder may also induce the galloping excitation in circular cylinders [27]. These studies on galloping of circular cylinders confirm that flow asymmetry caused by body geometric asymmetry, attachments, or upstream flow disturbance may cause galloping in a circular cylinder.

In this work, in order to enhance FIM and to excite galloping oscillation of a circular cylinder, a circular cylinder is geometrically modified using straight roughness strips placed on the cylinder surface following guidelines presented in the PTC-to-FIM Map [28]. Roughness on the cylinder can effectively change the flow properties, such as pressure distribution, boundary layer transition, and separation point. Past experimentation has shown that different parameters of the roughness can influence the fluid motion, including the roughness location, the roughness height, and the roughness coverage [6, 20, 21]. Passive Turbulence Control (PTC), which consists of selectively distributed surface roughness in the form of sand strips, has been studied extensively in the MRELab to alter the FIM of a single cylinder in the transverse flow. With the application of PTC, the flow over the cylinder surface is altering in a way that generates higher lift, which is shown to be better synchronized with the motion. These studies culminated in the development of the PTC-to-FIM Map for a single circular cylinder [21]. Results showed that

when the strip thickness is about equal to the thickness of the boundary layer, the PTC affects the FIM profoundly.

In this paper, the FIM of a circular cylinder with PTC is studied using two-dimensional Unsteady Reynolds-Averaged Navier-Stokes (2D-URANS) simulations and the results are verified by experimental data conducted in the MRELab. The objective of this study is to establish the capability of a numerical tool to simulate the VIVACE system with passive turbulence control in FIM and to enable the VIVACE converter to attain higher power generation with higher efficiency for energy harnessing. Tests are conducted in the range of $30,000 < Re < 110,000$, which falls in the high-lift TrSL3 regime. TrSL stands for Transition in Shear Layer and “3” indicates the third region where the shear layer is fully saturated resulting in stronger vortices, shorter formation length, and highest lift [19]. The physical model and running parameters are presented in Section 2. In Section 3, the numerical approach and grid generation are described. The mathematical model for the harnessed and dissipated power is summarized in Section 4. In Section 5, numerical results for amplitude response, frequency response, vortex structures, and energy harvesting are compared with experiments conducted in the Low Turbulence Free Surface Water (LTFSW) Channel of the MRELab and discussed. Conclusions are drawn at the end.

2. PHYSICAL MODEL

In the MRELab of the University of Michigan, a virtual damper-spring (V_{CK}) system has been introduced in order to replace the physical dampers and springs of the VIVACE converter.

The V_{CK} system provides the damper and spring forces based on displacement and velocity feedback and thus simulates two of the three elements of the mechanical oscillator; the third being the mass. The hydrodynamic force exerted on the cylinder due to the relative flow is not in the closed loop and thus is not biased [8]. In the present work, the sketch depicted in Figure 1 represents the V_{CK} VIVACE system and oscillator. The elements of this model are a rigid circular cylinder of diameter D and length L , a virtual spring of stiffness $K_{virtual}$, and a system damping C_{system} . The cylinder is constrained to oscillate in the y -direction, which is perpendicular to its axis and the flow velocity in the x -direction. Details on the V_{CK} system are provided by Lee et al.[8].

In the present study, simulations are verified by experimental measurements on FIM and energy harvesting of one circular cylinder with PTC. The experiments on harnessing energy using the V_{CK} VIVACE model were conducted in the LTFSW Channel at the MRELab of the University of Michigan. This channel has maximum flow speed of 2m/s with a test section 2.44m long, 1m across and 0.8m deep. It enables a continuous flow of water through the test section. The sidewalls of the test section are transparent to allow for visualization of the flow past cylinders. A photograph of the mounted test cylinder in the LTFSW Channel is shown in Figure 2(a). The cylinder oscillatory displacements and velocities are measured by an encoder built in the motor for V_{CK} VIVACE model. The detailed experimental setup and procedures can be found in earlier research by Chang et al. [6] and Lee et al. [8]. The system parameters in the simulation are the same as those used in the corresponding experiments in the MRELab, as listed

in Table 2. The inertial mass of the V_{CK} system, m , is composed of the mass of the oscillating part m_{osc} and effective inertial mass of the motor-pulley-belt system m_{eff} . The m_{eff} can be estimated after performing the free-decay tests for the motor-pulley-belt system [29].

It should be noted that, Lee and Bernitsas [30] performed system identification for the VIVACE oscillator, identified the actual nonlinear damping model, which they subtracted from the oscillator using the V_{CK} controller. Then, they added actual linear damping thus conducting experiments with a nearly ideally linear system as it is simulated in the CFD calculations. The real damping proved to be very complex and strongly nonlinear particularly in small speeds of the oscillator. Specifically, besides the classical linear viscous damping, Lee et al [8, 30] showed that it included Coulomb damping, quadratic and cubic damping as well as hysteresis. In earlier studies, they showed that the linear viscous damping model used in the simulations matches well with the physical damping model and real springs when the velocity of oscillations is not near zero [31]. Therefore, the classical linear viscous damping model is assumed in the present simulations of the single PTC-cylidnder. It represents well the tests performed with the V_{CK} model in the LTFSW Channel as long as the cylinder is in motion. Nevertheless, before the cylinder starts moving in FIM in the experiments, there is still static friction which is not modeled in the CFD model. Thus, as it can be seen in the results of the amplitude and frequency responses, the motion in CFD starts earlier than in the experiments.

In order to enhance the FIM of the cylinder and to increase the efficiency and power density of VIVACE converter in harnessing the marine hydrokinetic energy, PTC in the form of

roughness strips was introduced in the MRELab [6]. As shown in Figure 2(b), a pair of straight roughness strips are pasted on the surface of the cylinder symmetrically with respect to the forward stagnation point, covering circumferential surface area equivalent to 16° ($1'' = 0.0254\text{m}$ wide PTC strip). The angle $\alpha_{PTC}=20^\circ$, is measured in degrees from the forward stagnation point in the corresponding ideal flow. Based on the previous research [20, 21, 28], the strips with roughness designation P60 have been used as PTC in this study. All modeling parameters of PTC are defined in Figure 2(b). Table 3 shows the details of the roughness strips used in the present study.

