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论文题目：**Freely rotating triangular elements for passive control of circular-cylinder vortex-induced vibration**

Freely rotating triangular elements for passive control of circular-cylinder vortex-induced vibration

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Abstract

Freely rotating triangular elements are proposed as a compact passive control concept for vortex-induced vibration (VIV) of a circular cylinder, while geometrically matched fixed elements are considered as reference configurations. Two-dimensional numerical simulations at a nominal Reynolds number of $Re = 10^4$ are combined with wind-tunnel force measurements for a bare cylinder and twelve controlled configurations at free-stream velocities of 2 m.s^{-1} and 5 m.s^{-1} . The results show that the aerodynamic response depends strongly on element position, number and mounting condition. Numerically, a single fixed upstream element reduces the peak-to-peak lift fluctuation by 76.9%, but increases the mean-lift magnitude to approximately 5.9 times that of the bare cylinder. A fixed top-bottom pair reduces the dominant frequency by 66.7% while increasing the peak-to-peak lift by 12.0%, demonstrating that a shift in shedding frequency does not necessarily imply attenuation of the unsteady loading. In the wind-tunnel measurements at 5 m.s^{-1} , the fixed upstream configuration produces the smallest recorded transverse force, 0.00 N at the available measurement resolution, with a 3.7% increase in drag. Among the freely rotating arrangements, the upstream-downstream pair gives the most favourable combined performance, reducing the transverse force by 25% while increasing drag by 3.7% relative to the bare cylinder. Passive rotation also reduces the drag of the two- and four-element arrangements by approximately 20–31% relative to their fixed counterparts. These results identify the fixed upstream element and freely rotating upstream-downstream pair as promising configurations for passive VIV control. Longer coupled simulations and repeated synchronous measurements of force, cylinder displacement and element rotation are required to determine their statistically stationary vibration-suppression performance.

Keywords: circular cylinder; vortex-induced vibration; passive flow control; triangular control element; freely rotating fin

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1. Introduction

Subsea pipelines connect offshore wells, processing facilities and export infrastructure. Local scour, uneven seabed topography or installation tolerances can leave portions of a nominally supported pipeline suspended as free spans. Flow past such an approximately circular span produces separated shear layers and alternating vortices. The resulting unsteady transverse force can excite repeated cross-flow oscillations and accelerate fatigue damage accumulation. Similar physics affects risers, mooring components, bridge cables, chimneys and other slender bluff bodies. Vortex-induced vibration (VIV) is therefore both a canonical nonlinear fluid–structure interaction and a practical structural-integrity problem (Williamson and Govardhan 2004; Sarpkaya 2004; Gabbai and Benaroya 2005).

The response of an elastically mounted cylinder cannot be inferred from the stationary-cylinder wake alone. When the vortex-shedding frequency approaches a structural natural frequency, the wake and body motion can synchronise over a finite reduced-velocity interval. Within this lock-in region, displacement, fluid force, shedding frequency and phase evolve together, often through distinct response branches and wake modes (Williamson and Govardhan 2004; Bearman 2011). Reynolds number, mass ratio, structural damping, stiffness, reduced velocity and the available mechanical degrees of freedom all affect the measured response. Consequently, a decrease in mean lift, peak force or shedding frequency does not by itself demonstrate VIV suppression. The statistically stationary structural amplitude must also decrease under matched structural and flow parameters.

Passive control remains attractive offshore because it does not require continuous power or a feedback controller. Established devices include helical strakes, shrouds, fairings, splitter plates and other near-wake stabilisers (Zdravkovich 1981; Choi, Jeon, and Kim 2008). Helical strakes disrupt the coherence of separation along the span and can reduce the VIV response, but their additional projected area usually imposes a drag penalty and can increase installation loads (Zhou et al. 2011). Streamlined fairings offer another route, yet their performance depends on alignment, rotational stability and operating range. The engineering problem is therefore multi-objective: transverse excitation should be reduced without unacceptable drag penalties, steady transverse loading, galloping, or added mechanical complexity.

Small auxiliary bodies placed around a main cylinder provide a more local form of wake control. A control body can intercept a separated shear layer, change its roll-up distance and

alter the communication between the two sides of the wake. The sensitive-region experiments of Strykowski and Sreenivasan (1990) showed that a small secondary cylinder can suppress vortex shedding at low Reynolds number when placed in a restricted region of the near wake. Later studies examined two, four and eight control cylinders, demonstrating that number, diameter, clearance and azimuthal placement can shift a system among attenuation, ordinary VIV and galloping-like response (Chen, Ji, and Xu 2019; Silva-Ortega and Assi 2017a). External rotation of the auxiliary cylinders adds momentum near separation and can substantially reduce vibration, although the direction and rotation ratio remain critical control parameters (Silva-Ortega and Assi 2017b; Zhu and Gao 2017).

Free rotation is physically different from prescribed rotation. It adds no direct power input; the element orientation emerges from the competition among the hydrodynamic torque, rotational inertia, mechanical friction and any restoring torque. Free-to-rotate plates and short-tail fairings can suppress VIV when they settle into a stable deflected state, but low rotational resistance can permit persistent oscillation and high resistance can prevent alignment or promote galloping (Assi, Bearman, and Kitney 2009; Assi, Bearman, and Tognarelli 2014). A numerical study of a single triangular fairing similarly found that a stable orientation could attenuate the cylinder response, whereas strong fairing flapping could amplify vibration (Zhu et al. 2017). These studies show that “self-adaptation” is not an inherent property of a rotating device. It is a dynamic state that must be demonstrated through the angular history and its coupling to the cylinder motion.

The present concept differs from a continuous fairing surrounding the cylinder. Multiple spatially separated equilateral triangles are positioned at selected azimuths around the main body, and each movable element is intended to rotate independently about its centroid. The sharp vertices fix the separation points more strongly than a smooth circular rod, so even a small angular change can alter the projected blockage and the directions of the shear layers shed from the elements. This gives a plausible mechanism for disturbing the phase and strength of the main-cylinder wake. It also creates new failure modes: independent elements may lock at unequal angles, respond out of phase, collide with the cylinder if clearance is insufficient, or introduce low-frequency load modulation.

The objectives are to (i) quantify the effects of fin number and azimuthal placement on the reported mean lift, peak-to-peak lift and dominant frequency; (ii) compare fixed and freely rotating variants without conflating geometric blockage with rotational dynamics; (iii) use all available experimental lift and drag measurements to identify configuration-dependent trade-offs.

2. Theoretical and analysis framework

2.1. Flow and force coefficients

For an incompressible flow, the Reynolds and Strouhal numbers are

$$Re = \frac{\rho U_\infty D}{\mu} = \frac{U_\infty D}{\nu}, \quad St = \frac{f D}{U_\infty}. \quad (1)$$

For a two-dimensional computation reported per unit span, the drag, lift and moment coefficients are

$$C_D = \frac{F_D}{1/2 \rho U_\infty^2 D}, \quad C_L = \frac{F_L}{1/2 \rho U_\infty^2 D}, \quad C_M = \frac{M_z}{1/2 \rho U_\infty^2 D^2}. \quad (2)$$

The surface forces follow from integration of pressure and viscous traction. In vector form,

$$\mathbf{F} = \int_{\Gamma_b} [-p \mathbf{n} + \boldsymbol{\tau} \cdot \mathbf{n}] \, ds, \quad M_z = \int_{\Gamma_b} (\mathbf{r} - \mathbf{r}_p) \times [-p \mathbf{n} + \boldsymbol{\tau} \cdot \mathbf{n}] \, ds, \quad (3)$$

where Γ_b is the relevant body surface and $\mathbf{r}_p$ is the pivot position.

For a stationary record, the minimum useful set is the mean, root-mean-square fluctuation and peak-to-peak range,

$$\overline{C_L} = \frac{1}{T} \int_{t_0}^{t_0+T} C_L(t) \, dt, \quad C_{L,rms} = \left[\frac{1}{T} \int_{t_0}^{t_0+T} (C_L - \overline{C_L})^2 \, dt \right]^{1/2}, \quad C_{L,pp} = C_{L,max} - C_{L,min}. \quad (4)$$

Peak-to-peak range is retained because it is the only fluctuation metric supplied for the CFD cases. It is more sensitive than an rms value to isolated excursions and window selection.

2.2. Cylinder and element dynamics

An elastically mounted cylinder restricted to cross-flow translation obeys

$$m\ddot{y} + c\dot{y} + ky = F_L(t), \quad (5)$$

with natural frequency $f_n = (2\pi)^{-1} \sqrt{k/m}$, damping ratio $\zeta = c/(2\sqrt{km})$, mass ratio and reduced velocity

$$m^* = \frac{m}{\rho \pi D^2/4}, \quad U^* = \frac{U_\infty}{f_n D}. \quad (6)$$

The sign of the cycle-averaged power

$$\overline{P}_L = \frac{1}{T} \int_{t_0}^{t_0+T} F_L(t) \dot{y}(t) \, dt \quad (7)$$

distinguishes net fluid-to-structure energy input from net dissipation. This phase-sensitive quantity is more mechanistically informative than force amplitude alone.

Each triangular element rotating about a fixed centroid satisfies

$$I_\theta \ddot{\theta} + c_\theta \dot{\theta} + k_\theta (\theta - \theta_0) = M_z(t). \quad (8)$$

Ideal free rotation has $k_\theta = 0$; a real bearing still has inertia and friction. Useful nondimensional groups are

$$I_\theta^* = \frac{I_\theta}{\rho D^4}, \quad c_\theta^* = \frac{c_\theta}{\rho U_\infty D^3}, \quad k_\theta^* = \frac{k_\theta}{\rho U_\infty^2 D^2}, \quad \frac{g}{D}, \quad \frac{P}{D}. \quad (9)$$

The angle response should be classified as convergence to a stable angle, bounded periodic oscillation, intermittent switching or continuous rotation. These states can generate very different wakes even when the external geometry is identical.

