

Research on the control effect of control rods on vortex-induced vibration of an upstream square cylinder

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Abstract: Numerical simulations were used to explore how control rods affect the hydrodynamic forces and vortex-induced vibration (VIV) in a square cylinder. These control rods were placed close to the point of highest pulsating velocity in the cylinder's wake. Systematic analysis was conducted using five control rods of different sizes and three different gap ratios, with the Reynolds number fixed at 100. Through time-domain and frequency-domain experiments, evaluation of the flow characteristics, and proper orthogonal decomposition analyses, the physics governing the behavior of a square cylinder with control rods was revealed. The results show that the arrangement of the control rods significantly alters the VIV response of the square cylinder, where effective control depends on the size and gap ratio of the rods. Using the optimal control parameters, the maximum reduction in the vibration amplitude, fluctuating lift coefficient, and mean drag coefficient are 83.10%, 78.13%, and 28.90%, respectively. These significant reductions are attributed to the elongation and downstream shift of the wake vortices caused by the control rods. Additionally, vortices on the surface of the square cylinder merge with counter-rotating vortices generated on the control rods, reducing both the energy input from the fluid to the system and the vortex shedding intensity. Consequently, a more uniform pressure distribution is formed. Furthermore, suppression of the square cylinder's peak vibration amplitude, mean drag and fluctuating lift improves as the size of the control rod increases. Proper orthogonal decomposition analysis demonstrates that the first and second modes domi-

nate the flow characteristics, and their conjugacy remains consistent regardless of the presence of control rods.

Key words: square cylinder; vortex-induced vibration; control rod; flow dynamics; proper orthogonal decomposition; numerical simulation

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Flow-induced vibration (FIV) refers to the deformation or movement of a structure resulting from fluid flow, highlighting the complex interaction between fluid dynamics and structural behavior^[1-2]. The investigation of FIV in bluff bodies has long been a topic of interest in fluid mechanics and structural engineering. Such phenomena commonly occur in various engineering structures, including bridges, offshore risers, and high-rise buildings, where they can pose significant safety risks. Consequently, extensive research has been conducted to elucidate the mechanisms underlying FIV, providing valuable insights and references for addressing related engineering challenges^[3-4].

FIV can be further classified into vortex-induced vibration (VIV), galloping, and flutter^[5-6]. VIV is an interaction phenomenon characterized by multiple degrees of freedom and nonlinear fluid-structure interaction, caused by feedback coupling between alternating vortex shedding and structural vibration as fluid flows around a structure^[7]. Williamson et al.^[8] reviewed the literature and highlighted the frequent observation of VIV in various bluff structures. Kou et al.^[9] also found that VIV can induce significant structural vibration even at relatively low flow velocities. Compared with galloping and flutter, VIV is more common and can easily lead to structural fatigue, posing risks to structural safety. Consequently, considerable attention has been devoted to investigating the underlying flow mechanisms and developing effective control methods.

Circular or square cylinders are among the most extensively studied bluff bodies, with numerous researchers focusing on their VIV characteristics. For example, Pasto^[10] explored the factors influencing the VIV responses of cylinders in laminar and turbulent flows, revealing that the mass-damping parameter is not the sole parameter affecting the responses. The Reynolds number (Re) also plays a significant role. Kataoka^[11] examined FIV of a square cylinder in a turbulent boundary layer

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and found that the negative damping effect occurs intermittently; thus, the vibration amplitude of the square cylinder fluctuates unpredictably over time. Sun et al.^[12] studied VIV in a square cylinder with shear flow having two degrees of freedom, revealing that the shear ratio and dimensionless speed significantly affect the vibration pattern of the cylinder. Singh et al.^[13] explored VIV in a square cylinder with a low Reynolds number ($Re \leq 50$) and low mass ratio, demonstrating that VIV is feasible under these conditions. They also observed that the maximum amplitude and phase difference are strongly influenced by the dimensionless mass ratio, speed, and frequency. The characteristics and influencing parameters of VIV have been extensively investigated to facilitate the development of control technologies.

Techniques for controlling VIV in bluff bodies can be categorized as active and passive control, depending on whether external energy input is required^[14]. Active control techniques require energy input. Existing active measures include steady blowing and suction^[15-16], oscillatory morphing surfaces^[17], synthetic jets^[18-20], dual synthetic jets^[21], and so on. These methods are characterized by strong adaptability and high control efficiency. For instance, Wang et al.^[15] investigated a circular cylinder with two degrees of freedom and examined the effect of windward-suction-leeward-blowing (WSLB) on VIV control. Their research demonstrated that the WSLB combination is more effective than suction or blowing alone and can suppress VIV in the cylinder more rapidly and efficiently. Chen et al.^[16] experimentally evaluated the application of suction in controlling VIV in a circular cylinder and found that the angular position (θ) of the suction holes crucially affects the control performance. The control effect was most pronounced at 90° or 270° , and the lift-force fluctuations were reduced by up to 80%. Wang et al.^[18] studied the effect of synthetic jets on VIV control in circular cylinders. Under all controlled conditions, the fluctuating lift coefficient was lower than in the uncontrolled case, and optimal control reduced it to nearly zero.

Although active control techniques are highly effective in managing flow around bluff bodies, their reliance on continuous energy supply and the associated maintenance challenges limit their practical application in engineering^[22]. In contrast, passive control methods offer advantages such as low cost and ease of implementation and are thus currently more widely used. Common passive control techniques include the use of splitter plates^[23-25], superhydrophobic coatings^[26], helical grooves^[27-28], flexible plates^[29], small rods^[30], and flapping films^[31]. Sahu et al.^[24] evaluated circular cylinders equipped with flow deflectors, where the flow deflectors significantly reduced the amplitude of VIV in the cylinder and expanded the lock-in range. Wang et al.^[27] investigated the effect of helical grooves on controlling VIV in a circular cylinder. They reported that for a circular cylinder

with a single degree of freedom, the amplitude of VIV can be reduced by around 94.4%, almost eliminating the lock-in phenomenon. Hao et al.^[28] studied the impact of triple helical grooves on VIV suppression in circular cylinders through wind tunnel experiments. The grooves reduced the peak VIV by approximately 36% and narrowed the lock-in range.

Among existing control techniques, control rods have the advantages of a simple structure and easy installation, thus demonstrating considerable application potential^[32]. Existing studies have shown that control rods can effectively govern the vortex shedding of circular cylinders. Lu et al.^[33] explored the effect of the spatial arrangement of control rods on VIV in circular cylinders. Their study indicated that the angle of attack associated with different spatial layouts is an important influencing parameter, and that using four control rods provides more effective vibration control than using three. Using numerical simulation, Song et al.^[34] studied the influence of the number of control rods and the gap ratio on the vibration response of circular cylinders, demonstrating that the effectiveness of multiple control rods for vibration control is closely related to the gap ratio and the angle of attack. When nine control rods were arranged in the second symmetric configuration with $L/D = 1.4-1.5$ (L is the distance between the centers of the square cylinder and the control rod, and D is the width of the square cylinder), VIV in the circular cylinder was completely suppressed. Wu et al.^[35] investigated the effects of control rods on VIV in circular cylinders with different gap ratios. The control rods exhibited superior effectiveness in VIV control when the gap ratio was small, and the control effect gradually weakened as the gap ratio increased.

Nevertheless, existing studies rarely address VIV control in square cylinders using control rods. As another typical bluff body, a square cylinder exhibits many distinctive flow characteristics compared to circular cylinders. First, the effect of the Reynolds number on the square cylinder is more pronounced, and its flow characteristics and hydrodynamic forces are more strongly affected by the Reynolds number. Second, the square cylinder is prone to galloping, which complicates vibration suppression. Lastly, the flow around the square cylinder is susceptible to corner interference^[36]. Considering these characteristics, the efficacy of control rods for VIV control in square cylinders has not been studied, especially near the region of maximum pulsation in the cylinder wake.

This study employs numerical simulations to investigate the influence of control rods on VIV in square cylinders. A previous study demonstrated that placing control rods near the region of maximum pulsation in the wake of a circular cylinder can significantly delay vortex interactions and enhance the stability of the shear layer^[37]. Thus, in this study, control rods are strategically placed near the region of the maximum pulsation in the wake of

a square cylinder. First, the VIV characteristics of the square cylinder are studied at a low Reynolds number ($Re = 100$). The variation in the amplitude, mean drag force, and fluctuating lift force of the square cylinder with control rods at different reduced velocities is analyzed. The effects of controlling the hydrodynamic forces and VIV of the square cylinder are examined. The cases in this study are generally designed based on the experimental setup of Egitmen et al. [37]. Three gap ratios and five control rod sizes are considered, resulting in 15 control cases. The effects of the gap ratio (L/D) and control rod size (d/D , where d is the diameter of the control rod) on the efficacy of VIV control are systematically studied. Moreover, three typical cases, including the “Baseline” case and two cases with $L/D = 2.0$ and $d/D = 0.15$ and 0.35 , are evaluated to analyze the fluid behavior. Additionally, the proper orthogonal decomposition (POD) method is used to extract and analyze the dominant modes, enabling identification of the key factors affecting the flow dynamics.

