

Passive Vibration Control of Pipe-in-pipe (PIP) Systems Subjected to Vortex Induced Vibration (VIV)

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ABSTRACT

This paper proposes using modified pipe-in-pipe (PIP) system to mitigate vortex induced vibration (VIV). Numerical simulations are carried out to examine the effectiveness of the proposed method. Firstly, a semi-empirical oscillator model is developed and validated by an experimental test of a single pipe. The validated model is then extended to the modified PIP system, which is simplified as a structure-tuned mass damper (TMD) system in present study. The governing equation of vibration is solved in the time domain using MATLAB/Simulink programming. The results demonstrate that the carefully designed PIP system can significantly suppress VIV of offshore cylindrical structures up to 84%.

KEY WORDS: *Pipe-in-pipe (PIP); VIV; vibration control; optimal design.*

INTRODUCTION

The ongoing energy industry is heavily involved in using offshore floating structures which conventionally comprise of many vertical and horizontal cylindrical components such as risers, conductors and pipelines. Many structural failures in these structures are associated with the fatigue damages caused by Vortex Induced Vibration (VIV) (Williamson and Govardhan, 2008; Kamble and Chen, 2016). Various vibration control methods have been then suggested to suppress such destructive vibrations. These methods can generally be divided into two broad categories: Passive (require no external power to operate) and Active (additional power input/sensors is required for operation) (Gadel-Hak, 2000). The former approach is more applicable, much less expensive and easier to install particularly in the field of offshore and marine structures. The literature on the passive vibration control of marine cylindrical structures is quite rich (Gabbai and Benaroya, 2005; Liming, Xingfu and Yingxiang, 2012). All proposed approaches can mitigate VIV to a certain extent. However, they have some limitations which can considerably affect their performances particularly in deep and ultra-deep waters. Table 1 summaries the advantages/disadvantages of the most well-known current VIV suppression devices (Owen, Bearman and Szewczyk, 2001; Kumar, Sohn and Gowda, 2008; Assi,

Bearman and Kitney, 2009; Kiu, Stappenbelt and Thiagarajan, 2011; Zhou, Razali, Hao and Cheng, 2011; Azmi, Zhou, Cheng, Wang and Chua, 2012; Sudhakar and Vengadesan, 2012; Bernitsas and Raghavan, 2014; Quadrante and Nishi, 2014; Zeinoddini, Farhangmehr, Seif and Zandi, 2015). It can be seen that many methods are still controversial from both structural and economical points of view. It is important to develop more efficient, cost effective and practical passive control devices.

In this study, an innovative PIP system is proposed to mitigate VIV. The proposed system takes advantage of the structural layout of traditional PIP system and uses optimized spring and dashpot to connect the inner and outer pipes. This system can then behave as a non-conventional structure-tuned mass damper (TMD) system, it therefore has the potential for vibration control. Analytical simulations are carried out to examine the effectiveness of the proposed method. To this end, the basic concept of modified PIP system is briefly introduced. Mathematical model of a single cylinder under vortex shedding is then presented. The equation of motion of this single pipe system is solved and validated by the experimental data. This single pipe system is then extended to the modified PIP system. The equation of motion of the modified system is derived and solved in order to examine the effectiveness of the proposed system is discussed.

CONCEPT OF MODIFIED PIP SYSTEM

There are generally two common types of pipe-in-pipe systems used in the offshore industry (Bi and Hao, 2016): (i) compliant PIP, in which the entire annulus is filled with insulation material, and (ii) non-compliant PIP, in which the insulation is achieved by wrapping standard size insulation pads onto the inner pipe. In the latter type, the inner and outer pipes can move relative to each other. The whole system has therefore the potential to be designed as a structure-TMD system to control different sources of vibrations. (Bi and Hao, 2016) has recently proposed modified PIP system in which the hard polymeric centralizers were replaced by optimized springs and dashpots to connect the inner and outer pipes (Fig.1). For the proposed pipe-in-pipe system, the outer pipe can act as the main system and the inner pipe can be considered as the TMD mass. The stiffness and damping of the main system are determined by the surrounding

environment. The optimized springs and dashpots provide stiffness and damping to the TMD system. They showed that by optimizing the spring stiffness and damping coefficient, the inner pipe can vibrate out of phase with the outer pipe and the vibration of the systems therefore can be significantly suppressed (Bi and Hao, 2016).