3. COMPUTATIONAL METHOD

The basic equations and turbulence model used in the present work are discussed in this section. The integration scheme, computational domain, and grid generation are provided as well.

3.1. Governing Fluid Equations and Turbulence Model

In the present work, the flow around the cylinder of the VIVACE converter is studied by solving the two-dimensional Unsteady Reynolds-Averaged Navier-Stokes (URANS) equations together with the Spalart-Allmaras turbulence model. Simulations were performed by open source CFD tool OpenFOAM, which is composed of C++ libraries solving continuum mechanics problems with a finite-volume discretization method. The two-dimensional, incompressible, unsteady 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 x_i are the Cartesian coordinates, U_i are the corresponding velocity components, ν is the molecular kinematic viscosity, and S_{ij} is the mean strain-rate tensor

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

The Reynolds-stress tensor $\tau_{ij} = -\rho \overline{u'_i u'_j}$ is modeled through the Boussinesq eddy-viscosity approximation as $2\mu_t S_{ij}$, where μ_t is the turbulence eddy viscosity.

The original Spalart-Allmaras model [32] is employed in this work, which solves a transport equation for the kinematic eddy viscosity. The Spalart-Allmaras model is different from other one-equation models and algebraic models in the sense that it is a local model, which means the equation in one location does not depend on the solutions at other points [32]. It is therefore compatible with grids of any structure [31]. It is numerically forgiving, in terms of near-wall resolution and stiffness. This model has been shown to give acceptable results for a wide variety of situations and is known for its stability. In the Spalart-Allmaras model, the turbulent eddy viscosity is computed from:

$$\mu_t = \rho \tilde{\nu} f_{\nu 1} \quad (4)$$

where

$$f_{\nu 1} = \frac{\chi^3}{\chi^3 + c_{\nu 1}^3} \quad (5)$$

$$\chi = \frac{\tilde{\nu}}{\nu} \quad (6)$$

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

$$\frac{\partial \tilde{\nu}}{\partial t} + u_j \frac{\partial \tilde{\nu}}{\partial x_j} = c_{b1} \tilde{S} \tilde{\nu} - c_{w1} f_w \left(\frac{\tilde{\nu}}{d} \right)^2 + \frac{1}{\sigma} \left\{ \frac{\partial}{\partial x_j} \left[(\nu + \tilde{\nu}) \frac{\partial \tilde{\nu}}{\partial x_j} \right] + c_{b2} \frac{\partial \tilde{\nu}}{\partial x_i} \frac{\partial \tilde{\nu}}{\partial x_i} \right\} \quad (7)$$

Additional definitions of the functions and constants are given by Spalart and Allmaras [32]. The trip terms f_{t1} and f_{t2} are turned off and the “trip-less” initial condition [33, 34] for ν_t and $\tilde{\nu}$, which was successfully used in earlier work [31], is adopted in this study.

3.2. Oscillator Equation of Motion

Mathematically, the mass-spring-damper system of the single PTC-cylinder system can be modeled by a second order linear differential equation as

$$m\ddot{y} + C_{system}\dot{y} + K_{virtual}y = F_{Y,total}(t) \quad (8)$$

where $F_{Y,total}$ is the fluid force acting on the cylinder in the y-direction, m is the total inertial mass of the V_{CK} system

$$m = m_{osc} + m_{eff} \quad (9)$$

and C_{system} is the system damping coefficient defined as

$$C_{system} = C_{structure} + C_{harness} \quad (10)$$

where $C_{structure}$ is the damping coefficient due to losses in the transmission system, $C_{harness}$ is the damping coefficient added through the generator internal resistance to convert mechanical energy to electrical energy.

3.3. Numerical Schemes

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 used for solving the momentum and continuity equations together in a segregated way. The equation of motion for the circular cylinder is solved using a second-order mixed implicit and explicit time integration scheme.

3.4. Computational Domain

The computational domain is $50D \times 9D$ for the single PTC-cylinder system. As shown in Figure 3, the entire domain includes five boundaries: inflow, outflow, top, bottom, and the PTC-cylinder. The initial position, which is also the equilibrium position of the PTC-cylinder, is located at the center of the domain. The distance between the inlet boundary and the center of the cylinder, l_{up} , is set at $25D$. A case with larger size of l_{down} has also been used to make comparison. The results from the larger grid do not generate better results. Thus, the downstream length of the domain, l_{down} , is also set at $25D$. This is to ensure that the results of the numerical model are accurate and that the conditions at the flow outlet are close to the assumed conditions.

At the inlet boundary, an inflow with uniform and constant velocity is considered for the fluid field. For the outflow boundary, a zero gradient condition is specified for velocity as well as $\tilde{v}$. The specification of the outflow condition is justified because the distance l_{down} is sufficiently long for this boundary condition to have negligible influence on the near-wake flow. The bottom condition is defined as a wall boundary to match the experimental condition. In the experiments, the safety-stops are placed on both sides of the mean position with a distance of 2.8 times the

diameter. If the cylinder reaches the maximum displacement ($2.8D$) in the experiment, there is a distance between the free surface and the cylinder, which is $1.7D$. Therefore, the effect of the free surface on the FIM was not considered in the numerical study. Thus, the top is simplified by modeling it as a wall. A moving wall boundary condition is applied for the cylinder when the cylinder is in FIM. In addition, the “trip-less” initial condition for the eddy viscosity is considered: a zero value is applied for the upstream region corresponding to the domain of l_{up} and a nonzero value is used for the downstream region of l_{down} . The PTC strip is composed of paper and sand grits, as shown in Figure 2(b). In the computational grid for the cylinder with PTC, the paper component of PTC is modeled as part of the cylinder surface, and treated as wall boundary. The rough sand grits effects are modeled by the standard wall function [20, 35-36].