1 Physical Problem and Numerical Methods

1.1 Physical problem

In this study, the incoming flow velocity is kept constant, and the Reynolds number is fixed at 100. The reduced velocity U_r is defined as $U_r = U_\infty / (f_n D)$, where f_n is the natural oscillation frequency of the square cylinder. U_r is varied from 2 to 14 by adjusting f_n . Thus, the maximum pulsation in the wake of the stationary square cylinder at a given Re occurs at the position where $L/D = 2.0$. Therefore, three gap ratios are selected, namely $L/D = 1.5, 2.0, 2.5$. For each gap ratio, five control rod sizes are utilized as follows: 0.15, 0.20, 0.25, 0.30, and 0.35, as shown in Fig. 1. In the subsequent sections

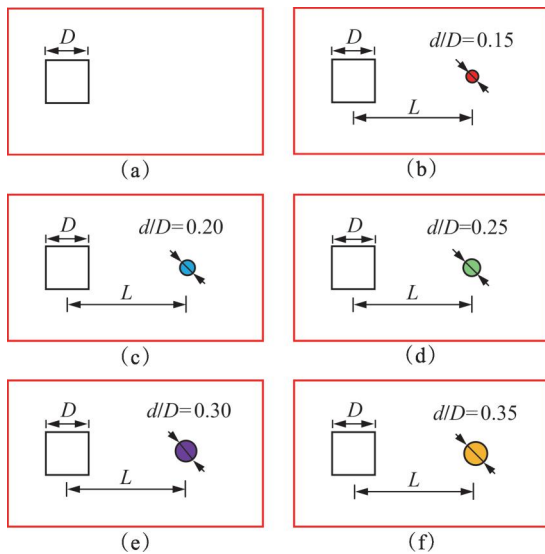


Fig. 1 Schematic diagram of control rod size and gap ratio for each case. (a) Baseline; (b) Case 1; (c) Case 2; (d) Case 3; (e) Case 4; (f) Case 5

of this study, unless otherwise specified, Baseline, Case 1, Case 2, Case 3, Case 4, and Case 5 represent the tested cases corresponding to different control rod sizes.

1.2 Computational domain and framework validation

As shown in Fig. 2, the overall size of the computational domain is $75D \times 50D$. The inlet is defined as a velocity inlet, located $25D$ to the left of the square cylinder. The side boundaries are located $25D$ from the center of the square cylinder and are set as symmetric conditions. The outlet is located $50D$ downstream of the square cylinder and is defined as a pressure outlet. The surfaces of the square cylinder and control rods are subject to no-slip boundary conditions. Specifically, the square cylinder is a single-degree-of-freedom system capable of vibrating vertically, while the downstream control rods are fixed. k is the stiffness, which is not constant and varies with U_r , and c is the damping coefficient.

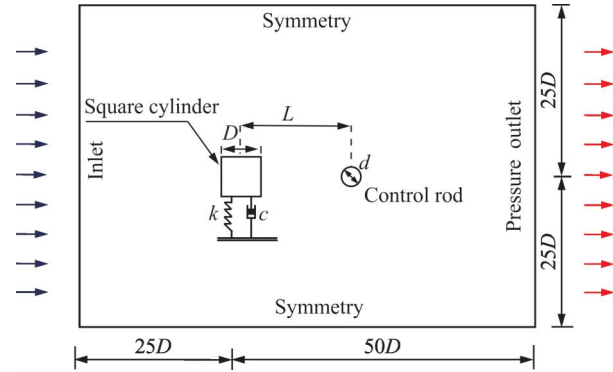


Fig. 2 Schematic diagram of computational domain and boundary conditions

This study employed overset and dynamic mesh techniques to accurately capture boundary layer flows and effectively simulate the vibration of the square cylinder. The grid distribution and local refinement near cylinder walls are illustrated in Fig. 3. The computational domain was split into a background and foreground grids, with data exchanged via an overset boundary. The foreground

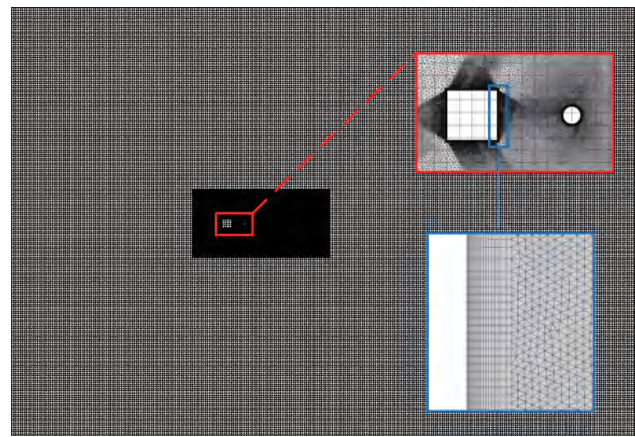


Fig. 3 Schematic diagram of the overall grid and local near-wall grid

region measured $16D \times 8D$. Key regions, including the wake and the areas around the square cylinder and control rods, were refinement to create the foreground grid, enabling precise capture of critical flow zones while allowing independent movement of the foreground grid. To ensure reliable and accurate results, the first grid layer height near the square cylinder and control rod walls was set to 0.01 mm to guarantee $y^+ \leq 1$, with a grid transition growth rate of 1.1 near the walls and 20 boundary layers. A time step of 0.02 s was chosen to balance the computational efficiency and solution accuracy^[38].

The numerical framework was verified by selecting three meshes with different grid numbers. Numerical validation was conducted for the uncontrolled square cylinder at $Re = 100$ and $U_r = 5$. The grid information and calculation results are presented in Table 1, including the mean drag coefficient (C_{d_mean}), root-mean-square lift coefficient (C_{l_rms}), Strouhal number (St), and root-mean-square vertical amplitude (y_{rms}). The differences between the calculation results corresponding to different grid quantities are negligible. Therefore, all three types of grids yield sufficiently accurate results. Based on the computational efficiency and result precision, the medium-density grid was chosen for subsequent computations.

Table 1 Grid independence verification

Mesh	Element number	Re	C_{l_rms}	C_{d_mean}	St	y_{rms}
Coarse	100 300	100	0.66	2.05	0.15	0.21
Medium	123 000	100	0.68	2.05	0.15	0.22
Fine	149 000	100	0.68	2.04	0.15	0.21

The lift coefficient, drag coefficient, Strouhal number, dimensionless amplitude, and reduced velocity are denoted as C_l , C_d , St , y , and U_r , respectively, and are defined as follows:

$$C_l = \frac{2F_y}{\rho U_\infty^2 D} \quad (1)$$

$$C_d = \frac{2F_x}{\rho U_\infty^2 D} \quad (2)$$

$$St = \frac{fD}{U_\infty} \quad (3)$$

$$y = \frac{Y}{D} \quad (4)$$

where F_y and F_x are the forces in the y and x directions, respectively; ρ is the fluid density; f is the vortex shedding frequency of the square cylinder; and Y is the vertical displacement of the square cylinder.

To further validate the developed numerical framework, we selected the same parameters as used by Zhang et al.^[39] and Bearman^[40] for comparison. Fig. 4 presents y_{rms} of the square cylinder under uncontrolled conditions. y_{rms} initially increases with U_r , peaks at $U_r = 5$, and then gradually decreases. The numerical results show good

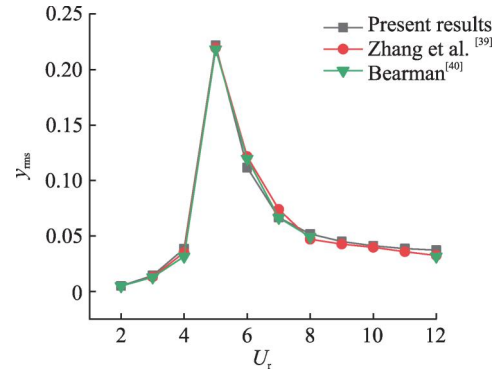


Fig. 4 Variation of y_{rms} for an uncontrolled square cylinder

agreement with those reported by Zhang et al.^[39] and Bearman^[40], which further verifies the accuracy and reliability of the numerical framework.