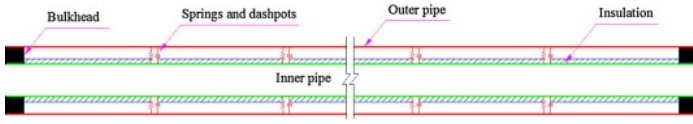



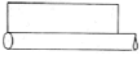
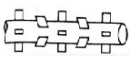
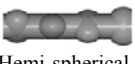


Fig. 1. Schematic drawing of modified PIP system (Bi and Hao, 2016)

(Bi and Hao, 2016) showed that modified PIP system can considerably mitigate vibration of pipelines when they are subjected to ground motions. In this paper, as a step forward, the effectiveness of the modified PIP system to control VIV is investigated.

Table 1. Advantages/disadvantages of some proposed passive control devices for suppressing VIV

Proposed Approach	Definition/Advantages	Disadvantage
 Helical strake	<ul style="list-style-type: none"> - widely-used in industry - simple and easy to install - reasonable cost 	<ul style="list-style-type: none"> - additional drag coefficient; - adversely affect the stress and then cause an increase in deflection and buckling - efficiency decrease with increasing the turbulence
 Fairing	<ul style="list-style-type: none"> - widely-used in industry - has higher efficiency than strake approach 	<ul style="list-style-type: none"> - expensive to manufacture; Difficult to handle and time consuming for installation - Susceptible to storm damage - Required rotation around the structure is a great challenge
 Perforated Shroud	<ul style="list-style-type: none"> - consist of a mesh screen cylinder which affects entrainment layers - can curb VIV up to 30-50% 	<ul style="list-style-type: none"> - is comparatively expensive - will lost its efficiency in the course of time because of growing marine growth - only appropriate in air
 Splitter Plate	<ul style="list-style-type: none"> - a flat plate attached to the rear of structure - can reduce drag reduction up to 40% 	<ul style="list-style-type: none"> - only appropriate where the current direction does not vary significantly - rotatable flat is also expensive and difficult to implement
 Spoiler/ribbon plates	<ul style="list-style-type: none"> - a number of spoiler plates attached on the cylinder - can reduce VIV up to 70% 	<ul style="list-style-type: none"> - will cause significant drag forces on the structure (then need more structural strength) - difficulty to maintenance
 Hemi-spherical bumps	<ul style="list-style-type: none"> - can reduce drag force up to 25% 	<ul style="list-style-type: none"> - high cost and difficult to construct - will lost its efficiency in the course of time because of growing marine growth

MATHEMATICAL MODELLING OF A SINGLE CYLINDER UNDER VIV

Extensive efforts have been made to systematically study the nature of VIV (Bearman, 1984; Pantazopoulos, 1994; Sarpkaya, 2004; Williamson and Govardhan, 2008). There are various approaches to investigate the VIV phenomena. The main method of numerical simulation is solving the equation of fluid motion, known as Navier-stokes, by direct numerical simulation (DNS) which is quite expensive in terms of computational cost and time consumption. Consequently, several alternative models have been also proposed to study the structures undergoing VIV such as Wake Oscillator Models (WOM), Single Degree of Freedom (SDOF) and force-decomposition models (Gabbai and Benaroya, 2005). The main difference between the alternative approaches with the DNS is that the time-dependant behaviour of the fluid at the rear of cylinder is modelled rather than being computed. In WOM, the displacement of mounted body and wake oscillations are coupled through the equations of motion and van der Pol, respectively (Hartlen and Currie, 1970; Facchinetti, De Langre and Biolley, 2004; Farshidianfar and Zanganeh, 2010). SDOF model employs a single ordinary differential equation to describe in-plane cross-flow oscillation of the body (Goswami, Scanlan and Jones, 1993; Chaudhury, 2011). In force-decomposition models, the fluid force is split into an excitation part and reaction part which defines all the motion-dependent force components (Griffin and Koopmann, 1977; Gabbai and Benaroya, 2005). The latter approach, in fact, is a SDOF which relies on the measurement of certain component of fluid forces from the experimental result (Gabbai and Benaroya, 2005). It can accurately explain and simulate experimental results and thus is of great interest to study the nature of VIV, especially when strong computational and practical limitations arise for numerical and experimental simulations. Moreover, it is believed that SDOF models are reasonably accurate for the purpose of evaluating maximum VIV response, which is sufficient for most wind and ocean engineering (Goswami, Scanlan and Jones, 1993).

