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Multi-high-frequency perturbation effects on flow-induced vibration control

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Abstract

High-frequency perturbation effects on the performance of suppressing flow-induced vibration (FIV) have been investigated based on the study of a phenomenological model of FIV. It was predicted that the nonlinear damping term in the model, describing the behavior of vortex shedding, could be increased via active perturbation in a frequency range well-exceeding the resonant frequency of the structure. Following the direction, a multi-high-frequency perturbation controller is proposed with a feedback closed-loop system, which has achieved the best experimental performance on the vibration control. Aside from the performance comparison between the multi-high-frequency controller and traditional control methods for FIV, detailed experiment has also been carried out to study the effects of some key parameters of the proposed controller, which is composed of a low-frequency active resonator and a hard limiter. Finally, robustness study on the multi-high-frequency controller has also been carried out subject to various flow conditions.

1. Introduction

Flow-induced vibration (FIV) of a structure in cross-flow has been regarded as an important research topic because of its significant value in engineering applications, such as construction of bridges and skyscrapers in civil engineering, design of oil-supplier tubes in ocean engineering, and design of aircraft and land vehicles in astronautical and mechanical engineering. Numerous fundamental research results on FIV have been reported, which can be found from a series of comprehensive review papers [1–4]. Under the broad area of FIV, the development of control technique for suppressing FIV has also attracted attentions from many researchers.

In order to suppress both vortex shedding and structural vibration, various active control methods have been explored in the past. Acoustic excitation [5–7] and oscillating cylinder [8–10] are recognized as precursory methods for active flow control. Recently, Cheng et al. [11] introduced a novel perturbation technique using embedded piezoelectric actuators for elastically mounted cylinders, which was found to be able to efficiently alter the interaction between the flow and the structure. In that work, a square cylinder was selected due to the

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comfortability of actuator installation. Three piezoelectric ceramic actuators, called the THin layer composite UNimorph piezoelectric Driver and sEnsoR (THUNDER), were embedded in series in a slot on one side of the cylinder to support a thin plastic plate, which was flush with the cylinder surface. Driven by the actuators, this plate will oscillate to create the local perturbation to the flow. At the earlier stage, this technique did not employ any feedback signal. The control was purely open-loop by only adjusting perturbation frequency. Nevertheless, this method can effectively reduce both structural vibration and wake velocity within certain small frequency range near the resonance region (vortex shedding frequency $\omega_s \approx$ structural nature frequency ω_n). Moreover, the frequency range of the perturbation being explored was relatively narrow, limited at lower frequencies up to 2–3 times of the shedding frequency. In Ref. [12], they further extended the idea to feedback control to improve control performance. The wake velocity was used as a feedback signal and a closed-loop control was implemented based on a manual-tunable PID controller. Though the performance was greatly enhanced, the tuning of PID parameters proved to be a very tedious task and time consuming. In this work, we try to extend the study in Ref. [11] by analyzing control effects when perturbation frequency is far away from the synchronization region. Moreover, an efficient closed-loop controller will be proposed while keeping the simplicity in its design.

Conventionally, in order to design a stable and effective controller, a dynamic model should be available. There are several analytical works on control problems with regard to the typical Navier–Stokes equations in fluid dynamics [13–16]. However, for FIV, the full enclosure of model requires the full solution of Navier–Stokes equations with a moving structure as boundary conditions. Although the development of computing technology brings the possibility to direct numerical simulation (DNS) in fluid dynamics, DNS is still a painstaking and computationally demanding task that is not suitable for real-time control design. The search for a simple but representative model for FIV was quite popular in the 1970s [1]. A phenomenological model based on a wake oscillator idea [17] is commonly adopted. The Van der Pol equation was first introduced to represent a wake oscillator [18]. The idea was improved by several researchers later [19–23]. Unfortunately, due to the difficulty of identifying all unknown parameters of FIV, most available models are based on a qualitative perception. Therefore, there has been a lack of a satisfactory FIV model for controller design. Consequently, there is only a little research work employing available models for control purposes. Researchers in Ref. [24] designed a direct velocity feedback controller using electromagnetic actuators to reduce structural vibration. They employed a wake oscillator model, as recommended by Skop and Griffin [19], for the lift coefficient to predict possible results of the controller. However, as mentioned above, the difficulty of identifying model parameters of the real apparatus constituted major obstacles in obtaining good comparison result between theoretical predictions and experimental results. Moreover, researchers in Ref. [25] designed a more advanced control scheme by using such a wake oscillator from a purely numerical approach. Since the coupling between fluid force and structural motion is still a controversial topic at the present research stage, to implement such a method is quite difficult.

Therefore, in this study, we only extract and explore the qualitative information from the wake oscillator model to guide our controller design. Based on it, a simple design rationale is proposed only by analyzing the perturbation frequency. This research work actually studies the problem via a model-independent approach. Although the attempt is made to link a qualitatively developed model to practical controller design for FIV, no detailed parameters of the model need to be identified. According to the model available in the literature, a feedback closed-loop control scheme is proposed by using multi-high-frequency perturbations. The organization of this paper is as follows. Section 2 provides some preliminary knowledge on the phenomenological model. Section 3 addresses the controller design, which starts from an analysis of traditional control methods. Results and discussions are presented in Section 4. The whole work is based on the control technique developed by Cheng et al. [11]. The main focus is put on the effects of multi-high-frequency controller to suppress wake vortex and structural vibration in the *lock-in* regime, in which case the vortex shedding frequency coincides with the natural frequency of the structure.

2. Preliminary studies

Most control design problems are based on the availability of plant model. In FIV case, however, the problem is more difficult since numerical solution of the Navier–Stokes equations does not provide much analytical directions for controller design, while the existing phenomenological models are only qualitative and applicable for the understanding of fluid–structure interaction. Thus, a special design routine has to be followed. In this section, a typical model to represent FIV is briefly reviewed and the key characteristics in the model are discussed. The prediction on the direction of controller design is also made. After that, experiments are conducted to verify the prediction. Since the controller only depends on the qualitative information of the model, only experimental validation is adopted here rather than numerical simulations.

2.1. Phenomenological model

Consider a typical 1-dof elastically mounted rigid cylinder with diameter D and length l. The cylinder is restricted to oscillate transversely to the uniform and stationary flow with free stream velocity U as shown in Fig. 1.

A commonly used model to describe FIV is a double oscillator model, which consists of a structure oscillator $y(t)$ for cylinder transverse motion and a wake oscillator $w(t)$ for vortex shedding. The equation of structure oscillator can be written as

$$m_0(\ddot{y} + 2\zeta\omega_0\dot{y} + \omega_0^2 y) = \tfrac{1}{2}\rho U^2 C_L D l, \tag{1}$$

where m_0 is the sum of cylinder mass and added fluid mass, ζ is the damping ratio, ω_0 is the undamped natural frequency of cylinder, ρ is the fluid mass density and C_L is the non-dimensional lift coefficient of the fluid flow around the cylinder. For the wake oscillator, all available phenomenological models use the Van der Pol equation with a nonlinear damping term to describe the self-sustainable fluid oscillation. There are different modeling approaches in the literature. Here, we adopt the simplest form from Ref. [22]. The model is written as

$$m_f\left[\ddot{w} - 2\zeta_f\omega_s\left(1 - \frac{w^2 + \dot{w}^2/\omega_s^2}{w_0^2}\right)\dot{w} + \omega_s^2 w\right] = -\frac{1}{2}\rho U^2 F_C D l, \tag{2}$$

where F_C is the force term imposed by the cylinder, m_f is the equivalent mass of wake oscillator, which is proportional to the fluid mass density ρ and the structure characteristic volume $D^2 l$, ζ_f is the damping ratio, ω_s is the vortex shedding frequency expressed in terms of $2\pi\, St\, U/D$ with St being the Strouhal number. In the vibration interval $(0, w_0)$, the damping term is negative to excite wake oscillation, and outside $(0, w_0)$ it is positive. The reason for choosing this damping model will be discussed in the next section, which is related to the controller design.