2.3. Performance metrics

The fixed-fin CFD values are normalised by the bare-cylinder values from the same reported dataset:

$$R_A = \frac{C_{L,pp}}{C_{L,pp,0}}, \quad R_f = \frac{f}{f_0}, \quad R_L = \frac{|\overline{C_L}|}{|\overline{C_{L,0}}|}. \quad (10)$$

For the experiment, comparison is restricted to a common speed:

$$R_L(U) = \frac{F_L(U)}{F_{L,0}(U)}, \quad R_D(U) = \frac{F_D(U)}{F_{D,0}(U)}. \quad (11)$$

The workbook does not state whether F_L is signed, mean, mean absolute, rms or peak; it is therefore described as a recorded lift-force magnitude. For a diagnostic of the speed dependence, a two-point apparent exponent is defined from drag,

$$n_D = \frac{\ln[F_D(5)/F_D(2)]}{\ln(5/2)}. \quad (12)$$

If reference area, density and C_D were unchanged, one would expect $F_D \propto U^2$ and $n_D = 2$. With only two speeds, n_D is not fitted as a physical law.

3. Numerical methodology

3.1. Geometric configurations and computational cases

The system considered here consists of a circular cylinder of diameter D surrounded by one or more passive triangular control elements. Each element has an equilateral triangular cross-section of side length $P = 0.1D$ and height $\sqrt{3}P/2 = 0.0866D$. The elements are placed at selected azimuthal positions to alter the separating shear layers and the subsequent near-wake development.

The azimuthal angle θ is measured clockwise from the upstream stagnation point. Accordingly, $\theta = 0^\circ, 90^\circ, 180^\circ$ and 270° denote the upstream, top, downstream and bottom

positions, respectively. The investigated layouts comprise a single element, two diametrically opposed elements and four elements distributed at the four cardinal positions.

Two mounting conditions are considered. In the fixed configurations, the triangular elements are attached directly to the cylinder and retain a prescribed orientation. In the freely rotating configurations, each element is mounted on a pin passing through its centroid, allowing its angular position to respond passively to the instantaneous hydrodynamic torque. The base of each triangular element is initially tangential to the cylinder and is located at a radial distance of $0.6D$ from the cylinder centre. Since the cylinder radius is $0.5D$, the corresponding initial clearance between the cylinder surface and the triangular base is $g = 0.1D$.

The complete set of configurations used in the experiments is listed in Table 1. The CFD dataset covers the bare cylinder and five fixed configurations: F-1-U, F-1-S, F-2-UD, F-2-TB and F-4. Two freely rotating configurations, R-2-TB and R-4, are additionally considered in the moving-cylinder simulations. The remaining configurations are included only in the experimental comparison.

Table 1: Control-element configurations and available data sources.

Code	Mounting	N_f	Azimuthal arrangement	Available data
Bare	None	0	None	CFD, experiment
F-1-U	Fixed	1	Upstream	CFD, experiment
F-1-D	Fixed	1	Downstream	Experiment
F-1-S	Fixed	1	Top or bottom	CFD, experiment
F-2-UD	Fixed	2	Upstream–downstream	CFD, experiment
F-2-TB	Fixed	2	Top–bottom	CFD, experiment
F-4	Fixed	4	Four cardinal positions	CFD, experiment
R-1-U	Freely rotating	1	Upstream	Experiment
R-1-D	Freely rotating	1	Downstream	Experiment
R-1-S	Freely rotating	1	Top or bottom	Experiment
R-2-UD	Freely rotating	2	Upstream–downstream	Experiment
R-2-TB	Freely rotating	2	Top–bottom	CFD, experiment
R-4	Freely rotating	4	Four cardinal positions	CFD, experiment

The numerical simulations consider flow past a two-dimensional circular cylinder of diameter $D = 1$ m in water at a free-stream velocity of $U_\infty = 0.01 \text{ m} \cdot \text{s}^{-1}$. These conditions correspond to the reported nominal Reynolds number $Re = U_\infty D / \nu \simeq 10^4$. The fixed elements are attached directly to the cylinder surface, whereas the freely rotating elements are separated from it by the initial clearance $g = 0.1D$. Consequently, comparisons between the fixed and freely rotating families involve simultaneous changes in both the rotational

constraint and the radial position of the elements. Their individual effects therefore cannot be isolated in the present configurations.

The height of the triangular elements is the smallest geometric scale in the fixed-element cases, so local mesh refinement is required near the sharp corners. For the freely rotating elements, the clearance introduces an additional narrow passage between the cylinder and the triangular base. Accurate resolution of the pressure and viscous stresses within this passage is important because they contribute to the torque about the pivot. The moving-grid treatment must also preserve adequate cell quality as the elements rotate.



Top-bottom freely rotating pair (R-2-TB). Four freely rotating elements (R-4).

Figure 1: Freely rotating triangular-element configurations considered in the moving-cylinder simulations. The sketches show the initial positions of the elements.

3.2. Numerical formulation and fluid-body coupling

The flow around the cylinder and triangular control elements was simulated using an unsteady pressure-based finite-volume method. The fluid was treated as incompressible and Newtonian. For the moving-body cases, the governing equations can be written in an arbitrary Lagrangian-Eulerian (ALE) form as

$$\nabla \cdot \bar{\mathbf{u}} = 0, \quad (13a)$$

$$\rho \left[\frac{\partial \bar{\mathbf{u}}}{\partial t} \right]_x + (\bar{\mathbf{u}} - \mathbf{u}_g) \cdot \nabla \bar{\mathbf{u}} = -\nabla \bar{p} + \nabla \cdot [2(\mu + \mu_t) \bar{\mathbf{S}}], \quad (13b)$$

where $\bar{\mathbf{u}}$ and $\bar{p}$ denote the resolved velocity and pressure fields, respectively, $\mathbf{u}_g$ is the local grid velocity, and $\bar{\mathbf{S}} = [\nabla \bar{\mathbf{u}} + (\nabla \bar{\mathbf{u}})^T]/2$ is the mean strain-rate tensor. The molecular and turbulent dynamic viscosities are denoted by μ and μ_t , respectively. For stationary meshes, $\mathbf{u}_g = \mathbf{0}$.

The cylinder motion was restricted to transverse translation, whereas each freely rotating triangular element was allowed to rotate only about its centroidal axis normal to the two-

dimensional flow plane. All other rigid-body degrees of freedom were constrained. The corresponding equations of motion are

$$m_y \ddot{y} + c_y \dot{y} + k_y y = F_y(t), \quad (14a)$$

$$I_{\theta,i} \ddot{\theta}_i + c_{\theta,i} \dot{\theta}_i + k_{\theta,i} (\theta_i - \theta_{i,0}) = M_{z,i}(t), \quad i = 1, \dots, N_f, \quad (14b)$$

where m_y , c_y and k_y are the cylinder mass, damping and stiffness in the transverse direction, respectively. The quantities $I_{\theta,i}$, $c_{\theta,i}$ and $k_{\theta,i}$ denote the moment of inertia, rotational damping and rotational stiffness of the i th triangular element. For a freely rotating element, $k_{\theta,i} = 0$, while a fixed element satisfies $\theta_i = \theta_{i,0}$.

The hydrodynamic force and moment acting on each body were obtained by integrating the surface traction over the corresponding body surface:

$$\mathbf{F} = \int_{\Gamma_b} (-\bar{p}\mathbf{n} + 2(\mu + \mu_t)\bar{\mathbf{S}} \cdot \mathbf{n}) d\Gamma, \quad (15a)$$

$$M_{z,i} = \int_{\Gamma_i} [(\mathbf{r} - \mathbf{r}_{p,i}) \times (-\bar{p}\mathbf{n} + 2(\mu + \mu_t)\bar{\mathbf{S}} \cdot \mathbf{n})] \cdot \mathbf{e}_z d\Gamma, \quad (15b)$$

where $\mathbf{r}_{p,i}$ is the pivot position of the i th element. At each time step, the fluid loads were transferred to the rigid-body solver, after which the body positions and computational mesh were updated. This two-way coupling allows the instantaneous orientation of the freely rotating elements to be determined by the balance among hydrodynamic moment, rotational inertia and mechanical resistance.

3.3. Computational domain and numerical implementation

The two-dimensional computational domain is shown in 2. The cylinder centre was located $15D$ downstream of the inlet and $25D$ upstream of the outlet. The upper and lower boundaries were each positioned $15D$ from the cylinder centre, resulting in a domain of $40D \times 30D$. This arrangement provides sufficient upstream distance for imposing a uniform inflow and sufficient downstream extent for the cylinder wake to develop.

A uniform velocity U_∞ was prescribed at the inlet, while a zero-gauge-pressure condition was imposed at the outlet. Symmetry boundary conditions were applied to the upper and lower boundaries. The cylinder and triangular-element surfaces were treated as no-slip walls. In the moving-body calculations, the wall velocity was updated according to the instantaneous translational or rotational motion of the corresponding body.

The cylinder diameter and free-stream velocity were $D = 1 \text{ m}$ and $U_\infty = 0.01 \text{ m.s}^{-1}$, respectively. With the kinematic viscosity of water used in the simulations, the Reynolds number was

$$Re = \frac{U_\infty D}{\nu} = 10^4. \quad (16)$$

The inlet turbulence intensity and turbulent-viscosity ratio were specified as 5% and 10, respectively.

Pressure–velocity coupling was achieved using the SIMPLE algorithm. The momentum convection terms were discretised using a second-order upwind scheme, while the diffusion terms were evaluated by central differencing. A second-order implicit scheme was employed for temporal discretisation. The dynamic-mesh procedure updated the fluid domain in response to cylinder translation and element rotation.