The equation of transverse motion (y) for an elastically-mounted rigid cylinder of diameter (D) under VIV oscillations can be written as (Griffin and Koopmann, 1977):

$$m\ddot{y} + 2m\omega_n\zeta_s\dot{y} + m\omega_n^2y = F_{fluid}(t) = F_L - F_R \quad (1)$$

In which m is the mass of oscillating system, ω_n is natural frequency of system and ζ_s refers to the structural damping ratio. The term F_{fluid} on the right hand of the equation can be subdivided into two terms; time-dependent excitation or lift force F_L and fluid reaction F_R which represent the sum of fluid dynamic damping and potential added mass force. In non-dimensional form, the above equation can be re-written as follow:

$$\ddot{y} + 2\omega_n\zeta_s\dot{y} + \omega_n^2y = \beta\omega_s^2C_{L0}(t) \quad (2)$$

Where $C_{L0}(t)$ is the observed time-varying lift coefficient on a structure under vortex shedding defined by $C_{L0}(t) = C_L - C_R$. C_L and C_R are the vortex lift and reaction coefficients, respectively. β is a mass parameter or inverse Skop-Griffin parameter:

$$\beta = \rho D^2 / 8\pi^2 S_t^2 m \quad (3)$$

and ω_s is Strouhal frequency which is defined as below:

$$\omega_s = 2\pi f_s = 2\pi S_t U / D \quad (4)$$

where S_t is the Strouhal number and U is the free-stream velocity of the flow. When the cylinder is under resonant vibrations, the

fluctuating transverse and total fluid force coefficient is nearly sinusoidal. So, the total fluid force can be expressed as:

$$C_{L0}(t) = C_{L0} \sin(\omega t + \phi) \quad (5)$$

Where ϕ is the phase difference between the fluid force and displacement of cylinder. After substituting the Eq. 5 to Eq. 2, the final form of equation of motion becomes:

$$\ddot{y} + 2\omega_n \zeta_s \dot{y} + \omega_n^2 y = \beta \omega_s^2 C_{L0} \sin(\omega t + \phi) \quad (6)$$

VIBRATION SIMULATION OF A SINGLE CYLINDER UNDER VIV

In this section, the SDOF model is used to simulate an available experimental test carried out by (Rahman and Thiagarajan, 2013; Rahman, 2015) aiming at developing a validated analytical model for further analysis (the analysis of the proposed PIP system). Fig. 2 shows the schematic view of the experiments.

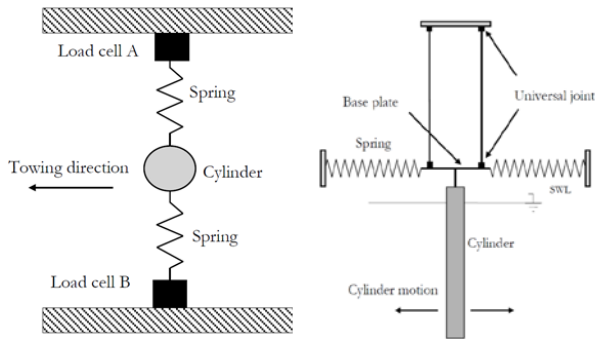


Fig. 2. Experimental setup of a single cylinder under VIV and Schematic diagrams of the pendulum rig (Rahman, 2015).

