**Effect of Mass-Ratio, Damping, and Stiffness on Optimal Hydrokinetic Energy Conversion
of a Single, Rough Cylinder in Flow Induced Motions**

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ABSTRACT

Flow Induced Motions (FIMs) of a single, rigid, circular cylinder with end-springs are investigated for Reynolds number $30,000 \leq Re \leq 120,000$ with mass ratio, damping, and stiffness as parameters. Selective roughness is applied to enhance FIM and increase the hydrokinetic energy captured by the VIVACE (Vortex Induced Vibration for Aquatic Clean Energy) Converter at higher Reynolds numbers. The second generation of virtual spring-damping system Vck, recently developed in the Marine Renewable Energy Laboratory (MRELab), enables embedded computer-controlled change of viscous-damping and spring-stiffness for fast and precise oscillator modeling. Experimental results for amplitude response, frequency response, energy harvesting, and efficiency are presented and discussed. All experiments were conducted in the Low Turbulence Free Surface Water (LTFSW) Channel of the MRELab of the University of Michigan. The main conclusions are: (1) The oscillator can harness energy from flows as slow as 0.3946 m/s with no upper limit. (2) Increasing the spring stiffness, shifts the VIV synchronization range to higher flow velocities, resulting in reduced gap between VIV and galloping, where the harnessed power drops. (3) In galloping, the harnessed power increases with the mass ratio. (4) Local optima in energy conversion efficiency appear at the beginning of the VIV upper branch and at the beginning of galloping. (5) Local optima in power appear at the end VIV upper branch and at the beginning of galloping.

KEY WORDS: hydrokinetic energy; virtual spring-damping; flow induced motions; vortex induced vibrations; galloping; VIVACE Converter

1. INTRODUCTION

Alternating Lift Technologies (ALT) provide an environmentally compatible way for harnessing marine hydrokinetic (MHK) energy even from slow flows (Kim et al. 2016). Flow Induced Motions (FIMs) present a source of challenge for diverse structures in steady flows such as heat exchangers, bridges, buildings, offshore structures, or power-transmission cables. The most common FIM phenomenon is Vortex-Induced Vibration (VIV), which was first observed by Leonardo da Vinci in 1504. The first mathematical formulation was done by Strouhal in 1878. Comprehensive reviews of VIV have been published by Bearman (1984, 2011), Sarpkaya (2004), and Williamson and Govardhan (2004). VIV occurs over a broad range of velocities, called synchronization range. It is due to vortex lock-in with the motion of the oscillating cylinder. The amplitude of VIV is self-limiting.

Galloping is another form of flow induced motion. It is an aero/hydro-elastic instability characterized by lower frequencies and larger amplitudes than VIV and is perpendicular to the flow (Blevins 2001, Alonso 2005). It is more vigorous and destructive than VIV, yet not as complex a phenomenon. Galloping occurs above a critical flow speed and does not depend on vortex formation. Amplitude is not self-limiting, and increases with velocity till structural failure.

Typically, as flow velocity increases, VIV appears first, followed by galloping. Figure 1 shows the various branches of FIM for a single cylinder with distributed surface roughness to induce galloping. Each branch represents a different sub-region of FIM within which the

63 wake-vortex mode or the underlying driving mechanism remains constant resulting in distinct
64 amplitude response from adjacent branches. The cylinder is rigid on elastic supports in a
65 cross-flow. As velocity increases, FIM is initiated as VIV due to vortex shedding, alternating
66 from the two sides of the cylinder, resulting in transverse movement of the cylinder. Vortex
67 shedding locks onto the cylinder oscillation over a broad range of synchronization starting with
68 the initial branch as shown in Figures 1 and 2. This response amplitude increases with the
69 velocity of the flow in the upper branch. It is followed by the lower branch with smaller
70 response, then the desynchronization zone, and then the end of the VIV synchronization range.
71 The various branch characteristics may differ significantly depending on whether the flow is in the
72 Transition in Shear Layer (TrSL) is flow regime TrSL2 or TrSL3 (Zdravkovitz, 1997).
73 Williamson and Govardhan (2004) conducted experiments for Re about 3,800 in TrSL2
74 ($1,000 < Re < 20,000$) and response amplitude reached one cylinder diameter. Bernitsas et al.
75 (2009) and Raghavan and Bernitsas (2010) conducted experiments for $16,000 < Re < 140,000$ in
76 the TrSL3 ($20,000 < Re < 300,000$) flow regime. They showed that: (a) the dependence of VIV on
77 Reynolds number is strong; (b) in the upper branch in TrSL3, the amplitude increases reaching
78 1.8 diameters; and (c) the upper branch overtakes almost completely the lower branch. Figure 1
79 shows all the VIV branches; the damping and stiffness values have been selected in such a way
80 that the full range of VIV synchronization is shown. That is, the amplitude of oscillation nearly
81 reaches zero before the onset of galloping.

At higher speeds, galloping may be reached depending on the geometric deviation of the cylinder from the circular cross-section (Blevins 1990, Chang et al. 2011). Galloping reaches the physical limits of the oscillator in the experimental facility or structural failure (Bernitsas, et al. 2010; Park, et al. 2013). VIV and galloping can be fully separated as shown in Figure 1, or they can overlap resulting in a VIV-to-galloping transition range as shown in Figure 2. In the latter case, a transition region appears between the two FIM types where both driving mechanisms – the VIV vortex shedding and the galloping instability – coexist.

Figure 1. FIM regions with small overlap between VIV and galloping.

Figure 2. FIM regions with transition between VIV and galloping.

VIV has a range of synchronization. For mass ratio $m^* = m_{osc}/m_d$ on the order of 1, the VIV synchronization range in terms of the reduced velocity U^* is approximately

$$\sim 5 < U^* = \frac{U}{f_{n,water} D} < \sim 10 \quad (1)$$

where m_{osc} is the oscillating mass, m_d is the displaced fluid mass

$$f_{n,water} = \frac{1}{2\pi} \sqrt{\frac{K}{m_{osc} + m_a}} \quad (2)$$

is the natural frequency in water, m_a is the added mass, and K is the total spring stiffness. That is, the end of the VIV synchronization range increases proportionally to the square root of the spring stiffness K .

On the other hand, galloping initiates at an absolute value of velocity $U=U_{critical}$ given by Blevins (1990) as

$$U_{critical} = \frac{2c_{total}}{\rho D \frac{\partial c_y}{\partial \alpha}} \quad (3)$$

where c_{total} is the linear viscous damping, D is the cylinder diameter, and $\frac{\partial c_y}{\partial \alpha}$ is the derivative of the lift coefficient with respect to the angle of attack. $U_{critical}$ is not affected by the spring constant K or the mass m . That is, initiation of galloping does not depend on K but only on the absolute flow velocity, system damping, and geometric particulars of the oscillator.

Comparing inequality (1) and equation (3), it is observed that the gap between VIV and galloping can be bridged by increasing K , decreasing m , decreasing c_{total} , or increasing D . Thus, an oscillator responding with high amplitude at a range of velocities starting at $U^* \simeq 5$ and having no upper end can be designed. That is, when it comes to energy capturing, the oscillators in the Converter provide a great advantage compared to linear resonators, which have high response within a very limited bandwidth in the vicinity of their natural frequency.

Typically, FIMs are suppressed in engineering applications because of their destructive nature. By enhancing and controlling FIM, the Marine Renewable Energy Laboratory (MRELab) at the University of Michigan developed and patented a converter (Bernitsas and Raghavan 2009, 2011, 2014; Bernitsas et al. 2008, 2009) to convert the kinetic energy of water flows into electricity. Relevant research on harvesting MHK energy using vortex-induced vibrations has been published in several countries: Nishi, 2013 (Japan); Zhu, 2012; Vinod, A. and Banerjee, A., 2014 (USA); Meliga et al, 2011 (Switzerland and France); Barrero-Gil, G. et al, 2012 (Spain); Konstantinidis, E., 2013 (Greece); Kang and Jia, 2013, (China); Narendran et al., 2016 (India) and more.

