EFFECT OF TIP-FLOW ON VORTEX INDUCED VIBRATION OF CIRCULAR CYLINDERS FOR $Re < 1.2 * 10^5$

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ABSTRACT

In two-dimensional experimental setups, tip-flow cannot be eliminated completely. In one degree-of-freedom Flow Induced Motions (FIM) of circular cylinders placed perpendicular to a uniform flow, three-dimensional effects may become significant. An ideal setup extends the cylinder to the limits of the flow-channel to minimize tip vortices, which reduce the effective length of the cylinder. Depending on how close to two-dimensional the experimental setup is, obtained results may differ. It is difficult to avoid the tip-flow in nature as well. Applications involving Vortex-Induced Vibrations (VIV) have more or less three-dimensional flow characteristics and one of the manifestations of three-dimensionality is the tip-flow. In this paper, the effects of tip-flow on VIV are investigated both experimentally and computationally. It is found that the tip-flow reduces the lift force exerted on the cylinder and narrows down the range of synchronization. Two-dimensional computational simulations become insufficient to grasp the effects of the tip-flow for a cylinder in VIV as the Reynolds number increases. Computational results for vortex-induced vibrations at these relatively high Reynolds numbers (up to 1.2×10^5) in the TrSL3 flow regime are not satisfactory when compared with experimental results. To improve the CFD predictions by introducing three-dimensional (3D) flow characteristics in a two-dimensional (2D) computational environment, a parameter called *tip-flow correction factor* is defined and analyzed. This parameter is introduced to compensate for any deviations from 2D flow approximation that might arise due to the 3D nature of the flow. The tip-flow correction factor is implemented as a multiplier of the force term in the vibration equation to represent the lift-force losses caused by the tip vortex. When compared to the results obtained with straightforward use of the vibration equation, it is found that the tip-flow correction factor improves the agreement between 2D computational results and experimental measurements. This

method extends the validity of 2D-URANS simulations at least up to $Re = 1.2 * 10^5$ for which experimental results are available in this study.

Keywords: tip-flow correction, vortex induced vibrations, computational fluid dynamics, two-dimensional approximation, experimental validation.

NOMENCLATURE

c	mechanical damping	L_e	effective length of the cylinder
C_a	added mass coefficient	m^*	mass ratio
C_y	lift coefficient	m_{osc}	oscillating mass
D	cylinder diameter	U^*	reduced velocity
$f_{n,w}$	natural frequency in still water	U	flow velocity
$f_{n,v}$	natural frequency in vacuum	w_c	width of the test section
f_{osc}	oscillation frequency	α	tip-flow correction factor
F_y	fluid force exerted on the cylinder	ρ	fluid density
k	spring stiffness	ϕ	phase difference
L	cylinder length	ζ	damping coefficient

1. INTRODUCTION

A two-dimensional flow is less complex compared to the corresponding three-dimensional flow. The former lacks cross-flow, cellular shedding and tip-vortices, which complicate the flow and becloud our understanding of what is the physically underlying phenomenon. For those reasons, experimental setups are usually designed so that they minimize the three-dimensional effects of the flow, thus allowing isolation of the fundamental aspects of the phenomenon or flow under consideration. Two-dimensional flows are preferred not only in experimental measurements but in computational simulations as well, as the computational cost decreases substantially and the stability of the computational solution increases in 2D analysis.

Although a two-dimensional approach has many advantages, both in terms of understanding the underlying flow and simplifying the computational approach, some aspects of the flow are lost due to this assumption. It is impossible to completely eliminate three-dimensionality in laboratory setups because a moving cylinder cannot be extended to the boundaries of the flow domain, which are the walls of the recirculating channel. There is no established standard on eliminating three-dimensionality of flows in experiments. The traditional approach is to try to extend the cylinder to the limits of the flow regime or attach large endplates. Some researchers improve two-dimensionality of flows by the addition of endplates (Govardhan and Williamson, 2000; Hover et al., 2004) while the others minimize the gap between the boundary of the flow regime and the cylinder (Feng, 1968; Park et al., 2013). This uncertainty in experiments leads to modeling discrepancies making it harder to evaluate and compare results from different tests. For example, two dimensional flow assumption does not allow investigation of the forces (Hover et al., 1998) and the vortex pattern (Techet et al., 1998) of tapered cylinders. Another example would be the tip effect, which is usually neglected by many researchers but has a strong impact both on the maximum amplitude and the range of synchronization for a circular cylinder in VIV. The fundamental challenge addressed in this study is to provide an explanation of the difference between experimental results due to three dimensional flow effects for cylinders in vortex-induced vibrations. Hover et al. (2004) reported that low-aspect ratio cylinder VIV is dominated by the end-effects and aspect ratios of over 70 are required to observe a clear two dimensional vortex shedding at the mid-span of the cylinder. Norberg (1994) covered a wide range of Reynolds numbers for the flow around circular cylinders and concluded that cylinders that have aspect ratios less than 7 have “bi-stable flow switching between regular vortex shedding and irregular flow” in the subcritical regime.