1.3 Numerical methods

The single-degree-of-freedom motion in the cross-flow direction adheres to Newton's second law of motion. Modeling is performed using a linear mass-spring-damper system. The equation of motion for this system can be written as

$$m\ddot{Y} + c\dot{Y} + kY = F_y(t) \quad (5)$$

where m denotes the structural mass with a fixed value of 0.235 5 kg. The damping coefficient c is set to zero. Here $\ddot{Y}$, $\dot{Y}$, Y correspond to the acceleration, velocity, and position of the square cylinder, respectively.

This study uses ANSYS Fluent software to solve the Navier-Stokes (N-S) equations. The two-dimensional, incompressible, unsteady N-S equations are expressed as

$$\frac{\partial u}{\partial a} + \frac{\partial v}{\partial b} = 0 \quad (6)$$

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial a} + v \frac{\partial u}{\partial b} = -\frac{1}{\rho} \frac{\partial p}{\partial a} + \frac{\mu}{\rho} \left(\frac{\partial^2 u}{\partial a^2} + \frac{\partial^2 u}{\partial b^2} \right) \quad (7)$$

$$\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial a} + v \frac{\partial v}{\partial b} = -\frac{1}{\rho} \frac{\partial p}{\partial b} + \frac{\mu}{\rho} \left(\frac{\partial^2 v}{\partial a^2} + \frac{\partial^2 v}{\partial b^2} \right) \quad (8)$$

where a and b are coordinates in the streamwise and cross-flow directions, respectively; u and v represent velocity components in the x and y directions; t is the flow time; p denotes the pressure. Given the Reynolds number of 100, the laminar model is utilized. To ensure computational accuracy, the discretization scheme is set to the second-order upwind scheme.

2 Results and Discussion

2.1 VIV response and hydrodynamic characteristics

As illustrated in Fig. 5, the variations in y_{rms} , C_{d_mean} , and C_{l_rms} for the square cylinder with reduced velocity are compared for cases with and without control. As indicated in Fig. 5(a), under the uncontrolled condition, y_{rms} of the square cylinder first increases and then decreases

with increasing reduced velocity, peaking at $0.222D$ when $U_r = 5$. After installation, all five control rods of varying sizes effectively suppress the VIV amplitude of the square cylinder within the lock-in region, indicating their applicability. Figs. 5(a), (d), and (g) reveal that the vibration suppression effect is related to the specifications of the control rods. As the size of the control rod gradually increases from $0.15D$ to $0.35D$ within the lock-in region, the VIV suppression effect is progressively enhanced. The maximum suppression effect reaches 83.10% in the case of $L/D = 2.0$ and $d/D = 0.35$. Note that for larger control rods, at higher reduced velocities,

the vibration amplitude in the square cylinder sharply increases with U_r , which may indicate the occurrence of galloping^[41]. This observation indicates that although larger control rods can effectively suppress the VIV resonance in the square cylinder at lower U_r , they may induce galloping at higher U_r , requiring caution in practical engineering applications. The occurrence of galloping is also influenced by the specifications of the control rods. A comparison of Figs. 5(a), (d), and (g) shows that increasing the gap ratio effectively suppresses the galloping phenomenon in the square cylinder, indicating that the gap ratio also plays a significant role in controlling galloping.

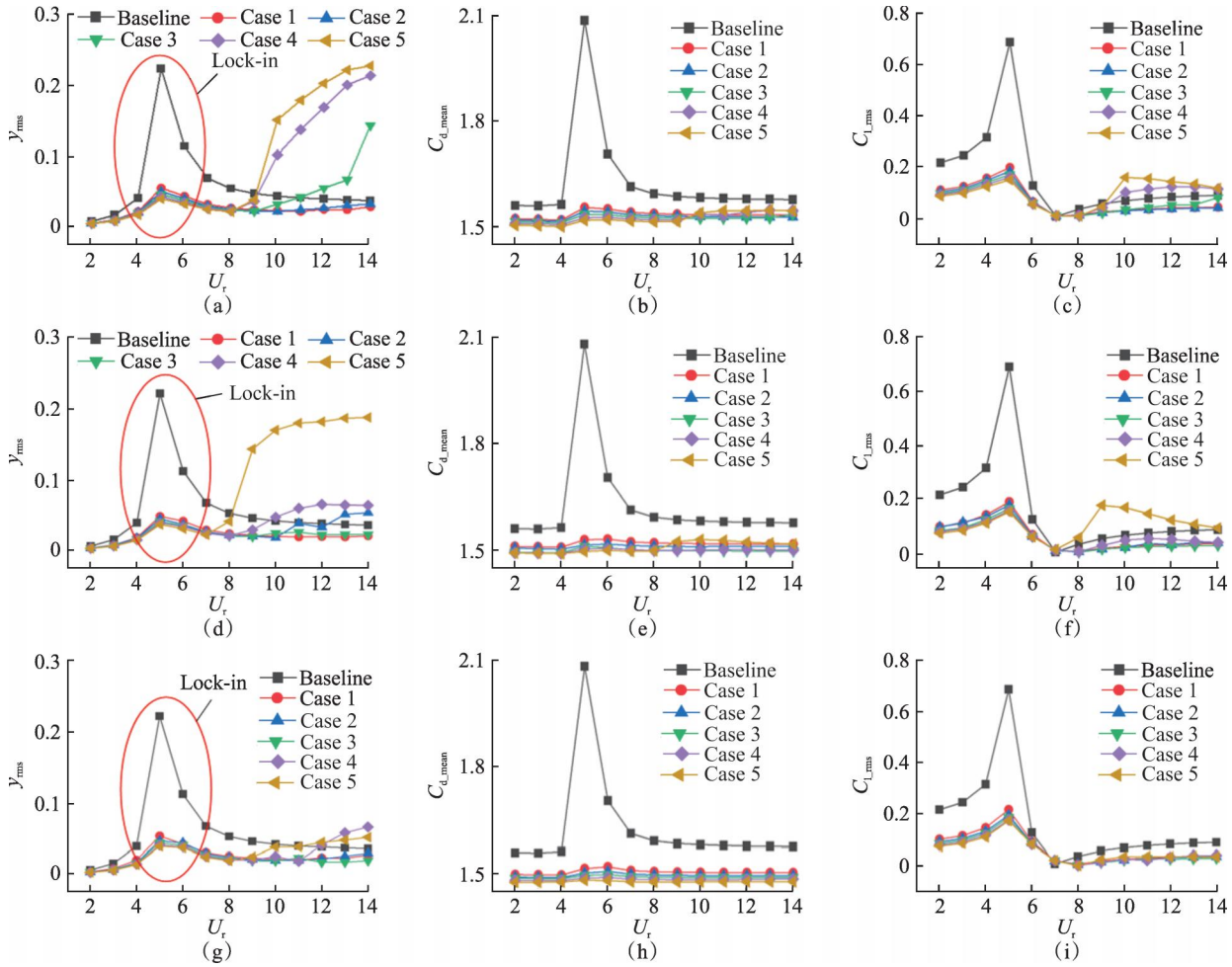


Fig. 5 Effects of control rods with different gap ratios and sizes on controlling VIV in the square cylinder. (a) y_{rms} , $L/D = 1.5$; (b) C_{d_mean} , $L/D = 1.5$; (c) C_{l_rms} , $L/D = 1.5$; (d) y_{rms} , $L/D = 2.0$; (e) C_{d_mean} , $L/D = 2.0$; (f) C_{l_rms} , $L/D = 2.0$; (g) y_{rms} , $L/D = 2.5$; (h) C_{d_mean} , $L/D = 2.5$; (i) C_{l_rms} , $L/D = 2.5$

As shown in Figs. 5(b), (e), and (h), at a fixed gap ratio, the C_{d_mean} of the square cylinder first increases and then decreases as U_r increases, reaching its maximum at $U_r = 5$. Comparison of C_{d_mean} before and after control under different reduced velocities reveals a consistent reduction. Focusing on the effect of control on suppressing the peak C_{d_mean} , it is found that the best drag reduction effect is achieved in Case 5 with $L/D = 2.5$, with a maximum reduction of 28.90%. After installing the control rods,

C_{d_mean} of the square cylinder is suppressed to varying degrees across different reduced velocities. This demonstrates that control rods of various sizes are effective in reducing drag, further validating the effectiveness of this passive control method.