3.5. Grid Generation

Two-dimensional, structured, computational grids were generated for all cases. The subdomain in the vicinity of the 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. 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. In the present study, y^+ is not a constant during the calculation for each case. When calculation for one case was finished, the values of y^+ were checked to ensure all values were in the range of 30-70. In order to determine the overall grid resolution to achieve a convergent and accurate solution in reasonable computational-time, a grid sensitivity study was conducted using three different grid densities for the stationary

PTC-cylinder. The grid parameters and selected results are listed in Table 4, where C_d is the time-average value of the drag coefficient, C_l is the average value of the peaks of the lift coefficient, and S_t is the Strouhal number. As shown in Table 4, the three grids produce similar results. Thus, the medium grid resolution (total 60,252 cells) for the PTC-cylinder was selected in the present work. A close-up of the medium grid is shown in Figure 4.

In the present work, the simulated Reynolds number range for which experiments were conducted in the MRELab is $30,000 < Re < 110,000$, which falls in the high-lift TrSL3 regime. Galloping was observed and the maximum amplitude reached in the experiments was $2.61D$. In the CFD simulations, large mesh deformations occur with the cylinder undergoing galloping. In order to minimize the mesh deformation, a dynamic mesh technique of topological change is used in the present study. The simple schematic of the topological mesh changes is shown in Figure 5. The moving layer in Figure 5 is located at the top of the $2D \times 2D$ square subdomain, and General Grid Interfaces (GGI) [37] are employed to connect the dynamic mesh region with the upstream/downstream domain. When the cylinder is in FIM, the $2D \times 2D$ square is moving up and down with the cylinder. The distance, h , which is between the moving layer and layer n , changes with the oscillation of the cylinder as well. When the moving layer moves up, h is compressed and shortened. Once $h < h_{minimum}$, the layer n will collapse. On the contrary, the new layer $n-1$ will be generated with the moving-down layer when $h > h_{maximum}$. Here, the $h_{minimum}$ and $h_{maximum}$ are set based on the overall quality of the computational grid. They are closely related to the minimum

grid space of the mesh. In the present work, $h_{minimum}=0.0028\text{m}$ and $h_{maximum}=0.0032\text{m}$. Thus, there is little deformation in the mesh when the cylinder undergoes large amplitude FIM.

4. MATHEMATICAL MODEL FOR POWER

In this section the simple mathematical model of the oscillator in Eq. (8) is used to calculate the harnessed and dissipated power of the VIVACE converter. In the experiments, the harnessed power in the V_{CK} system is dissipated in a resistor box (heat bank) and it is assumed as available electrical power in real life application of VIVACE. The power converted from the flow by the oscillating cylinder to mechanical in the oscillator during a cycle of period T_{osc} is given by the following equation [9]

$$P_{VIVACE} = \frac{1}{T_{osc}} \int_0^{T_{osc}} F_{Y,total} \dot{y} dt \quad (11)$$

Based on the information provided in section 3.2, the single PTC-cylinder oscillator is assumed as a mass-spring-damper system. The motion Eq. (8) can be rewritten as

$$F_{Y,total}(t) = m\ddot{y} + C_{system}\dot{y} + K_{virtual}y \quad (12)$$

Plugging Eq. (12) into Eq. (11), results in the following expression for the power in the VIVACE converter:

$$P_{VIVACE} = \frac{1}{T_{osc}} \int_0^{T_{osc}} [m\ddot{y} + C_{system}\dot{y} + K_{virtual}y] \dot{y} dt \quad (13)$$

In order to simplify Eq. (13) and to obtain the mathematical model of the harnessed and dissipated power of the VIVACE converter, the VIV and galloping responses of the rigid cylinder are both assumed to be approximatedly sinusoidal. This is a fairly accurate

approximation with the exception of the desynchronization VIV branch. The latter may be overtaken by the galloping range depending on the initiation velocity of galloping. Therefore, the displacement of the cylinder can be expressed as

$$y(t) = A \sin(\omega_{osc} t) \quad (14)$$

where A is the amplitude, ω_{osc} is the angular frequency of the oscillating cylinder and

$$\omega_{osc} = 2\pi f_{osc} \quad (15)$$

From Eq. (14), the velocity of the cylinder in the y -direction is

$$\dot{y}(t) = A \omega_{osc} \cos(\omega_{osc} t) \quad (16)$$

Based on the sinusoidal assumption of Eq. (14), the only nonzero term in Eq. (13) on the right hand side is the velocity term. Thus,

$$P_{VIVACE} = \frac{1}{T_{osc}} \int_0^{T_{osc}} C_{system} \dot{y}^2 dt \quad (17)$$

Inserting Eq. (16) into Eq. (17), the final expression for the power in the VIVACE oscillator can be achieved as

$$P_{VIVACE} = 8\pi^3 m \zeta_{system} f_{n,water} A^2 f_{osc}^2 \quad (18)$$

where the damping ratio ζ_{system} for the VCK VIVACE system is defined as

$$\zeta_{system} = \frac{C_{system}}{2\sqrt{mK_{virtual}}} \quad (19)$$

Based on the definition of the total damping coefficient in Eq. (10), the harnessed power and the dissipated power can be obtained as

$$P_{harness} = \frac{C_{harness}}{C_{system}} P_{VIVACE} \quad (20)$$

$$P_{dissipated} = \frac{C_{dissipated}}{C_{system}} P_{VIVACE} \quad (21)$$

Power Ratio: From the Bernoulli Equation, the kinetic pressure head in the fluid can be expressed as

$$p_T = \frac{1}{2} \rho U^2 \quad (22)$$

The projected area of the cylinder in the x -direction is DL . Thus, the fluid force acting on the cylinder in the same direction x is

$$F_x = \frac{1}{2} \rho U^2 DL \quad (23)$$

Then, the power in the fluid can be calculated as the product of the acting force times the velocity in the x -direction, that is

$$P_{fluid} = F_x U = \frac{1}{2} \rho U^3 DL \quad (24)$$

Thus, the ratio of harnessed power to the power available in the fluid is

$$\eta_{VIVACE} = \frac{P_{harness}}{P_{fluid}} \quad (25)$$

Eq. (25) provides a reference for evaluating the efficiency of VIVACE. It should be noted that other expressions may be used for the reference power in the fluid such as the power through the total area scanned by the cylinder. That would be $0.5\rho U^3(2A+D)L$ which would result in a considerably smaller efficiency. There are arguments in support of both expressions.