The scope of this study is to understand the effects of tip-flow in vortex-induced vibrations. Experimental demonstration of tip-flow is discussed in Section 2 and field-test verification is provided in Section 3. Mathematical formulation and CFD simulations for VIV are discussed in Sections 4 and 5, respectively. Results are presented in Section 6 and insight into the use of the tip-flow correction factor in Section 7. Conclusions are presented at the end of the paper.

2. EXPERIMENTAL DEMONSTRATION OF TIP-FLOW AND ITS EFFECTS

The free end of a cylinder can cause the flow perpendicular to the cylinder axis to be strongly 3D rendering basic 2D-flow assumptions invalid (Sumner et al. 2004). They reported in their study that the tip flow from the free ends can affect the whole flow around cylinders that have smaller aspect ratios. Their study covers the flow around stationary cylinders showing the different wake modes of cylinders with different aspect ratios. Schematic of the flow around the tip of a cylinder is given in Figure 1 (Sumner, 2013).

Figure 1. Illustration of tip-flow for a stationary cylinder in steady, uniform flow (Sumner, 2013)

Different experimental setups may generate different forms of amplitude graphs even for the same parameters used. Figure 2 reveals the amplitude curves generated by Feng (1968) and Govardhan and Williamson (2000) for the mass-damping parameter $(m^* + C_A) \cdot \zeta = 0.251$ taken from Govardhan and Williamson (2000). Their results varied from Feng's (1968) results even at the same mass damping parameter. The difference between the two experimental setups was that Govardhan and Williamson (2000) used end-plates “to encourage two-dimensional shedding” and reduce three-dimensionality effects. They have stated that the difference in results might be due to the fact that no end plates were used in Feng's (1968) experiments. The comparison of their results is shown in Figure 2.

Figure 2. Smooth cylinder in VIV in steady uniform flow. Comparison of amplitude response with respect to non-dimensional flow velocity. Govardhan and Williamson (2000) in circles and Feng (1968) in squares. The former attributed the difference to their use of end-plates to encourage two-dimensionality. $(m^* + C_A) \cdot \zeta = 0.251$

The utilization of end plates at the tips of the cylinders reduce 3D effects resulting in a broader range of synchronization in Govardhan and Williamson's (2000) experiments.

Another experimental demonstration of the effect of the tip-flow on VIV is provided by comparing the results of Park et al. (2013) using a VIV oscillator with very low 3D tip-flow in the old Low Turbulence Free Surface Water (LTFSW) Channel of the Marine Renewable Energy Laboratory (MRELab) of the University of Michigan with the experiments made in the new LTFSW Channel using a VIVACE (Vortex Induced Vibration for Aquatic Clean Energy – please refer to Bernitsas et al. (2009) for detailed explanation of the system) oscillator that has strong 3D tip-flow. Park et al. (2013) conducted their Flow Induced Motion (FIM) tests in the older version of the Channel with the VIV oscillator, which has struts above the water to position the cylinder and springs in place. The cylinder tips are located less than a ¼" from the Channel walls. The new LTFSW Channel of the MRELab has a deeper test-section to minimize free-surface effects and allow for larger amplitude oscillations of up to 7 diameters (D) for a cylinder with $D=3.5$ " (0.0889m). Such high amplitudes are typical of flow in the TrSL3 regime in comparison with the TrSL2 flow regime, where VIV amplitudes of 1D are expected. In these experiments, instead of a VIV oscillator a VIVACE Converter was used, with cartwheels and struts inside the water channel, making 3D tip-flow strong.. The two oscillators are extreme cases in terms of tip-flow effect. On one hand, the VIV oscillator has minimal tip-flow effect since the entire channel wall serves as end-plate and it is not even in motion with the cylinder. On the other hand, the converter oscillator has very strong tip-flow as all support struts and cartwheels including springs, belts, and pulleys are placed in the support struts, which have a rectangular (non-fared) profile and cross-section to the flow.

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. Accordingly; negative lift is induced by the two ends of the cylinder exposed to tip flow, while positive lift is generated by the rest of the cylinder. Positive lift enhances the oscillation and negative lift tries to suppress it. Figure 3 shows schematically the old setup with the VIV oscillator and the new setup with the VIVACE Converter. The parameters of the oscillators used in the experiments in both setups are summarized in Table 1. Although the parameters are not exactly equal, it is believed that they are sufficiently close to illustrate quantitatively the effect of the tip-flow.

Figure 4 compares the results of tests of the VIV oscillator vs. the VIVACE Converter. As it can be observed from this figure, the maximum amplitude and the range of synchronization obtained with the converter setup is quite different than that of obtained by Park et al. (2013). The reason for this is accounted to the renovated setup having relatively smaller cylinder effective length. Therefore, the flow leaks from the tips and does not contribute to lift generation. A photograph taken from the renovated lab is provided in Figure 10.

Table 1. Experimental parameters used for comparison in different lab setups.

	Old Channel and VIV oscillator Park et al. (2013)	New Channel and VIVACE Converter
m^*	1.725	1.685
$f_{n,w}$	1.1183	1.0215
ζ	0.0158	0.02

Figure 3. Schematic view of the VIVACE Converter used in the new LTFSW Channel and the VIV oscillator used in the older LTFSW Channel.