As observed in Figs. 5(c), (f), and (i), the C_{l_rms} values for the selected cases correspond to values at various reduced velocities. For the Baseline case, the overall trend of C_{l_rms} with U_r is similar to that of the VIV ampli-

tude and mean drag, but with some differences: Before $U_r = 5$, C_{l_rms} increases as the velocity decreases, and reaches a maximum at $U_r = 5$. For $U_r = 5-7$, the C_{l_rms} of the square cylinder decreases gradually but increases slightly in the range of $U_r = 7-14$. As observed in Figs. 5(c), (f), and (i), after introducing the control rod, the peaks of C_{l_rms} are significantly suppressed, with the maximum reduction in C_{l_rms} reaching 78.13% at $L/D = 1.5$ and $d/D = 3.5$. Except for Case 5, for which C_{l_rms} exhibits a downward trend at $L/D = 1.5$ and $L/D = 2.0$ (specifically, C_{l_rms} follows a downward trend for $U_r = 10-14$), C_{l_rms} remains invariant relative to the Baseline case. For $U_r = 10-14$, C_{l_rms} for the square cylinder is suppressed in all cases except for Case 5 at $L/D = 1.5$ and Cases 4 and 5 at $L/D = 2.0$, where it is larger than that of the uncontrolled case.

2.2 Analysis of energy input and power spectral density

The time-histories of y and C_l for the square cylinder in the controlled and uncontrolled configurations are presented in Fig. 6. For the square cylinder, y and C_l decrease after installing the control rod. At $U_r = 5$, the

time-history curves for all controlled cases exhibit an overall similar trend: C_l is in phase with y . A comparison of Figs. 6(a), (d), and (e) reveals that the peak points of the time-history curves for y and C_l of the square cylinder are significantly reduced after control. Thus, Cases 1 and 5 with $L/D = 2.0$ are selected as representative cases for further analysis of the energy input.

According to the definition of energy input for a vibrating system proposed by Dong et al. [42], within one vibration cycle, the dimensionless energy input to the system by the fluid is denoted as W , which is defined as follows:

$$W = \frac{1}{n} \int_0^{nT} C_l(t) \dot{y} dt \quad (9)$$

where n is the number of cycles, which is not less than 20; T represents the vibration period; $C_l(t)$ denotes the instantaneous lift coefficient of the square cylinder.

As shown in Figs. 6(d) and (e), the phase difference between y and C_l remains unchanged, but the lift force acting on the system decreases significantly. According to the definition of W , the energy input from the fluid to the system is reduced, as illustrated in Fig. 7, which ultimately leads to a decrease in the vibration amplitude in the square cylinder. W decreases gradually as the size of

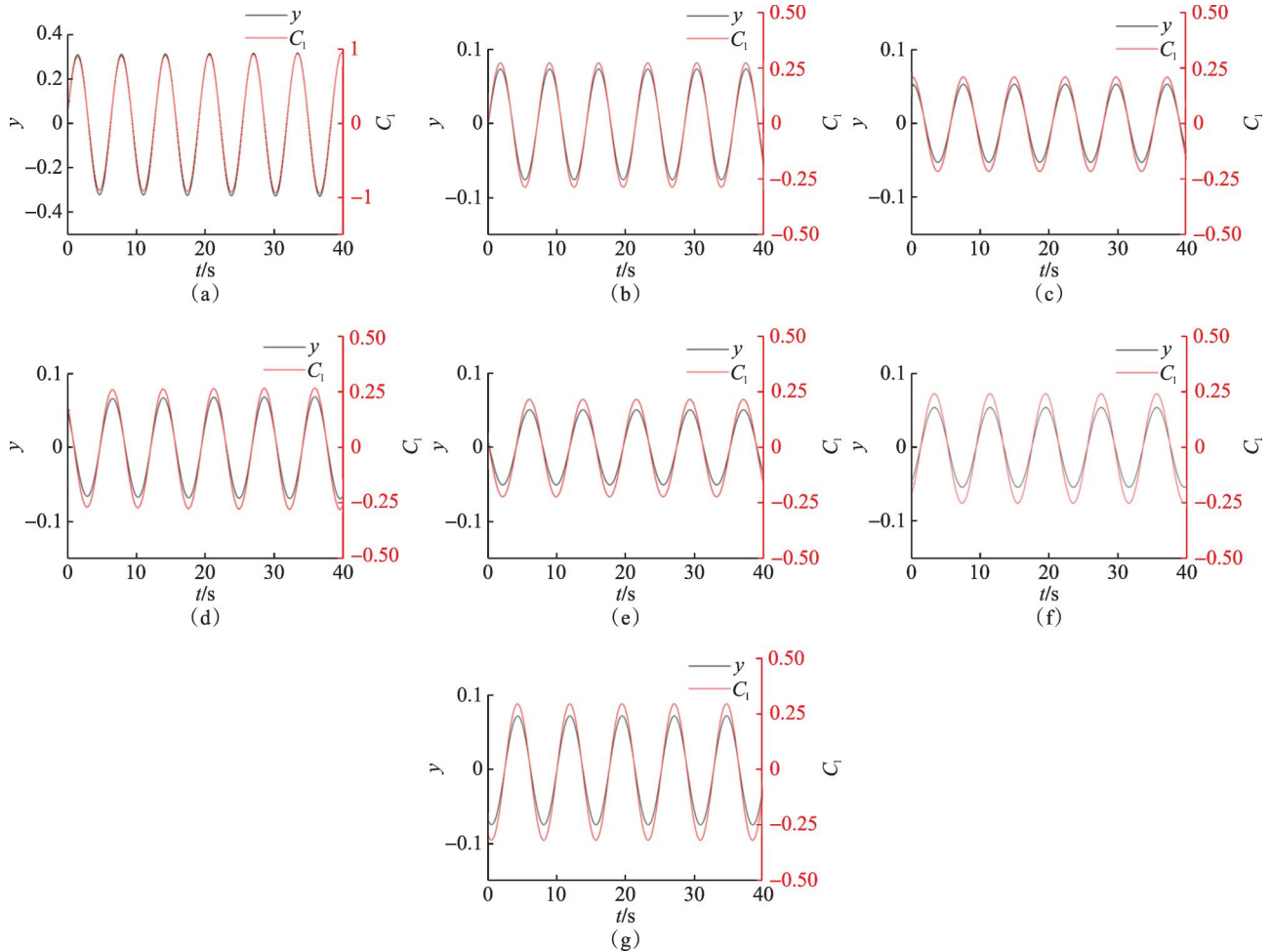


Fig. 6 Time-history curves of C_l and y for selected cases at $U_r = 5$. (a) Baseline; (b) $L/D = 1.5$, Case 1; (c) $L/D = 1.5$, Case 5; (d) $L/D = 2.0$, Case 1; (e) $L/D = 2.0$, Case 5; (f) $L/D = 2.5$, Case 1; (g) $L/D = 2.5$, Case 5

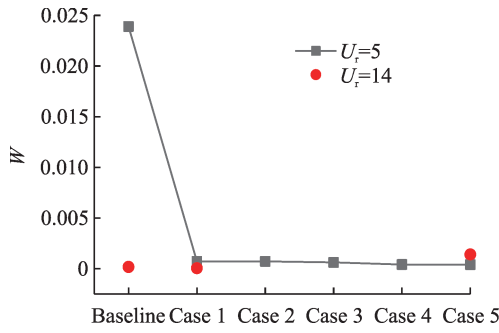


Fig. 7 Work done over one cycle for selected cases at $U_r = 5$ ($L/D = 2.0$)

the control rod increases. Thus, y_{rms} decreases with increasing control rod size, as illustrated in Fig. 5(a).

Analysis of Fig. 5 indicates that galloping may occur in Case 5 with gap ratios $L/D = 1.5$ and $L/D = 2.0$ at $U_r = 14$. Therefore, the phase characteristics of typical cases at high reduced velocities are also analyzed, specifically Cases 1 and 5 with three gap ratios at $U_r = 14$. As shown in Fig. 8(a), under the uncontrolled condition, C_l and y adopt anti-phase configurations, which

means the structure is only accelerated by the lift when it moves toward the equilibrium position ($Y = 0$), thereby suppressing vibration of the cylinder. By comparing the values of C_l and y at different moments in Figs. 8(b), (d), and (f) with those in Fig. 8(a), it is found that after the application of control, the phases of C_l and y remain in-phase, but the peak of C_l is significantly reduced. The vibration suppression mechanism is similar to that at $U_r = 5$. The decrease in C_l reduces the work done, and the vibration amplitude in the square cylinder decreases accordingly. The results in Figs. 8(c) and (e) indicate that the phases of C_l and y shift from anti-phase to in-phase when $d/D = 0.35$. This indicates that C_l promotes an increase in y , leading to a “positive feedback” mechanism between the structure and the fluid, which in turn triggers structural galloping^[43-44]. Another factor causing galloping is observed in Figs. 8(c) and (e). C_l increases when galloping occurs. According to the energy input mechanism proposed by Dong et al.^[42], the work done by the fluid increases, thereby increasing y for the square cylinder, as shown in Fig. 7.