5. RESULTS AND DISCUSSION

A series of simulations of FIM of a single cylinder with selectively distributed surface roughness is conducted. The Reynolds number range is $30,000 < Re < 110,000$, and the corresponding reduced velocity range is $2.75 < U_{water}^* < 10.09$. In the experimental tests, the virtual spring stiffness $K_{virtual}$ is 1,600N/m and the damping ratio for energy harnessing ζ_{harn} is 0.04. In spite of solving 2D-URANS equations using CFD, this is a challenging task at high

Reynolds number. Specifically, the separation point is not predicted properly for smooth cylinders for $Re > 10,000$, thus, limiting agreement between CFD and experiments in FIM to $Re < 10,000$ [38]. For $Re > 10,000$, the separation points are predicted to be fixed at 90° by CFD calculation. Experimental data show, however, that the separation points and the forward stagnation point oscillate due to vortex shedding alone by $\pm(5^\circ-10^\circ)$ even for a stationary cylinder. Therefore, it is hard to use 2-D URANS to simulate properly this unique feature of cylinder flows when $Re > 10,000$. In this paper, however, the PTC applied to modify the surface geometry of the cylinder sets the separation point both in the simulations and in the experiments. With proper modeling of PTC, the 2D-URANS simulations exhibit several of the salient features of the flow resulting in an excellent agreement with experiments [31]. Thus, the code developed for FIM of a single cylinder with PTC can be used with confidence to predict certain flow properties, which are difficult to be measured experimentally at such high Reynolds numbers.

The good agreement between simulations and experiments is evident in the results of the present study. Within the test range of experiments and simulations, the CFD study correctly predicts five branches for the oscillation of the single PTC-cylinder system: no-FIM branch, the initial branch of VIV, the upper branch of VIV, transition from VIV to galloping, and galloping. In this section, the results of the amplitude and frequency response for the single PTC-cylinder are provided first. The vortex structures at different Reynolds number and the harnessed energy calculation results are discussed as well.

5.1. Amplitude Ratio

The amplitude ratio A_{peaks}/D graphs obtained by the numerical study and experimental measurements are compared in Figure 6. The amplitude, A_{peaks} , of the PTC-cylinder is calculated by averaging the absolute values of the 60 highest negative and positive peaks. In the present simulation, the bottom of the channel is set as a wall boundary to match the condition in the experimental test. However, in earlier simulations by Wu et al. [31], the effect of the bottom wall on FIM was ignored. The results reported by Wu et al. [31] are plotted in Figure 6 as well.

Compared with the experimental data, the amplitude ratio plot from the present 2D-URANS results follows the same trend. Within the test Reynolds number range, five regions are clearly observed in the amplitude ratio curve. It should be noted that back-to-back VIV and galloping is achieved. This is important from the point of view of hydrokinetic energy conversion as the response amplitude operator (RAO) becomes very broad-banded contrary to simple linear oscillators which have high amplitude near resonance only. The maximum displacement of the oscillating cylinder reached in galloping is $3.5D$.

The observed branches are:

- (a) $Re < 30,000$: No FIM takes place in this range experimentally or numerically.
- (b) $30,000 \leq Re < 60,000$: This is the VIV initial branch. The 2D-URANS results show that the amplitude ratio increases from 0.58 at $Re=30,000$ ($U_{water}^*=2.75$) to 1.47 at $Re=60,000$ ($U_{water}^*=5.50$). But in the experiments, the oscillation of cylinder can only be observed when $Re \geq 50,000$ ($U_{water}^* \geq 4.59$), the initial branch is initiated at a higher velocity ($Re=50,000$) than that in the numerical simulation ($Re=30,000$). This is due to the difference between the damping models used in the numerical simulation and the experimental apparatus.

Specifically, the mathematical damping model is assumed as classical linear viscous damping model in CFD study, while the physical model in the experiments is a much more complex viscous model in spite of the implementation of the V_{CK} system as explained in Section 2 [8].

(c) $60,000 \leq Re < 90,000$: This is the upper branch in VIV. The 2D-URANS results over-predict the response compared with the experimental data in the amplitude curve for $60,000 \leq Re < 90,000$ ($5.50 \leq U_{water}^* < 8.26$). We observe however that the trend is predicted very accurately including the starting and ending points of the upper branch. The oscillation of the cylinder is stable and the amplitude ratio measured experimentally remains in the vicinity of 1.0. For the simulation results, the amplitude of the cylinder varies with Re and stays around $1.40D$.

(d) $90,000 \leq Re < 100,000$: This is the transition region from the VIV upper branch to galloping. The amplitude of the PTC-cylinder increases more rapidly for $90,000 \leq Re < 100,000$ ($8.26 \leq U_{water}^* < 9.17$). This indicates the transition from VIV to galloping is successfully initiated at $U_{water}^* \approx 8.26$. That is in very good agreement with experiments with the PTC-cylinder. Without the PTC but some other form of geometric asymmetry, galloping is initiated typically at reduced velocity $U_{water}^* \approx 20$.

(e) $Re \geq 100,000$: This is the galloping region. Galloping is known to be a high amplitude and low frequency FIM. The galloping generation mechanism is based on the lift instability due to asymmetry [22]. Non-circular cross-section could be subjected to high amplitude

galloping oscillations. In the present study, the circular cylinder is altered by attaching roughness strips, which are the cause of geometric asymmetry inducing galloping. The maximum amplitude of the PTC cylinder reaches $3.5D$ at $Re=110,000(U_{water}^* \approx 10.09)$ in the CFD results while $2.8D$ is achieved in the experimental test. In the experiments conducted in the LTFSW channel in the MRELab, the cylinder would hit the safety stops limiting its travel in galloping, which lead to the smaller amplitude in the tests than in the simulation. In the numerical simulations by Wu et al. [31], the amplitude increased further at higher speeds which were not considered in this work.