Figure 4. Comparison of VIV response of a smooth cylinder using VIV oscillator and VIVACE Converter with parameters shown in Table 1. ($m^* \approx 1.7$, $f_{n,w} \approx 1.1$, $\zeta \approx 0.02$)

3. FIELD EXPERIENCE ON THE EFFECTS OF TIP-FLOW

When the flow perpendicular to the cylinder axis is not uniform along the entire span (length) of the cylinder, the cylinder may not even achieve VIV. Then, the shedding vortices from the cylinder will not reach full correlation along the cylinder span. The parts of the cylinder exposed to a non-uniform or even transient flow may not be subjected to positive lift in which case the force transversely to the flow may be negative lift opposing VIV or FIM in general. Such a case occurred in field-tests of the converter oscillator in a canal in The Netherlands conducted by TAUW and Vortex Hydro Energy (VHE) in 2013. Figure 5 shows the converter oscillator in full swing, at 2.5knots, in the towing tank of the University of Michigan.

This VIVACE hydrokinetic energy converter was installed in a canal in The Netherlands, where the flow velocity was adjusted by varying the opening of a gate. The first installation was unsuccessful and the cylinder did not oscillate. After closer examination of the flow, it was found that the wall opening, where the oscillator was placed, would not allow a 2D flow perpendicular to the cylinder. The flow deviated from 2D in three ways. First, the depth of the opening (in the flow direction) was no bigger than the device. Thus, the flow was uniform in less than about 1/3 of the opening from top to bottom. That is, the middle 1/3 of the cylinder was be subjected to positive lift inducing VIV, while the other 2/3 of the cylinder would subjected to negative lift suppressing VIV. Second, for the same reason, the cylinder was subjected to positive lift only when centered in its frame in about the middle 1/3 of the swing. Third, the free surface boundary effect allowed for very strong 3D effect resulting in the vertical whirl in Figure 6. This resulted in a smaller cylinder effective length and the generated lift due to vortex shedding was not enough to move the cylinder.

Figure 5. VIVACE converter oscillator shown during VIV oscillations in the Towing Tank at the University of Michigan; towing speed of 2.5 knots.

To improve the flow around the converter oscillator, openings were cut out symmetrically to the left and right of the oscillator. The main idea behind this was to subject the cylinder to a more uniform flow so that oscillations could be induced. The cylinder started oscillating in VIV and at higher speeds in galloping in spite of the fact that only its middle $1/3$ was subjected to positive lift while $1/3$ from each tip was subjected to negative lift. The cutouts allowed for the cylinder to be subjected to a uniform flow inducing motion over more than $1/3$ of the horizontal travel of the cylinder. The cutouts increased the efficiency of the usage of the cylinder span, which resulted in positive lift over a bigger fraction of the cylinder travel. Figure 7 demonstrates the implementation of the cutouts in the second trial.

Figure 6. Upstream view of the flow *without wall cut-outs* in a canal, The Netherlands; courtesy of TAUW and VHE.

Figure 7. Downstream view of the flow *with wall cut-outs* in a canal, The Netherlands courtesy of TAUW and VHE.

4. USE OF CFD FOR VIV OF CIRCULAR CYLINDERS

In many engineering applications, three-dimensional (3D) flows are simplified and confined in one direction to approach the solution from a two dimensional (2D) perspective. This also holds true for flow-induced motions. With this assumption, though, important 3D aspects of the flow may be left out of the problem. A 2D approach for vortex induced vibration (VIV) will miss the effects of cross flows, cellular shedding, and tip vortex, which will result in discrepancies between CFD calculations and experimental measurements and inefficacy in capturing the real physics of the flow. Williamson (1996) discusses the 3D wake instabilities playing an important role in the whole flow for flows $Re > 200 - 300$. Zhao et al. (2014) made both 2D and 3D CFD simulations on VIV and noted that 3D simulations return more accurate results. Shur et al. (2005) stated that 2D problems are just models for real 3D flows and 2D approach is a “strategic” choice if the resulting accuracy is within the acceptable limits. This is the underlying reason behind many CFD based papers to be limited to very low Reynolds numbers. 3D flow requires high computational capabilities and CPU time and, thus, 2D flow assumptions are rather preferred in design where extensive parametric analysis is required. At higher Reynolds numbers, it is even harder to derive meaningful results by 2D CFD, per Saltara et al. (2011).

CFD methods offer quite extensive and useful ways to calculate the amplitude of a cylinder in FIM. An experienced CFD user with good computing capabilities might build up a three-dimensional model and to solve the problem and assess quantitatively the drawbacks associated with two-dimensional approaches. However, this would be costly and time consuming. Dong and Karniadakis (2005) have managed to solve 3D VIV at $Re = 10^4$ with Direct Numerical Simulation (DNS) and used parallel computing to tackle the high computational cost. In his review on vortex induced vibrations, Sarpkaya (2004) quotes Michelassi et al. (2003) stating that

the DNS and Large Eddy Simulation (LES) must be used to investigate the wake – boundary layer interaction with a greater insight. Although Unsteady Reynolds-Averaged Navier-Stokes (URANS) simulations may not provide accurate information at the boundary layer scale, they provide accurate results at the von Kármán scale on wake vortex modes of the cylinder, amplitude, lift coefficient, or oscillation frequency under certain conditions. For example, using turbulence stimulation Wu et al. (2014), Ding et al. (2013a, 2013b, 2015) achieved very good agreement between 2D-URANS simulations and experimental measurements.