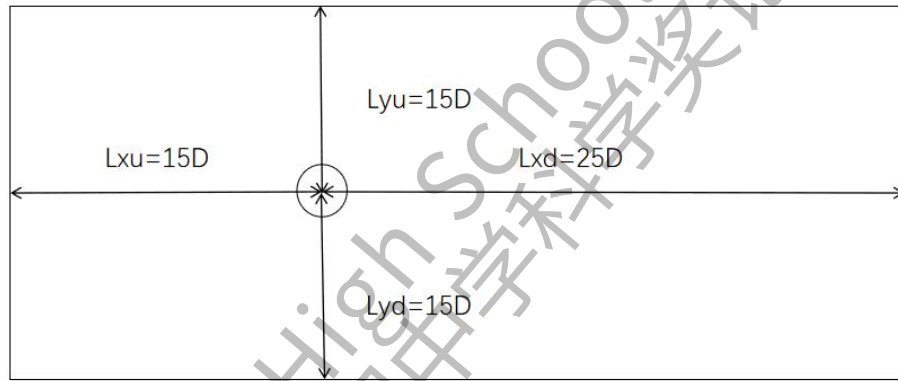


Figure 2: Computational domain and boundary conditions used for the cylinder–control–element simulations.

4. Experimental methodology

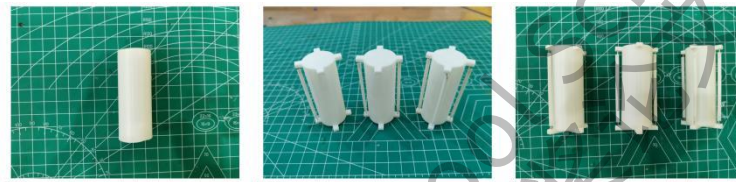
4.1. Wind-tunnel models and test conditions

Wind-tunnel experiments were performed to examine whether the force-modification trends identified numerically also occur for physical cylinder models. The experimental programme covered the bare cylinder and the fixed and freely rotating triangular-element configurations defined in Table 1. The models were fabricated by additive manufacturing, as shown in 3. In the fixed configurations, the triangular elements were integral to the cylinder model. In the

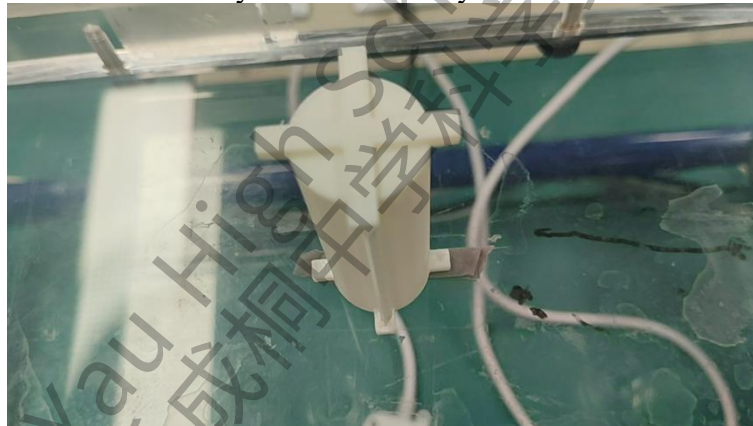
freely rotating configurations, the elements were mounted on external pins through their centroids, allowing rotation about axes parallel to the cylinder axis.

The term “freely rotating” denotes the mechanical mounting condition: the triangular element was not constrained to maintain a fixed orientation. No actuator or imposed rotational motion was used. Its instantaneous orientation was instead governed by the combined effects of aerodynamic torque, rotational inertia and mechanical resistance at the pivot.

The models were installed in a transparent wind-tunnel test section, as shown in 3. Each configuration was tested at nominal free-stream velocities of $U_\infty = 2 \text{ m.s}^{-1}$ and 5 m.s^{-1} . The bare-cylinder measurement at the same velocity served as the reference for evaluating the effect of each control-element arrangement.



Additively manufactured cylinder models.



Installation of a model in the wind-tunnel test section.

Figure 3: Experimental models and representative wind-tunnel installation. The model set includes the bare cylinder, fixed triangular elements and pin-supported rotating elements.

4.2. Force measurements

The streamwise and transverse forces acting on each cylinder model were measured at the two free-stream velocities. The measured streamwise and transverse forces are denoted by F_D and F_L , respectively. The transverse force is represented by its magnitude because the retained measurements are non-negative and do not distinguish between positive and negative lift directions.

At each velocity, measurements were first made on the bare cylinder and then repeated for the fixed and freely rotating configurations. This permits direct comparison between models under the same nominal flow condition. Force readings were recorded to a precision of 0.01 N. A recorded value of 0.00 N was retained as reported and was not replaced by an assumed small positive value.

The freely rotating elements were installed without active angular control. Accordingly, the measured force represents the coupled response of the cylinder and the passively oriented elements over the measurement interval. The experiments did not include synchronous measurements of element angle, angular velocity or aerodynamic torque; the force data therefore quantify the overall performance of each configuration but do not resolve its rotational dynamics independently.

4.3. Force normalisation and comparison metrics

Because the experiments were conducted at two free-stream velocities, the effect of each control-element arrangement was evaluated relative to the bare cylinder at the same velocity. For configuration j , the normalised lift and drag ratios are defined as

$$R_{L,j}(U_\infty) = \frac{F_{L,j}(U_\infty)}{F_{L,0}(U_\infty)}, \quad (17a)$$

$$R_{D,j}(U_\infty) = \frac{F_{D,j}(U_\infty)}{F_{D,0}(U_\infty)}, \quad (17b)$$

where the subscript 0 denotes the bare cylinder. Values of $R_{L,j} < 1$ and $R_{D,j} < 1$ indicate reductions in the measured transverse and streamwise forces, respectively. The corresponding percentage changes are

$$\Delta F_{L,j} = (R_{L,j} - 1) \times 100\%, \quad (18a)$$

$$\Delta F_{D,j} = (R_{D,j} - 1) \times 100\%. \quad (18b)$$

A configuration that reduces the transverse force but incurs a large drag penalty is not uniformly beneficial. The experimental results are therefore also presented in the (R_D, R_L) plane. The lower-left region ($R_D < 1, R_L < 1$) represents simultaneous reductions in both force components, whereas the lower-right region represents a reduction in transverse force accompanied by increased drag.

For matched geometric arrangements, the effect of releasing the rotational degree of freedom is quantified by

$$R_{L,R/F} = \frac{F_{L,\text{rotating}}}{F_{L,\text{fixed}}}, \quad (19a)$$

$$R_{D,R/F} = \frac{F_{D,rotating}}{F_{D,fixed}}. \quad (19b)$$

These ratios are evaluated only when the corresponding fixed-element force is non-zero.

The variation of drag force between the two velocities is further characterised by an apparent velocity exponent,

$$n_D = \frac{\ln [F_D(5 \text{ m.s}^{-1})/F_D(2 \text{ m.s}^{-1})]}{\ln(5/2)}. \quad (20)$$

For dynamically similar incompressible flow with an approximately constant drag coefficient, one expects $F_D \propto U_\infty^2$ and hence $n_D \simeq 2$. In the present study, n_D is used only as a consistency diagnostic rather than as a fitted drag law, because it is based on two velocity conditions.

5. Numerical results

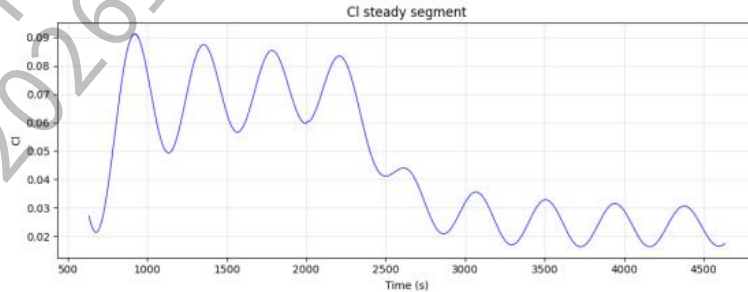
5.1. Baseline cylinder response

For the bare cylinder, the mean lift coefficient, peak-to-peak lift fluctuation and dominant frequency are $\overline{C_L} = 0.0452$, $C_{L,pp} = 0.0748$ and $f_s = 0.00225$ Hz, respectively. With $D = 1$ m and $U_\infty = 0.01$ m.s⁻¹, the corresponding Strouhal number is

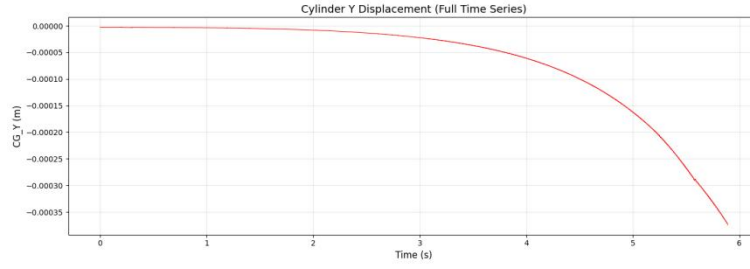
$$St = \frac{f_s D}{U_\infty} = 0.225, \quad (21)$$

which is within the range expected for vortex shedding from a circular cylinder (Norberg 2003).

The lift coefficient exhibits an oscillatory response superimposed on a slowly varying mean component, as shown in 4. The non-zero value of $\overline{C_L}$ indicates that the wake is not perfectly symmetric over the evaluated interval.



Lift-coefficient history.



Initial transverse displacement.

Figure 4: Aerodynamic and transverse responses of the bare cylinder.