It is important to note that CFD predicts very well the shape of the VIV and galloping response as well as the start and end points in this range of Reynolds number, which falls in the TrSL3 regime [19] which is very different from the corresponding shape in the TrSL1 regime and below for $Re < 10,000$ [15, 16]. All branches are present in both ranges, but for the shape, the upper branch is much wider than the upper branch for $Re < 10,000$. The upper branch is flat and relatively narrow for higher mass ratio m^* compared to the lower branch. In TrSL3, the upper branch slopes up and overtakes the lower branch, which is very narrow [5]. In the CFD results in this paper, this is predicted well. The PTC triggers galloping at lower U_{water}^* leading to transition from VIV to galloping before the lower branch is reached. Thus, the RAO of the VIVACE converter does not exhibit any low values between VIV and galloping.

5.2. Frequency Ratio

In Figure 7, the frequency ratio $f_{osc}/f_{n,water}$ of the PTC-cylinder is plotted versus Reynolds number Re , reduced velocity U_{water}^* , and flow velocity U along with the experimental results. Using Fast Fourier Transform (FFT), the oscillation frequency f_{osc} is calculated as the dominant frequency of the displacement of the cylinder over the recorded period. The displacement history and FFT spectral analysis for different Reynolds numbers are shown in Figure 8.

The following observations can be made based on Figure 7.

- (a) The frequency ratio curve of the present numerical results shows a clear increasing trend as the flow velocity increases in the range $30,000 \leq Re < 60,000$ which is the initial branch of VIV. The major harmonic frequency in the numerical simulations differs from the experimental tests at the beginning of this range. This is attributed to the difference in damping models, which are explained in Subsection 5.1 resulting in earlier initiation of VIV in the CFD computations compared to experiments.
- (b) At $Re=60,000$ ($U_{water}^*=5.50$), the VIV upper branch begins. The frequency drops off at the first part of the upper branch in the CFD results, that is for $60,000 \leq Re < 70,000$ ($5.50 \leq U_{water}^* < 6.42$). The frequency ratio reaches the maximum value at $Re=80,000$ ($U_{water}^*=7.34$) for the numerical results as well as the experimental data. As Re approaches 90,000, the frequency sudden drop indicates the beginning of the transition to galloping. The frequency drop is accompanied by a rapid jump in the amplitude of the cylinder, due to progressing from the VIV upper branch to the transition region from VIV to galloping.

(c) In the galloping branch for $Re \geq 100,000$ ($U_{water}^* > 9.17$), one of the important features is the frequency reduction that is well demonstrated by the drop in the frequency of the PTC-cylinder. In addition, different branches are clearly observed in the earlier work by Wu et al. [31] as well, such as VIV initial branch, VIV upper branch, galloping, and the transition between the branches. In summary, the present simulation results of the oscillation frequency for the single PTC-cylinder compare well with experimental data.

5.3. Vortex Patterns

There are several forms of the near-wake vortex-pattern related to flow induced motions. Williamson and co-researchers [16, 39-41] summarized experimental results and introduced the vortex pattern terminology, e.g. 2S and 2P. Here 2S and 2P indicate Two Single and Two Pairs of vortices shed per vibration cycle. The 2D-URANS results for the single PTC-cylinder match well with the experiments. In the earlier work by Wu et al. [31], the simulation results showed that transition between branches is accompanied by vortex pattern change and non-uniform amplitude response. On the other hand, the vortex pattern is stable and the amplitude is uniform when the oscillating cylinder is in one branch. Therefore, the amplitude and frequency response of the PTC-cylinder are closely related to the near-wake structure.

This subsection presents the vorticity field observations in the results of 2D-URANS simulations. In order to compare the vortex patterns at different Reynolds numbers, the simulated real time is non-dimensionalized by the corresponding system oscillating period T_{osc} . The

vortices shed in one cycle are tagged as V1~Vn (n=2, 3, 4, ...) indicating the order of their genesis.

The following observations can be made based on the wake patterns shown in Figures 9-16.

(a) $Re=30,000$ ($U_{water}^*=2.75$): The oscillating PTC-cylinder is in the initial branch of VIV. The 2S mode of vortex shedding can be clearly observed at $Re=30,000$ as shown in Figure 9. This mode is the classical von Kármán Street. Two single vortices, V1 and V2, with opposite direction of rotation, shed from the oscillating PTC-cylinder during one oscillation cycle.

(b) $Re=40,000$ ($U_{water}^*=3.67$): Figure 10 shows the vortex patterns when $Re=40,000$. Two dominant vortices, V1 and V2, are observed shedding from the cylinder in a cycle. However, the vortex pattern is not 2S at $Re=40,000$. Comparing Figure 10 with the 2S mode in Figure 9, there is a significant difference between the vortex patterns of $Re=40,000$ and $Re=30,000$. An additional vortex can be captured near each dominant vortex. This vortex pattern is called Quasi-2P. It means that two pairs of vortices shed per cycle, but in each pair, one vortex is much weaker than the other [31]. Actually, the Quasi-2P mode can be observed during the mode transition from 2S to 2P.

(c) $Re=50,000$ ($U_{water}^*=4.59$): The transition from the initial branch to the upper branch of VIV occurs around $Re=50,000$. As shown in Figure 11, a pair (P) of vortices, V1 and V2, is shed during the downward travel ($t/T_{osc}=0.000\sim0.250$) of the PTC-cylinder. When the cylinder moves up at in the time span $t/T_{osc}=0.374\sim0.748$, vortex V3 sheds from the lower side of the cylinder. Vortex V4, which is much weaker than V3, sheds following vortex V3. But V4

disappears as soon with the fluid flowing to downstream. For the upward travel, the vortex pattern is Quasi-Pair (QP) mode formed by vortex V3 and V4. Thus, in one oscillation cycle at $Re=50,000$, the vortex pattern is the P+QP mode.