CFD based papers in the literature cover VIV simulations up to $Re = 10^4$ for smooth cylinders in VIV. It is harder to go farther because the 3D wake instabilities play an important role in the flow after $Re > 200 - 300$ as stated by Williamson (1996). As the Re number is increased and 2D CFD approach is deployed, eventually the results generated by the CFD approach become less accurate. Experiments of Khalak and Williamson (1996) are usually used as a benchmark to evaluate CFD methods for $Re < 10^4$. Many authors in the literature validated their computational studies by their experiments. Pan et al. (2007) used the $k - \omega$ turbulence model but could not manage to capture the upper branch. Capturing the upper branch with RANSE seems to be problematic but Guilmineau and Queutey (2004) partially managed to get it by introducing the *increasing velocity condition*. Wanderley et al. (2008) used the $k - \varepsilon$ turbulence model to find a good match between their results and the experiments. Wu et al. (2014) also partially managed to get the upper branch by using the Spalart – Allmaras turbulence model. However, these experiments and all these related CFD studies have relatively low Re for industrial purposes. Risers and other offshore structures usually work in higher Re numbers, which can reach up to or exceed 10^6 , limiting the applicability of CFD to challenge the essential problems faced by the industry. One of the targets of this paper is to extend CFD applicability to

a higher Re number for a transversely oscillating smooth cylinder. However, 2D solution approach misses the vortices formed at the tips, as explained before. Saltara et al. (2011) also argued that as Re gets higher, the applicability of CFD codes to the problem becomes harder. The flow that escapes from the tips of the cylinder in the form of vortices, disables its ability to contribute to lift generation. Direct application of the vibration equation in Section 5 will include these dead zones eliminating from the lift force model the parts of the cylinder that do not contribute to lift. This process leads to the *tip-flow correction factor* which can be calculated by CFD simulations. This parameter is then introduced into the vibration equation.

5. MATHEMATICAL FORMULATION OF VORTEX INDUCED VIBRATION

To solve for the response of a smooth circular cylinder in VIV, a CFD approach using 2D-URANS method is implemented. The details of the computational results generated are provided in this section.

5.1. Theoretical background

The vibration equation for a cylinder oscillating in a fluid is:

$$m_{osc}\ddot{y} + c\dot{y} + ky = F_y(t) \quad (1)$$

To accurately predict the amplitude A of an oscillating body, the displacement $y(t)$ at each time step must be calculated. F_y is the fluid force exerted on the cylinder and it is a function of time t . The system damping is denoted by c and k is the spring stiffness. If we assume that the cylinder is subjected to a steady fluid flow it will apply a force F_y on the cylinder. It is possible to

calculate the pressure distribution by solving 2D-URANS equations where the applied fluid force is calculated by integrating the pressure distribution on the cylinder. The $k - \omega SST$ turbulence model is used to solve using the 2D-URANS approach. URANS stands for unsteady RANS and has been a powerful tool to investigate VIV response of cylinders in fluid flow. This model is a two-equation turbulence model especially suitable for solving flows around bluff bodies, which we generally expect to have a high adverse pressure gradient. The details and some applications regarding the compatible results produced with this turbulence model are given by Menter (1994).

5.2.Numerical implementation

The right hand side of Equation (1), which involves the force acting on the cylinder, is mathematically solved at each time step. This force moves the cylinder transversely to the fluid flow and the cylinder axis. The new position of the cylinder is determined by solving for the displacement $y(t)$ in the same equation. A user defined function (UDF) that numerically calculates the displacement at each time step is developed and implemented in ANSYS-Fluent. A finite difference method is used to calculate the displacement, the velocity, and the acceleration of the cylinder. The UDF is applied in four steps:

- (a) Define the system of the oscillator in water using the parameters: $m_{osc}, R, \rho, C_a, c, k, \Delta t$.
- (b) Set the initial velocity and the position of the cylinder.
- (c) Use 2D-URANS equations to solve for the fluid force F_y exerted on the cylinder at time t .
- (d) Determine the new velocity and the new displacement of the cylinder required in the new iteration.

About 12,000 elements are used in the whole fluid domain at the start of the simulation with a

two dimensional finite volume method. Depending on the response amplitude of the oscillating cylinder, the number of elements may increase. The $k - \omega$ SST turbulence model is implemented with SIMPLE numerical scheme. Momentum, turbulent kinetic energy and turbulent dissipation rate are all selected to be first order upwind. Due to the requirements of the turbulence model, y_{max}^+ is set to be around 10 and the CFL number is set to be 1 although an implicit time integration method is used. A real-time simulation of 10 seconds takes around 1.5 hours of computation time with an Intel Xeon CPU E5-2630@2.30GHz. The workstation that is used for these simulations has a 64-bit operating system with 64GB of installed memory (RAM). The code uses 12 cores with 2 processors at each core.

The reason for selecting first order upwind numerical scheme rather than second order is due to the better results obtained. VIV is a fluid-structure interaction problem with a couple of different frequencies involved in it. A slight miscalculation of phase difference varies the results greatly. The difference in the results obtained between a first order and a second order upwind numerical scheme is explained in Section 7.