For Eqs. (1) and (2), C_L and F_C are usually regarded as coupling candidates in the literature. Extensive research has been carried out on how to choose the combinations of $\{w, \dot{w}, \ddot{w}\}$ or $\{y, \dot{y}, \ddot{y}\}$ to represent C_L and F_C, respectively [18,22,23]. While in this paper, due to the uncertainty about the coupling format of these two oscillators, we only concentrate on the qualitative information in the model to guide the controller design for FIV.

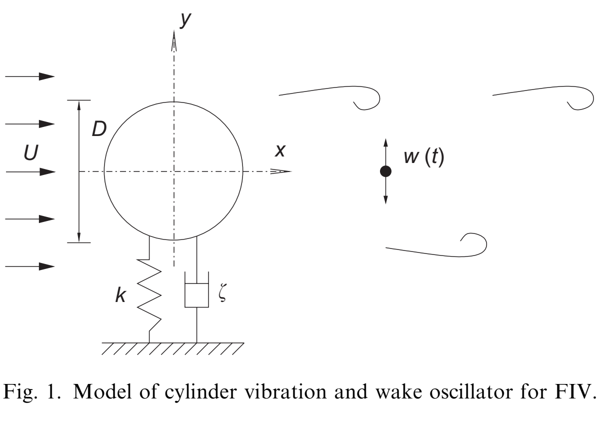


Fig. 1. Model of cylinder vibration and wake oscillator for FIV.

2.2. Pre-design: high-frequency perturbation effect

Consider the control technique developed in Ref. [11] as shown in Fig. 2. A local perturbation p is imposed on the flow around a square cylinder by the excitation of a movable plastic plate driven by a series of embedded piezoelectric actuators. Details on the embodiment of the actuators on the cylinder can be found in Ref. [11]. Suppose p causes a sinusoidal vibration component at the perturbation frequency ω_p to the wake oscillation signal $w(t) = A_p \sin(\omega_p t)$. Obviously, it should also affect the FIV model described by Eqs. (1) and (2). Here, we only consider the perturbation effect on the damping term in the wake oscillator. As a result of the perturbation, the damping term becomes

$$2\zeta_f \omega_s \left(\frac{A_p^2 \omega_s^2 \sin^2 \omega_p t + A_p^2 \omega_p^2 \cos^2 \omega_p t}{w_0^2 \omega_s^2} - 1 \right), \tag{3}$$

with values between

$$2\zeta_f \omega_s \frac{A_p^2 \omega_s^2 - w_0^2 \omega_s^2}{w_0^2 \omega_s^2} \quad \text{and} \quad 2\zeta_f \omega_s \frac{A_p^2 \omega_p^2 - w_0^2 \omega_s^2}{w_0^2 \omega_s^2}.$$

Clearly, if the perturbation frequency ω_p is much higher than vortex shedding frequency ω_s, the maximum damping value of wake oscillator will be larger. Now, we make the following proposition.

Proposition 1. *Assume that flow-induced vibration can be described as the phenomenological model given by Eqs.* (1) *and* (2). *Higher frequency perturbation can produce larger damping term in the wake oscillation. Correspondingly, in the local flow around structure, higher frequency perturbation will provide better performance on the suppression of vortex shedding in the wake, leading to a reduction of structural vibration when the direct perturbation effect on the structural motion is neglected.*

We conduct an experiment to validate this proposition. Based on the perturbation technique in Fig. 2, an open-loop control is implemented by using sinusoidal signals with a fixed amplitude but different frequencies. The results are monitored by a laser vibrometer for structural vibration and a hotwire for wake vortex shedding, which are shown in Fig. 3. The flow condition is set at FIV lock-in regime, where $\omega_s \approx \omega_0$. Define the dimensionless frequency as St_f, the dimensionless shedding frequency as $St_s \approx 0.13$, and the dimensionless perturbation frequency as $St_p = f_p D/U$. In the experiment, power spectrum densities (PSD) of wake velocity and structural vibration signals are obtained under different perturbation frequencies $St_p \in [0.07, 2.80]$. The peak level trends of PSD are plotted in Fig. 4. It is found that for $St_p \geqslant 0.7$, the peak level of PSD drops significantly and the overall peak level in this St_p region is much lower than that in the region of $St_p < 0.7$. Besides the peak level, the rms reduction value of signals between controlled and uncontrolled cases is also plotted as shown in Fig. 5 to verify the proposition. As for wake velocity reduction, the rms reduction level for $St_p \geqslant 0.7$ is higher than the rms reduction level for smaller St_p. The experimental results matches the former part of Proposition 1.

This phenomenon also agrees with other findings in flow control using acoustic excitation. In Refs. [6,7], it was found that effective perturbation frequency range for acoustic excitation corresponded to the shear-layer instability frequency, which was of higher order than vortex shedding frequency and was dependent on Reynolds number [5]. When St_p is in the order of 1, as mentioned by [6], another 'lock-in' phenomenon occurs.

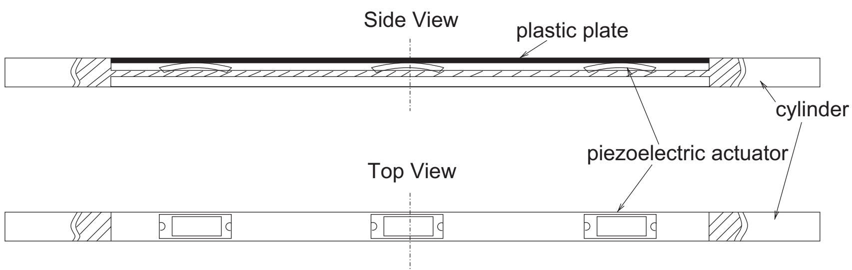


Fig. 2. The layout of control technique [11].

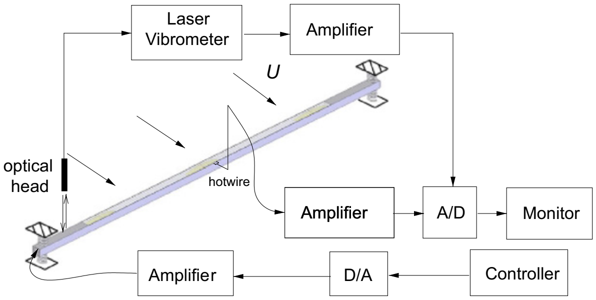


Fig. 3. Open-loop control experimental setup.