A towing tank with the length, width and depth of 50m, 1.25m and 1.10m, respectively was used. The testing cylinder was towing by the towing cable in the still water. The cylinder was mounted to a four-armed pendulum system, and two linear springs were used to connect the cylinder in the transverse direction. Only cross-flow vibration of the cylinder is considered. Moreover, the experiment was conducted in the subcritical flow region with Reynolds numbers ranging from 7.4×10^3 to 2×10^5 , corresponding to the range of reduced velocity $U_r (U_r = U/f_n D)$ from 2 to 14 (Rahman and Thiagarajan, 2013; Rahman, 2015). To consider the effect of aspect ratio (the proportion of cylinder length to the diameter), two cases (with $L/D=10$ and 5) are examined in the present study using the parameters listed in Table 2.

Table 2. Experimental matrix performed in (Rahman and Thiagarajan, 2013; Rahman, 2015).

	Exp. 1 ($L/D=10$)	Exp. 2 ($L/D=5$)
L (m)	0.60	0.40
D (m)	0.06	0.08
K (N/m)	245	245
m (kg)	2.71	3.21
f_n (Hz)	1.12	1.035
ζ	0.031	0.032
S_t	0.130	0.098
U (m/s)	[0.14-1.10]	[0.17-1.37]

A time-domain Simulink model (Fig. 3) is developed in MATLAB to solve Eq. 6. On the left side of the Simulink model, the external fluid

force including two feedback loops is entered to the system. The first one refers to the cylinder velocity and it is multiplied by the damping, and the second loop stands for the cylinder displacement, which is multiplied by the stiffness of the spring.

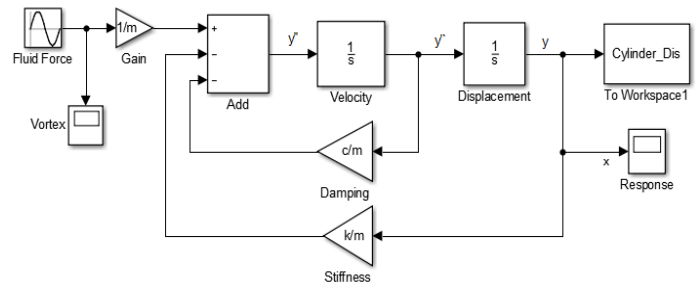


Fig. 3. Schematic of the dynamic Simulink model for a single cylindrical structure under VIV.

Fig. 4 compares the normalized amplitudes of the cylinder oscillations (A_y/D) obtained from the experiments and the analytical results over a range of reduced velocities. It can be seen that there is a good agreement between the experimental and semi-empirical results, which demonstrate the accuracy of the dynamic Simulink model as shown in Fig. 3. The comparisons also indicate that the accuracy of the SDOF model is slightly decreased for lower aspect ratios (especially in Fig. 4 (b)). This is possibly due to the dependency of lift frequency on the Strouhal number as discussed in (Srinil and Zanganeh, 2012; Rahman, 2015).