5.3. Mesh structure

The cylinder is a bluff body that creates turbulent fluid flow in its wake. The wake is completely turbulent at higher Reynolds numbers and, thus, a high resolution mesh structure is required to accurately model the fluid flow. The density of mesh structure should be finely tuned with respect to the demands of the used turbulence model. The grid used in this study is given in Figure 8. Quadrilateral elements are used in the inner domain that surrounds the cylinder and its wake while triangular elements are used for the rest of the domain. The inner domain has 5D (D represents the diameter) height and 20D width (including the wake region). There are around 12000 elements (7000 triangular, 5000 quadrilateral) but this number is subject to change as the

dynamic mesh structure remeshes the outer domain (which has triangular elements) at each time step. The outer region is deformable while the inner domain is not. The inner domain is allowed to move with the cylinder so that the elements near the cylinder have high quality at any time in the simulation.

Figure 8. Grid structure around a circular cylinder subjected to VIV.

While setting the density of the mesh, special attention to Courant-Friedrichs-Lewy (CFL) condition is given because the time step size decreases with respect to increasing mesh density.

CFL condition dictates that

$$C = \frac{u \cdot \Delta t}{\Delta x} + \frac{v \cdot \Delta t}{\Delta y} \leq C_{max} \quad (2)$$

C_{max} changes with the method used to solve the problem. The general tendency is to set this value close to 1 (ANSYS Fluent Manual).

6. 2-D CFD SIMULATIONS FOR 3-D REAL FLOWS

Comparing the results of the experiments directly with the results generated by CFD may not be of value unless we are comparing results under similar conditions. Two-dimensional flows that are simulated with CFD are assumed to be perfectly 2D flows, where there is no leakage of flow from the tips. However in real 3D flows, cylinders do not cover the total width of the lab setup. Even large endplates, which limit tip-flow, introduce other problems such as increased friction drag. In order to allow the cylinder to move, the ends of the cylinder are let free so the flow that escapes from the tips cannot be avoided.

6.1. *Tip-flow correction factor*

In order to compensate for this specific three-dimensionality issue (the tip-flow) in experimental setups, the tip-flow correction factor is introduced. The effective length of the cylinder over the test section width (L_e/w_c) is less than 1 in real flows while it is strictly 1 in two-dimensional flows that are assumed to be ideally 2D. Therefore, the test section was modeled in a 3D environment first and the streamlines following the cylinder surface were counted at a specified Reynolds number. This Reynolds number is selected so that it falls in the range of measured data. After finding the effective length of the cylinder this way, the ratio of this value to the total width of the test section was found and used as the tip-flow correction factor of the experimental setup. Multiple values of Reynolds number can be used. So Equation (1) becomes;

$$m_{osc}\ddot{y} + c\dot{y} + ky = \alpha \cdot F_y(t) \quad (3)$$

where

$$\alpha = L_e/w_c \quad (4)$$

α is the tip-flow correction factor, L_e is the effective length of the cylinder, and w_c is the width of the test section. It should be noted that the length of the cylinder not subjected to the uniform flow perpendicular to the axis of the cylinder may contribute to positive or negative lift.

Figure 9. Graphical representation of the effective cylinder length for a stationary cylinder using 3D CFD. (a) $Re = 25,000$ (left), (b) $Re = 75,000$ (right).

A more advanced tip-flow correction factor would include the Reynolds number dependence of the flow around the cylinder as investigated in Figure 9. 3D CFD simulations for flows at $Re = 25,000$ and $Re = 75,000$ were generated to observe possible effects that the Reynolds number may have on the effective cylinder length. Comparing the two cases in Figure 9, it is observed that the change in the effective length is negligible. It should be noted though, that both values selected fall in the TrSL3 flow regime with similar values of formation length. It is possible that over the entire range of Re values the dependence of the tip-flow factor may be stronger. The

portion of the cylinder length, which is considered to be “*effective*”, is where the velocity along the cylinder length does not exceed 10% of the fluid flow velocity.

6.2. Lab setup

All experiments were conducted in the old and new Low Turbulence Free Surface Water (LTFSW) Channels of the Marine Renewable Energy Laboratory (MRELab) of the University of Michigan. The Channel was renovated in 2014 and new oscillators were built as well. The old oscillators were VIV oscillators with small tip-flow leakage, but limited to three-diameter amplitudes by the depth of the old Channel.

The new oscillators were part of the VIVACE Converter model for energy extraction with much higher tip-flow effects, but up to seven-diameter amplitude allowed by the bigger depth of the test section. A virtual spring-damping system was developed and installed inside the struts (Figure 10) keeping the hydrodynamic force out of the control loop (Lee et al. 2011, Sun et al. 2015). Measurements of the setup are provided in the same figure. The struts are inside the water and take a considerable amount of space of the test cross-section. The width of the channel is $w_c = 100.8\text{cm}$ with 3.8cm wide struts at each side. The struts are attached to the plexiglass side-walls of the channel. There is 1.8cm of free space between the strut and the cylinder at each side. The length of the cylinder is 89.6cm . That is, the converter oscillators serve a different purpose than the VIV oscillators in the old channel. At the same time, comparison between these two extreme cases of tip-flow effect provides valuable information on tip flow.