5.2. Force characteristics of fixed elements

The lift statistics obtained for the bare cylinder and five fixed triangular-element configurations are compared in Table 2 and Figure 5. Here, $R_A = C_{L,pp}/C_{L,pp,0}$ denotes the peak-to-peak lift ratio and $R_f = f_s/f_{s,0}$ denotes the dominant frequency ratio, where the subscript 0 refers to the bare cylinder.

Table 2: Lift statistics of the fixed triangular-element configurations.

Configuration	$\overline{C_L}$	$C_{L,pp}$	R_A	f_s (Hz)	St	R_f
Bare	0.0452	0.0748	1.000	0.002250	0.2250	1.000
Top-bottom	0.0062	0.0838	1.120	0.000750	0.0750	0.333
Upstream-downstream	0.1116	0.1309	1.750	0.000508	0.0508	0.226
Top single	-2.5550	2.3107	30.892	0.002250	0.2250	1.000
Upstream single	0.2664	0.0173	0.231	0.002250	0.2250	1.000
All four	0.0308	3.4726	46.425	0.001750	0.1750	0.778

R_A and R_f are normalised by the corresponding bare-cylinder values.

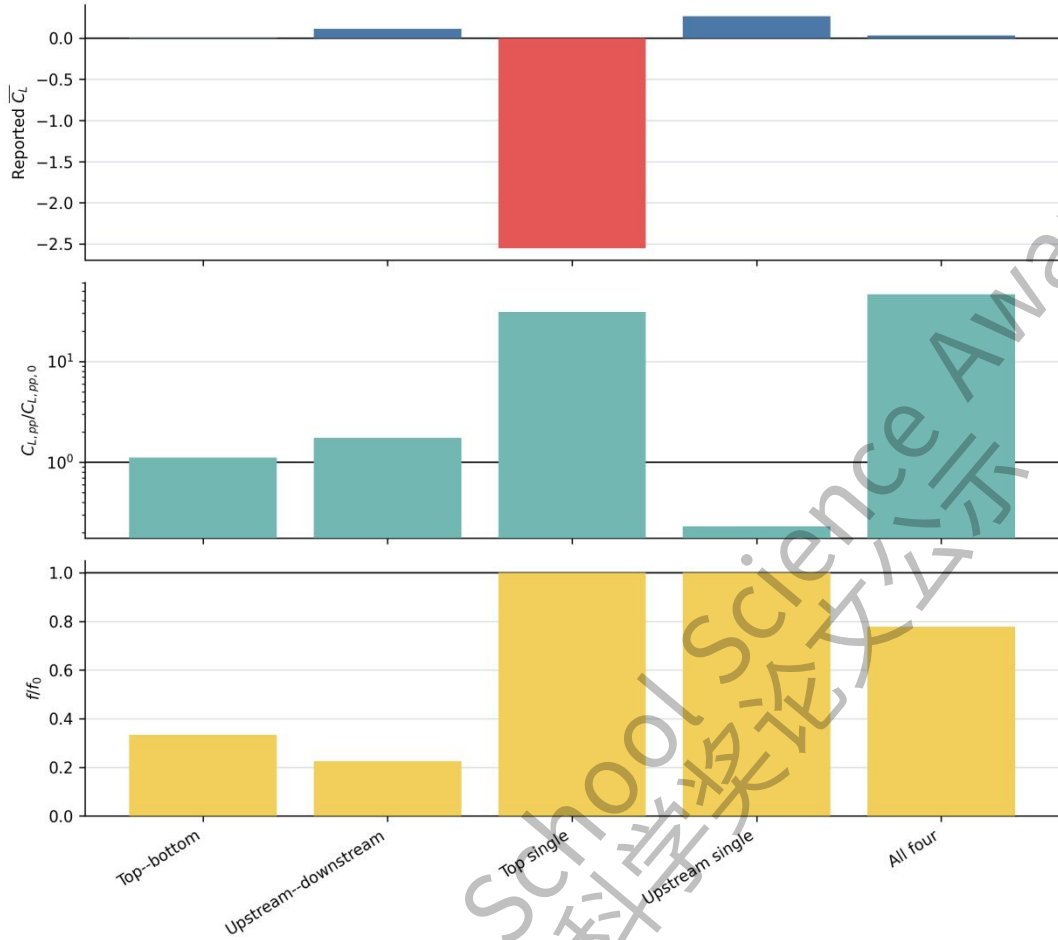


Figure 5: Effects of the fixed triangular-element arrangements on mean lift, peak-to-peak lift fluctuation and dominant frequency.

The fixed elements produce three distinct types of aerodynamic response. First, the upstream single element substantially attenuates the fluctuating component of lift. Its peak-to-peak lift ratio is $R_A = 0.231$, corresponding to a reduction of 76.9% relative to the bare cylinder. However, its mean lift increases from 0.0452 to 0.2664, or approximately 5.89 times the baseline magnitude. The reduced fluctuation is therefore accompanied by a pronounced transverse-force bias.

Second, the two symmetric-element arrangements mainly alter the dominant wake frequency. For the top-bottom pair, $|\overline{C_L}|$ decreases to 0.0062 and the dominant frequency is reduced by 66.7%. Its peak-to-peak lift, however, increases by 12.0%. The upstream-downstream pair produces an even lower frequency ratio, $R_f = 0.226$, but its mean-lift magnitude and peak-to-peak fluctuation increase to 2.47 and 1.75 times the corresponding bare-cylinder values. A reduction in dominant frequency is therefore not necessarily accompanied by attenuation of the unsteady transverse load.

Third, strong amplification occurs for the top single-element and four-element arrangements. The top element breaks the transverse symmetry of the geometry, producing $\overline{C_L} = -2.555$ and a peak-to-peak fluctuation approximately 30.9 times the bare-cylinder value. Although the four-element configuration is geometrically symmetric, it produces the largest fluctuation ratio, $R_A = 46.4$. Its dominant frequency is reduced by 22.2%, but the frequency reduction is accompanied by a substantial increase in lift variation.

The corresponding lift histories are shown in 6. The upstream single-element case exhibits the smallest oscillation range, whereas the top and four-element configurations exhibit much larger variations. The growth observed in the four-element signal also indicates a longer transient development than in the upstream single-element case.

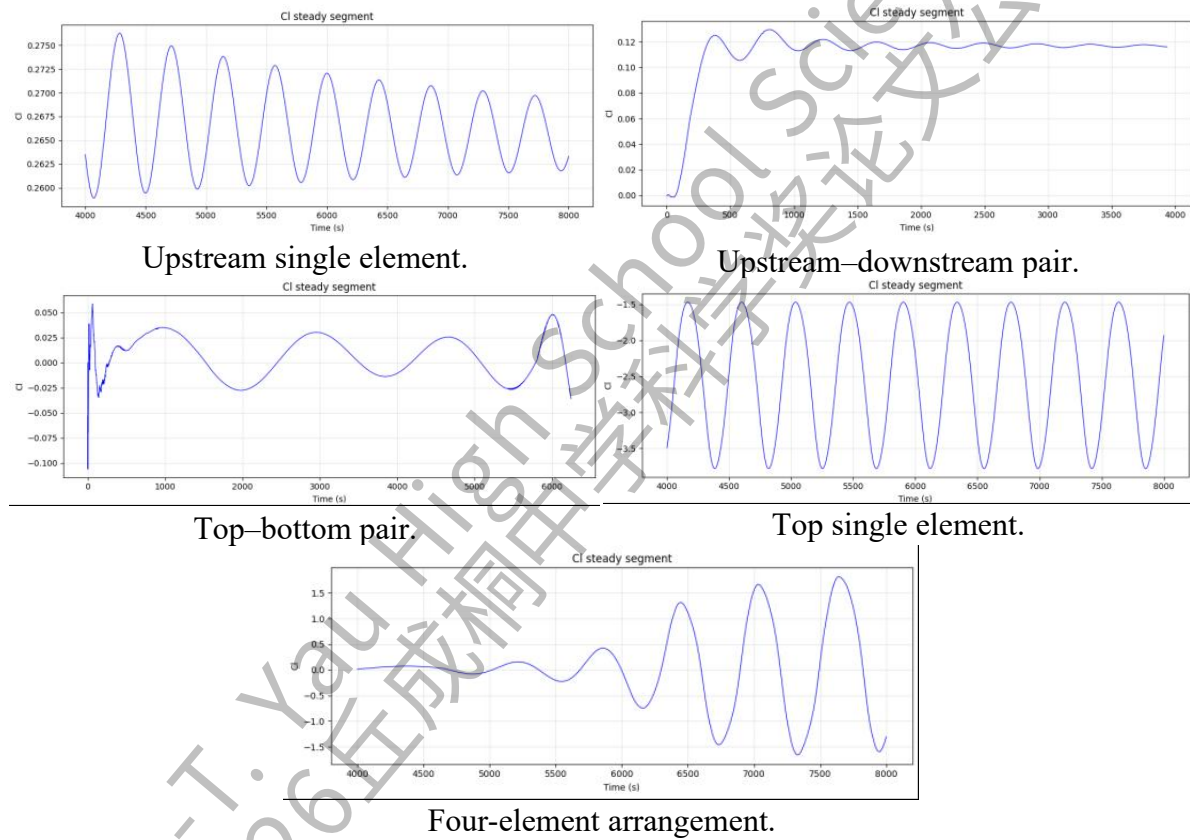


Figure 6: Lift-coefficient histories for the fixed triangular-element configurations.