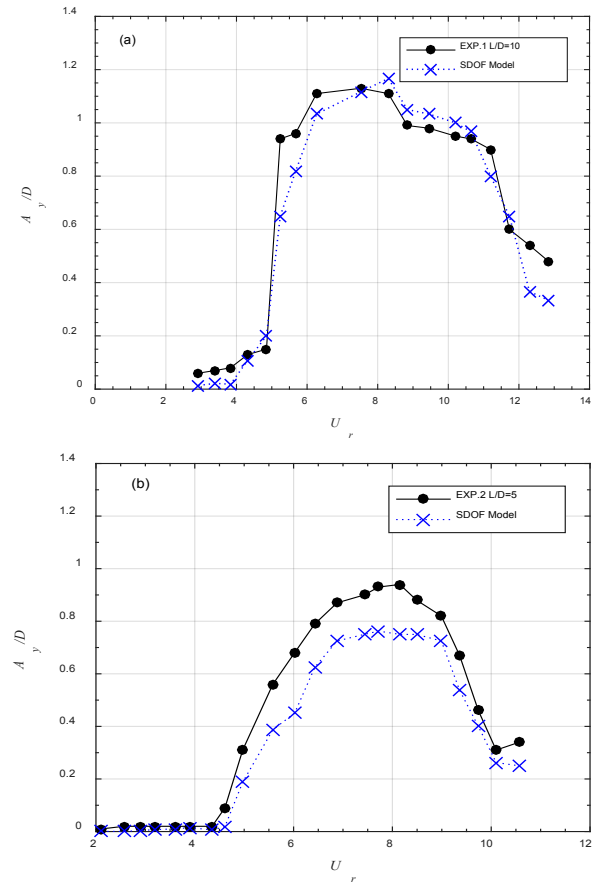


Fig. 4. Comparisons of the normalized amplitudes obtained from the experimental data and the analytical solutions for different aspect ratios (a) $L/D=10$ (b) $L/D=5$.

Fig. 4 also shows that within a certain reduced velocity range (normally known as the lock-in or synchronization region), the oscillating amplitudes of the cylinder are significant, which means there is a broad range, but not just one particular frequency, that can cause significant vibrations to the cylinder. The suppression of VIV is therefore important for the cylindrical offshore structures. The lock-in regions are different for different aspect ratios, but generally speaking for the investigated aspect ratios, lock-in phenomena take place when the reduced velocity is between 5 and 9.

It should be noted that some assumptions are made in the derivation of Eq. 6 (Gabbai and Benaroya, 2005). They are: 1) the cylinder is rigid and the flow around the body is fully correlated and two-dimensional; and 2) the dynamic behaviour of the flow field around the oscillating body and the effect of geometry are not considered. The comparisons in Fig. 4, however, indicate that these simplifications do not significantly influence the accuracy of the results. This semi-empirical model will be extended to study the behaviour of modified PIP system under VIV in the following sections.

BASIC GOVERNING EQUATIONS OF PIP SYSTEM AND METHOD OF SOLUTION

Fig. 5 shows the structural and simplified analytical models of the modified PIP system. As shown in Fig. 5(a), the coaxial inner and outer pipes are connected by the springs and dashpots. This system can be simplified as a two-degree-of-freedom (Two-DOF) system consisting of a primary oscillator (the outer pipe) and an auxiliary TMD (the inner pipe) as shown in Fig. 5(b). It should be noted that for a conventional structure-TMD system, the mass ratio between the TMD and the main structure is normally in an order of 1% to a few percent. For this modified PIP system, the mass ratio, however, can be much larger and in certain cases the mass of the inner pipe can be almost the same as the outer pipe. This system is normally regarded as a non-conventional TMD system (Bi and Hao, 2016).

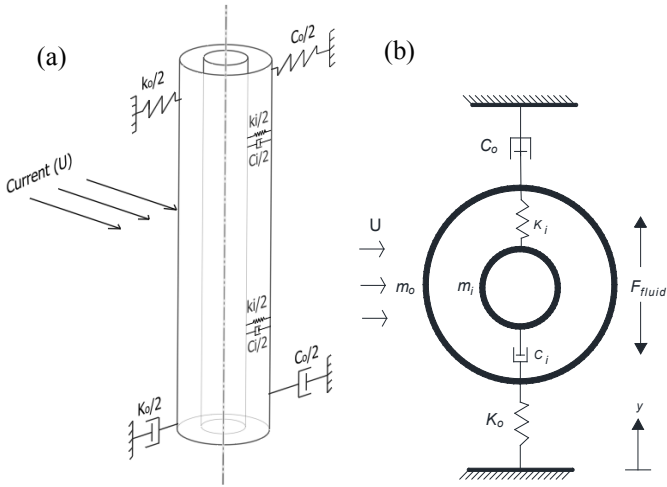


Fig. 5. Structural and analytical models of the modified PIP system (a) structural model and (b) analytical model.