Figure 10. LTFSW Channel and details of the experimental setup.

The effective length of the cylinder, which is generating lift, can be detected by studying the streamlines produced by the 3D steady CFD analysis (Figure 11). It is concluded that $15.8cm$ of the cylinder at each end do not contribute to generating lift, reducing the effective length of the cylinder. So the effective length, L_e becomes:

$$L_e = 100.8cm - 2 * (3.8cm + 1.8cm + 15.8cm) = 58cm \quad (5)$$

Substituting the effective length of the cylinder L_e and the Channel width w_c into Equation (4), the tip-flow correction factor α , for this experimental setup, is calculated to be

$$\alpha = 0.5752 \quad (6)$$

Figure 11. Effective cylinder length from 3-D URANS simulations for the new LTFSW Channel at $Re = 75,000$.

7. RESULTS FOR $3 \times 10^4 \leq Re \leq 1.2 \times 10^5$

The generated results using CFD simulations with and without the tip-flow correction factor are presented in this section. Comparisons are made with measurements from the experiments conducted in the new LTFSW Channel of the MRE-Lab of the University of Michigan. The parameters used in the experiments are summarized in Table 2. The amplitude and the frequency response obtained with URANS, both with and without the tip-flow correction factor, are given in Figure 12 and Figure 13, respectively. It should be noted here that the amplitude is non-dimensionalized by the diameter and the frequency by the natural frequency in still water. The

amplitude used in the experiments is the average of the sixty highest positive or negative peaks after discarding the transient phase.

Initial, upper and lower branches, both with and without the tip-flow correction factor, are captured with URANS as can be clearly seen in Figure 12. For both CFD results, the upper branch starts at $U^* = 4$ while in the experiments it starts at around $U^* = 5$. CFD calculates the maximum amplitude around $A^* = 0.9 - 1$ and cannot capture the achieved value in the experiments which is around $A^* = 1.2$. The cylinder response in FIM (VIV and galloping) can be captured accurately when passive turbulence control is used as shown by (Wu et al. 2013, Ding et al. 2013). The tip-flow correction factor, though, which aims to account for the three-dimensionality in the flow, has a significant effect in correcting the range of synchronization.

The bias of results between the experiments and CFD in the upper branch can be in relation with the implementation of a first order spatial approximation numerically. A second order upwind numerical scheme was also tested with our code and we obtained higher lift exerted on the cylinder. This might lead to the idea that higher fluid force would increase the maximum amplitude achieved; however phase difference between the displacement and the vortex shedding was not properly simulated with a second order spatial approximation. As a result, we obtained lower amplitudes in the upper branch as opposed what we have been expecting. On the contrary, Wanderley et al. (2008) have obtained a perfect match with the benchmark experiments of Khalak and Williamson (1996) using a higher order TVD numerical scheme. They have accurately captured the upper and lower branches for a cylinder in VIV in the TrSL2 flow regime. Our study covers a flow regime which is inherently different than TrSL2. The Reynolds number range covered in this study is substantially higher than the range covered by Wanderley et al. (2008). Raghavan and Bernitsas (2010) investigated the Reynolds number effect on VIV

and concluded that there is a significant dependence of response of cylinders in VIV on Reynolds number. Even though the basic branches of VIV exist both in the TrSL2 and TrSL3 flow regimes, their nature is very different. Although current state of research is able to numerically predict the VIV response of a cylinder at relatively low Reynolds numbers accurately, it is still a challenge today to exceed the Reynolds number of around 10,000. As Sarpkaya (2004) states, *“it appears that the question of three-dimensional numerical simulation of VIV even for a smooth circular cylinder in the range of Reynolds number from a few thousand to over a million will remain unresolved indefinitely. Higher order modes caused by secondary instabilities resulting from the changes in phase and amplitude in self-excited oscillations will certainly complicate the simulations with any turbulence model in any indirect numerical simulation”*.

Table 2. Oscillator particulars for experiments.

Cylinder Diameter	D	8.89cm
Cylinder Length	L	89.535cm
Total oscillating mass including 1/3 of the spring mass	m_{osc}	7.45kg
Mass ratio	m^*	1.34
Natural frequency in still water	$f_{n,w}$	1.0812
Total damping ratio	ζ	0.06

Developing a method to post-process FIM data utilizing the tip-flow correction factor in the vibration, would eliminate the need for end-plates in Govardhan and Williamson’s experiments (2000) which introduce skin friction, or the need for clean cylinder ends as in the VIVACE Converter oscillator, or the need for near-contact of the cylinder ends with the Channel walls as in the VIV oscillators.

Figure 12. Amplitudes obtained by URANS with and without the tip-flow correction factor for smooth cylinder; $m^* = 1.34$, $f_{n,w} = 1.0812$, $\zeta = 0.06$.

Figure 13. Frequency response obtained by URANS with and without the tip-flow correction factor for smooth cylinder; $m^* = 1.34$, $f_{n,w} = 1.0812$, $\zeta = 0.06$.