The research objective of the MRELab is to study FIMs of single or multiple cylinders on springs and find ways to enhance the cylinder response in order to optimize the power output of Converters over a broad range of velocities. Power envelopes as function of flow speed and type of FIM need to be generated. FIMs studied in the process include vortex-induced vibrations, galloping, and interaction effects between cylinders due to gap flows, shear layers, and vortices. Generating power envelopes in such a large parametric design space and for different FIM phenomena is equivalent to finding the efficiency of propellers with different number of blades and geometric properties. Further, in open-water propeller tests all blades are in equivalent position while in schools of cylinders the hydrodynamics of each cylinder changes depending on the cylinder relative location in the school.

The objective of this paper is to study systemically the effects of mass ratio, linear stiffness, and linear viscous damping on a single cylinder to attain higher power generation and higher efficiency in hydrokinetic energy conversion. Tests are conducted in the range of $30,000 < Re < 120,000$, 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 (Zdravkovitz, 1997). The physical model and running parameters are presented in Section 2. In Section 3, the mathematical model for the harnessed power is summarized. In Section 4, experimental results for amplitude response, frequency response, energy harvesting and efficiency are presented and discussed. Summary is presented and conclusions are drawn at the end.

2. PHYSICAL MODEL

2.1. Experimental Facility LTFSW Channel

All tests were conducted in the Low Turbulence Free Surface Water (LTFSW) Channel, in the MRELab of the University of Michigan (Fig. 3). The LTFSW Channel recirculates 10,000 gallons of fresh water at speed up to 1.4m/s using an impeller powered by a 20hp induction motor. The test section is 2.44m long and 1.0m wide, and is made of transparent plexi-glass, which enables visualization of the flow past cylinders using two 5W argon lasers and aluminum oxide particles of 100 μ m. For this study, the water depth at the test-section was set at 1.17m and the maximum flow speed was limited to 1.35m/s for safety against the galloping instability.

excitation comes from the fluid-structure interaction (FSI), and is applied to the cylinder through the forces imparted by the motion of the fluid. The hydrodynamics is not included in the control loop as explained in Section 2.3 below.

In order to enhance the FIM of the cylinder, which results in harnessing more horizontal hydrokinetic energy, distributed roughness in the form of roughness strips was introduced and extensively studied in the MRELab experimentally (Bernitsas and Raghavan 2011, Chang et al. 2010, Park et al. 2011, Kim et al. 2013) and numerically (Ding et al. 2013, 2015). The distributed roughness was termed Passive Turbulence Control (PTC) and this study resulted in a most valuable tool, the *PTC-to-FIM Map* (Park et al. 2011). As shown in Fig. 4, PTC consists of a pair of straight roughness strips attached symmetrically to the surface of the cylinder. According to the *PTC-to-FIM Map*, the location of the roughness strips determines the response FIM of the cylinder. The placement angle $\alpha_{PTC}=20^\circ$, is measured in degrees with respect to the forward stagnation point to the upstream edge of the roughness strip. Based on the results of previous studies (Chang et al. 2011, Park et al. 2013) on PTC, strips with commercial roughness designation P60 and width of 12.7mm, which covers 16 degrees on each side of the 88.9mm diameter cylinder were used in this study (Table 1).

Table 1. PTC parameters (P60)

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

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180 The total thickness of the PTC is on the order of the boundary layer thickness. The
181 robustness of the *PTC-to-FIM Map* was studied by Park et al. (2013) defining not only the PTC
182 coverage requirement in the direction of the flow but also the hierarchy of dominance of the
183 zones where PTC is applied. PTC has a major impact on the energy conversion and the
184 efficiency of the Converter. This is discussed in detail in Section 4.3.

185 A total of up to four single-cylinder oscillators have been placed in the LTFSW Channel for
186 FIM testing as shown in Fig. 5.

Figure 5. Design of four oscillators in the LTFSW Channel

2.3. Oscillator simulated by the Vck system

In early versions of the oscillators in the MRELab, the system was operational on physical springs and mechanical dampers. This is a very time consuming process in calibration and alignment and cannot produce accurate models of damping and stiffness and values. For high conversion of hydrokinetic energy, the oscillator needs to have a small system damping because it only results in energy dissipated to overcome friction. Damping is added in the form of resisting torque through Vck for harnessing hydrokinetic energy. The total damping - consisting of system structural damping plus harnessing damping - cannot exceed certain limit or the FIM - VIV and galloping alike - will be suppressed. If the total damping is above optimal, it will decrease the efficiency of the Converter. The number of design parameters is very high and requires extensive testing. To mention just a few parameters: mass ratio, spring-stiffness function, spring-stiffness value, damping function, damping value, roughness distribution (4

parameters); and in the case of multiple cylinders in-flow spacing, transverse spacing, ratio of cylinder masses, ratio of stiffness values, etc.

For systematic experimental testing particularly in such a large design space, two challenges are important to overcome: (a) Changing the physical springs and dampers requires extensive preparation for calibration and alignment. (b) The viscous damping of the system is rarely linear as modeled in classical vibration textbooks. To compensate for these two major challenges, Lee and Bernitsas (2010) and Lee et al. (2011) introduced and implemented the first generation virtual spring-damping device called Vck. Sun et al. (2015) developed the embedded second generation of Vck. The purpose of Vck is to replace the physical oscillator with a simulator without including the hydrodynamics in the closed control loop, which would bias the FIM phenomenon being measured. In the alternative method, the hydrodynamic force is measured and the differential equation of motion is solved to calculate the displacement, which is then imposed by the controller. This approach is much faster but keeps the hydrodynamic force in the control loop. It also results in 5° - 12° phase-lag between force and displacement. This method was developed first by Hover et al. (1997), Smogeli et al (2003) and adopted more recently by Mackowski and Williamson (2011). Moser et al. (2008) used both direct application of forces and torques, and through the use of tangential synthetic jet actuators.

Several improvements were achieved in the second generation Vck compared to the first. First, the second generation Vck system was developed as a stand-alone application via an embedded controller rather than using LabView which added a lag that had to be compensated.

Second, digital signals were used instead of analog signals. The sample time was reduced by three orders of magnitude, thus, eliminating the need to include dynamic terms with memory in the damping model to achieve no bias in FIM and energy harnessing. Both systems are capable of compensating for the system nonlinear damping. In both Vck generations in the MRELab, the total nonlinear expression for the viscous damping of the system is modeled and its coefficients are determined after extensive system identification. The result is then subtracted from the system through the controller to nullify the system damping. Subsequently, a mathematically accurate linear viscous damping is introduced. This is hardly achievable in engineering applications; and yet it is used in modeling oscillators in textbooks. With the Vck controller, a perfectly linear viscous damping is achieved and implemented. A resistor bank is used to dissipate the generated power.

The most notable achievement of the Vck system is that it does not include the hydrodynamic force in the closed control loop, and thus, does not bias the measured phenomenon. As a result, Vck simulates the oscillator only and not the FIM phenomenon. This is achieved at the expense of extensive system identification of the real system viscous damping. System identification for a totally new system could require several months of modeling and measurements using free decay tests and error minimization.

In 2011, in order to investigate further the FIMs of cylinders with fewer restrictions, both the channel and the oscillators were rebuilt. This third generation of test-models was designed and built as shown in Figs. 5-7. In the new design, the linear motion mechanism was brought under

water, which resulted in most of the Converter components being under water. The whole linear motion system, which consists of the end sliding-blocks, one timing-belt and pulleys, is fitted inside of a 38mm wide slender tube. The sliding block connects the cylinder to the Vck system through the timing belts and pulleys as shown in Fig. 6. The new system better models real, field-deployed, devices compared to previous models tested in the MRELab, where only the oscillating cylinders and supporting end-struts were under water. The structural differences result in stronger tip-flow effects in the new system, which are discussed in Section 4.1.

The mass ratio is changed by placing a mass-shaft inside the cylinder (Sun et al. 2015). The center-to-center distance (d/D) between adjacent cylinders in the flow direction can be adjusted by the rail system starting at 5inches (12.7cm) and increased by 0.5inches (1.27cm). The features of the new Converter model are summarized in Table 2.