As shown in Fig. 5(b), the primary oscillator is characterized by a mass m_o , a linear stiffness k_o and a constant viscous damping coefficient c_o .

The natural frequency and viscous damping ratio of the primary

structure are therefore $\omega_o = \sqrt{k_o/m_o}$ and $\zeta_o = c_o/2\sqrt{k_o m_o}$. Similar to the main system, the corresponding parameters for the TMD system are m_i, k_i, c_i , with $\omega_i = \sqrt{k_i/m_i}$ and $\zeta_i = c_i/2\sqrt{k_i m_i}$.

When this system is subjected to the vortex shedding excited cross-flow vibration, the equation of motion can be expressed as:

$$\begin{bmatrix} m_o & 0 \\ 0 & m_i \end{bmatrix} \begin{Bmatrix} \ddot{y}_o \\ \ddot{y}_i \end{Bmatrix} + \begin{bmatrix} c_o + c_i & -c_i \\ -c_i & c_i \end{bmatrix} \begin{Bmatrix} \dot{y}_o \\ \dot{y}_i \end{Bmatrix} + \begin{bmatrix} k_o + k_i & -k_i \\ -k_i & k_i \end{bmatrix} \begin{Bmatrix} y_o \\ y_i \end{Bmatrix} = \begin{Bmatrix} F_{fluid}(t) \\ 0 \end{Bmatrix} \quad (7)$$

where y_o and y_i are the in-plane transverse displacements of the outer and inner pipes, respectively. To facilitate the optimization of the spring stiffness and damping coefficient, it is convenient to define the mass ratio μ and tuning frequency ratio f as follows:

$$\mu = m_i/m_o \quad (8)$$

$$f = \omega_i/\omega_o = \sqrt{\frac{1}{\mu}(k_i/k_o)} \quad (9)$$

EFFECTIVENESS OF OPTIMIZED PIP SYSTEM TO MITIGATE VIV OF CYLINDRICAL STRUCTURES

This section investigates the effectiveness of using modified PIP system to mitigate VIV. The cylindrical structure tested in (Rahman and Thiagarajan, 2013; Rahman, 2015) is adopted again. However, the single pipe in the test is divided into two pipes in order to form a PIP system. For these two systems, the total mass of the pipe(s) is kept as the same. For example, if the mass of the single pipe is m , the masses of the outer and inner pipes can be determined by the mass ratio as $m_o = m/(1 + \mu)$ and $m_i = m - m_o$. As discussed before, the mass ratio of a PIP system can be quite large and without losing generality, a mass ratio of $\mu = 0.8$ is assumed in the present study to demonstrate the effectiveness of the proposed design. It should be noted that by dividing a single pipe into two, the thicknesses of each pipe becomes smaller, which may cause certain problems in real applications. For example, the thinner external pipe may not be enough to withstand the external hydrostatic pressure when it is located in the subsea. However, the purpose of this paper is to investigate the effectiveness of the proposed concept, such problems are not considered in the present study.

Fig. 6(a) shows the single pipe tested in (Rahman and Thiagarajan, 2013; Rahman, 2015). Fig. 6(b) and Fig. 6(c) show the modified PIP systems. A very stiff spring ($k \sim \infty$) is used to connect the inner and outer pipes in Fig. 6(b). These two pipes will therefore vibrate together when it is subjected to VIV and this system actually becomes the same as that in Fig. 6(a). In Fig. 6(c), the optimal spring and dashpot are used to connect the inner and outer pipes. Relative displacement is allowed between the inner and outer pipes and this system can be simplified as a structure-TMD system as mentioned above. In the present study, based on a numerical optimization study, the optimal tuning frequency and damping ratio are estimated as $f^{opt} = 0.65$ and $\zeta_i^{opt} = 0.35$.

The equation of motion of the proposed PIP system (a two-DoF system) can be represented by Eq. 7. To solve this equation, a time-domain Simulink model is developed in MATLAB again and shown in Fig. 7. To validate this model, the vibration of the system shown in Fig. 6(b) is calculated and compared with those obtained in Fig. 4 (the blue

curves). Exactly same results are obtained. The accuracy of this Simulink model is therefore validated and it is applied to calculate the PIP response shown in Fig. 6(c).