(d) $Re=60,000$ ($U_{water}^*=5.50$): The 2P vortex pattern is observed at $Re=60,000$ which is the beginning of the VIV upper branch. One pair of vortices is shed on each side of the cylinder. Vortices V1 and V2 shed during $t/T_{osc}=0.000\sim0.362$ as presented in Figure 12. In the time span $t/T_{osc}=0.483\sim0.845$, another pair consisting of vortices V3 and V4 is shed resulting in the 2P pattern.

(e) $Re=70,000$ ($U_{water}^*=6.42$): The PTC-cylinder is still in the VIV upper branch at $Re=70000$. The vortex structure is 2P which is the same as $Re=60,000$, as shown in Figure 13. Four vortices are shed from the cylinder in one oscillation cycle.

(f) $Re=80,000$ ($U_{water}^*=7.34$): The number of shedding vortices increases with the increase in the flow velocity. The vortex pattern observed is 2P+2S when $Re=80,000$ as presented in Figure 14. The vortex pair, V1 and V2, sheds from the upper side of the cylinder during the time interval $t/T_{osc}=0.000\sim0.366$. A single vortex V3 can be clearly observed as the cylinder moves farther down. Thus, the P+S mode is produced during the downward travel of the cylinder. On the other side of the cylinder, one pair (V4 and V5) sheds first and a single (V6) vortex follows, which can be seen in the interval $t/T_{osc}=0.488\sim0.856$ in Figure 14.

(g) $Re=90,000$ ($U_{water}^*=8.26$): For $Re=90,000$, seven vortices are observed in one oscillation cycle as shown in Figure 15. The vortex pattern is P+2S+P+S. In this transitional regime

between the VIV upper branch and galloping, the vortex pattern shows a transitional behavior. As the cylinder moves downward from the maximum displacement point ($t/T_{osc}=0.000\sim0.472$), a pair (P) of vortices (V1 and V2) is shed first, followed closely by a single vortex V3 (S) and then a second single V4 (S). On the other hand, the behavior is a little different during the upward travel of the cylinder; one pair of vortices (V5 and V6) is shed first and a single one (V7) follows. Thus, for the case of $Re=90,000$, the ordering of the vortex shedding is P+S+S+P+S in one cycle, which has one more vortex than the P+S+P+S mode observed in the case of $Re=80,000$.

(h) $Re=100,000(U_{water}^*=9.17)$: Figure 16 shows the vortex pattern for the PTC-cylinder in fully developed galloping at $Re=100,000$. The vortex structure is not stable and changes between $2P+3S$ and $P+S+T+S$, in the sense that the number and the format of the vortices shed during each cycle are not exactly the same throughout the time history. This makes it difficult to generalize the vortex pattern over all. As shown in Figure 16 for one cycle at $Re=100,000$, 6 vortices are observed when the cylinder moves upward from $t/T_{osc}=0.000$ to $t/T_{osc}=0.494$. On the other hand, the velocity of the cylinder is faster and only two vortices shed during the downward travel.

For all cases, the number of the vortices shed in one cycle increases as the velocity increases. The vortex pattern changes from 2S to $2P+2S$ mode for $30,000\leq Re\leq80,000$ in the VIV initial branch and upper branch. When $Re\geq90,000$, the transition from VIV to galloping is initiated and

the galloping is fully developed at $Re = 100,000$, more than 7 vortices are observed for these cases.

5.4. FIM Driving Mechanisms

A very important flow feature of FIM can be observed in Figures 9-16. The hydrodynamic mechanism that drives the cylinder in VIV is completely different from the mechanism that drives the cylinder in galloping. These are discussed next in relation to the observed vortex patterns.

(a) VIV (Figures 9-14): The VIV mechanism whether explained as “lock-in” of vortex shedding on the frequency of oscillation of the cylinder [16] or as “resonance of variable natural frequency” due to added mass variation [17], it is due to the synchronization between the oscillatory pressure from the shed vortices and the cylinder motion. Vortices push the cylinder away from the side they shed. In VIV, regardless of the branch, there is synchronization between the cylinder oscillation and the wake frequency. That is, whether the pattern is 2S (two vortices), 2P (four vortices), or 2P+2S (six vortices) the wake frequency matches the cylinder oscillation resulting in “lock-in” or “variable natural frequency resonance”.

(b) Galloping (Figure 16): The mechanism driving the cylinder in galloping is an instability of the lift due to asymmetry caused by flow or geometry. The shear layers of course follow the motion of the cylinder. The amplitude is high and many vortices are shed in each half cycle. It can be noticed in Figure 16 that some of these vortices push the cylinder in the wrong

direction. That is the vortex shedding pattern and the resulting wake pattern frequency do not synchronize with the cylinder motion any longer and thus are ineffective - if not obstructive - in driving the cylinder in oscillatory motion.

(c) Transition from VIV to galloping (Figure 15): In this region, the two driving mechanisms co-exist and contribute to the cylinder oscillation. As velocity increases and the galloping instability kicks in on top of the VIV mechanism, the amplitude increases resulting in more vortices shedding and eventually desynchronization of the wake frequency.

5.5. Energy Harvesting

The power that can be harnessed by a single, PTC-cylinder VIVACE converter is the power extracted by the generator, which is equal to the power that VIVACE extracts from the fluid minus the power dissipated by structural, transmission, and internal generator losses. Based on the numerical results of amplitude and frequency responses, the harnessed power ($P_{harness}$) by the single PTC-cylinder VIVACE system is calculated using the mathematical model presented in Section 4. The harnessed power versus Reynolds number (Re), reduced velocity (U_{water}^*), and flow speed (U) is depicted in Figure 17. The experimental results are also presented for comparison.