The instability waves are amplified, and momentum transport and flow mixing are enhanced, which affects wake characteristics significantly when compared to no perturbation condition. In our investigation, the experimental results from the proposition also agrees with this point. In addition, considering the frequency effect of perturbation p on the reduction of structural vibration, in practice, it is impossible to neglect the energy imposition from the actuator to the structure. Higher frequency results in more energy imposed on the structure in the same time interval, even if the energy amplitude is small. Therefore, the control of FIV must balance between counteracting flow energy on structure by increasing damping and imposing perturbation energy on structure by increasing frequency. Indeed, in Fig. 5, there is an optimal frequency region $St_p \in [0.5, 2]$, where both wake and structural velocity have considerable reduction. Loosely speaking, the latter part of Proposition 1 is also valid. This finding on the high-frequency perturbation effect gives one possible control design direction for FIV based on the phenomenological model.

3. Controller design

Based on the perturbation technique, we aim at finding a more effective control scheme for the reduction of both structure and wake oscillations in the *lock-in* regime. It is known that closed-loop control usually provides more satisfactory performance than open-loop [12]. Thus, this paper pursues a closed-loop control method for FIV by using structural vibration signal as feedback signal as shown in Fig. 6. In order to provide reasonable objects for comparison, we first study traditional control methods, and then give a multi-high-frequency control scheme considering the limitations in traditional methods and the inspiration from the finding in the pre-design section.

3.1. Traditional control methods

3.1.1. Variable structure control

A possible solution for structure vibration control under unknown disturbance is called variable structure control. For FIV problems, structure vibration is induced by the uncertain but bounded flow excitation. The structure vibration model is given by

$$\ddot{y} + 2\zeta\omega_0\dot{y} + \omega_0^2 y = \frac{d+u}{m_0}, \tag{4}$$

where d is the bounded fluid force and u is the actuator control input. We consider a Lyapunov energy function $L = \frac{1}{2}(\dot{y}^2 + \omega_0^2 y^2)$. Its derivative, evaluated along the trajectory of Eq. (4), can be derived as

$$\dot{L} = \dot{y}(\ddot{y} + \omega_0^2 y) = -2\zeta\omega_0\dot{y}^2 + \frac{\dot{y}(d+u)}{m_0}. \tag{5}$$

We substitute a typical variable structure control law $u = -\text{sgn}(\dot{y})F_u$ into Eq. (5), where sgn is sign function and F_u is peak force generated by actuator. Eq. (5) becomes

$$\dot{L} = -2\zeta\omega_0\dot{y}^2 + \frac{\dot{y}d - |\dot{y}|F_u}{m_0}. \tag{6}$$

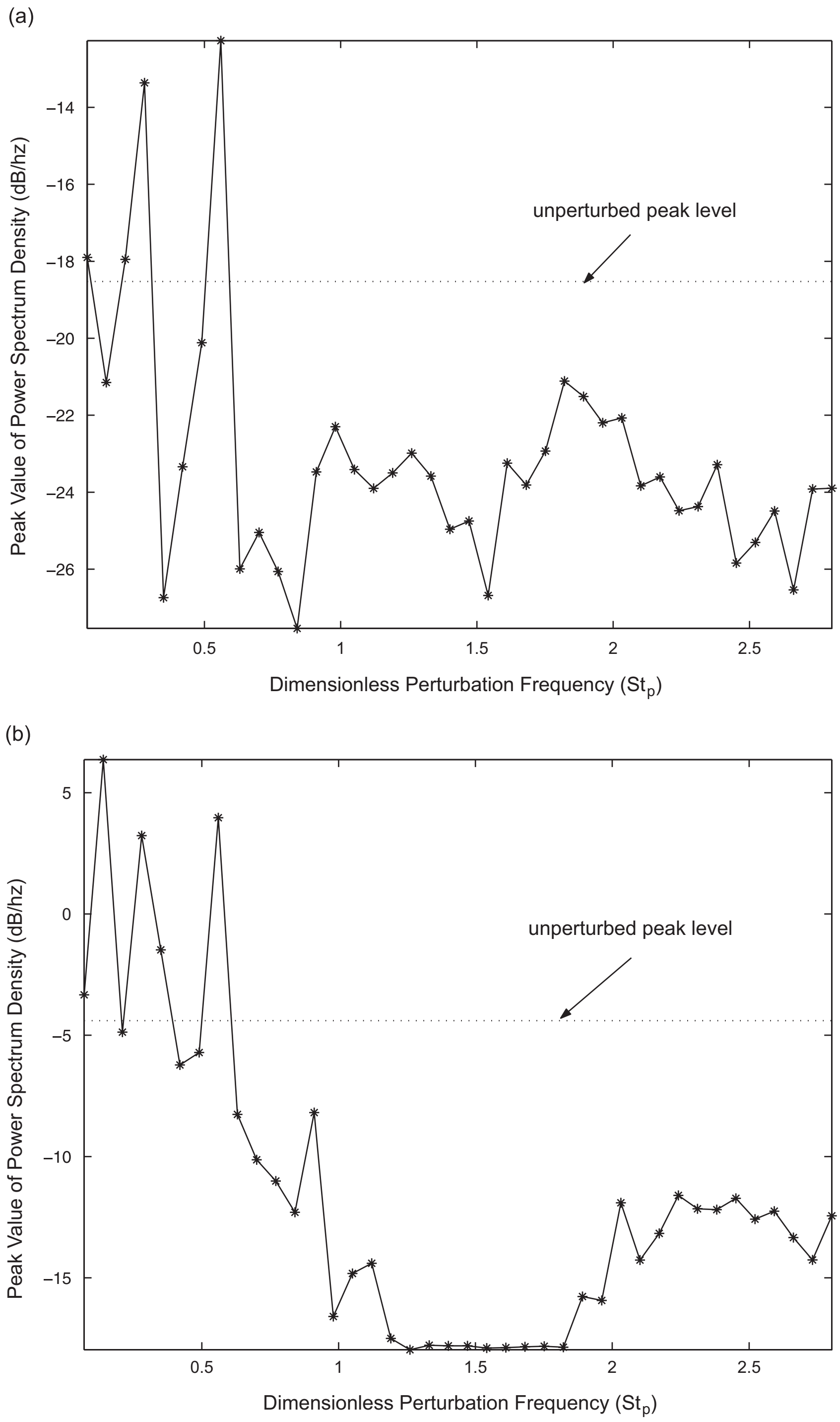


Fig. 4. Power spectrum density peak level VS different perturbation frequencies ($St_p \in [0.07, 2.8]$).

Ideally, this control law forces the derivative of L negative definite if $F_u \geqslant \|d\|$. Since L is positive definite, by the direct method of Lyapunov theory, system (4) is asymptotically stable at $(\dot{y}, y) = (0, 0)$. However, the experimental results shown in Fig. 7 indicates that this ideal design cannot work. This phenomenon results from a commonly known chattering problem in variable structure control [26]. In practical applications,