Fig. 8 compares the normalized cross-flow fluctuations of the single pipe and the optimized PIP systems. The dark curves are the results obtained from the single pipe model and the blue curves are from the optimized PIP system. Fig. 8 clearly shows that with the optimized PIP system, the vibration induced by vortex shedding can be significantly suppressed within all the considered reduced velocities. In other words, this system is not sensitive to the frequency of excitation. This is because the mass ratio of this PIP system is quite large, and it is not sensitive to the external vibration sources.

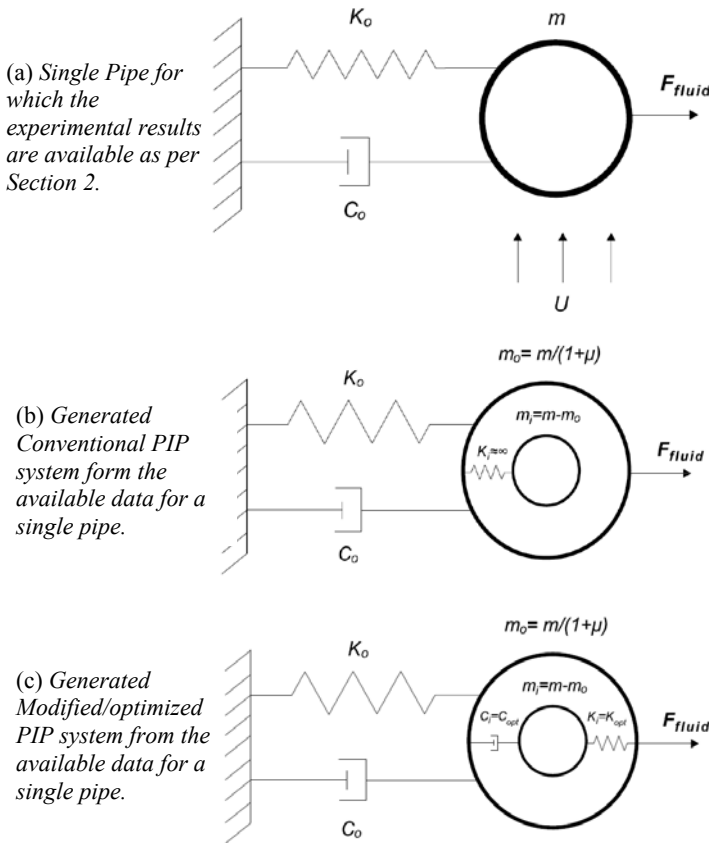


Fig. 6. Different pipe models under VIV: (a) A single pipe as presented in section 2, (b) a PIP system connected by a very rigid spring and (c) a modified PIP system with the optimized parameters.

Table 3 tabulates the maximum amplitudes for different systems and the corresponding reduction ratios. It can be seen that the maximum reduction ratio can reach 87% when the aspect ratio is 13. Aspect ratio seems only slightly influence the control efficiency. For the investigated cases the reduction ratios are more or less the same with an average value about 84%.

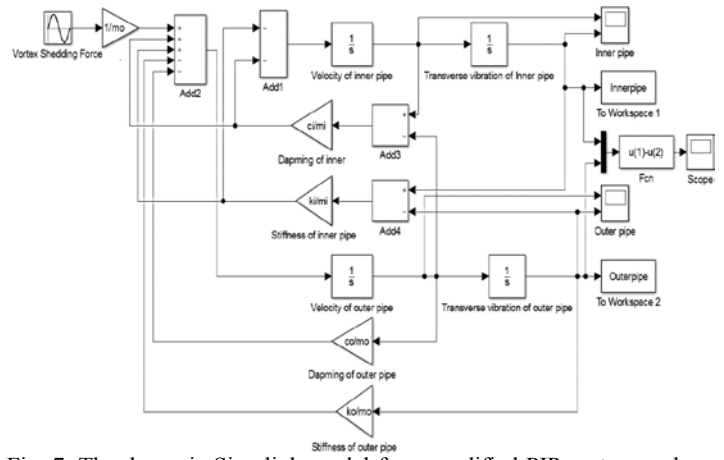


Fig. 7. The dynamic Simulink model for a modified PIP system under VIV.