As shown in Figure 17, power can be harnessed over a broad range of velocity from 0.38m/s to 1.52m/s. The upper limit is only due to limitations of the LTFSW Channel. In the initial branch of VIV, the harnessed power of the single PTC-cylinder VIVACE is very small and only 0.14W is achieved when $Re=30,000$ ($U_{water}^*=2.75$). This is due not only to the fact that the

available power in the fluid is small at such low speeds but also due to the low amplitude and frequency in the initial branch. With the increasing in flow velocity, the power in the fluid increases by the cube of the velocity - per Eq. (24) - and at the same time the VIV upper branch begins, resulting in a steep increase in energy conversion. This can be observed in the CFD calculations and the experimental measurements in Figure 17. A second steep increase in converted power comes at the point of transition from VIV to galloping. In the present CFD results, it climbs from 9.67W ($Re=90,254$, $U_{water}^*=8.28$) to 20.96W ($Re=100,000$, $U_{water}^*=9.17$). In the experimental measurements converted power rises from 8.76W ($Re=90,254$, $U_{water}^*=8.28$) to 23.54W ($Re=101,536$, $U_{water}^*=9.31$). The power calculated by 2D-URANS simulation is close to that obtained by experiments. Even when the values differ a little the slop of the two graphs in Figure 17 are similar rendering the graphs to have the same trend and thus capturing the important variations in power conversion vs. flow speed. Further, Eq. (20) shows that the harnessed power of VIVACE is proportional to the linear viscous damping, thus making it easy to calculate the dissipated power due to frictional losses.

The power conversion efficiency based upon the numerical and experimental results are calculated using Eq. (25), and the results are shown in Figure 18. Two peaks of power conversion efficiency are observed in the numerical simulations and experiments. The two points are located at about $Re=60,000$ ($U_{water}^*=5.50$) and $Re=100,000$ ($U_{water}^*=9.17$), which correspond to the starting point of the VIV upper branch and galloping, respectively. When $Re \approx 60,000$ ($U_{water}^* \approx 5.50$), as the FIM of the PTC-cylinder transits from the VIV initial branch to the VIV

upper branch, the amplitude and frequency increase greatly. The energy conversion efficiency reaches 37% ($Re=60,000$, $U_{water}^*=5.50$) in the CFD results and 26% ($Re=59229$, $U_{water}^*=5.43$) in experiments. When Re is increased to 100,000 ($U_{water}^*=9.17$), galloping occurs and the amplitude of the cylinder increases sharply, but the oscillation frequency decreases. The energy conversion efficiency of the single PTC-cylinder reaches 25% at $Re \approx 100,000$ ($U_{water}^* \approx 9.17$). The corresponding experimental measurement shows efficiency $\eta_{VIVACE}=28\%$ at $Re=95,895$ ($U_{water}^*=8.80$). The discrepancy appears for $Re < 60,000$, because there is difference between the mathematical damping model in numerical simulations and the actual physical damping model in the experimental apparatus [7, 8]. The difference in damping model is more pronounced at low oscillator speed (dy/dt). As shown in the mathematical model in Section 4, the power and efficiency are closely related to the amplitude and frequency of the cylinder. Combining Figure 18 with Figures 6 and 7, the initial branch is initiated numerically earlier than in the experiments. Thus, the simulation results including amplitude, frequency, and efficiency present higher values than experimental data for $Re < 60,000$. However, the variation tendency can be predicted properly by CFD calculations. In addition, Figure 17 shows that the power in the galloping branch is higher than that of VIV. However, the power conversion efficiency can reach higher value in VIV upper branch. It should be noted that VIV can be initiated at a much lower velocity than galloping. Therefore, both the ranges of VIV and galloping are appropriate for operation of the VIVACE converter with PTC.

6. CONCLUSIONS

The flow induced motion of a single, rigid, circular cylinder with PTC is investigated using two-dimensional unsteady Reynolds-Averaged Navier-Stokes equations with the Spalart-Allmaras turbulence model at high Reynolds numbers ($30,000 < Re < 110,000$, $2.75 \leq U_{water}^* \leq 11.92$). The simulations are verified by experimental measurements conducted in the LTFSW Channel of the MRELab of the University of Michigan. The results including amplitude response, frequency response, vortex structures and wake patterns, and energy conversion of the single-cylinder VIVACE as function of Reynolds number are analysed. PTC proved to be the key factor in achieving agreement between numerical and experimental results. The following conclusions can be drawn.

(1) The General Grid Interface with Topological Mesh Changes is an effective method for handling high-amplitude FIM response.

(2) The amplitude ratio of 2D-URANS results follows the same trend with the experimental data. Within the test Reynolds number range, five regions were clearly observed, including the no-FIM range, the VIV initial branch, the VIV upper branch, transition from VIV to galloping, and galloping. The maximum amplitude achieved in the simulations is $3.5D$ at $Re=110,000$ ($U_{water}^*=10.09$) when galloping occurs. This value could not be reached experimentally because the oscillator hit the safety stops in the LTFSW Channel.

(3) The computational results for the frequency of oscillation are in excellent agreement with the experimental data. They also show the initial branch, upper branch, transition from VIV to

633 galloping, and galloping. With the application of PTC, cylinder FIM was enhanced and
634 back-to-back VIV and galloping was achieved.

635 (4) The number of vortices shed in one oscillation cycle increases with the inflow velocity in
636 VIV. The vortex pattern changes from 2S to 2P to 2P+2S mode for $30,000 \leq Re \leq 80,000$ in the
637 VIV initial branch and upper branch.

638 (5) When $Re=90,000$, transition from VIV to galloping is initiated and the P+2S+P+S pattern is
639 developed. In transition two driving mechanisms coexist: the wake pattern frequency lock-in and
640 the galloping instability.

641 (6) In fully developed galloping, the galloping instability driving mechanism takes over. The
642 vortex pattern is not in synchronization with the cylinder oscillations. More than 7 vortices are
643 observed when galloping is fully developed at $Re = 100,000$ and the vortex pattern is not stable
644 and changes between 2P+3S and P+S+T+S.