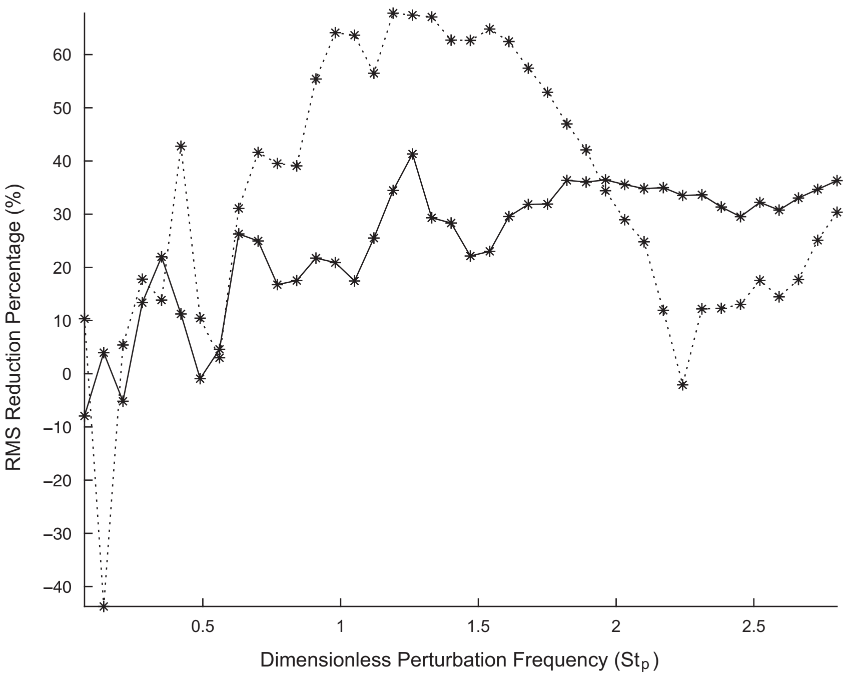


Fig. 5. rms Reduction percentage: structural velocity and wake velocity ('- -' structural velocity, '—' wake velocity).

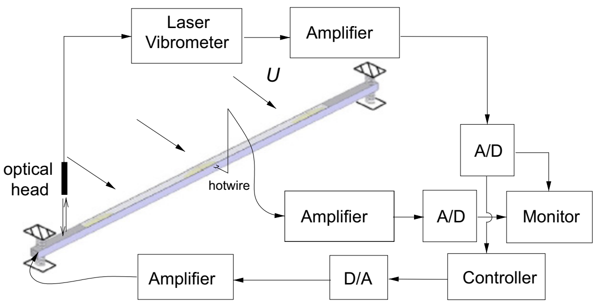


Fig. 6. Closed-loop control experimental setup.

unmodeled dynamics in closed-loop system such as actuator often fail to response to the control demand instantaneously, and the imperfect control signal will cause oscillations in the system, which causes local instability at the equilibrium point.

3.1.2. Active resonator control

Active resonator control is to synthesis an electronic resonator in the closed-loop system, which is able to counteract the peak of plant vibration at its natural frequency. This idea is used for structure vibration control in FIV problem. Suppose the cylinder has a transfer function $P(s) = s/(s^2 + 0.25s + 125^2)$ with undamped natural frequency at 20 Hz. A possible control law may be $C(s) = (s^2 + 50000s + 125^2)/(s^2 + 1000s + 125^2)$ and the block diagram is shown in Fig. 8. The simulation results are given in Fig. 9. However, experimental results in Fig. 10 only exhibit moderate control effect. The rms reduction value of structural vibration velocity is 49%, which is worse than those obtained by open-loop control in Section 2. In our observation, the resonator frequency matches the natural frequency of structure vibration, but also coincides with shedding frequency of wake vortex. The unknown interaction between the control signal and flow shedding may weaken original control effect on the structure. Furthermore, since the objective of this method is only to suppress structure vibration, flow excitation will always exist. While vibration

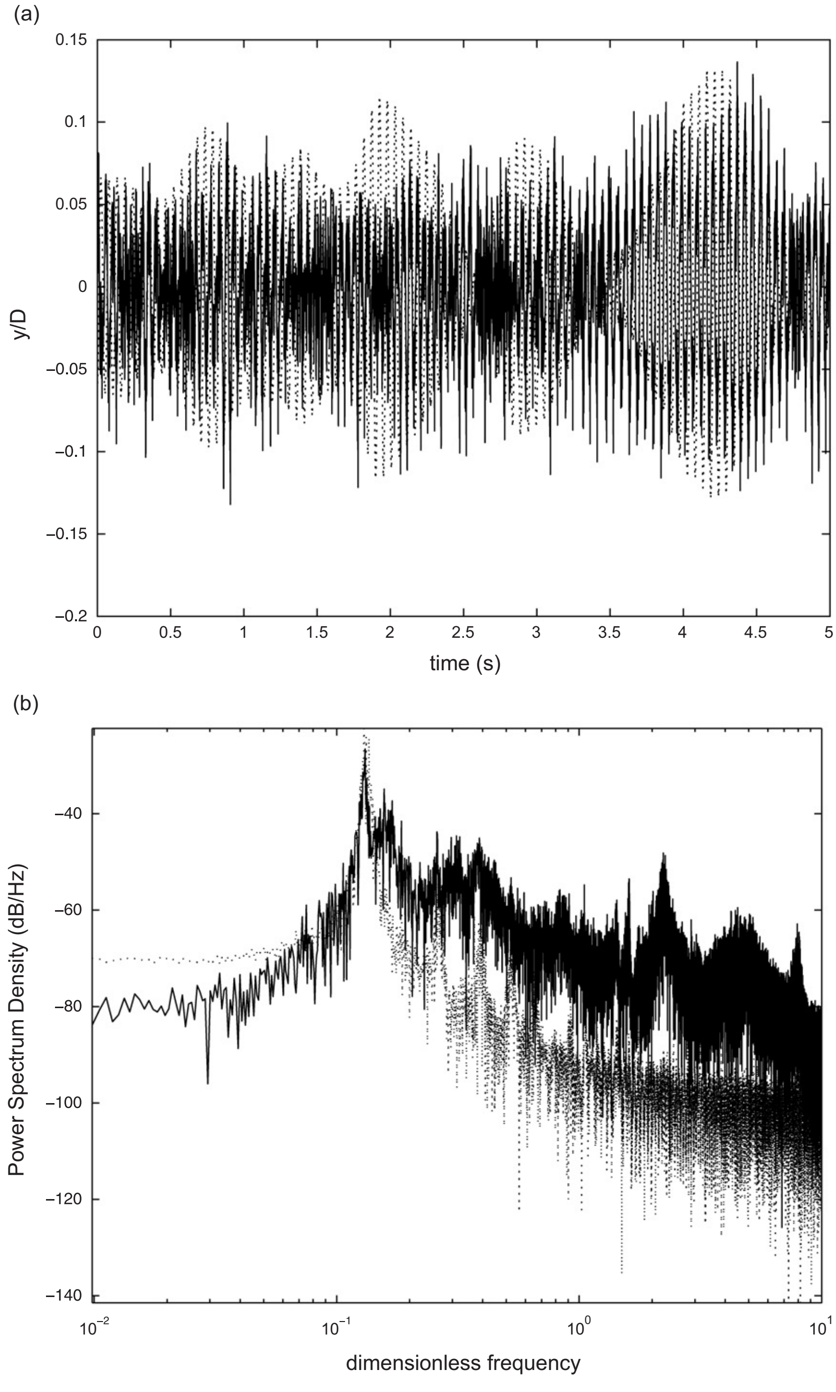


Fig. 7. Experimental results of structural vibration by no control and variable structure control ('- -' no control, '—' variable structure control).

amplitude is approaching zero, due to feedback control, the active resonator also generates ''near zero'' control input on the structure. Thus, flow excitation will dominate the control effect again and consequently structure will tend to vibrate. This suggests that FIV control problem should focus on the control of flow excitation.