Without the tip-flow correction factor, the cylinder goes into de-synchronization later which seems to happen around $U^* = 12$. This is like the ideal two-dimensional flow, which is hard to accomplish experimentally. The experiments return a much narrower range of synchronization and the additional parameter that is presented in this study helps to narrow the calculated synchronization range. Figure 12 also points to the possibility that the actual value for the tip-flow correction factor may be lower than the value calculated above because the range of synchronization is still a little broader than the experimental results. This implies that a more accurate method of calculating the tip-flow correction factor can be developed. The main target of this work is to explain the need for this parameter computationally; therefore not a lot of effort is put into perfecting this value for the used tests. An in-depth analysis of the calculation of this parameter would include integration of the pressure around several stations along the span of the cylinder to determine which cross-sections provide lift and which oppose it. The negative lift (drag in the lift direction) produced by the tip-zones would be subtracted from the generated lift by the rest of the cylinder.

The tip-flow correction factor compensates for three-dimensionality affects in two-dimensional CFD simulations. Without this parameter, the generated results would be valid only for ideal case of 2D flows. Examining Figure 12, one may observe that if the mass ratio of the cylinder was a little bit smaller, it would be in *infinite resonance*. The infinite resonance regime was determined experimentally by Govardhan and Williamson (2000) to occur for $m^* < 0.54$. This value is debatable since it might not be possible to reproduce the same result at each laboratory depending on how close to ideal the experimental setup is.

The oscillations and the lift coefficient that is exerted on the cylinder by the fluid flow are given in Figure 14. Vortex shedding modes are presented by the vorticity distribution in the fluid

domain. Figure 15 shows the vortices at each reduced velocity in the wake of the cylinder. At $U^* = 3.5$, the cylinder is in the initial branch. The lift coefficient and the cylinder motion are in accordance with each other and apparent oscillations start. The vortex mode is 2S. At $U^* = 5$ which corresponds to the critical value of $St = 0.2$ (the vortex shedding frequency for stationary cylinder), the cylinder and the vortex shedding are in resonance which makes the cylinder move into the upper branch in the amplitude graph (Figure 12). It can be said that the vortex mode is 2P but the second vortex is very weak and disappears quickly. $U^* = 9$ is an example falling in the lower branch. The lower branch at these relatively high Reynolds numbers (TrSL3 flow regime) is actually overtaken by the upper branch (Bernitsas et al., 2009; Raghavan and Bernitsas, 2010) and differs from Khalak and Williamson's (1997) conclusion about the branches in the amplitude curves in the TrSL2 flow-regime. However, the vortex shedding mechanism and the relation between the lift force and the displacement are in agreement. The lift force generated by the fluid flow is out of phase with the cylinder movement and therefore the amplitude is suppressed when compared to the upper branch. The vortex mode is expected to be 2P in this branch but CFD generates 2S. At $U^* = 11$, the cylinder is in de-synchronization. The vortices shedding from the cylinder are much stronger increasing the lift coefficient, which is totally out of phase with the cylinder. This restricts the oscillations and eventually the vibrations start to die out at higher reduced velocities.

Figure 14. Fluid force vs. smooth cylinder motion in VIV at different reduced velocities at $m^* = 1.34$, $f_{n,w} = 1.0812$ and $\zeta = 0.06$.

Figure 15. Vorticity field at different reduced velocities corresponding to Figure 14. Smooth cylinder in VIV at different reduced velocities; $m^* = 1.34$, $f_{n,w} = 1.0812$ and $\zeta = 0.06$.

8. HOW THE TIP-FLOW CORRECTION FACTOR NORMALIZES RESULTS

To further investigate the effectiveness of the tip-flow correction factor, it is hypothesized that implementing it in the simple FIM math model used in the literature will improve the agreement between calculated amplitude curve and experimental measurements. When the tip-flow correction factor, α , is introduced into the mathematically derived amplitude equation, the hypothesis is that there is no need to introduce this new parameter into new CFD simulations. The hypothesis further holds that it is reasonable to expect that the tip-flow correction factor would modify the lift coefficient in the math model, so that its predictions would match experimental data. To test this hypothesis, we first have to derive the mathematical model.

Algebraic transformation of Equation (1) gives:

$$m_{osc}(\ddot{y} + \frac{c}{m_{osc}}\dot{y} + \frac{k}{m_{osc}}y) = F_y \quad (7)$$

Introducing damping ratio ζ_v and natural frequency in vacuum $\omega_{n,v} = 2\pi f_{n,v}$ in Equation (7) gives:

$$m_{osc}(\ddot{y} + 4\zeta_v\pi f_{n,v}\dot{y} + 4\pi^2 f_{n,v}^2 y) = F_y \quad (8)$$