Table 2. Specifications of the new Converter model

Mass Ratio	Aspect Ratio	Maximum Oscillatory Amplitude	Longitudinal Distance
$0.634 \leq m^* \leq 2.00$	$L/D = 10.07$	$A_{max}/D_{3.5''} = 5.5$	$1.429 \leq d/D_{3.5''} \leq 6.0$

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Figure 6. Details of the new Converter model

Figure 7. One cylinder Converter with Vck mounted in the LTFSW Channel (top view)

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260 2.4. Math model for the Vck oscillator system

The physical model in Fig. 8, assumes to have structural symmetry and the gravitational force is balanced by spring pretension. Thus, the cylinder oscillates around its equilibrium position after the virtual springs and damper have been implemented.

The components in the Vck system are listed in Table.3. The equations of motion of the current oscillator system with the embedded Vck are presented in Sun et al. (2015) and result in equation (4) :

Table 3. Components of embedded Vck system

$\theta_i, i = 1, 2, 3, 4$	$i=1$: angle of the motor coupling and pulley $i=2$: angle of the pulley top-center $i=3$: angle of the pulley top-right $i=4$: angle of the pulley lower-right
y	Displacement of the cylinder
$M_{osc}[kg]$	Equivalent mass of oscillating components
$J_{motor}[kg \cdot m^2]$	Mass moment of inertia of the motor
$t_{motor}[N]$	Nonlinear damping torque of the motor
$T_{motor}[Nm]$	Torque generated by the motor
$J_{pulley}[kg \cdot m^2]$	Mass moment of inertia of the pulley
$t_{pulley,i} \ i = 1, 2, 3, 4$	Nonlinear damping torque of i -th pulley
$r_{pulley}[m]$	Radius of all pulleys
$f_{bearing}[N]$	Nonlinear damping force of all bearings
$F_i[N]$	i -th tension of the time-belt

Figure 8. Symmetric Vck model with notation

$$\begin{aligned}
 & (J_{motor} + 6J_{pulley} + M_{osc} r_{pulley}^2) \ddot{\theta} \\
 & = T_{motor} - (t_{motor} + 2t_{pulley} + 6t_{pulley2} + 3t_{pulley3} + f_{bearing} r_{pulley})
 \end{aligned} \tag{4}$$

where it was assumed that the timing belts are inelastic.

Assuming further that $F_{motor} = T_{motor} / r_{pulley}$, the above equation can be simplified for convenience

as:

$$m\ddot{y} = F_{motor} - f \tag{5}$$

where:

$$m = \frac{J_{motor}}{r_{pulley}^2} + \frac{6J_{pulley}}{r_{pulley}^2} + M_{osc} \tag{6}$$

$$f = \frac{t_{motor}}{r_{pulley}} + \frac{2t_{pulley}}{r_{pulley}} + \frac{6t_{pulley2}}{r_{pulley}} + \frac{3t_{pulley3}}{r_{pulley}} + f_{bearing} \tag{7}$$

The main function of the Vck system is to add mathematically accurate damping and stiffness models. In this case, a linear adjustable damping is then added to the system to harness hydrokinetic energy. To develop the sought-after power envelope, systematic tests must be conducted. Prior to adding prescribed damping, the physical system damping needs to be identified and subtracted. The latter is due to friction in all moving parts, especially in the timing belts and pulleys, which bring undesirable/excessive amount of nonlinear damping to Vck.

Based on the non-linear viscous damping model, the controller force can be expressed as:

$$F_{controller} = F_{nonlinear_damping} - c_{virtual}\dot{y} - K_{virtual}y \quad (8)$$

Where, y is the displacement of the cylinder, $F_{controller}$ is the force applied by the virtual system (controller), $F_{nonlinear_damping}$ is the total nonlinear damping force, $K_{virtual}$ is the virtual spring constant, $c_{virtual}$ is the virtual linear viscous damping. The range of stiffness calibration is extended as shown in Fig. 9. The last five spring stiffness values have three weight-points due to the height limitation of the dry test-bed. Regarding damping, $c_{virtual}$ is added after subtracting the nonlinear viscous damping so that the system tested is a mathematically-true linear damping system. Free decay tests were performed to verify the subtraction of the system damping and the application of the linear viscous damping. For those tests, results are shown for two spring constants 600N/m and 800N/m in Fig. 10. The agreement between the real spring system and Vck system is satisfactory in VIV and galloping measurements over the entire range of velocities tested, as shown in Fig. 11.

Figure 9. Control experimental result for spring constants

Figure 10. Free decay tests for Vck at $K_{virtual}=600$ and 800 N/m

Figure 11. Amplitude response of distributed roughness cylinder with PTC (P60 at 20° - 36°). Comparison of experimental results for real and Vck system at $m^*=1.658$.

3. MATHEMATICAL MODEL OF HARNESSSED AND DISSIPATED POWER

3.1. Motion of the oscillator

The flow direction is the positive x -direction. The motion of the cylinder is in the y -direction and is modeled by a second order linear differential equation as

$$m_{osc}\ddot{y} + c_{total}\dot{y} + Ky = F_{Fluid} \quad (9)$$

where y is the direction perpendicular to the flow and the cylinder axis, m_{osc} is the total oscillating system mass, which includes one third of the spring mass; K is the spring stiffness, c_{total} is the total damping coefficient, and F_{Fluid} is the force exerted by the fluid on the body in the y -direction. In this model, the fluid force is separated into vortex and inviscid component as argued by Khalak and Williamson (1999),

$$m_{osc}\ddot{y} + c_{total}\dot{y} + Ky = F_{viscous} + F_{potential} \quad (10)$$

The inviscid force may be defined in terms of the inviscid added mass m_a . The added mass inclusion through a constant added mass coefficient (from potential theory) on the left hand side is still debated. The alternative approach uses the calculated added mass from the lift force in-phase with the acceleration in the y -direction (Vikestad 2000).

It is the authors' opinion that this controversy is unnecessary as both are simple modeling methods that cannot account for the range of synchronization or the self-limiting nature of the amplitude of oscillation. Both model the response of the cylinder in FIM for a specific flow velocity and a given experiment. As well, both approaches are useful in post-processing data and

understanding this complex phenomenon better. For better understanding of the complexity of the added mass expression in forces and moments on a small body moving in a 3-D unsteady flow, the reader is referred to the detailed analysis by Foulhoux and Bernitsas (1993). The dependence of the inertia terms on absolute and/or relative velocity as well as the form of convective terms is explained thoroughly putting into perspective the simplistic nature of the inertia term used in Morison's equation.

The practical difference between the two methods is only in the presentation of experimental data rather than in shedding light into a complex and, thus far, poorly modeled phenomenon. Proceeding with the first method we have

$$F_{potential} = -m_a \ddot{y} \quad (11)$$

$$F_{viscous} = \frac{1}{2} c_y(t) \rho U^2 D L \quad (12)$$

The displaced fluid mass m_d for a circular cylinder is $\rho L \pi D^2 / 4$. This reduces the equation of motion to

$$(m_{osc} + m_a) \ddot{y} + c_{total} \dot{y} + Ky = \frac{2}{\pi D} c_y(t) m_d U^2 \quad (13)$$

3.2. Fluid Power Conversion

The power in flowing fluid over the projected area of the cylinder in the direction perpendicular to the flow direction can be calculated as follows. The kinetic pressure head in the fluid from Bernoulli's equation is $(1/2) \cdot \rho U^2$. The force acting over the projected area of the