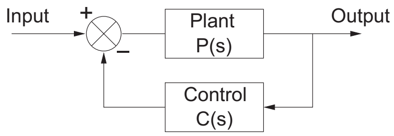


Fig. 8. Block diagram of closed-loop control.

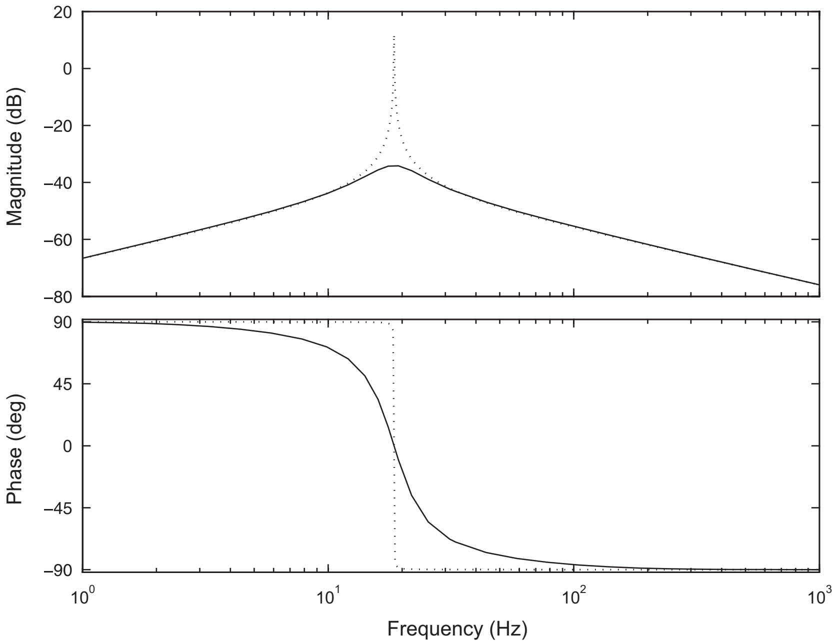


Fig. 9. Bode diagrams of P(s) by no control and active resonator control ('- -' no control, '—' active resonator control).

3.2. Proposed control method: multi-high-frequency perturbation

Considering the limitations in traditional control methods, a high-frequency perturbation approach based on the phenomenological model is preferred to design a simple but effective controller for FIV problem in the lock-in regime. One merit of the high-frequency control is that since St_p is far away from the lock-in regime, the perturbation phase effect can be neglected. Recalling the results in Fig. 5, there is an optimal perturbation frequency region $St_p \in [0.5, 2]$. An active resonator with an optimal frequency St_p should be adopted. However, considering the limitations of active resonator in closed-loop feedback control for FIV, the negative damping term has to be used to provide a 'limit-cycle' behavior of the controller. This is also reasonable by analyzing the phenomenological model. In the wake oscillator, a nonlinear damping term is used to allow self-sustainable oscillation in the FIV model system. As a result, a similar measure to allow self-sustainable oscillation in the controller should be applied so that the controller can counteract FIV continuously. Furthermore, in order to prevent the controller input from diverging due to negative damping, a hard limiter should be added before feeding control input. In practice, it is possible to use a low-frequency resonator, which actually produces multi-high-frequency perturbation signals when integrated with a hard limiter. The proposed control method is shown in Fig. 11.

The design factors are k_D and ω for the active resonator, and bound b for the hard limiter. Several experiments are conducted to assess the control effect of each parameter. In all the experiments, the testing environment is set at the lock-in regime, where $St_s \approx St_0 = 0.13$ and $Re \approx 1800$. Structural vibration velocity is used as the feedback signal. We use the rms reduction values of both structure velocity and wake velocity signals, for the cases without control and with the proposed control method, to evaluate control performance as shown in Fig. 12. Most results in Fig. 12 demonstrate better control performance than those achieved by

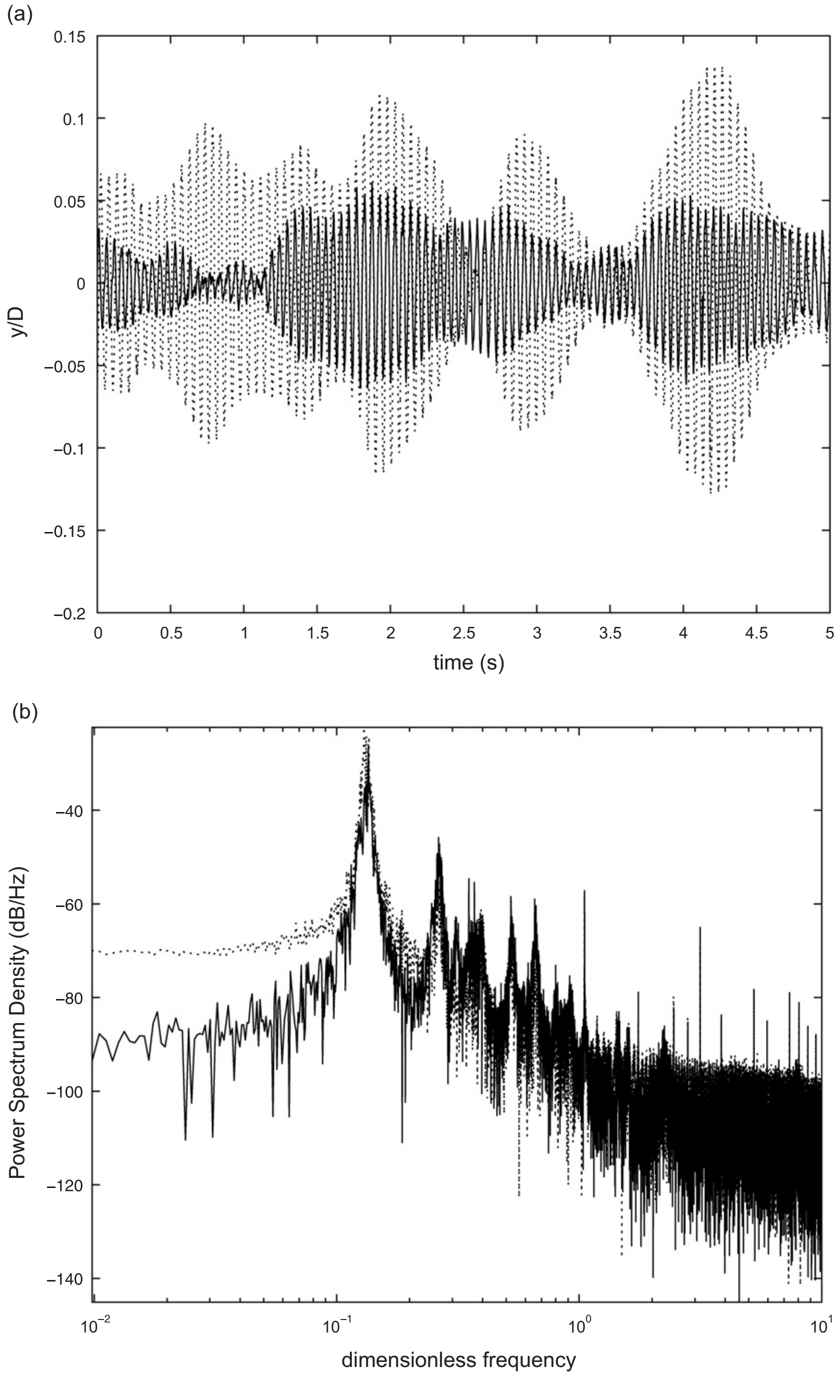


Fig. 10. Experimental results of structural vibration by no control and proposed control ('- -' no control, '—' proposed control).