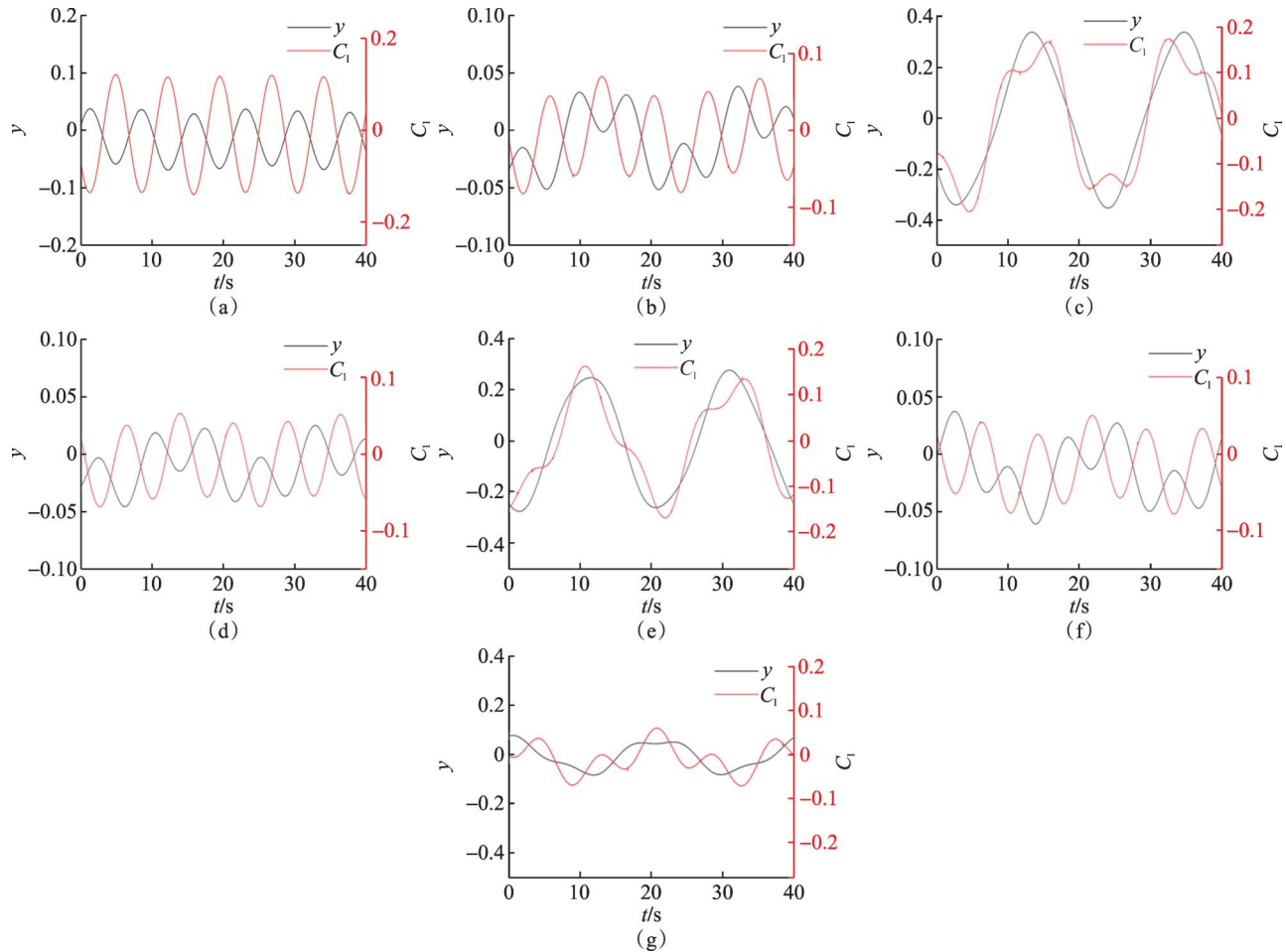


Fig. 8 Time-history curves of C_l and y for selected cases with $U_r = 14$. (a) Baseline; (b) $L/D = 1.5$, Case 1; (c) $L/D = 1.5$, Case 5; (d) $L/D = 2.0$, Case 1; (e) $L/D = 2.0$, Case 5; (f) $L/D = 2.5$, Case 1; (g) $L/D = 2.5$, Case 5

As depicted in Fig. 9, the PSDs of C_l and y reflect the vortex shedding and vibration frequencies for selected

cases at a gap ratio of $L/D = 2.0$. Both the horizontal and vertical axes are presented on logarithmic scales. The ab-

scissa at the peak value represents the primary vibration frequency, and the corresponding dominant frequency is highlighted in the figure. After installing the control rods, the dominant frequency of the square cylinder de-

creases and continues to decrease as d/D increases. This indicates that the control rods influence the vortex shedding frequency, thereby altering the vibration frequency of the square cylinder^[45].

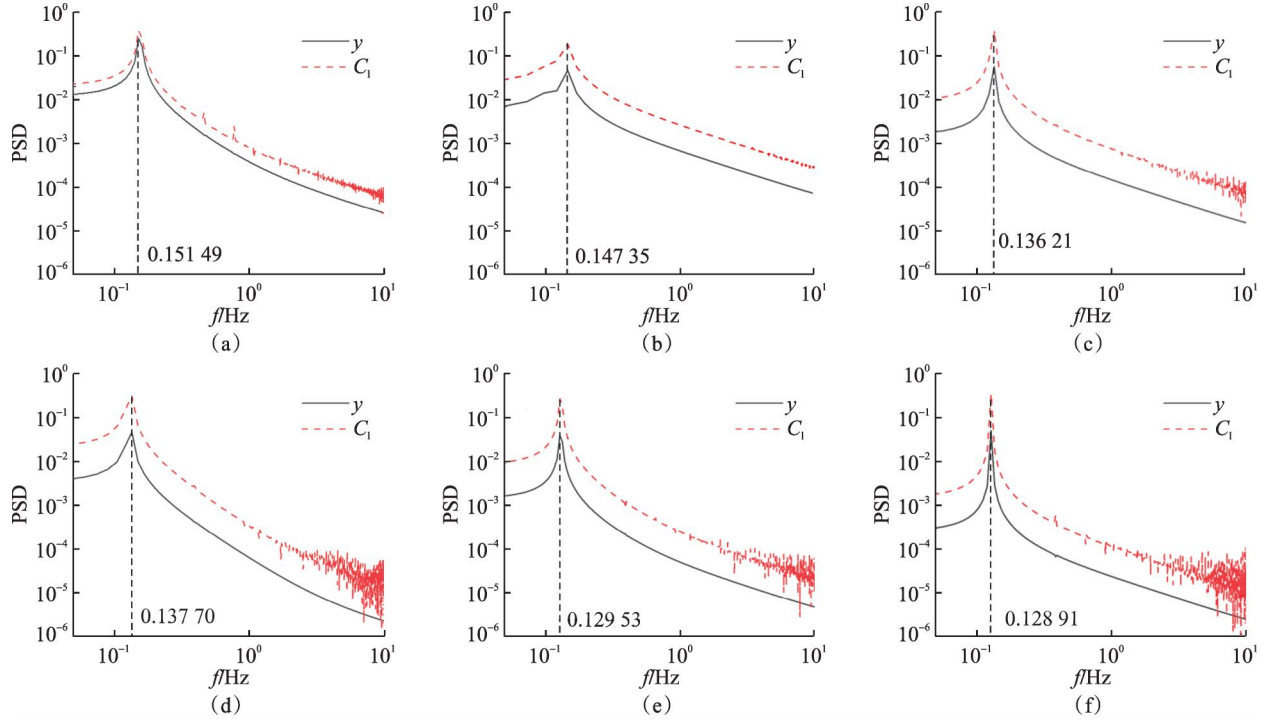


Fig. 9 Spectra of typical cases at $L/D = 2.0$ and $U_r = 5$. (a) Baseline; (b) Case 1; (c) Case 2; (d) Case 3; (e) Case 4; (f) Case 5

2.3 Flow dynamics

To elucidate the underlying physics by which control suppresses VIV and hydrodynamic forces, the velocity, vorticity, and pressure distributions around the cylinder are evaluated for selected cases, as shown in Figs. 10-13.