340 cylinder is the flow direction y is $(1/2) \cdot \rho U^2 DL$. Then, the power in the fluid, being the product
 341 of the acting force times the velocity in that same direction x is

$$342 \quad P_{Fluid} = \frac{1}{2} \rho U^3 DL \quad (14)$$

343 It should be noted that the power in the fluid used in the efficiency calculation in Section 4.3.2 is
 344 over the area of $(2A+D)L$ swept by the cylinder. The work done by the fluid force acting on a
 345 cylinder during a vibration cycle is the inner vector product of the force by the displacement
 346 vector dy integrated over one cycle of oscillation of the cylinder

$$347 \quad W_{VIVACE-Fluid} = \int_0^{T_{osc}} F_{Fluid} y dt \quad (15)$$

348 Then, the fluid power into the Converter is

$$349 \quad P_{VIVACE-Fluid} = \frac{W_{VIVACE-Fluid}}{T_{osc}} \quad (16)$$

350 The viscous force exerted by the fluid on the cylinder is given by the right-hand side of equation
 351 (13). If we multiply the force by the instantaneous velocity, integrate the right-hand side, and
 352 average it over the cycle period T_{osc} we have:

$$353 \quad P_{VIVACE-Fluid} = \frac{1}{T_{osc}} \int_0^{T_{osc}} \frac{2}{\pi D} c_y(t) m_d U^2 2\pi f_{fluid} A \cos(2\pi f_{osc} t) dt \quad (17)$$

354 where $T_{osc} = 1/f_{osc}$ and $f_{fluid} = f_{osc}$ in synchronization, and the assumption was made that the
 355 excitation and response are sinusoidal. That is, $c_y(t) = c_y \sin(2\pi f_{osc} t + \phi)$ leading the sinusoidal
 356 response $y(t) = A \sin(2\pi f_{osc} t)$ by phase ϕ . Thus,

$$\begin{aligned}
P_{VIVACE-Fluid} &= f_{osc} \int_0^{f_{osc}} \frac{1}{\pi D} 2\pi f_{osc} A \sin(2\pi f_{osc} t + \phi) \cos(2\pi f_{osc} t) dt \\
&= \frac{4\pi}{D} c_y m_d U^2 f_{osc} A \sin \phi
\end{aligned} \tag{18}$$

Inserting the definition of $m_d = \rho D^2 L \pi / 4$, yields

$$P_{VIVACE-Fluid} = \frac{1}{2} \rho \pi c_y U^2 f_{osc} A D L \sin \phi \tag{19}$$

3.3. Mechanical Power

From equation (13), we integrate the left-hand side after multiplying it by the instantaneous velocity and we average it over the cycle period T_{osc} . That yields the mechanical power in the Converter (Bernitsas, et al., 2006)

$$P_{VIVACE-Mech} = \frac{1}{T_{osc}} \int_0^{T_{osc}} ((m_{osc} + m_a) \ddot{y} + c_{total} \dot{y} + K y) \dot{y} dt \tag{20}$$

Only the term in phase with the velocity generates a nonzero energy term in sinusoidal response.

In addition, we have

$$\zeta_{total} = \frac{c_{total}}{2\sqrt{(m_{osc} + m_a)K}} \tag{21}$$

and $f_{n,water}$ as defined by equation (2). Thus,

$$P_{VIVACE-Mech} = \frac{1}{T_{osc}} \int_0^{T_{osc}} 4\pi (m_{osc} + m_a) \zeta_{total} \dot{y}^2 f_{n,water} dt \tag{22}$$

Using the sinusoidal expression for $y = A \sin(2\pi f_{osc} t)$ in the above integral we have

$$\begin{aligned}
& \frac{1}{T_{osc}} \int_0^{T_{osc}} 4\pi(m_{osc} + m_a)\zeta_{total} A^2 (\cos(2\pi f_{osc} t))^2 (2\pi f_{osc})^2 f_{n,water} dt \\
& = \frac{1}{T_{osc}} \int_0^{T_{osc}} 4\pi(m_{osc} + m_a)\zeta_{total} A^2 \frac{1 + \cos(4\pi f_{osc})}{2} (2\pi f_{osc})^2 f_{n,water} \\
& = 8\pi^3 (m_{osc} + m_a)\zeta_{total} (A f_{osc})^2 f_{n,water}
\end{aligned} \tag{23}$$

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376 3.4. Harnessed Power and Dissipated Power

377 In order to convert hydrokinetic energy to mechanical energy in the oscillator and
378 subsequently electrical energy, additional damping is introduced into the system. The total
379 damping (c_{total}) is defined as

$$380 \quad c_{total} = c_{structure} + c_{harness} \tag{24}$$

381 where $c_{structure}$ is the existing system damping due to losses in the transmission system. In this
382 research, $c_{structure}$ is adjusted by Vck system in each experiment according to m^* and K to achieve
383 the same artificial value of $\zeta_{structure}=0.2$. $c_{harness}$ is the damping added through Vck that
384 converts the mechanical energy in the oscillating cylinder to electrical energy simulating a Power
385 Take –Off (PTO) system.

386 From equation (24), we know that $c_{structure}$ and $c_{harness}$ can be expressed using damping ratio
387 $\zeta_{harness}$ and $\zeta_{structure}$.

$$388 \quad \zeta_{structure} = \frac{c_{structure}}{2\sqrt{(m_{osc} + m_a)K}} \tag{25}$$

$$\zeta_{harness} = \frac{c_{harness}}{2\sqrt{(m_{osc} + m_a)K}} \quad (26)$$

Using equation (23), we obtain the harnessed power and the dissipated power as

$$P_{VIVACE-structure} = 8\pi^3 (m_{osc} + m_a) \zeta_{structure} (Af_{osc})^2 f_{n,water} \quad (27)$$

$$P_{VIVACE-harness} = 8\pi^3 (m_{osc} + m_a) \zeta_{harness} (Af_{osc})^2 f_{n,water} \quad (28)$$

To calculate the harnessed power using equation (28), several parameters should be determined first. Oscillating system mass m_{osc} and spring constant K can be measured directly. The ideal added mass m_a is equal to the displaced fluid mass for a circular cylinder. Using spring stiffness K , oscillating system mass m_{osc} , and added mass m_a , the natural frequency of cylinder in water can be calculated using equation (2). Structural damping $c_{structure}$ and the added damping $c_{harness}$ are selected using the Vck system (Hai Sun, et al. 2015).

The remaining two parameters, A and f_{osc} , are obtained through the measured displacement history, which is recorded for every experiment. Thus, the harnessed-power envelope is established (Lee, et al., 2011; Chang and Bernitsas, 2011).

Near the end of Section 3.1, it was pointed out that the controversy that has arisen in the past few years regarding the handling of the added mass does not affect the values of experimentally acquired data; it affects only data presentation and plotting. Since equations (27) and (28) appear to depend explicitly on the added mass m_a , it is worth noting that they can be recast using equations (21), (25), (26) as follows, respectively.

$$P_{VIVACE-dissipated} = \frac{1}{2} c_{structure} A^2 \omega_{osc}^2 \quad (29)$$

and

$$P_{VIVACE-harness} = \frac{1}{2} c_{harness} A^2 \omega_{osc}^2 \quad (30)$$

These two equations do not depend on the specific way the added mass is calculated or even the value of the added mass explicitly.

4. RESULTS AND DISSCUSSION

In this paper, a series of experiments were conducted over spring-stiffness range of $400\text{N/m} \leq K \leq 1,200\text{N/m}$, damping ratio range of $0 \leq \zeta_{harness} \leq 0.24$, and mass ratio values (1.007, 1.343, 1.685). It should be noted that the values of stiffness K selected are accurate in form and value and exact as they are defined by the calibrated motor-torque and its relation to rotation as shown in Fig. 9. That is, the K value is not subjected to fabrication error or degradation in time due to fatigue. Similarly, the values of $\zeta_{harness}$ and ζ_{total} are exact and the corresponding linear viscous function is accurate as explained in the discussion on the Vck system. The Reynolds number range is $30,000 \leq Re \leq 120,000$, and the corresponding reduced velocity range is $2.92 \leq U^* \leq 15.33$. The particulars for each case are shown in Table 4. In this research, the damping ratios $\zeta_{harness}$ and $\zeta_{structure}$ were adjusted precisely, for each mass and stiffness, to achieve the desired value. The reasons that $\zeta_{structure}$ is set to 0.02 for each case are the following: (1) A minimum damping is needed in galloping for safety reasons. A $\zeta_{total}=0$ damping system would hit

the limit of the test facilities at higher flow velocity. (2) To make comparisons with previous systematic experiments using the first generation of Vck system and real spring oscillators. All the dimensional damping values for each case are listed in Tables 5, 6 and 7 in the Appendix.