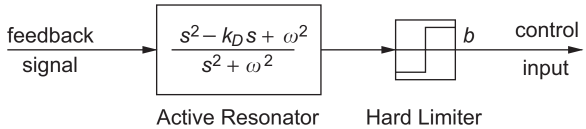


Fig. 11. Proposed control method: multi-high-frequency perturbation.

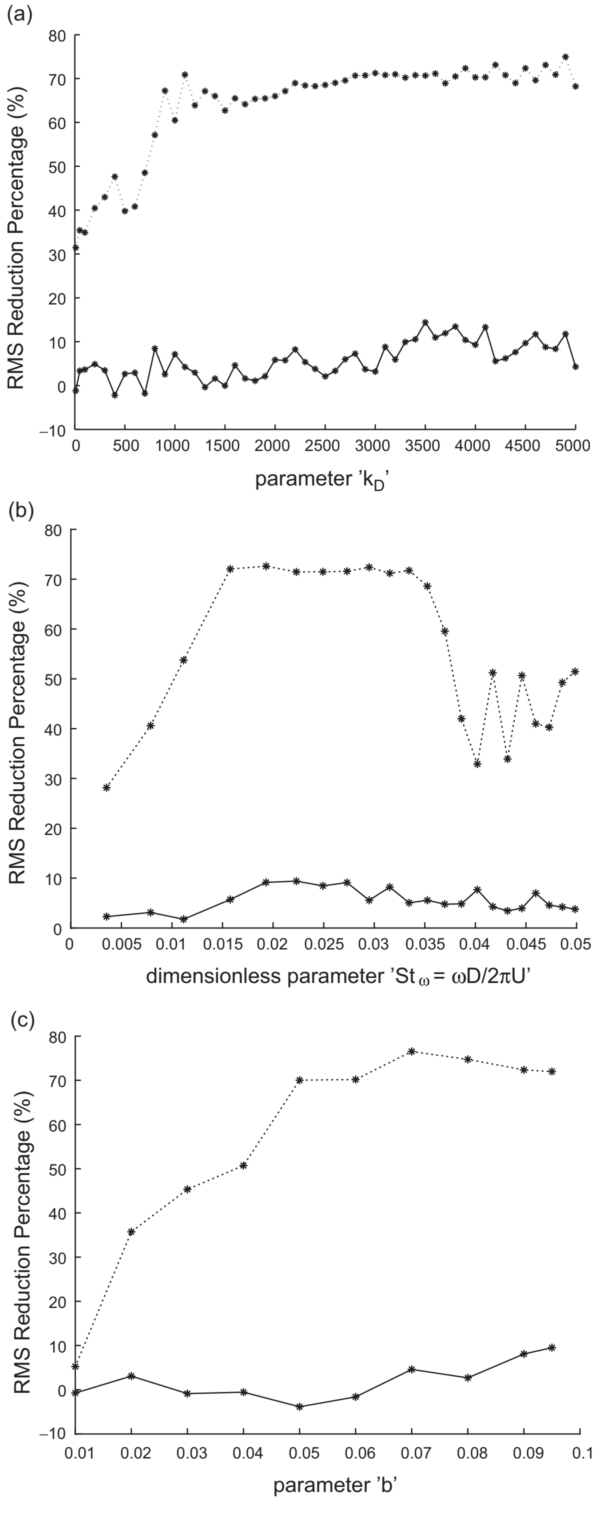


Fig. 12. Effects of proposed controller parameters: rms reduction percentage ('- -' structural velocity, '—' wake velocity).

other methods studied in the paper. The best performance of suppressing structure vibration is over 70%. Considering individual parameter effects, from Fig. 12(a), a larger k_D gives better result due to its contribution on increasing controller's damping. From Fig. 12(b), there also exists a certain optimal region for the selection of ω, which agrees well with the findings on the pre-design testing using open-loop perturbation in Section 2. Closed-loop control is more steady and robust with the variation of ω. From Fig. 12(c), it is seen that a larger bound b means more energy can be used to compensate the FIV system energy, and in turn leads to a better result.

4. Discussions

4.1. Supplementary model interpretation

By now, an effective controller has been designed and implemented by using the phenomenological model as a design guide. Together with the pre-design study, this controller also verifies Proposition 1 derived from the model study. Take the proposed controller with $(k_D, St_\omega, b) = (3500, 0.02, 0.095)$ as an example. Fig. 13 gives experimental results of structure vibration. Clearly, in Fig. 13(b), there are multi-high-frequency responses around the region $St_p \in [0.5, 2]$, which has been identified as the optimal perturbation frequency region in Fig. 5. Furthermore, considering Eq. (3) for the wake oscillator damping term, if $\omega_p = \omega_s$, which means perturbation frequency is equal to shedding frequency, then this damping term will be $(A_p^2/w_0^2) - 1$. If perturbation amplitude A_p is moderately small, then $(A_p^2/w_0^2) < 1$ gives a negative damping term in all situations. This will increase wake oscillation, which also agrees with experimental analysis about St_p around St_s by Cheng et al. [11].

4.2. Control performance and analysis

The effect of various flow conditions is investigated with the following considerations in mind. Apart from the flow velocity, it is very difficult to change other physical parameters of the system in the experimental set-up. Therefore, any flow speeds other than $Re = 1800$ induce non-resonant system response. In most engineering practice, resonant case is usually the major concern, which is characterized by violent structural vibrations, causing serious structural problems. In this sense, non-resonant cases are of secondary importance. Nevertheless, for completeness, an attempt is made to verify whether the proposed controller is still effective within a certain flow velocity range.

To this end, four configurations are tested with $Re = 1000, 2500, 3400, 4100$, respectively. Fig. 14 shows the respective PSD plots. It can be noticed that a reduction in PSD peaks can only be found in Figs. 14(b) and (c), corresponding to a reduction of 36% and 38%, respectively, in terms of rms values. This is much lower than the one obtained under resonant case (Fig. 13 with $Re = 1800$). Similarly, this reduction in the peak value is obtained at the expense of increasing high-frequency energy, a phenomenon which is commonly seen in many other control applications. For the other two flow speeds, Figs. 14(a) and (d) show no sign of reduction in the dominant shedding frequency. One plausible explanation is that for the low Reynolds number $Re = 1000$, the flow energy is not sufficient to be compensated by control input. Therefore, the surplus energy of control input generated by the negative damping of the controller is transferred to the cylinder and in return enhances its vibration. On the other hand, at very high Reynolds number (Fig. 14(d)), strong turbulence occurs such jeopardizing the validity of the simple wake oscillator model previously used. In fact, it was observed that a small variation in the model parameters can cause larger changes in experiments than what is predicted by the model [22]. Nevertheless, these results show that although the proposed controller is based on such a simple and sensitive model, it can still achieve acceptable robustness near the resonant region, despite some deficits in the control performance when flow condition is far from the resonant region.

Based on the results, a general analysis on the working principle of the control and its performance can be provided.