Fig. 10 shows the horizontal velocity contour of the square cylinder over one vibration cycle at $L/D = 2.0$ and $U_r = 5$ for $0, T/4, T/2$, and $3T/4$, which correspond to the instants when the square cylinder is at the equilibrium

position ($Y = 0$) while moving upward, at the highest position ($Y = Y_{\max}$), at the equilibrium position ($Y = 0$) while moving downward, and at the lowest position ($Y = -Y_{\max}$), respectively. X and Y in the figure are dimensionless coordinates. The velocity field distribution is represented by the dimensionless horizontal velocity U^* , where $U^* = U/U_\infty$. During the movement of the square cylinder from 0 to $T/4$, vortices form and are then shed from the upper surface of the square cylinder. In

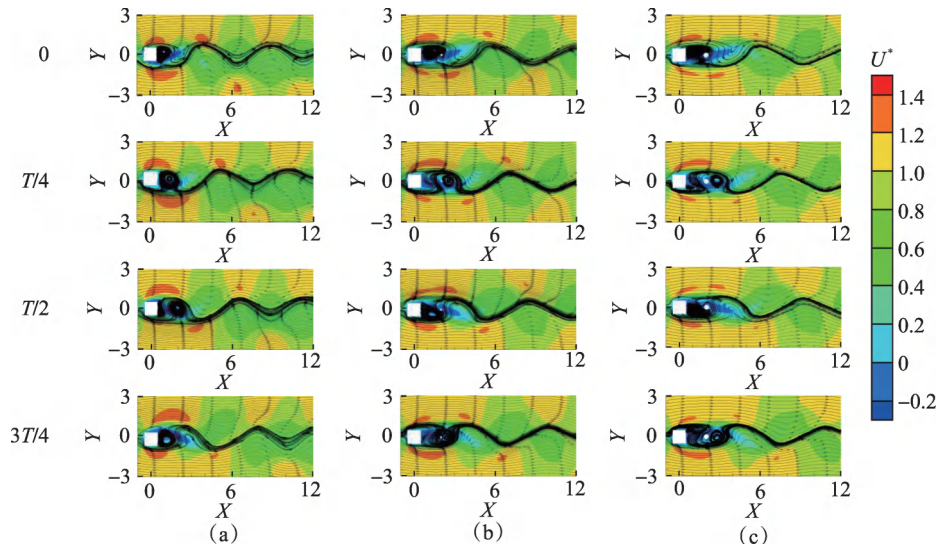


Fig. 10 Velocity contour distribution at $L/D = 2.0$ and $U_r = 5$. (a) Baseline; (b) Case 1; (c) Case 5

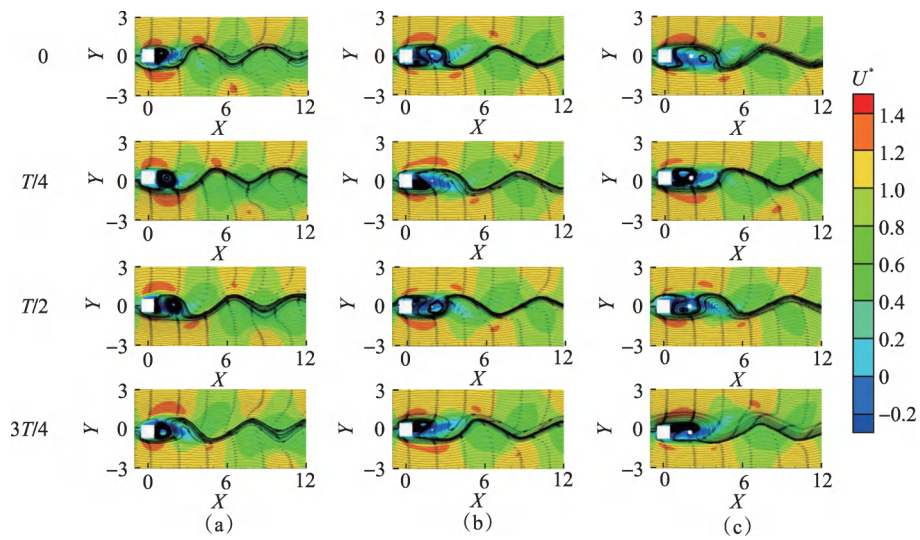


Fig. 11 Velocity contour distribution at $L/D = 2.0$ and $U_r = 14$. (a) Baseline; (b) Case 1; (c) Case 5

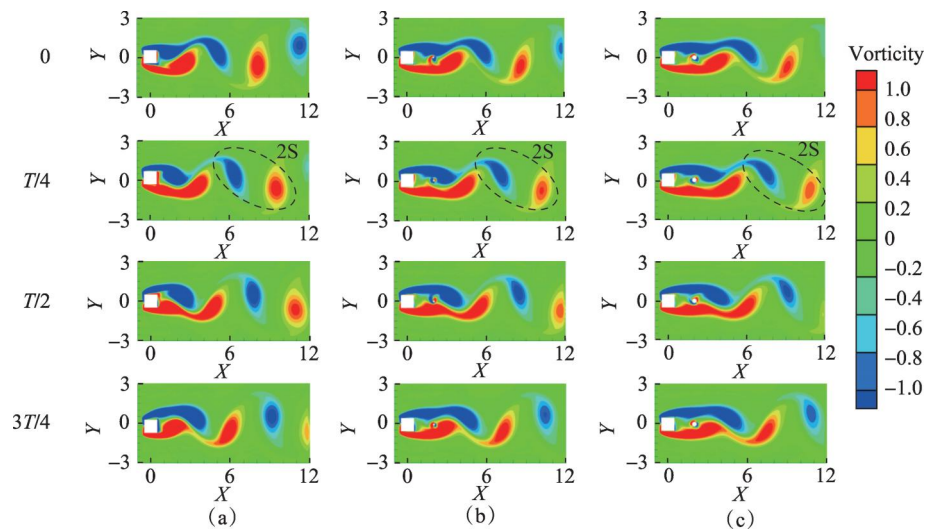


Fig. 12 Vorticity contour distribution at $L/D = 2.0$ and $U_r = 5$. (a) Baseline; (b) Case 1; (c) Case 5

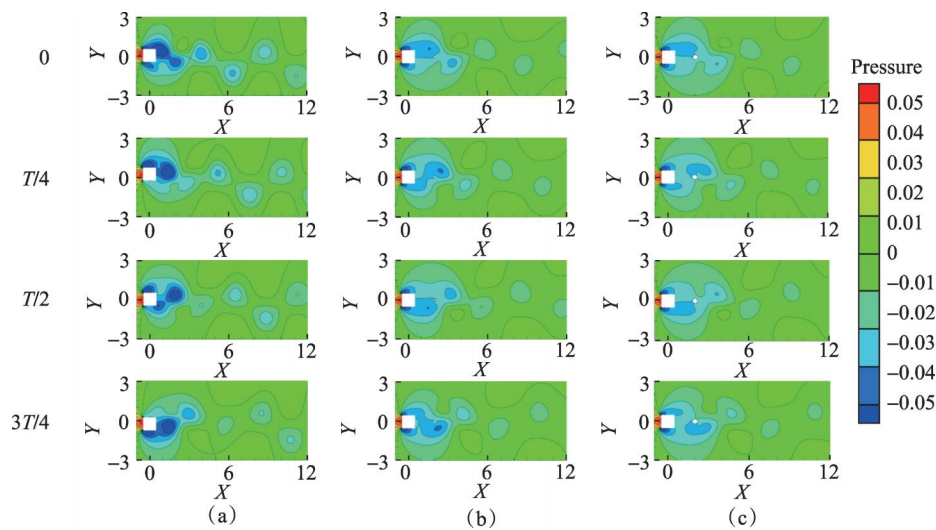


Fig. 13 Pressure contour distribution at $L/D = 2.0$ and $U_r = 5$. (a) Baseline; (b) Case 1; (c) Case 5

comparison, the control rods accelerate the movement of the vortices away from the surface of the square cylinder, reducing the duration of the vortices' action on the

square cylinder and decreasing the energy input to the system, in accordance with the phenomenon observed in Fig. 7.

As the size of the control rod increases, the vortex formation length increases, causing the vortices to move farther away from the cylinder surface. This reduces their interference with the cylinder surface, thereby enhancing the control efficiency^[46]. Fig. 10 further illustrates that although the range of the negative velocity region expands after implementing control measures, the minimum velocity in this region increases significantly, thereby decreasing the absolute value of the negative velocity. This observation indicates that compared to the uncontrolled case, the velocity distribution becomes more uniform after control. The control rod significantly alters the characteristics of the velocity distribution, effectively suppresses the development of low-velocity fluid in the wake region, and enhances the stability of the flow wake.

As shown in Fig. 11, the velocity contours for the Baseline, Case 1, and Case 5 are presented at $L/D = 2.0$ and $U_r = 14$. For the Baseline case, two vortices are generated and shed within one vibration cycle. In comparison, four vortices are generated and shed within one vibration cycle in Case 5. Relative to the Baseline case, this represents a significant increase in the number of vortices shed per cycle, indicating greater energy input to the square cylinder. Thus, the vibration amplitude in the square cylinder increases, which is one of the factors contributing to galloping^[47].