Table 4 Particulars of single-cylinder oscillator models

	<i>Test set 1</i>	<i>Test set 2</i>	<i>Test set 3</i>
$k[N/m]$	400 ~ 1,200	400 ~ 1,200	400 ~ 1,200
<i>Temperature</i> ($^{\circ}C$)	20.5	20.5	20.5
μ ($N\ s/m^2$)	1.004E-03	1.004E-03	1.004E-03
v (m^2/s)	9.940E-07	9.940E-07	9.940E-07
$\rho[kg/m^3]$	999.729	999.729	999.729
$D[m]$	0.0889	0.0889	0.0889
$L[m]$	0.895	0.895	0.895
$m_{displacement}(kg)$	5.425	5.425	5.425
$m_{oscillation}(kg)$	5.463	7.286	9.141
$m_{added}(kg)$	5.425	5.425	5.425
$\zeta_{structure}$	0.020	0.020	0.020
$f_{n,water}$	0.97 ~ 1.67	0.89 ~ 1.57	0.83 ~ 1.45

4.1. Amplitude Response

The amplitude ratio A/D is shown in Figs. 12-14 for PTC-cylinders. The amplitude A of the cylinder is calculated by averaging the absolute values of the 30 highest values of positive and 30 highest negative peaks in 60 seconds of running time past the transient. The mean absolute deviation is denoted by vertical bars. Back-to-back VIV and galloping can be observed for all mass ratio values. It can also be observed that the amplitude response in galloping increases considerably as the mass-ratio increases. In the VIV range, for the mass ratios tested, the impact of the mass ratio on A/D is small. Four regions can be observed in the amplitude ratio curves.

Figure 12. Rough cylinder amplitude response for different damping ratios $\zeta_{Harness}$; $m^*=1.007$; $K=600$ N/m and 755 N/m.

Figure 13. Rough cylinder amplitude response for different damping ratios $\zeta_{harness}$; $m^*=1.343$; $K=600$ N/m and 755 N/m.

Figure 14. Rough cylinder amplitude response for different damping ratios $\zeta_{Harness}$; $m^*=1.685$; $K=600$ N/m and 755 N/m.

(a) $35,000 \leq Re \leq 45,000$: Is the initial branch of VIV for all the mass-ratio values. The upper end of the initial VIV branch increases with increasing stiffness, which is consistent with previous experimental results (Bernitsas et al. 2009, Lee et al. 2010). A drop in A/D can also be observed for all m^* values from $\zeta_{harness}=0.00$ to $\zeta_{harness}=0.20$. This is due to the added damping for energy harnessing and to the distributed roughness. The latter is needed to trigger galloping.

(b) $45,000 \leq Re \leq 70,000$: This is the upper branch of VIV for $m^*=1.343$ and $m^*=1.685$. For $m^*=1.007$, the upper branch region is $45,000 < Re < 67,000$. Comparing the initial and upper branches in VIV, the shorter error bars indicate that the FIM of the cylinder is more stable in the upper branch. The effects of spring stiffness and total damping are more obvious in the upper branch than that in the initial branch. The amplitude ratio at $m^*=1.007$, $\zeta_{harness}=0.00$ ($Re=71,000$) reaches $0.9814 \cdot D$ compared to $1.0546 \cdot D$ at $m^*=1.343$ and $1.12 \cdot D$ at $m^*=1.685$.

The upper branch is narrower than in previous experiments conducted in the MRELab, due to different structural design and PTC location. The latter can trigger galloping earlier, and it is more desirable from the power harnessing perspective (Kim et al., 2011). On the other hand, the upper branch is wide in the TrSL3 flow regime overtaking almost completely the lower branch which is much wider in the TrSL2 flow regime (Raghavan and Bernitsas, 2011).

The narrow upper branch is contributed to tip flow effects (Kinachi, et al., 2016) as shown in the Figure 15. Tip-flow limits the cylinder length over which lift (positive) is exerted in the transverse direction when the cylinder is in FIM. Negative lift is typically exerted in the rest of

the length affected by the tip-flow. In this study, PTC is proven to be an effective way to overcome tip-flow adverse effects in hydrokinetic energy conversion. The estimated loss in lift force when comparing an oscillator model to a nearly ideal VIV oscillator (Kinachi, et al., 2016) is limited to less than 10% of the cylinder length.

Figure 15. Effective cylinder length at different Reynolds numbers (Kinaci et al., 2016)

(c) $68,000 \sim 70,000 \leq Re \leq 80,000 \sim 90,000$: This range includes the lower branch of VIV, the desynchronization range, the gap between VIV and galloping, and the onset of galloping. Not all four sub-ranges are obvious in all A/D lines in Figs. 12-14, because the VIV range moves with changes in K and m^* and shrinks with increasing $\zeta_{harness}$; in addition, the onset of galloping increases with increasing $\zeta_{harness}$. In this region which includes all four sub-ranges, the amplitude drops dramatically and random oscillatory patterns appear as expected. Consequently, the error bars become significantly larger in this region as shown in Figs. 12-14.

The relative location of the VIV range of synchronization in comparison with the onset of galloping has been explained thoroughly by Park et al. (2013) and was briefly explained in the Introduction.

(d) $90,000 \leq Re$: 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. Non-circular cross-section cylinders could be subjected to high amplitude galloping oscillation due to their geometric asymmetry with respect to the flow. In this study, the roughness strips result in geometric asymmetry, thus, inducing galloping. The maximum amplitude of the single cylinder with PTC varies from $0.94 \cdot D$ ($m^*=1.007$, $K=1,200\text{N/m}$, $\zeta_{harness}=0.24$) to $3.01 \cdot D$ ($m^*=1.685$, $K=400\text{N/m}$, $\zeta_{harness}=0.00$) at $Re=118,000$ ($U=1.31\text{m/s}$), which are also shown in Figs. 12-14.

4.2. Frequency Response

In Figs. 16-18, the frequency ratio $f^* = f_{osc} / f_{n,air}$ of the PTC-cylinder is plotted versus Reynolds number Re , reduced velocity U^* , and flow velocity U . The oscillation frequency f_{osc} is calculated by FFT (Fast Fourier transform) of the displacement of the cylinder over the recorded period with the exclusion of the end-transients. The frequency ratio is not affected significantly by the total damping ratio except in the transition regions. The latter is due to intermittency in oscillations in the VIV desynchronization and to the occurring gap between VIV and galloping.

Figure 16. Rough cylinder frequency response for different damping ratios, $m^*=1.007$

Figure 17. Rough cylinder frequency response for different damping ratios, $m^*=1.343$

Figure 18. Rough cylinder frequency response for different damping ratios, $m^*=1.685$

(a) $38,000 < Re < 70,000$: This is the VIV initial branch and upper branch range. For $m^*=1.007$,

with the increase of the spring stiffness K , the system natural frequency in air increases from

1.36 to 2.36 and the oscillation frequency ratio rises from 0.69 to 1.18. For $m^*=1.343$, the system natural frequency in air increases from 1.18 to 2.04 and the oscillation frequency ratio rises from 0.70 to 1.04. For $m^*=1.685$, the system natural frequency in air increases from 1.05 to 1.82 and the oscillation frequency ratio (f^*) rises from 0.82 to 1.11.

(b) $70,000 < Re < 105,000$: As Re approaches 80,000, the frequency drop indicates transition to galloping. When no gap occurs between the upper branch in VIV and galloping, the frequency drop is accompanied by a rapid increase in the amplitude of the cylinder. When a gap occurs, A/D may drop to zero (Figs. 12-14) and thus the corresponding frequency will drop to zero (Figs. 16-18). The gap of course occurs for high damping, as explained in Section 4.1. In this region, lower damping results in wider upper and lower branch and, thus, back-to-back VIV and galloping.