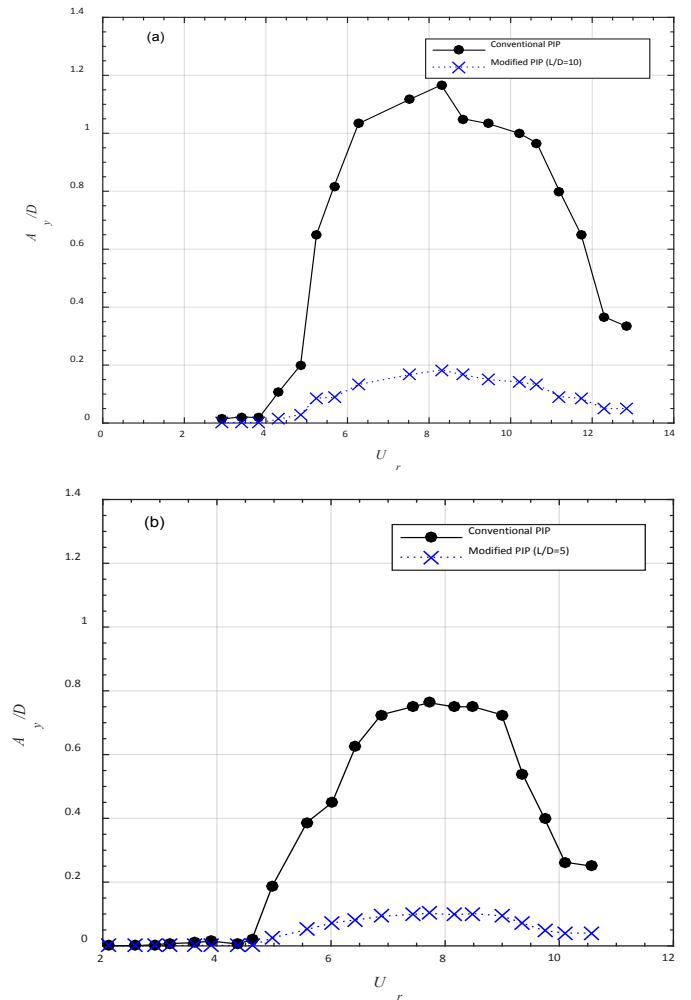


Fig. 8. Comparisons of normalized transverse vibration amplitudes of a single pipe system with the optimized PIP system. (a) $L/D=10$ (b) $L/D=5$.

Table 3. Peak normalized amplitudes of the single pipe model and the optimized PIP system under VIV and the corresponding suppression ratio

Aspect ratio (L/D)	Single pipe	Optimized PIP	Suppression ratio
13	1.04	0.13	87%
10	1.17	0.18	84%
7.5	1.02	0.17	82%
5	0.76	0.10	86%

CONCLUSION

This paper proposes using modified PIP system to mitigate vortex induced vibrations of cylindrical offshore structures. The effectiveness of the proposed method is investigated through analytical analyses by simplifying the system as a structure-TMD system. The equation of motion of the PIP system is derived and implemented into the MATLAB/Simulink code and validated by the experimental data. The optimal parameters for the connecting spring and dashpot are calculated and the explicit formulae for these parameters are derived. Analytical results show that VIV can be significantly suppressed by the proposed PIP system and this system is robust and not sensitive to the external excitation frequency and variations of the mass ratio. This system is believed having great application potentials to control VIV of cylindrical offshore structures.

ACKNOWLEDGEMENT

The authors would like to acknowledge the support from Australian Research Council Discovery Early Career Researcher Award (DECRA) DE150100195 for carrying out this research.

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