The instantaneous vorticity contours over one vibration cycle for the Baseline, Case 1, and Case 5 for $L/D = 2.0$ and $U_r = 5$ are presented in Fig. 12. Fig. 13 shows the instantaneous pressure contours at the same moments and for the same cases as in Fig. 12. As shown in Fig. 12, in the Baseline case, shear layers alternately appear on the surface of the square cylinder before rolling up to form a Kármán vortex street with a “2S” shedding pattern. However, with small cylinders installed, the shear layers on the surfaces of the square cylinder collide with those on the surfaces of the small cylinder, suppressing rolling-up of the shear layers and extending the vortex formation length. Corresponding to the pressure field in Fig. 13, after implementing control, the pressure in the upstream region of the square cylinder remains unchanged, while the absolute value of the negative pressure in the downstream region decreases. This indicates a reduction in the horizontal pressure difference across the square cylinder, which in turn decreases $C_{d,mean}$, consistent with the phenomenon shown in Fig. 5 (e). As shown in Fig. 13, when the square cylinder moves to the highest position, the negative pressure region on its upper surface is larger than that on the lower surface, resulting in an upward force. Similarly, when it moves to the lowest position, a downward force is exerted, consistent with the observations in Figs. 6 (d) and (e). The pressure distribution becomes more uniform in Cases 1 and

5, where the pressure difference between the upper and lower surfaces is reduced, weakening the upward or downward forces acting on the square cylinder during its motion and thus causing significant reductions in both its vibration amplitude and the fluctuating lift force. This explains the decrease in y_{rms} and $C_{l,rms}$ observed in Figs. 5 (d) and (i); namely, the control rod can significantly suppress the peaks of y_{rms} and $C_{l,rms}$ ^[48]. As shown in Fig. 12, in Cases 1 and 5, the control rod generates vortices during one vibration cycle of the square cylinder. When the square cylinder is at 0, the shear layer on the upper surface of the control rod rolls up to form a counterclockwise vortex, while a clockwise vortex is generated on the lower surface. In contrast, the vortices shed from the surfaces of the square cylinder rotate in opposite directions: a clockwise vortex forms on the upper surface, while a counterclockwise vortex forms on the lower surface. As the square cylinder moves from 0 to $T/4$, the vortex on its upper surface interacts with the vortex generated on the upper surface of the control rod. Because of their opposite rotation directions, the intensity of the original vortex on the upper surface of the square cylinders weakens after interacting with the vortex from the control rod. Similarly, as the square cylinder moves from $T/2$ to $3T/4$, the intensity of the vortex on its lower surface is also reduced by the counter-rotating vortices produced by the control rod^[49-50]. Ultimately, the vortices generated by the control rods, which rotate in directions opposite to those in the square cylinder’s wake, interfere with the evolution of the vortices. This interaction provides an effective mechanism for controlling VIV from a fluid perspective, further confirming the advantages and potential of this control method in suppressing such vibration.

Existing studies have indicated that the magnitude of $C_{l,rms}$ reflects the intensity of vortices^[51]. The $C_{l,rms}$ in Case 5 is smaller than that in Case 1, suggesting that the vortices in the wake of the former are weaker than those of the latter (Fig. 5(i)). As shown in Fig. 12, when the size of the control rod increases from $d/D = 0.15$ to $d/D = 0.35$, the vortex formation length increases significantly. Thus, the vortices develop further downstream, and their interference with the surface of the square cylinder becomes weaker. Therefore, as the size of the control rod increases, the vortex formation length increases significantly, the intensity of the vortex weakens, and the interference with the cylinder surface diminishes, leading to a gradual improvement in the control effect^[47].

2.4 Proper orthogonal decomposition analysis

POD is an order reduction tool for analyzing complex flow fields, which offers an efficient framework for analyzing coherent structures in wake flows^[52-53]. Since Lumley^[54] first introduced POD into the field of fluid me-

chanics, POD analysis has been widely used for identifying coherent structures in flow fields and extracting the dominant modes governing the flow fields.

The POD method was employed to analyze the Baseline, Case 1, and Case 5 at $L/D = 2.0$. Fig. 14 illustrates the energy distribution for the first 25 modes, with the abscissa denoting the mode number and the ordinate representing the energy contribution of each mode relative to the total energy. As indicated in Fig. 14, in all cases, the first few modes account for a large proportion of the energy, while the energy contribution decreases rapidly with increasing mode number. The installation of the control rod leads to the redistribution of the energy contribution of each mode. For the Baseline case, the first-order mode accounts for 51% of the energy. After introducing the control rod, the corresponding values are 51.7% and 50.5% for Cases 1 and 5, respectively, indicating little variation. In contrast, the energy contribution for the second-order mode changes considerably, increasing from 38% in the Baseline case to 44.3% and 44.5% for Cases 1 and 5, respectively. Consequently, the first two modes account for more than 90% of the to-

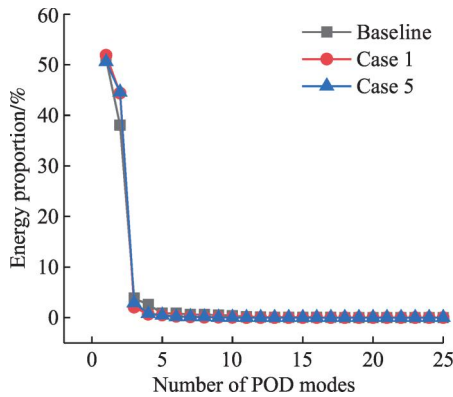


Fig. 14 Energy proportion of each POD mode under the conditions of $L/D = 2.0$ and $U_r = 5$

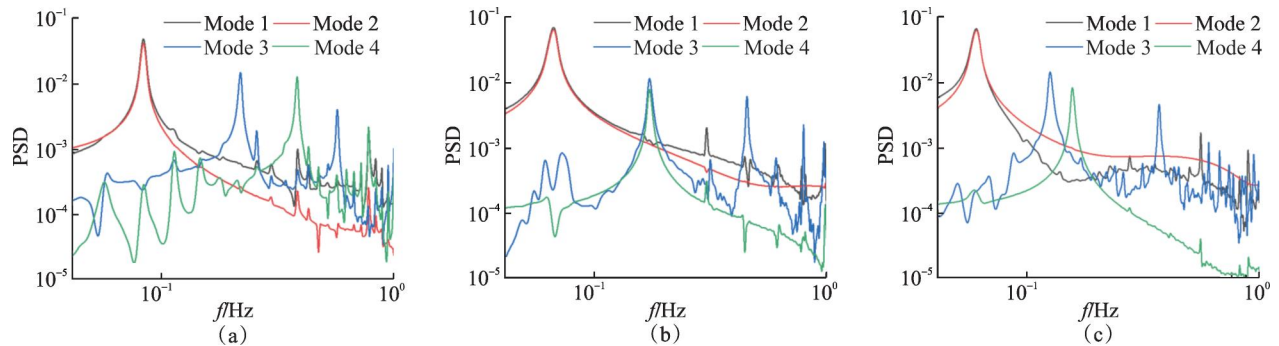


Fig. 15 Power spectral density of the first four POD modes of the square cylinder at $L/D = 2.0$ and $U_r = 5$. (a) Baseline; (b) Case 1; (c) Case 5

Fig. 16 presents the vorticity distributions of the first four POD modes, along with a comparison among the three cases under the conditions of $L/D = 2.0$ and $U_r = 5$. The vorticity contours of the POD modes effectively reveal the spatial characteristics of the flow field. As

shown in Fig. 16, in all three cases, both Modes 1 and 2 exhibit large-scale, periodically alternating vortex pair structures that closely resemble the classic Kármán vortex street. Based on the principle that low-order POD modes dominate the fundamental characteristics of the

total energy. As such, these first two modes are the primary contributors to the main fluid structure. Accordingly, it is tentatively postulated that the control rods primarily influence the characteristics of the first two modes in the flow field, thereby inducing changes in the overall flow dynamics^[55-56]. Based on the quantitative energy proportion of each mode, the first four modes collectively account for 95% of the total energy across all three cases, which is sufficient for analyzing the major flow characteristics of the fluid. Therefore, an in-depth analysis was performed for these four dominant modes. Fig. 15 shows the distribution characteristics of the fluid energy in terms of the frequency, with logarithmic coordinates on the vertical and horizontal axes. The peak frequency corresponds to the dominant frequency of the mode, while the number and amplitude of peak points indicate the degree of energy concentration. The energy distributions of the 1st and 2nd POD modes are concentrated for the three cases, whereas the peaks of the 3rd and 4th POD modes are relatively scattered with lower amplitudes. Comparison of the data in Figs. 8(a), (c), and (d) shows that the dominant frequency of vortex shedding aligns well with that of the 1st- and 2nd-order modes. Examination of the PSD curves of each mode reveals that the 1st- and 2nd-order modes are consistently conjugated, indicating that the dominant frequency of the vortex remains governed by a single frequency after control. As shown in Figs. 5(a), (d), and (e), the 3rd- and 4th-order modes display conjugate distribution with small control rods (Case 1) but lose conjugacy when the control rods are excessively large (Case 5). Because low-order modes largely govern vortex shedding and the influence of high-order modes is negligible, variations in the conjugacy of the 3rd- and 4th-order modes do not affect the vortex shedding frequency^[57].