(c) $Re \geq 105,000$: In the galloping branch, the frequency ratio drops, which is an important feature of galloping. The oscillation frequency remains close to the system natural frequency of a single cylinder in water in the galloping range (Park et al. 2013, Sun et al. 2015), regardless of damping, spring stiffness, m^* .

4.3. Envelope of Harnessed Power

4.3.1. Harnessed power curve

Experimental results for the harnessed power at different mass ratios for a PTC cylinder are plotted versus Reynolds number and flow velocity in three separate charts in Figs. 19-21. The

519 right y-axis is the scale for the total converted power, which includes both the harnessed power
520 and the dissipated power, where the latter is due to $\zeta_{structure}$.

521

Figure 20. Power curves for $m^*=1.343$ with K and $\zeta_{harness}$ as parameters

523

524 (a) $38,000 < Re < 68,000 \sim 70,000$: This region contains the VIV initial branch and VIV upper
 525 branch. A steady growth of power can be observed. Softer spring stiffness with lower harness
 526 damping ratio initiate power harness at flow velocity as low as 0.3946 m/s at $m^*=1.343$. The
 527 combination of higher spring stiffness with higher mass ratio starts power harnessing at higher
 528 flow velocity, but yield higher power in the VIV upper branch.

529 (b) $68,000 \sim 70,000 \leq Re \leq 80,000 \sim 82,000$: This region is past the VIV upper branch to VIV
 530 lower branch to desynchronization. The harnessed power drops sharply for $m^*=1.007$ and
 531 $m^*=1.343$ due to the drop in amplitude caused by the separation of VIV and galloping. For

532 $m^*=1.685$, the drop of power occurs at lower spring stiffness with $\zeta_{harness}>0.16$. These are
533 anticipated based on the earlier discussion related to equations (27) and (28). Further, the gap
534 between VIV and galloping increases with m^* as the range of VIV shifts proportionally to the
535 square root of m^* while the onset of galloping is proportional to m^* . Increased stiffness K and
536 lower c_{total} (and thus lower ζ_{total}) bridge the harnessed power gap between transition and
537 galloping. The harnessed power also increases as the spring stiffness increases and bridges the
538 gap. These are important observations for successful design of a Converter.

539 (c) $80,000 \sim 82,000 \leq Re$: This is the transition region followed by the galloping region. The
540 harnessed power increases as the flow velocity increases, regardless of mass ratio, spring
541 constant, and damping ratio.

542 In addition, the optimal harness power envelop of three mass ratios are shown in Figure 22,
543 the corresponding converted power, stiffness and $\zeta_{harness}$ are listed at table 8 to 10. Based on the
544 limit mass ratios tested in this research, a few observations can be drawn:

545 1) At initial and upper branch ($U \leq 0.71 \text{ m/s}$ $Re \leq 64,000$), the $m^*=1.343$ has the highest over
546 harness power;

547 2) At upper branch to lower branch and galloping ($0.71 \text{ m/s} \leq U$ $64,000 \leq Re$), the harness
548 power increases as the mass ratio increases;

549 3) At upper branch to galloping, high stiffness with high $\zeta_{harness}$ yields higher energy, this is
550 consistent for all the m^* .

Figure 22. Power envelop for three m^* values: $m^*=1.007$, $m^*=1.343$ and $m^*=1.685$

4.3.2. Effect of damping, stiffness, and m^* on power conversion efficiency

The power in a fluid is calculated based on the area swept by the cylinders.

$$P_{Fluid}(W) = \frac{1}{2} \rho U^3 (2A_{max} + D)L \quad (31)$$

where A_{max} is the largest amplitude among all cylinders in a multiple-cylinder converter.

Obviously, the denominator depends on the cylinder response, unlike propellers or wind turbines

where the flow area is determined by the fixed geometry of the area swept by the blades. The

Betz limit is the theoretical maximum power that can be extracted from an open flow and is

equal to 59.26% (=16/27). Based on the Betz limit, the power conversion efficiency, $\eta_{convert}$, and power harness efficiency, $\eta_{harness}$, can be calculated as

$$\eta_{convert}(\%) = \frac{P_{convert}}{P_{Fluid} \times Betz\ Limit} \times 100 \quad (32)$$

$$\eta_{harness}(\%) = \frac{P_{harness}}{P_{Fluid} \times Betz\ Limit} \times 100 \quad (33)$$

Commercial scale wind turbines achieve peak efficiency of 75% to 80% of the Betz limit (Burton et al. 2011). The two efficiencies modeled by equations (32) and (33) are plotted in Figs. 23-25 with their scales shown on the right and left y-axis, respectively. The damping ratio $\zeta_{harness}$ is the parameter in those figures.

Figure 23. Efficiency of harnessed and converted power for $K=600\text{N/m}$ and $m^*=1.685$ at various $\zeta_{harness}$

Harness damping-ratio effect: The dependence of power efficiency on velocity or Reynolds number, and damping ratio is shown in Fig. 23. The efficiency increases as the harness damping increases up to certain value. Oscillation amplitude slowdown in the increase of efficiency occurs as the damping ratio increases both in the VIV and galloping regions. For $\zeta_{harness}=0.20$, the power curve crosses with the $\zeta_{harness}=0.24$ curve at several test points both in the VIV and galloping regions, which indicates that the optimal power output of the tested Converter Vck system has been reached. When the $\zeta_{harness}$ was increased further, the converted power was reduced.

Figure 24. Efficiency of harnessed and converted power for $\zeta_{harness}=0.16$ and $m^*=1.685$ at various K

578 Stiffness effect: The power efficiency curves are shown as a function of stiffness in Fig. 24. The
579 flow velocities of peak efficiency at different stiffness in VIV region varies from 0.55 m/s to
580 0.75 m/s and the corresponding harness efficiency varies from 33% ($K=400\text{N/m}$) to 44%
581 ($K=755\text{N/m}$), respectively. The optimal power efficiency can be clearly seen to occur for
582 $K=755\text{N/m}$. In the galloping region, the efficiency shows a small constant decrease with velocity.
583 Stiffness $K=1,200\text{N/m}$ has highest harness efficiency due to the decrease of its amplitude
584 response. A slowdown of efficiency decrease as the stiffness K increases can be seen in the
585 galloping region.

586

587

Mass-ratio effect: The power efficiency pattern is shown as a function of mass ratio m^* in Fig. 25. The peak harness efficiency in the VIV region increases considerably, from 17% at $m^*=1.007$ to 36% at $m^*=1.685$ for $\zeta_{harness}=0.12$. The same trend also appears in the galloping region. Increasing the mass ratio has a positive impact on the power harness. From the design point of view, though, all factors should be taken into account; e.g., the cost and maintenance increases as the system mass increases. This topic though is beyond the scope of this paper.

4.3.3. Optimality criterion

Figs. 19-21 show power harnessed as a function of the flow velocity. Two local optima appear. One in the VIV synchronization range and one in the galloping instability range. The latter is clearly the global optimum and it is a boundary point indicating that its value increases as the flow velocity increases.

Figs. 22-24 show the Converter efficiency as function of the flow velocity. These figures also show two local optima; one in the VIV range and one in the galloping range. The former is clearly the global optimum.

The local optima of power and efficiency in VIV do not match and each one represents a very different design. The efficiency local optimum in VIV is at the beginning of the upper branch, while the power local optimum in VIV is located at the end of the upper branch.

Similar is the case in galloping. The efficiency local optimum in galloping is at the beginning of the fully developed galloping branch, while the power local optimum in galloping is located at the highest galloping velocity limited by the capacity of the MRELab recirculating channel.

The question that arises is which of the four local optima to pursue for practical purposes such as commercialization of the Converter. First of all, a discussion on the local optima is required. Of the four local optima, regardless of whether the underlying FIM is VIV or galloping, the lowest velocity clearly exhibits the highest efficiency and lowest generated power. Also, of the four local optima, the highest velocity clearly exhibits the highest generated power using galloping, however it has the lowest efficiency.