The force exerted on the body by the fluid due to vortex shedding in VIV, which is F_y , can be expressed in terms of a non-dimensional coefficient C_y and the phase difference ϕ between the force F_y and the body oscillation y at the frequency of oscillation of the body $\omega_{osc} = 2\pi f_{osc}$ as:

$$F_y = \frac{1}{2}\rho DLU^2 C_y \sin(2\pi f_{osc}t + \phi) \quad (9)$$

If it is assumed that the oscillatory motion of the body is purely sinusoidal, the displacement can be expressed as:

$$y(t) = A \sin(2\pi f_{osc} t) \quad (10)$$

Accordingly, the expressions for velocity and acceleration are:

$$\dot{y}(t) = A \omega_{osc} \cos(2\pi f_{osc} t) \quad (11)$$

$$\ddot{y}(t) = -A \omega_{osc}^2 \sin(2\pi f_{osc} t) \quad (12)$$

When Equations (9) – (12) are inserted into Equation (8) and rearranged they yield:

$$\begin{aligned} \sin(2\pi f_{osc} t) \cdot \left[-4m_{osc} A \pi^2 f_{osc}^2 + 4m_{osc} A \pi^2 f_{n,v}^2 - \frac{1}{2} \rho D L U^2 C_y \cos(\phi) \right] \\ + \cos(2\pi f_{osc} t) \cdot \left[8m_{osc} A \zeta_v \pi^2 f_{n,v} f_{osc} - \frac{1}{2} \rho D L U^2 C_y \sin(\phi) \right] = 0 \end{aligned} \quad (13)$$

To satisfy Equation (13) at all times, the coefficients of both the *sin* and *cos* terms have to be equal to zero. This yields two simultaneous equations:

$$-4m_{osc} A \pi^2 f_{osc}^2 + 4m_{osc} A \pi^2 f_{n,v}^2 - \frac{1}{2} \rho D L U^2 C_y \cos(\phi) = 0 \quad (14a)$$

$$8m_{osc} A \zeta_v \pi^2 f_{n,v} f_{osc} - \frac{1}{2} \rho D L U^2 C_y \sin(\phi) = 0 \quad (14b)$$

When the non-dimensional terms $A^* = A/D$, $m^* = m_{osc}/(\pi D^2 L \rho/4)$, $f^* = f_{osc}/f_{n,v}$ and $U_v^* = U/f_{n,v} D$ are introduced into Equations (14a) and (14b) and rearranged, the following two equations with two unknowns (C_y and ϕ) are obtained.

$$C_y \cdot \cos(\phi) = \frac{2\pi^3 A^* m^* (1 - f^{*2})}{U_v^{*2}} \quad (15a)$$

$$C_y \cdot \sin(\phi) = \frac{4\pi^3 A^* m^* \zeta_v f^*}{U_v^{*2}} \quad (15b)$$

When the calculated CFD amplitudes and frequency without the tip-flow correction factor are introduced into Equations (15a) and (15b), C_y and ϕ can be calculated. As said before, the tip-flow correction factor basically affects C_y . If we denote the new lift coefficient by C_y'

$$C_y' = \alpha \cdot C_y \quad (16)$$

and use this new lift coefficient to calculate the new amplitude using Equation (15), the amplitude curve will move up but the range of synchronization will not change. The mathematically modified amplitude curve is given in Figure 16.

Figure 16. Mathematical modification of amplitude of a smooth cylinder in VIV using tip-flow correction factor; $m^* = 1.34$, $f_{n,w} = 1.0812$, $\zeta = 0.06$.

It is obvious from Figure 16 that the mathematically modified amplitude curve does not agree well with the experimental data. An interesting question here would be whether 2D-URANS could manage to change the range of synchronization while the math models are insufficient to correct it. When introduced into 2D-URANS, the tip-flow correction factor α takes into account the changes in the phase difference ϕ as well as the lift coefficient C_y ; whereas in the math model only C_y is changed.

Figure 12 shows the results of 2D-URANS with the tip-flow correction factor. Obviously ϕ is affected by C_y . It may be said that ϕ is a function of C_y but the existing literature does not have an equation relating these two parameters. This is the reason why the math model fails to grasp the experimental results when 2D-URANS is partially able to do so.

9. CONCLUSIONS

In this paper, 2D-URANS results for a smooth circular cylinder in VIV up to $Re = 1.2 \cdot 10^5$ are presented in comparison with experimental data. At these Re numbers the flow is highly 3D. Thus, the tip-flow correction factor was introduced to account for the effects of three-dimensionality in the results. Based on some field-tests and laboratory tests produced by different oscillators and experimental setups, it was concluded that the cylinder in VIV was unable to take full advantage of the lift force due to vortex shedding because of leaks from the tips. The cylinder tips do not contribute to lift and may even be subjected to negative lift opposing VIV. The tip-flow correction factor removes these deadzones from the calculations and compensates for the over-estimation of the lift force when used either with a 2D mathematical model or with 2D-URANS simulations by CFD.

2D-RANS simulations show that the tip-flow correction factor is not dependent on Reynolds number in the range covered in this study, which falls in its entirety in the TrSL3 regime. It is likely that it will depend on Re , when other flow regimes like TrSL2 are considered, given the differences in FIM between these two regimes.

The tip-flow correction factor has a different impact when used in mathematical models, which are linear, because tip-flow correction factor only modifies the lift applied on the cylinder by the fluid. It does not affect the phase difference, which limits its usage with the existing mathematical models.

When the tip-flow correction factor was introduced into 2D-URANS simulations, it affected the results in two ways; both by scaling the force and modifying the phase resulting in a modified amplitude and synchronization range.

In future research, more advanced computational methods and experimental measurements and visualization will be used to calculate the tip-flow correction factor more accurately.

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