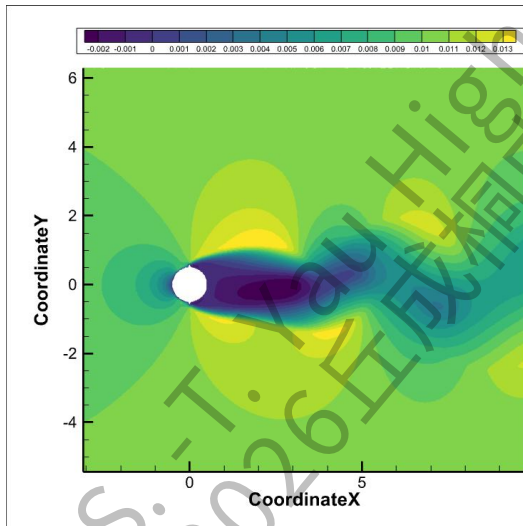
5.3. Modification of the wake structure

The instantaneous streamwise-velocity fields in 7 demonstrate that the triangular-element position substantially changes the velocity deficit and the large-scale organisation of the near wake. Because the contours show the streamwise velocity rather than vorticity, the discussion focuses on wake width, deflection and recovery.

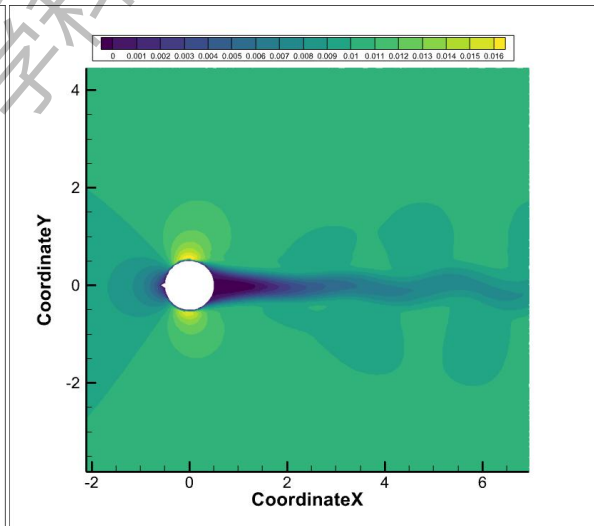
The top-bottom pair maintains an approximately centred near wake, consistent with the small mean lift obtained for this configuration. However, the velocity deficit remains spatially extensive, in agreement with the absence of a reduction in peak-to-peak lift. The main effect of this arrangement is therefore a modification of the characteristic wake time scale rather than complete suppression of the unsteady wake.

The upstream single element produces a comparatively narrow velocity deficit over the displayed downstream region. By disturbing the flow before it reaches the cylinder, this element modifies the development of both separating shear layers and weakens the alternating component of the transverse loading. The asymmetric time-averaged force, however, indicates that attenuation of the oscillatory component is accompanied by an unequal distribution of surface pressure.

For the top single-element case, the wake is strongly deflected from the centreline. This asymmetric flow organisation is consistent with the large negative mean lift. The four-element arrangement produces large alternating high- and low-speed regions downstream of the cylinder, consistent with its strongly amplified lift fluctuations. These results show that increasing the number of control elements does not necessarily stabilise the wake; interactions among multiple local shear layers can instead reinforce large-scale unsteadiness.



Top-bottom pair.



Upstream single element.

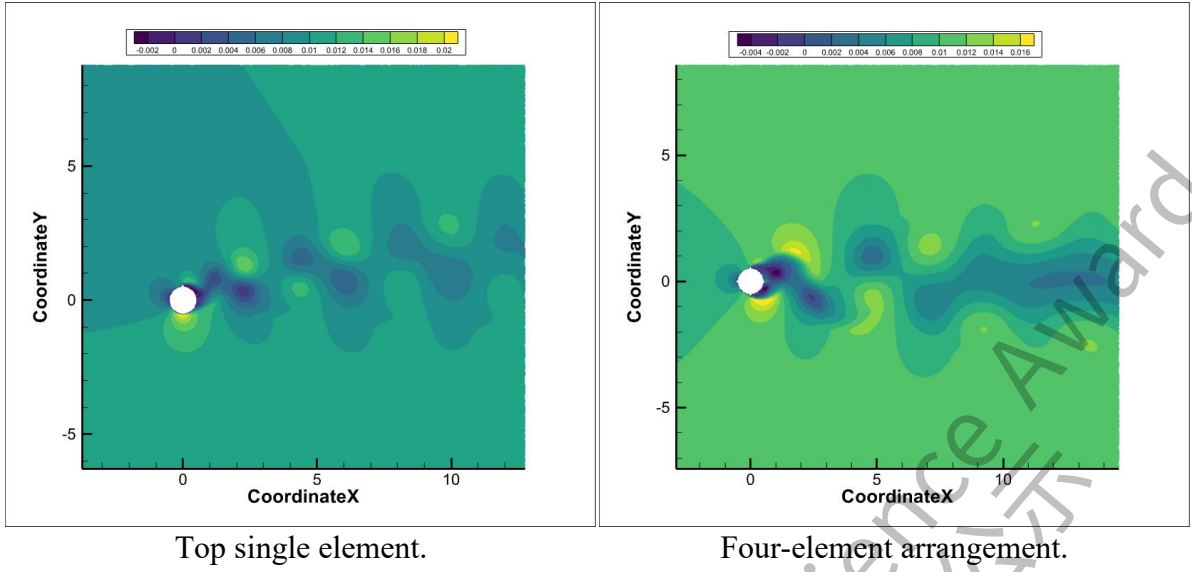
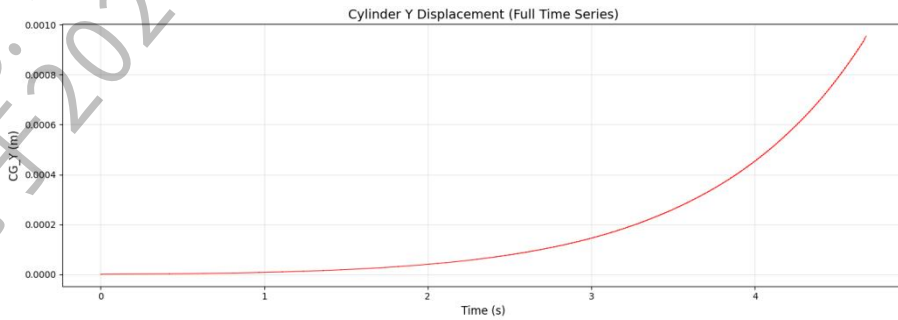


Figure 7: Instantaneous streamwise-velocity fields for representative fixed triangular-element configurations.

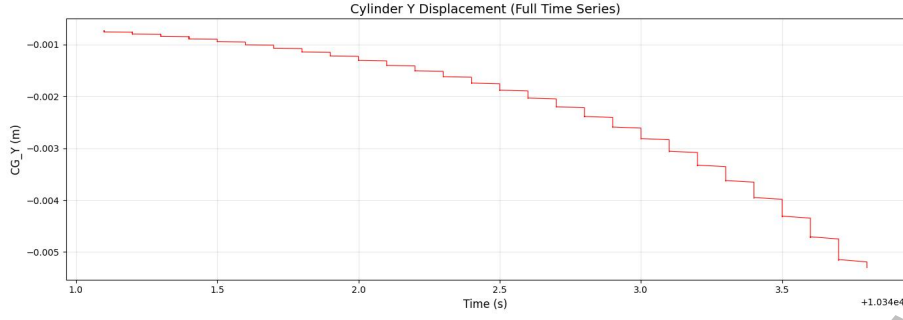
5.4. Transient response with freely rotating elements

The initial transverse-displacement histories of the R-2-TB and R-4 configurations are presented in 8. Both histories exhibit a monotonic response over the available time interval, but the displacement directions are opposite. The difference in sign indicates that the two arrangements generate different initial transverse-force biases, despite both being nominally symmetric about the wake centreline.

The R-2-TB record extends to approximately 6 s, whereas the R-4 record covers approximately 3.5 s. These durations correspond to less than 1.5% of the bare-cylinder shedding period. The histories consequently describe the initial rigid-body response rather than the long-time vibration state. Within these intervals, neither configuration develops a complete oscillation cycle from which a response amplitude or dominant structural frequency could be determined.



Top-bottom rotating pair (R-2-TB).



Four rotating elements (R-4).

Figure 8: Initial transverse-displacement histories for the freely rotating configurations.

6. Experimental results

6.1. Overall force characteristics and velocity dependence

The measured lift and drag forces for all configurations are shown in 9. The drag force increases from $U_\infty = 2 \text{ m} \cdot \text{s}^{-1}$ to $5 \text{ m} \cdot \text{s}^{-1}$ for every configuration. In contrast, the transverse force does not exhibit a uniform velocity dependence: its magnitude increases for some configurations, decreases for others and remains nearly unchanged for the remainder.