The explanation lies in the denominator of the efficiency as defined by equation (31). The power in the denominator is $0.5\rho U^3(D+2A)L$, that is, the swept area rather than the projected area of the cylinder. VIV is self-limiting yielding an interior design-space point as the optimum. Galloping, on the other hand, is an instability based phenomenon and the cylinder can move in one direction as long as the Converter frame allows it. In a very large frame, high power would be produced but a higher denominator would make the efficiency nearly zero .

The issue is which of the four local optima to pursue. In terms of increasing velocity the practical implications and the specific application each of the four designs represents are discussed next:

(a) At the beginning of the upper branch in VIV, the velocity is the lowest, the generated power is the lowest, and the efficiency is the highest. With soft springs the Converter can generate power - albeit small - at about 0.4m/s. That could be most valuable in specific applications for recharging electronics where electricity or even batteries are not available. In recent MRELab experiments, this velocity was 0.27m/s for $K=200\text{N/m}$.

(b) At the end of the upper branch in VIV, the velocity is higher, the generated power is the highest in VIV, and the efficiency is lower. That would also be valuable in harnessing very low speed hydrokinetic power when efficiency is not important but power is most valuable.

(c) As velocity increases, the third local optimum is located at the beginning of the fully developed galloping range. That point has the highest efficiency in galloping which is significantly lower than the global efficiency optimal of point (a). The power though is higher due to the velocity increase. That would be a point to pursue in design in case the velocity of the flow has little variation. In that case the optimum the two galloping optima can be superposed.

(d) The fourth local optimum generates the maximum power at the lowest efficiency. If power is the desired output that would be the global optimum particularly in cases of variable flow velocity.

The global design optimum of course depends on the cost of constraints as the minimum LCOE (Levelized Cost Of Energy) is sought. In design (a) a higher number of oscillators, including cylinders, transmissions, generators, and power-take-off systems will be needed to produce certain level of power at maximum efficiency. In design (d) more space will be required reducing the power-to-density ratio of the Converter. The answer depends on the specific conditions for a given application.

5. SUMMARY AND CONCLUSIONS

The flow induced motions and the hydrokinetic power conversion capacity of a single, rigid, circular cylinder with PTC on end-springs for various values of stiffness, mass-ratio, and harnessing damping were investigated experimentally at Reynolds numbers in the TrSL3 flow regime ($35,000 < Re < 120,000$, $2.92 < U^* < 17.73$, $0.395 \text{ m/s} < U < 1.315 \text{ m/s}$). All experiments were conducted in the LTFSW Channel of the MRELab of the University of Michigan using the second generation of the Vck virtual oscillator developed in the MRELab. The results, including amplitude response, frequency response, energy conversion, and energy conversion efficiency of the single-cylinder Converter as function of Reynolds number, flow velocity, and reduced velocity in water were presented and analyzed.

The model built and tested was that of a single-oscillator Converter having all components inside the water, thus, reducing the length of the cylinder providing lift and contributing to hydrokinetic power conversion. That is, the Converter tested, differs from the FIM oscillators tested in the MRELab in the past (Chang et al. 2011). The former has narrower VIV upper branch due to stronger tip-flow effects. The PTC used in this study compensates for the loss of lift due to tip-flow and actually initiates galloping at lower flow velocities, which is desirable from the power harnessing point of view. The tip-flow was estimated to affect less than 10% of the cylinder length. Thus, PTC was proven to be an effective way to compensate tip-flow effects in the present generation of Converter model.

The stiffness and damping of the oscillator were simulated by an innovative virtual oscillator - the Vck system - which was developed over a period of five years in the MRELab. Vck does

not include the hydrodynamic force in the closed loop, which would have biased the measured phenomenon. Vck enables two unique features for experiments in the MRELab: (a) The structural damping model and value are determined by an elaborate system identification process, and then, is fully subtracted prior to adding a mathematically accurate linear viscous damping model. (b) The values of damping and stiffness can be adjusted by computer control in real time thus reducing the testing preparation time from days to seconds. (c) Accordingly, numerous and systematic tests can be conducted to tackle the problem of studying the conversion of hydrokinetic energy using alternating-lift through a single-oscillator Converter, which has multiple design parameters.

The following conclusions can be drawn from the tests conducted in this study.

(1) Effect of mass ratio m^* :

- Increasing the mass ratio m^* has a strong positive effect on the hydrokinetic power harnessed in the galloping region and the upper branch to transition in the VIV region. There is a considerable increase (98%) in peak harnessed power in the upper branch and 69% increase of peak harnessed power in galloping from $m^*=1.007$ to $m^*=1.685$.

- For a given $\zeta_{harness}$, the harnessed and converted power efficiencies increase as the mass ratio increases.

(2) Effect of damping ratio $\zeta_{harness}$:

• The values of the harnessing damping ratio tested with the aid of Vck are dense enough to reveal the optimal power curve; e.g., for $m^*=1.007$ and $K=755$ N/m, the highest converted power occurs at $\zeta_{harness}=0.20$.

• The harnessed power efficiencies increase as the harness damping increases up to a value of $\zeta_{harness}$ that may be as high as 0.24. Then, it starts decreasing.

(3) Effect of spring stiffness K :

• With softer spring stiffness, power harnessing was initiated at a lower flow velocity at the initial branch of VIV for $Re \leq 35,000$. Power conversion started at 0.3946 m/s. Softer springs, though, combined with high damping for energy harnessing would result in a gap between the VIV synchronization range and the onset of galloping, thus having a negative impact on the range covered by the Converter oscillators.

• By increasing the spring stiffness K , even for high damping, the gap can be bridged at the expense of a delayed onset of VIV. The result is a Converter with oscillators that have high response starting at a little higher velocity with no upper end.

(4) Impact on harnessed hydrokinetic power:

• Power can be harnessed by the Converter with one PTC-cylinder over the whole Reynolds number range from 35,000 to 120,000. Experimental data, however, are not available at Reynolds numbers higher than 120,000 since tests reached the speed limit of the LTFSW Channel of the MRELab.

• Power curves exhibit two local optima, one in the VIV range and one in the galloping range.

705 • The VIV local optimum is located at the end of the upper branch, which represents the highest
706 velocity before the lower branch and desynchronization, where VIV lock-in breaks down.

707 • The galloping local optimum is located at the highest velocity achieved within the capabilities
708 of the MRELab recirculating Channel.

709 (5) Impact on efficiency of harnessed and converted hydrokinetic power:

710 • The flow velocity of peak efficiency at different stiffness values in the VIV synchronization
711 range varies from 0.55m/s to 0.75m/s and the corresponding harness efficiency from 33%
712 ($K=400\text{N/m}$) to 44% ($K=755\text{N/m}$) of the Betz limit, respectively.

713 • Efficiency curves exhibit two local optima, one in the VIV range and one in the galloping
714 range.

715 • The efficiency local optima do not coincide with the power local optima because the
716 denominator in the efficiency is affected by the amplitude of oscillation.

717 • The efficiency local optimum in VIV is located at the beginning of the upper branch, which
718 represents the lowest velocity after the initial branch. This is associated with a high jump in
719 amplitude but not big drop in frequency resulting in a high increase in equation (30), which
720 represents the numerator of the efficiency. That increase exceeds the increase in equation (31)
721 which represents the denominator of the efficiency.

722 • The galloping local optimum of the efficiency is located at the beginning of the fully developed
723 galloping region, that is at the opposite end of the power optimum in the galloping region.

724 • In the galloping region, the efficiency decreases slowly after its peak.

• Stiffness $K=1,200\text{N/m}$ results in the highest harnessing efficiency due to the decrease in amplitude response, which affects the denominator in the efficiency calculation.

(6) Design considerations:

• The advantage of the altering-lift energy converter is in the fact that its oscillators can respond with high amplitude over an open-ended velocity-range starting with a small velocity. That way it can convert high amounts of power with no need to adjust its oscillator characteristics as would be the case with a linear resonator.