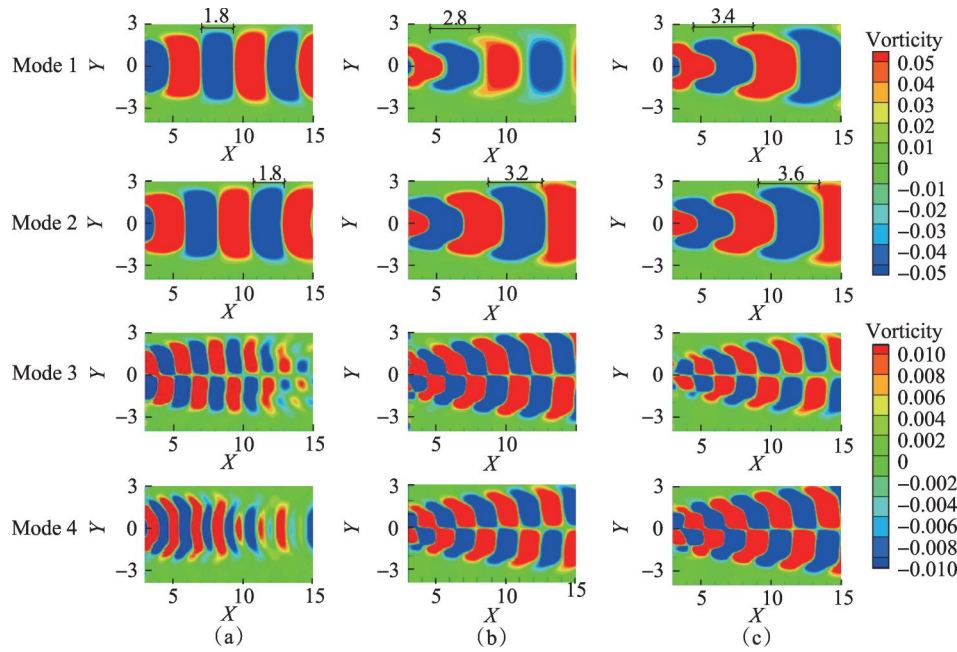


Fig. 16 Vorticity distribution of the first four POD modes at $L/D = 2.0$ and $U_r = 5$. (a) Baseline; (b) Case 1; (c) Case 5

flow field, and considering the variation in the energy distribution among different modes, a detailed comparison of the vortex structures in the first two modes before and after implementing control is presented. The application of the control rod does not change the overall pattern of vortex shedding but significantly alters the spacing between vortices. In the Baseline case, the streamwise length^[58] of the vortex in the first mode is $1.8D$ (Fig. 16). Under the effect of the small control rod, the streamwise length of the vortex stretches to $2.8D$, and with the larger control rod, it is further elongated to $3.4D$. The second mode shows a similar trend with an even more remarkable variation. The streamwise length of the vortex increases from $1.8D$ in the Baseline case to $3.2D$ and $3.6D$ for Cases 1 and 5. This variation aligns well with the extended vortex formation length observed in the flow field, demonstrating that the control rod operates by modifying the spatial scale of the vortices. The observed phenomenon is consistent with that presented in Fig. 12.

In contrast, Modes 3 and 4 exhibit fragmented, multi-scale vortex structures with significantly lower energy proportions, primarily reflecting the nuances of the local flow. In summary, low-order modes dominate the macroscopic core flow mechanism, whereas high-order modes reflect secondary details. Implementation of the control rod mainly modifies the characteristics of the shear layer by regulating the vortex structures in low-order modes, thereby suppressing VIV in the square cylinder. In contrast, the high-order modes play an auxiliary role and have a relatively minor impact on the overall flow dynamics^[59].

3 Conclusions

This study investigated the effectiveness of control rods in suppressing VIV and hydrodynamic forces in a square cylinder. The influence of the size of the control rod and the gap ratio on the amplitude and hydrodynamic forces experienced by the square cylinders was evaluated through flow dynamics and POD analyses, revealing the underlying control mechanisms. The main conclusions are as follows:

(1) Installing control rods can significantly reduce the VIV amplitude, where the vibration suppression effect is closely related to both the size and spacing of the rods. Optimal control is achieved at $L/D = 2.0$ and $d/D = 0.35$, where the maximum suppression rate of y_{rms} reaches 83.10%. C_{d_mean} is reduced by up to 28.90% at $L/D = 2.5$ and $d/D = 0.35$, while the maximum reduction in C_{l_rms} reaches 78.13% at $L/D = 2.0$ and $d/D = 0.35$. For a fixed gap ratio, the control efficacy increases with the size of the control rod. However, at high reduced velocities, larger control rods may cause the amplitude of the square cylinder to increase with the reduced velocity, potentially triggering galloping.

(2) Time-domain and frequency-domain analyses reveal that at $U_r = 5$, the control rod does not alter the phase relationship between y and C_l but significantly reduces the lift force acting on the system. This reduction decreases the work done on the system by the fluid, ultimately reducing the vibration amplitude of the square cylinder. At $U_r = 14$, however, the control rods alter the phase between y and C_l , inducing the “positive feedback” mechanism between the structure and fluid, which triggers structural galloping.

(3) Observations of the wake fluid in the square cylinder reveal that the control rods suppress the development of low-velocity regions in the wake, resulting in a more uniform velocity distribution. The vortex formation length in the wake increases and moves away from the cylinder surface, thereby reducing the forces exerted on the cylinder. The control rods generate vortices that rotate in the opposite direction to those on the corresponding surface of the square cylinder. The interaction of these vortices weakens the intensity of the vortices on the surface of the square cylinder. This ultimately suppresses y_{rms} and $C_{l_{rms}}$. Additionally, the pressure distribution in the wake becomes more uniform, with a significant reduction in the negative pressure zone. These changes in the pressure distribution provide a basis for the excellent drag reduction achieved by the control rods.

(4) POD analysis reveals that Modes 1 and 2 dominated the fluid with and without control, indicating their dominant role in vortex shedding in the square cylinder. The application of control rods primarily increases the vortex formation length by lengthening the vortex in the first two modes. Furthermore, the vortex shedding frequencies of the first two modes are conjugated and are consistent with the actual vortex shedding frequency. These observations confirm that low-order modes dominate the flow field characteristics, with high-order modes playing an auxiliary role.

This study has certain limitations. First, this work focuses on relatively low Reynolds numbers; it does not account for higher regimes where inertial effects become dominant. Subsequent studies will explore these higher-Reynolds-number conditions to broaden the scope of the findings. Furthermore, the influence of three-dimensional effects is neglected in this study; thus, subsequent research will focus on three-dimensional calculations. Finally, this study investigates only the effect of a single method for controlling VIV. In future research, we will explore the differences in the control mechanisms of other control methods and control rods, and seek a composite control strategy based on the synergistic effects of such mechanisms.

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控制棒对上游方柱涡激振动控制效果的研究

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摘要: 采用数值模拟方法, 探究了控制棒放置于方柱尾流中脉动速度最大的位置附近对其水动力和涡激振动(VIV)的抑制效果。系统研究了5种不同的控制棒尺寸和3种不同的间隙比。 Re 数固定为100。综合采用时域与频域分析、流场特性分析和本征正交分解法, 揭示了控制棒作用下方柱流动特性的内在物理机理。结果表明, 布置控制棒能显著改变方柱VIV响应, 控制效果与尺寸和间距有关。选择最佳的控制棒参数, 振幅、脉动升力、平均阻力的最大降幅可达83.10%、78.13%、28.90%。施加控制棒后, 尾流中旋涡拉长并向下游发展, 并且方柱表面的涡与控制棒表面产生的反向涡相互融合, 使流场对系统输入的能量与旋涡脱强度降低, 这是导致方柱振幅减小的主要原因。同时, 控制后方柱尾流的压力分布更加均匀。此外, 控制棒尺寸越大, 对方柱振幅、平均阻力、脉动升力的峰值点抑制效果越好。本征正交分解结果表明, 前2阶模态的流场特性具有主导作用, 且二者的共轭关系不受控制棒影响。

关键词: 方柱; 涡激振动; 控制棒; 流场特性; 本征正交分解法; 数值模拟

中图分类号: O358