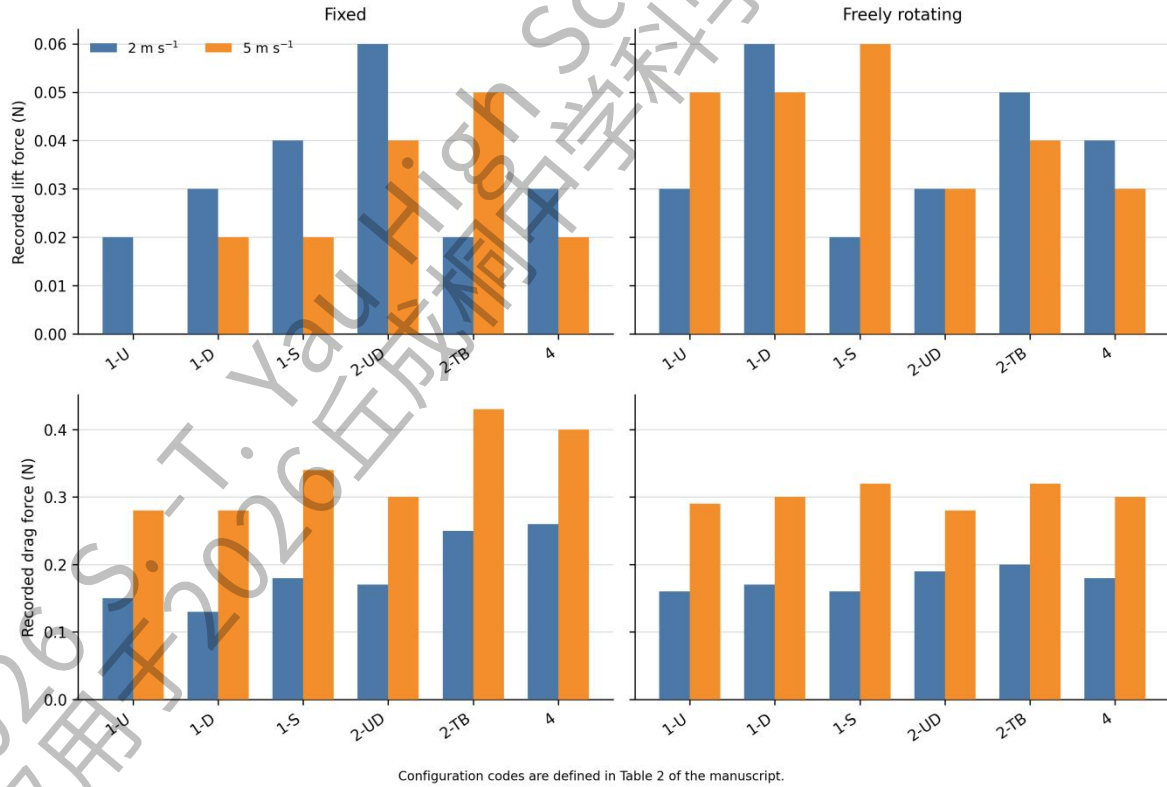


Figure 9: Measured lift and drag forces for the bare, fixed-element and freely rotating configurations at the two free-stream velocities.

The different degree of consistency between the two force components is also reflected by their rank correlations. Across the 13 configurations, the Spearman rank correlation between the two velocities is $\rho_s = 0.01$ for lift and $\rho_s = 0.75$ for drag. Thus, the relative ordering of the drag measurements is substantially more stable than that of the lift measurements. This distinction is consistent with the small absolute magnitude of the transverse-force readings and their greater sensitivity to slight geometric and flow asymmetries.

The measured drag increase between the two velocities is quantified in 10. If the drag coefficient remained constant, increasing the velocity from 2 to 5 m.s⁻¹ would increase drag by a factor of 6.25. The measured ratios $F_D(5)/F_D(2)$ instead range from 1.47 to 2.15, with a median value of 1.77. The apparent velocity exponent defined by [eq:drag-exponent] ranges from 0.42 to 0.84, with a median of 0.62. The measured cross-velocity scaling is therefore substantially weaker than the quadratic dependence associated with a constant drag coefficient.

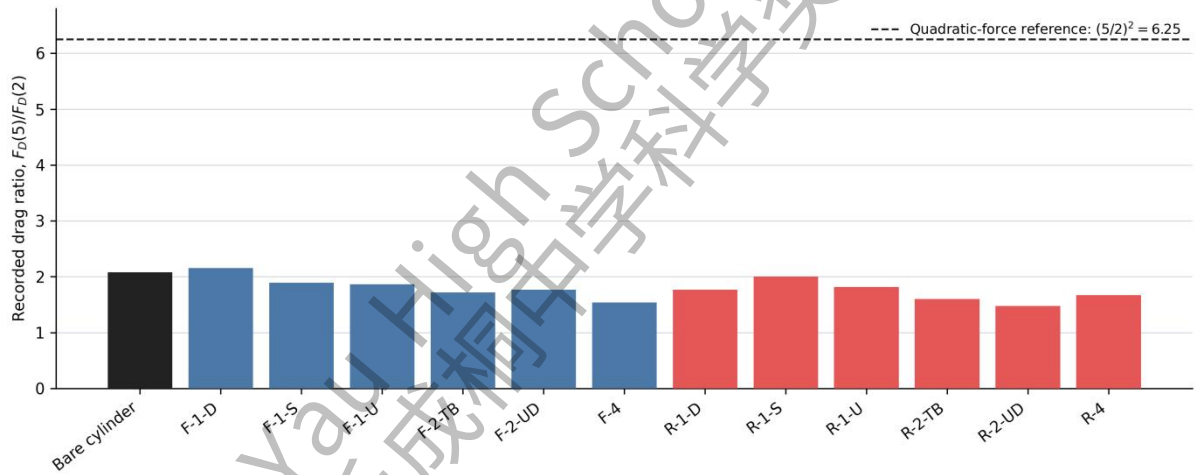


Figure 10: Increase in measured drag force between $U_\infty = 2 \text{ m.s}^{-1}$ and 5 m.s^{-1} . The dashed reference denotes the quadratic increase of 6.25 expected for a constant drag coefficient.

6.2. Combined lift–drag performance

The lift and drag ratios of the triangular-element configurations relative to the bare cylinder are presented in 11. Since each ratio is referenced to the bare-cylinder measurement at the same free-stream velocity, the comparison isolates the relative change caused by the control-element arrangement at each test velocity. Values below unity indicate a reduction in the corresponding measured force.

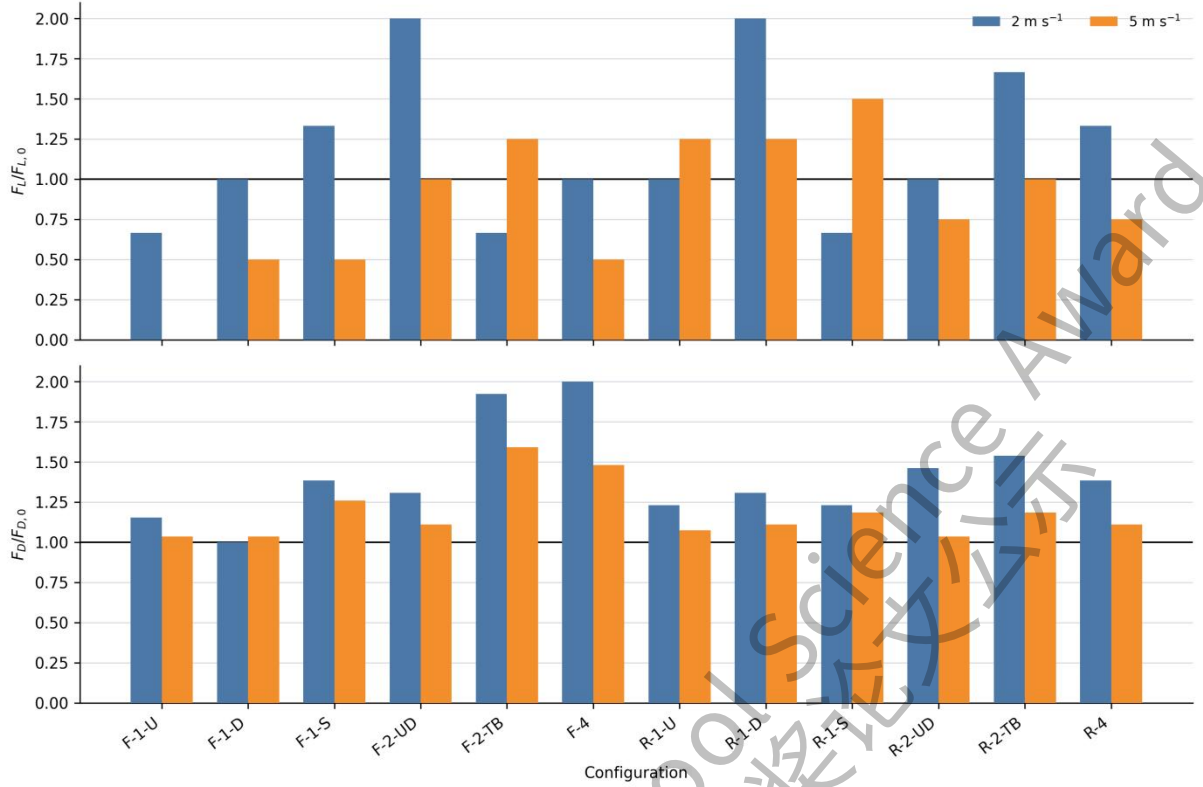


Figure 11: Lift- and drag-force ratios relative to the bare cylinder at the same free-stream velocity. The horizontal line at unity represents the bare-cylinder reference.

At $U_\infty = 2 \text{ m} \cdot \text{s}^{-1}$, the measured transverse force of the bare cylinder is 0.03 N. The F-1-U, F-2-TB and R-1-S configurations each produce a transverse force of 0.02 N, corresponding to

$$R_L = \frac{0.02}{0.03} = 0.67. \quad (22)$$

Although these three configurations give the same measured reduction in transverse force, their effects on drag differ considerably. Their drag ratios are 1.15, 1.92 and 1.23, respectively. The F-1-U configuration therefore provides the most favourable lift–drag combination among the three low-lift cases at this velocity. In contrast, F-2-UD and R-1-D produce transverse forces of 0.06 N, twice the bare-cylinder value.

At $U_\infty = 5 \text{ m} \cdot \text{s}^{-1}$, the measured transverse force of the bare cylinder is 0.04 N. The F-1-U value is recorded as 0.00 N at the available force resolution. This result represents the smallest recorded transverse-force magnitude, although it should not be interpreted as an exactly zero instantaneous or time-averaged lift. The F-1-D, F-1-S and F-4 configurations each produce 0.02 N, corresponding to $R_L = 0.50$. Among the freely rotating configurations,

R-2-UD and R-4 produce 0.03 N, giving $R_L = 0.75$, whereas R-1-S produces 0.06 N, or 1.5 times the bare-cylinder value.