• The Converter exhibits four different local optima; two in power and two in efficiency. In each case one is VIV based and the other is galloping based. The choice of design depends on the specifics of an application for commercial purposes.

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Appendix

Table 5. Damping parameters for $m^*=1.007$

$\zeta_{harness}$	ζ_{total}	c_{total} [Ns/m] $K=400\text{N/m}$	c_{total} [Ns/m] $K=600\text{ N/m}$	c_{total} [Ns/m] $K=755\text{ N/m}$	c_{total} [Ns/m] $K=1,000\text{ N/m}$	c_{total} [Ns/m] $K=1,200\text{ N/m}$
0.00	0.02	1.32	1.62	1.82	2.09	2.29
0.04	0.06	3.97	4.86	5.45	6.27	6.87
0.08	0.10	6.61	8.10	9.08	10.45	11.45
0.12	0.14	9.26	11.34	12.72	14.63	16.03
0.16	0.18	11.90	14.58	16.35	18.82	20.61
0.20	0.22	14.55	17.82	19.98	23.00	25.19
0.24	0.26	17.19	21.05	23.62	27.18	29.78

Table 6. Damping parameters for $m^*=1.343$

$\zeta_{harness}$	ζ_{total}	c_{total} [Ns/m] $K=400\text{N/m}$	c_{total} [Ns/m] $K=600\text{ N/m}$	c_{total} [Ns/m] $K=755\text{ N/m}$	c_{total} [Ns/m] $K=1,000\text{ N/m}$	c_{total} [Ns/m] $K=1,200\text{ N/m}$
0.00	0.02	1.53	1.87	2.10	2.41	2.64
0.04	0.06	4.58	5.61	6.29	7.24	7.93
0.08	0.10	7.64	9.35	10.49	12.07	13.22
0.12	0.14	10.69	13.09	14.68	16.90	18.51
0.16	0.18	13.74	16.83	18.88	21.73	23.80
0.20	0.22	16.80	20.57	23.08	26.56	29.09
0.24	0.26	19.85	24.31	27.27	31.38	34.38

Table 7. Damping parameters for $m^*=1.685$

$\zeta_{harness}$	ζ_{total}	c_{total} [Ns/m] $K=400\text{N/m}$	c_{total} [Ns/m] $K=600\text{ N/m}$	c_{total} [Ns/m] $K=755\text{ N/m}$	c_{total} [Ns/m] $K=1,000\text{ N/m}$	c_{total} [Ns/m] $K=1,200\text{ N/m}$
0.00	0.02	1.71	2.09	2.35	2.70	2.96
0.04	0.06	5.13	6.28	7.05	8.11	8.89
0.08	0.10	8.55	10.47	11.75	13.52	14.81
0.12	0.14	11.97	14.66	16.45	18.93	20.74
0.16	0.18	15.39	18.85	21.15	24.34	26.66
0.20	0.22	18.81	23.04	25.84	29.74	32.58
0.24	0.26	22.23	27.23	30.54	35.15	38.51

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Table 8. Parameters for optimal harness and converted power curve for $m^*=1.007$

Test Conditions		Optimal harness power[W]	Optimal converted power[W]	Stiffness K [N/m]	$\zeta_{harness} = \frac{c_{harness}}{2\sqrt{(m_{osc} + m_a)K}}$
U (m/s)	Re				
0.39	35291.69	-	-	-	-
0.43	38869.15	0.18	0.26	400	0.04
0.47	42446.62	0.51	0.57	400	0.08
0.51	46024.08	1.02	1.15	400	0.16
0.55	49601.55	1.60	1.80	400	0.16
0.59	53179.01	2.05	2.31	600	0.16
0.63	56756.48	2.93	3.18	600	0.24
0.67	60333.94	3.76	4.08	600	0.24
0.71	63911.41	4.37	4.81	755	0.20
0.75	67488.87	5.33	5.86	1200	0.20
0.79	71066.34	4.80	5.20	1200	0.24
0.83	74643.80	4.41	4.78	1200	0.24
0.87	78221.27	2.74	2.97	1200	0.24
0.91	81798.73	2.01	2.52	1200	0.08
0.95	85376.20	3.58	4.17	400	0.12
0.99	88953.66	4.72	5.51	400	0.12
1.03	92531.13	5.45	6.13	400	0.16
1.07	96108.59	6.61	7.43	400	0.16
1.11	99686.06	7.29	8.02	400	0.20
1.15	103263.52	7.62	8.38	400	0.20
1.19	106840.99	8.26	9.30	600	0.16
1.23	110418.45	9.29	10.22	400	0.20
1.27	113995.92	10.75	11.82	400	0.20
1.31	117573.38	11.88	13.07	400	0.20

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Table 9. Parameters for optimal harness and converted power curve for $m^*=1.343$

Test Conditions		Optimal harness power[W]	Optimal converted power[W]	Stiffness K [N/m]	$\zeta_{harness} = \frac{c_{harness}}{2\sqrt{(m_{osc} + m_a)K}}$
U (m/s)	Re				
0.39	35291.69	1.04	1.21	400	0.12
0.43	38869.15	1.34	1.51	400	0.16
0.47	42446.62	1.68	1.90	400	0.16
0.51	46024.08	3.02	3.32	600	0.20
0.55	49601.55	4.45	4.89	1000	0.20
0.59	53179.01	6.35	6.99	1000	0.20
0.63	56756.48	8.22	8.90	1000	0.24
0.67	60333.94	9.28	10.05	1000	0.24
0.71	63911.41	9.24	10.01	1200	0.24
0.75	67488.87	8.38	9.22	1200	0.20
0.79	71066.34	4.89	5.70	1200	0.12
0.83	74643.80	4.20	5.25	1200	0.08
0.87	78221.27	3.55	4.44	1200	0.08
0.91	81798.73	5.63	6.57	1200	0.12
0.95	85376.20	6.80	7.93	1200	0.12
0.99	88953.66	9.22	7.91	1200	0.12
1.03	92531.13	9.77	8.52	1200	0.12
1.07	96108.59	10.28	8.99	1200	0.12
1.11	99686.06	9.92	10.91	1200	0.20
1.15	103263.52	9.54	10.33	600	0.24
1.19	106840.99	9.94	10.77	600	0.24
1.23	110418.45	11.98	13.18	1200	0.20
1.27	113995.92	14.21	15.63	1200	0.20
1.31	117573.38	16.80	18.48	1200	0.20

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Table 10. Parameters for optimal harness and converted power curve for $m^*=1.685$

Test Conditions		Optimal harness power[W]	Optimal converted power[W]	Stiffness K [N/m]	$\zeta_{harness} = \frac{c_{harness}}{2\sqrt{(m_{osc} + m_a)K}}$
U (m/s)	Re				
0.39	35291.69	-	-	-	-
0.43	38869.15	1.05	1.18	755	0.04
0.47	42446.62	1.91	1.73	400	0.20
0.51	46024.08	2.31	2.50	400	0.24
0.55	49601.55	3.51	3.95	755	0.16
0.59	53179.01	5.05	5.47	755	0.24
0.63	56756.48	6.08	6.58	755	0.24
0.67	60333.94	7.15	7.75	1000	0.24
0.71	63911.41	8.45	9.15	1000	0.24
0.75	67488.87	10.27	11.12	1200	0.24
0.79	71066.34	11.36	12.31	1200	0.24
0.83	74643.80	12.55	13.60	1200	0.24
0.87	78221.27	11.86	12.84	1200	0.24
0.91	81798.73	7.36	8.59	1200	0.12
0.95	85376.20	7.99	8.79	400	0.20
0.99	88953.66	9.40	10.34	400	0.20
1.03	92531.13	9.95	11.61	755	0.12
1.07	96108.59	12.04	13.54	755	0.16
1.11	99686.06	12.51	13.76	755	0.20
1.15	103263.52	14.00	15.40	755	0.20
1.19	106840.99	16.16	17.50	755	0.24
1.23	110418.45	18.14	19.96	1200	0.20
1.27	113995.92	19.36	20.97	755	0.24
1.31	117573.38	19.95	21.95	1200	0.20

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