When flow blows over a bluff body, vortices separate alternately from the structure, giving rise to excitation forces and causing the structure to vibrate. The structural motion in turn influences the flow field, resulting in a highly nonlinear fluid–structure coupling. The Kármán vortex street behind the bluff body is unstable and

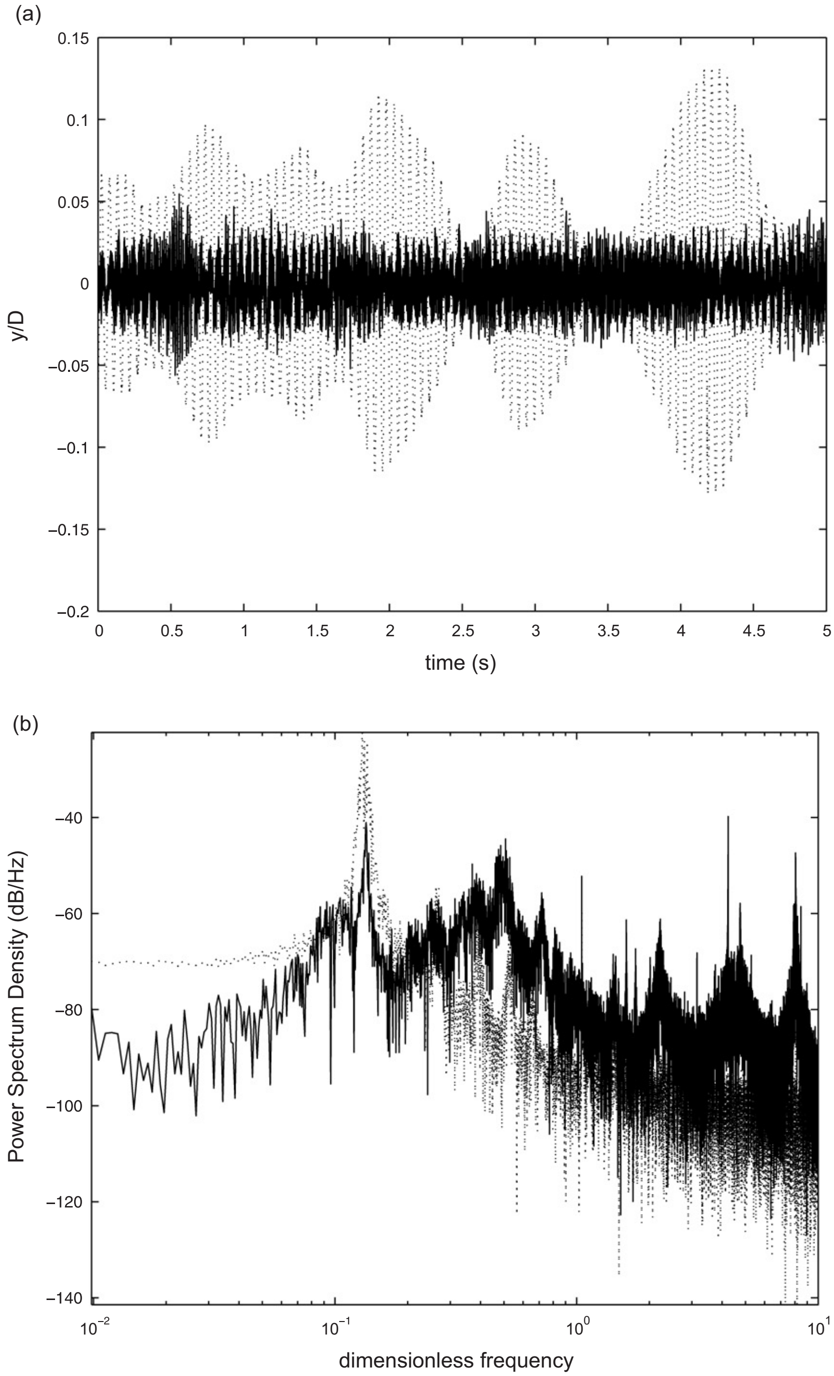


Fig. 13. Experimental results of structural vibration by no control and proposed control ('- -' no control, '—' proposed control).

depends on its infant form near the separation area. Any local perturbations into flow, if small enough to comply with linear theory, may grow exponentially, exerting a significant influence on the highly nonlinear unsteady Kármán vortex street [27,28]. These observations further imply that local perturbations may significantly influence the nature of fluid–structure interaction or the vortex-induced structural vibration. Based on these facts, the present perturbation technique is conceived to provide a possible control of both flow and structural vibration by using piezoelectric actuator-driven surface perturbations.

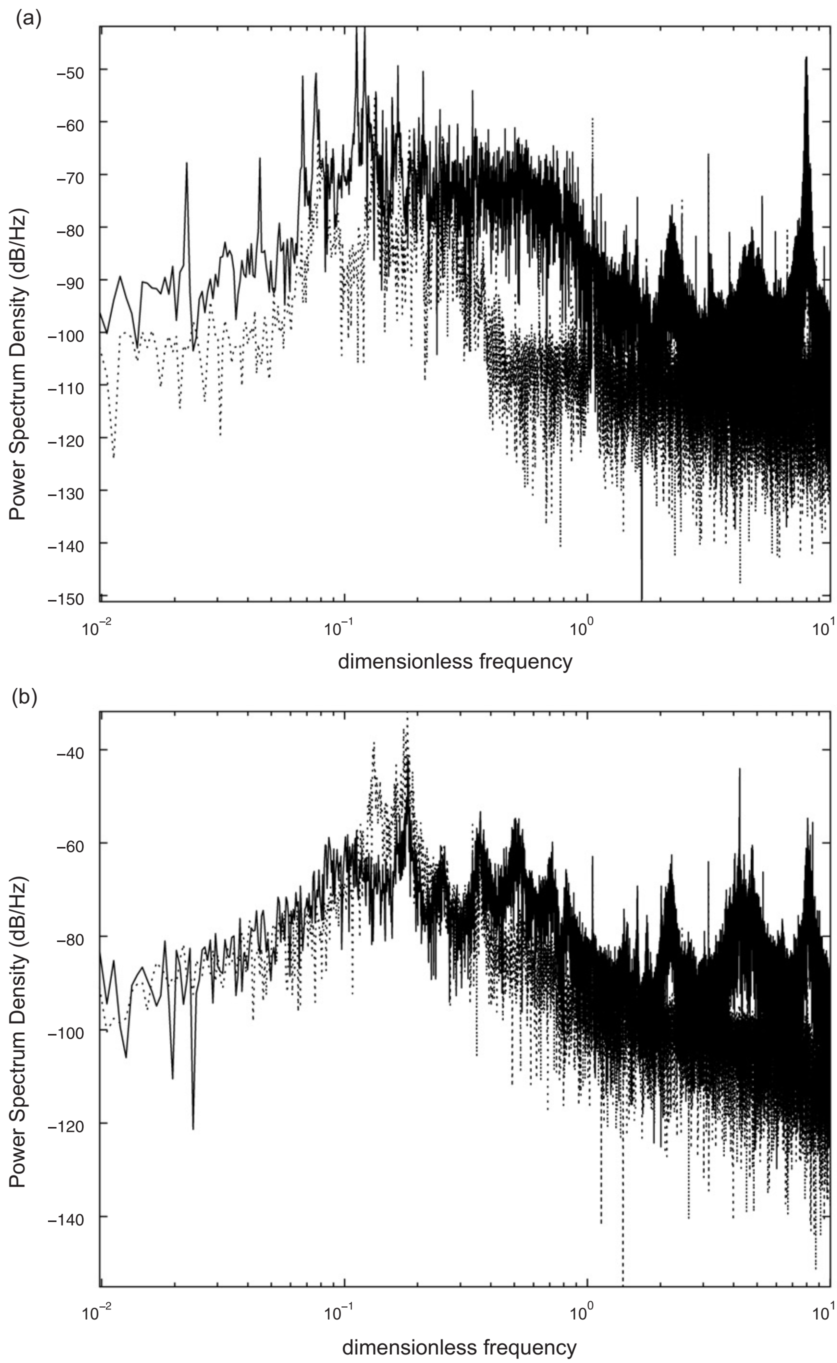
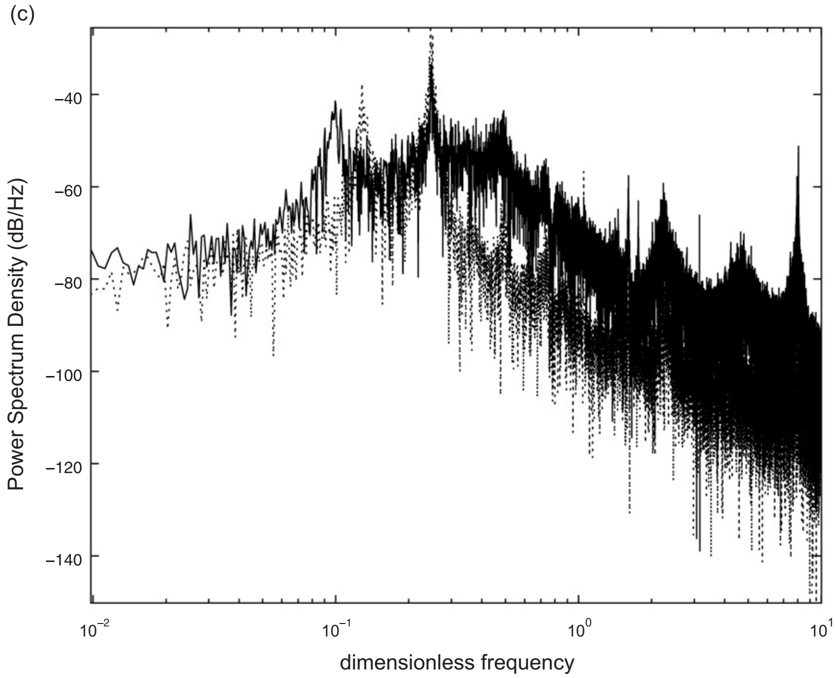