The drag measurements exhibit a more uniform response. None of the triangular-element configurations produces a drag force lower than that of the bare cylinder at either velocity. At $U_\infty = 2 \text{ m} \cdot \text{s}^{-1}$, F-1-D produces the same measured drag as the bare cylinder, 0.13 N. The drag forces of F-1-U, R-1-U and R-1-S are 15.4%, 23.1% and 23.1% higher than the bare-cylinder value, respectively. The largest relative increases occur for F-2-TB and F-4, whose drag forces are 92.3% and 100% above the bare-cylinder value, respectively.

At $U_\infty = 5 \text{ m} \cdot \text{s}^{-1}$, the bare-cylinder drag force is 0.27 N. The F-1-U, F-1-D and R-2-UD configurations each produce a drag force of 0.28 N, corresponding to

$$R_D = \frac{0.28}{0.27} = 1.037, \quad (23)$$

or a 3.7% increase relative to the bare cylinder. The largest drag increase is obtained for F-2-TB, whose measured value of 0.43 N is 59.3% above the bare-cylinder value.

To evaluate the two force components jointly, 12 presents each configuration in the (R_D, R_L) plane. The bare cylinder is located at $(1,1)$, and movement towards the lower-left corner represents simultaneous reductions in drag and transverse force. A configuration located below but to the right of the bare-cylinder point reduces transverse force while increasing drag.

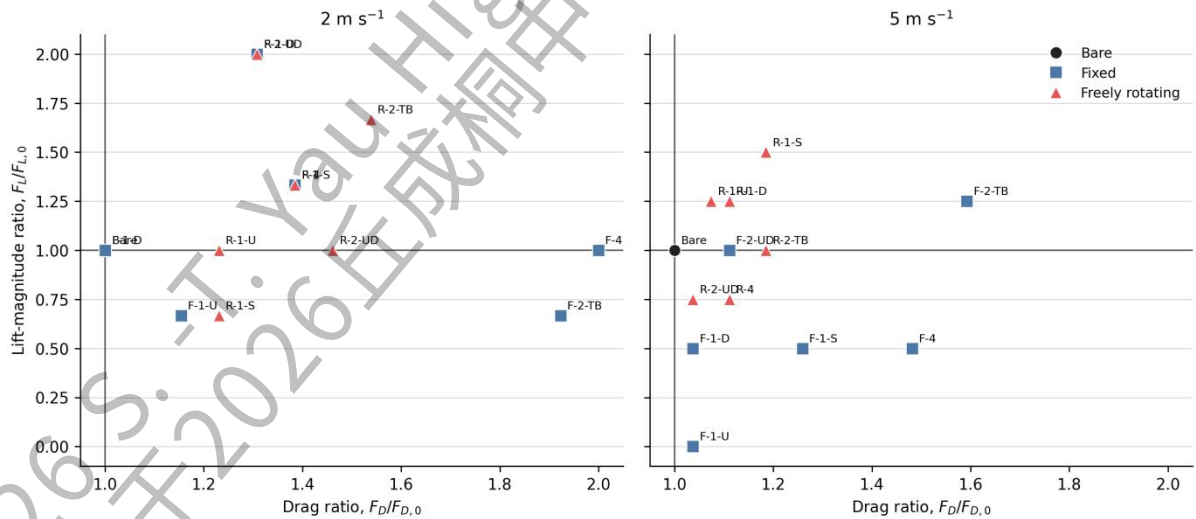


Figure 12: Combined lift–drag performance of the tested configurations. The bare cylinder is located at $(R_D, R_L) = (1,1)$. Movement towards the lower-left corner indicates simultaneous reductions in the measured drag and transverse-force magnitude.

At $U_\infty = 2 \text{ m.s}^{-1}$, none of the controlled configurations reduces both force components relative to the bare cylinder. Among the configurations with $R_L = 0.67$, F-1-U is closest to the bare-cylinder drag level, with $R_D = 1.15$. The low transverse force of F-2-TB is accompanied by a much larger drag ratio of 1.92, indicating that increasing the number of fixed elements does not necessarily improve the overall force response.

At $U_\infty = 5 \text{ m.s}^{-1}$, F-1-U gives the smallest recorded transverse force together with a drag ratio of 1.037. F-1-D also provides a favourable combination, with $R_L = 0.50$ and $R_D = 1.037$. Among the freely rotating configurations, R-2-UD gives the most favourable measured combination, with

$$(R_D, R_L) = (1.037, 0.75). \quad (24)$$

Thus, its transverse force is 25% below the bare-cylinder value, while its drag is 3.7% higher.

Several configurations show identical transverse-force readings but different drag forces. At 5 m.s^{-1} , F-1-D, F-1-S and F-4 all have $R_L = 0.50$, but their drag ratios are 1.037, 1.259 and 1.481, respectively. Consequently, F-1-S and F-4 do not provide an advantage over F-1-D in terms of either measured force component. Similarly, R-2-UD and R-4 both have $R_L = 0.75$, whereas their drag ratios are 1.037 and 1.111. R-2-UD therefore gives the better lift–drag combination of these two freely rotating arrangements.

The relative performance also changes with free-stream velocity. F-2-TB changes from $R_L = 0.67$ at 2 m.s^{-1} to $R_L = 1.25$ at 5 m.s^{-1} , while R-1-S changes from $R_L = 0.67$ to 1.50. In contrast, F-1-D and F-4 change from $R_L = 1.00$ at the lower velocity to $R_L = 0.50$ at the higher velocity. The configuration ranking is therefore velocity-dependent, and no single arrangement can be identified as universally optimal from the two tested conditions.

Overall, F-1-U and F-1-D provide the most favourable measured lift–drag combinations among the fixed configurations at 5 m.s^{-1} , while R-2-UD performs best among the freely rotating configurations.

6.3. Effect of passive rotation

The measured forces of the freely rotating elements are compared with those of the corresponding fixed arrangements in 13. A ratio below unity indicates that the freely rotating model produces a smaller force than the nominally matched fixed model.

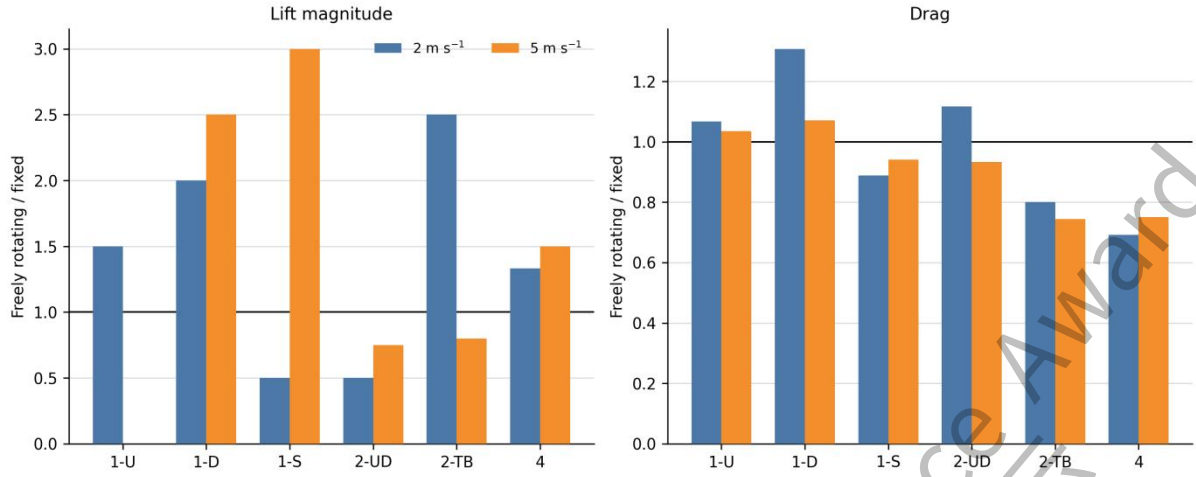


Figure 13: Ratios of freely rotating to fixed-element forces for nominally matched azimuthal arrangements. The R-1-U/F-1-U lift ratio at $5 \text{ m} \cdot \text{s}^{-1}$ is not evaluated because the corresponding fixed-element lift is recorded as 0.00 N .

Passive rotation does not produce a uniform reduction in transverse force. At $5 \text{ m} \cdot \text{s}^{-1}$, the lift forces of R-2-UD and R-2-TB are 25% and 20% below those of F-2-UD and F-2-TB, respectively. In contrast, the rotating downstream, side and four-element configurations produce larger transverse forces than their fixed counterparts.

A more systematic difference is observed in the drag of the multi-element arrangements. At $2 \text{ m} \cdot \text{s}^{-1}$, the drag forces of R-2-TB and R-4 are respectively 20.0% and 30.8% below those of the corresponding fixed models. At $5 \text{ m} \cdot \text{s}^{-1}$, the reductions are 25.6% and 25.0%. These results suggest that passive reorientation can reduce the effective flow blockage of densely distributed elements. However, the benefit is not observed consistently for the single-element configurations, indicating that the influence of rotation depends strongly on element number and azimuthal placement.

7. Discussion

7.1. Synthesis of the numerical and experimental trends

The numerical and experimental results consistently show that the aerodynamic response is controlled primarily by element placement and mounting condition rather than by element number alone. Increasing the number of triangular elements does not lead to monotonic improvement. In the numerical simulations, the four fixed elements produce the largest peak-to-peak lift fluctuation, whereas the upstream single element produces the strongest

fluctuation attenuation. Likewise, the experimental four-element configurations generally incur larger drag than the corresponding single-element arrangements.