645 (7) Power can be harnessed by the VIVACE converter with one PTC-cylinder over the whole
646 Reynolds number range from 30,000 to 110,000. Harnessing will increase past that range since
647 galloping does not cease until structural failure. Experimental data however are not available
648 since tests reached the limits of the LTFSW Channel on the MRELab.

649 (8) The peak of energy conversion efficiency reaches 37% when $Re=60,000$ ($U_{water}^* = 5.50$) in the
650 present numerical simulations. Efficiency of 28% is reached at $Re=95,895$ ($U_{water}^* = 8.80$) in the
651 experiments.

(9) The VIVACE system approaches optimal energy conversion efficiency at the beginning of the VIV upper branch and at the beginning of galloping.

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755

756 **FIGURE CAPTIONS**

757 **Figure 1.** Sketch of the physical model of a single-cylinder VIVACE converter

758 **Figure 2.** Lab module: (a) VIVACE converter in LTFSW Channel; (b) configuration of PTC[6]

759 **Figure 3.** Computational domain

760 **Figure 4.** Close-up of the grid in the vicinity of a cylinder with PTC

761 **Figure 5.** Topological mesh change

762 **Figure 6.** Amplitude ratio comparison of the cylinder with PTC

763 **Figure 7.** Frequency ratio comparison of the cylinder with PTC

764 **Figure 8.** Displacement history and frequency spectrum for different Re

765 **Figure 9.** 2S vortex pattern of the PTC-cylinder for $Re=30,000$

766 **Figure 10.** Quasi-2P vortex pattern of the PTC-cylinder for $Re=40,000$

767 **Figure 11.** P+QP vortex pattern of the PTC-cylinder for $Re=50,000$

768 **Figure 12.** 2P vortex pattern of the PTC-cylinder for $Re=60,000$

769 **Figure 13.** 2P vortex pattern of the PTC-cylinder for $Re=70,000$

770 **Figure 14.** 2P+2S vortex pattern of the PTC-cylinder for $Re=80,000$

771 **Figure 15.** P+S+S+P+S vortex pattern of the PTC-cylinder for $Re=90,000$

772 **Figure 16.** 2P+3S vortex pattern of the PTC-cylinder for $Re=100,000$

773 **Figure 17.** Harnessed power of VIVACE with PTC

774 **Figure 18.** Power conversion efficiency of single PTC-cylinder VIVACE converter

Table 1. Nomenclature

A	Amplitude
A_{peaks}	Mean amplitude of the peaks
C_d	Drag coefficient
$C_{harness}$	Added damping to harness energy
C_l	Lift coefficient
$C_{structure}$	Structural damping
C_{system}	System damping
D	Diameter of the cylinder
F_x	Fluid force in the x -direction
$F_{Y,total}$	Fluid force in the y -direction
$K_{virtual}$	Spring stiffness
L	Length of the cylinder
P	Thickness of sand paper
$P_{dissipated}$	Dissipated power
P_{fluid}	Power in the fluid
$P_{harness}$	Harnessed power
P_{VIVACE}	Power of VIVACE converter
S_{ij}	Mean strain-rate tensor
S_t	Strouhal number
T	Total thickness of strip
T_{osc}	Period of oscillation
U	Mean flow velocity
U_i	Velocity components
$U_{water}^* = U/(f_{n,water}D)$	Reduced velocity
$f_{n,water} = \frac{1}{2\pi} \sqrt{\frac{K_{virtual}}{(m + m_a)}}$	Natural frequency in water
f_{osc}	Oscillating frequency
k	Average grit height
m	Inertial mass of oscillating system
m_a	Added mass
m_d	displaced mass
m_{eff}	Effective mass of motor-pulley-belt
m_{osc}	Oscillating system mass
$m^* = m_{osc}/m_d$	Mass ratio
p_T	Kinetic pressure head
t	Time
$y(t)$	Displacement of cylinder
α_{PTC}	PTC placement angle
ζ_{system}	Damping ratio for VIVACE system
η_{VIVACE}	Efficiency of VIVACE
θ	Angular coverage of strip

μ	Dynamic viscosity of water
μ_t	Turbulence eddy viscosity
ν	Kinematic viscosity of water
$\tilde{\nu}$	Intermediate working variable
ρ	Water density
τ_{ij}	Reynolds-stress tensor
ω_{osc}	Angular frequency

Table 2. Physical model parameters

Diameter of the circular cylinder	D [m]	0.0889
Length of cylinder	L [m]	0.9144
Oscillating system mass	m_{osc} [kg]	10.75
Effective mass	m_{eff} [kg]	2.05
Displaced mass	m_d [kg]	5.6707
Mass ratio	m^*	1.896
Spring stiffness	$K_{virtual}$ [N/m]	1600
Structure damping	$C_{structure}$ [Ns/m]	3.2
Damping ratio for energy harnessing	$\zeta_{harness}$	0.04
Natural freq. in water	$f_{n,water}$ [Hz]	1.48
Dynamic viscosity of water	μ [Ns/m ²]	0.0011379
Kinematic viscosity of water	ν [m ² /s]	1.139E-06
Water density	ρ [kg/m ³]	999.1026

Table 3. PTC parameters (P60 sand paper)

Strip placement angle	α_{PTC} [degree]	20
Angular coverage of strip	θ [degree]	16
Sand paper plus tape thickness	P [mm]	0.587
Average grit height	k [mm]	0.26
Total thickness of strip	$T=P+k$ [mm]	0.847

Table 4

Table 4. Grid resolution study ($Re=30,000$)

Grid(Central square: circumferential×radial)	C_d	C_l	S_t
Coarse(180×40)	1.029	0.287	0.152
Medium(240×70)	1.039	0.299	0.152
Fine(360×100)	1.038	0.298	0.150

Figure 1

Figure 2

(a)

(b)

Figure 3

Figure 4

Figure 5

Figure 6

Figure 7

Figure 8

Figure 9

Figure 10

Figure 11

Figure 12

Figure 13

Figure 14

Figure 15

Figure 16

Figure 17

Figure 18