Fig. 14. Experimental results of structural vibration under different flow conditions ('- -' no control, '—' proposed control).

The present control scheme using multi-high-frequency perturbation differs from our previous work [11,12] using PID controllers in many aspects, leading to very different control mechanism. Previous work makes direct use of feedback signals and introduces a time delay between the feedback signals and control signals. Dominated by the shedding frequency, the control signal mainly contains low frequencies, such requiring a careful tuning of the control parameters to ensure a proper phase relationship between control signal and system responses. In fact, it has been demonstrated that a successful control should change the in-phased



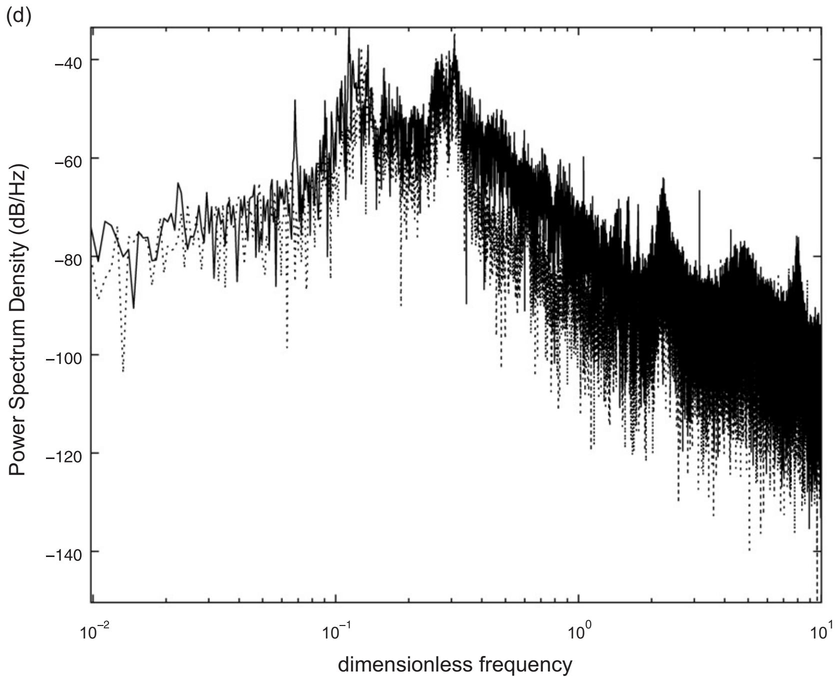


Fig. 14. (*Continued*)

fluid–structure interaction to anti-phased [12]. The present controller makes use of the negative damping property of the controller itself together with a hard limiter to create rich high-frequency control action. An electronic limit-cycle is induced by the flow to produce the perturbation through the thin plastic plate. Since the mass of the plastic plate is significantly smaller than that of the beam, a weak actuation signal is sufficient to cause multi-frequency perturbation to hinder the formation of the vortex near the beam surface. The phenomenological model clearly shows that, by doing so, significant damping will be introduced to the fluid–structure system. Therefore, the main working principle of the present control scheme is to change the system damping. Since all dynamics systems are very sensitive to the damping in the resonant region (lock-in region in the present case), the good performance of the controller in the lock-in region can be easily understood. By the same token, the requirement for the careful phase tuning by PID controllers is released, making the controller design much easier.

As mentioned before, the lack of accurate physical models for the fluid–structure system has always been the major obstacle towards the design of more effective controllers. The rapid development in CFD will certainly help to provide more reliable and useful information. Alternatively, a large quantity of tests should be carried out under various flow conditions so that some key parameters necessary for controller design can be identified experimentally. This will also give the possibility of exploring other control strategies to further improve the control performance.

5. Conclusion

In this paper, we extend the study of perturbation effects on FIV from synchronization region [11] to high-frequency region, in which the frequency is far away from the lock-in regime. The physical plant is an elastically mounted square cylinder and control input is the movable surface of cylinder driven by embedded actuators as in Ref. [11]. The challenge of FIV control problem is the lack of accurate analytical model for controller design. The high-frequency perturbation rationale roots from the study of a phenomenological model, which uses a Van der Pol oscillator to describe the behavior of wake vortex. In the literature, it is found that FIV behaves quite similarly to the model qualitatively. Based on the model and experimental studies at the lock-in condition, it is found possible to suppress both structural vibration and wake vortex by introducing a perturbation input at a high-frequency range well-exceeding resonant frequency of structure. This also agrees well with the findings of others using acoustic excitations. Based on this idea, an effective feedback control is designed to generate multi-high-frequency perturbation using a low-frequency active resonator and a hard limiter. The effects of controller parameters are studied experimentally. The performance of the proposed controller outperforms other traditional controllers. Moreover, the controller is moderately robust towards variations in both controller parameters and physical plant conditions. This work successfully links computational model for FIV to the controller design problem, and at the same time, verifies the physical aspects of the phenomenological model.

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