The upstream fixed single element is the most promising fixed configuration identified by the combined numerical and experimental results. Numerically, it reduces $C_{L,pp}$ by 76.9%. Experimentally, it produces the smallest transverse-force reading at $5 \text{ m} \cdot \text{s}^{-1}$ while increasing drag by only 3.7%. However, the numerical mean lift increases to 5.89 times the bare-cylinder magnitude. This configuration therefore appears effective in reducing the alternating lift component but may introduce an undesirable static transverse-force bias.

Of the freely rotating configurations, R-2-UD provides the most favourable experimental lift–drag compromise at $5 \text{ m} \cdot \text{s}^{-1}$. Its transverse force is 25% below that of the bare cylinder, while its drag penalty is only 3.7%. The freely rotating top–bottom pair and four-element arrangement also produce less drag than their fixed counterparts, suggesting that passive reorientation may be particularly beneficial when several elements are distributed around the cylinder.

7.2. Implications for vibration control

The present results distinguish between modification of the fluid forcing and suppression of the structural response. A reduction in $C_{L,pp}$ can reduce the available fluctuating excitation, whereas a shift in shedding frequency may move the wake either towards or away from the structural natural frequency. A non-zero mean lift can also shift the static equilibrium position. These effects must therefore be assessed jointly rather than represented by a single lift or frequency metric.

The top–bottom fixed pair illustrates this distinction. It reduces the dominant frequency by 66.7% but increases the peak-to-peak lift by 12.0%. Whether this configuration reduces cylinder vibration depends on the structural natural frequency, damping and reduced velocity. Similarly, the upstream fixed element strongly attenuates lift fluctuation but produces a substantial mean-force bias.

The short freely rotating displacement records capture only the initial cylinder motion. Consequently, the present fluid–structure simulations identify the direction of the early response but do not determine a stationary vibration amplitude. Long-time coupled calculations are required before the reduction in fluid force can be translated into a quantitative VIV-suppression efficiency.

7.3. Limitations and future measurements

The numerical and experimental results should at present be regarded as complementary configuration screening rather than direct quantitative validation. The simulations and experiments differ in working fluid, velocity, Reynolds number and force definition. In addition, the experimental comparison is based on a single retained force value for each condition, while synchronous time histories of lift, drag and element angle are unavailable.

The next experimental campaign should therefore focus on the non-dominated configurations F-1-U and R-2-UD, together with the bare cylinder and at least one unfavourable configuration such as F-4. Repeated force measurements should be acquired simultaneously with cylinder displacement and triangular-element angle. Reporting the mean, rms and spectral content of the force and displacement signals would permit consistent comparison with CFD and distinguish stationary response from initial drift.

For the numerical simulations, longer sampling intervals, grid and time-step refinement at $Re = 10^4$, and complete specification of the structural and rotational parameters are required. Phase-resolved vorticity, pressure and element-moment fields would further clarify how passive rotation alters shear-layer development and transfers energy among the fluid, control elements and cylinder.

8. Conclusions and future work

The effects of fixed and freely rotating triangular control elements on the wake forces and transverse response of a circular cylinder were investigated using numerical simulations and wind-tunnel measurements. The results demonstrate that the aerodynamic effect of the triangular elements depends strongly on their azimuthal position, number and mounting condition. Increasing the number of elements does not produce a monotonic improvement, and reductions in lift fluctuation, mean transverse force and shedding frequency do not necessarily occur simultaneously.

Among the fixed configurations, the upstream single element produces the largest reduction in peak-to-peak lift fluctuation. Its $C_{L,pp}$ is 76.9% below that of the bare cylinder, indicating that an upstream disturbance can substantially weaken the alternating component of the transverse loading. This reduction is, however, accompanied by an increase in mean-lift magnitude to approximately 5.9 times the bare-cylinder value. The configuration therefore converts part of the oscillatory loading into a pronounced mean transverse-force bias rather than eliminating the transverse load.

The top–bottom fixed pair mainly modifies the characteristic time scale of the wake. Its dominant frequency is reduced by 66.7%, while its peak-to-peak lift increases by 12.0%. The upstream–downstream pair produces a still lower dominant frequency but increases both the mean and fluctuating lift. These results show that shifting the shedding frequency cannot by itself be interpreted as suppression of the structural response. Whether such a frequency shift is beneficial depends on its position relative to the natural frequency and damping characteristics of the supported cylinder.

The top single-element and four-element fixed configurations produce strongly amplified lift fluctuations, with peak-to-peak values 30.9 and 46.4 times that of the bare cylinder, respectively. The associated velocity fields show pronounced wake deflection for the top single-element case and large alternating velocity regions for the four-element arrangement. Thus, interactions between the additional shear layers can destabilise the wake rather than suppress it, even when the overall element arrangement is geometrically symmetric.

The wind-tunnel measurements further confirm that the force response is highly configuration- and velocity-dependent. At $U_{\infty} = 5 \text{ m.s}^{-1}$, F-1-U produces the smallest recorded transverse force, 0.00 N at the available measurement resolution, while its drag is only 3.7% above that of the bare cylinder. F-1-D also gives a favourable measured combination, with a 50% reduction in transverse force and the same 3.7% increase in drag. Among the freely rotating configurations, R-2-UD provides the best combined result at this velocity: its transverse force is 25% below the bare-cylinder value, while its drag is 3.7% higher.

None of the triangular-element configurations produces a drag force lower than that of the bare cylinder at both tested velocities. Nevertheless, passive rotation reduces the drag of the denser multi-element arrangements relative to their fixed counterparts. The drag of R-2-TB is 20.0% and 25.6% below that of F-2-TB at 2 m.s^{-1} and 5 m.s^{-1} , respectively, while R-4 produces corresponding reductions of 30.8% and 25.0% relative to F-4. This result suggests that passive reorientation may reduce the effective flow blockage of multi-element configurations. The same benefit is not observed consistently for the single-element arrangements, indicating that rotational freedom alone does not guarantee improved performance.

The present numerical and experimental results should be interpreted as complementary configuration screening rather than direct quantitative validation. The simulations and wind-

tunnel tests were conducted under different working-fluid, velocity and Reynolds-number conditions, and the numerical and experimental force quantities do not have identical statistical definitions. Moreover, the increase in measured drag between $2 \text{ m} \cdot \text{s}^{-1}$ and $5 \text{ m} \cdot \text{s}^{-1}$ ranges from only 1.47 to 2.15, whereas the dynamic pressure increases by a factor of 6.25. This difference indicates that velocity calibration, force-balance calibration, tare correction, model blockage and force averaging require further examination before the measurements are converted into aerodynamic coefficients or used for quantitative CFD validation.

The freely rotating simulations presently describe only the initial transverse motion. Their displacement records cover less than 1.5% of the bare-cylinder shedding period and do not contain a complete oscillation cycle. They therefore do not yet determine stationary response amplitude, lock-in behaviour or vibration-suppression efficiency. A complete fluid–structure assessment requires longer simulations and explicit specification of the cylinder mass ratio, natural frequency and damping ratio, together with the moment of inertia, pivot friction and angular constraints of the rotating elements.

Future numerical work should be extended over sufficient shedding and structural-response cycles to obtain converged mean, rms and spectral quantities. Synchronously sampled lift, cylinder displacement, element angle and hydrodynamic moment would allow the phase relationship and cycle-averaged energy transfer to be quantified. Phase-resolved vorticity and surface-pressure fields would further clarify how element rotation modifies shear-layer development and wake symmetry.

Future experiments should concentrate initially on the bare cylinder, F-1-U and R-2-UD, together with an unfavourable reference configuration such as F-4. These cases represent, respectively, the baseline, the strongest numerically identified attenuation of lift fluctuation, the most favourable freely rotating experimental combination and a configuration associated with strong wake amplification. Repeated measurements of lift and drag should be acquired simultaneously with cylinder displacement and element angle over a range of reduced velocities. Reporting uncertainty, mean and rms response, dominant frequency and amplitude response curves would make it possible to determine whether the observed force modifications translate into robust suppression of vortex-induced vibration.

Overall, the present study establishes triangular control elements as a strongly configuration-dependent method for modifying the forces acting on a circular cylinder. The upstream fixed element and freely rotating upstream–downstream pair emerge as the leading candidates for further investigation. The results provide a rational basis for the next numerical and experimental campaign, while a definitive VIV suppression claim will require stationary

structural-response data, resolved rotational dynamics and matched nondimensional test conditions.

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Acknowledgements

When I first encountered the phenomenon of vortex-induced vibration—the flagpole cable at school shaking violently from time to time—I never expected that one day I would conduct a complete research project on such a common phenomenon in daily life. From the flagpole cables swaying in the wind on campus to the free-spanning pipelines in the ocean, vortex-induced vibration may appear ordinary, yet behind it lies complex fluid-structure interaction mechanisms. How to effectively suppress this harmful vibration became the starting point of my project.

I would like to thank my school supervisor. In the early stages of the project, the teacher helped me sort out my research ideas, reviewed my research plan. Whenever I encountered doubts, the teacher offered patient guidance and pointed out potential solutions, and provided valuable suggestions regarding academic writing standards. The teacher encouraged me to think independently and experiment boldly. Such trust and support gave me greater confidence when confronting challenges.

I would like to thank my father. He encouraged me when I encountered setbacks, helped me open up new ideas when I was stuck, and lent me his work computer when my own computing power was insufficient. Without his and my family's understanding and support, I could not have devoted myself to this research so efficiently.

No AI tools were employed during this research. I primarily learned the simulation software and relevant knowledge by referring to literature. Therefore, I would also like to thank all authors of the cited references. Their research findings provide an important theoretical foundation and inspiration for this project. At the same time, I would like to thank the creators on platforms such as GitHub and Bilibili who selflessly shared operation guides and taught me how to use these tools.

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