

On the load-shifting phenomenon of accelerating reconfigurable circular plates

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In this study, we investigate the dynamic behaviour of reconfigurable circular plates under acceleration as a model problem to understand the interplay between kinematics and shape deformation in biological propulsion. A high-resolution force transducer and time-resolved particle image velocimetry were employed to simultaneously capture both hydrodynamic forces and vortex dynamics. The results reveal that, unlike rigid plates that exhibit Reynolds number independence, the force evolution of reconfigurable plates is governed by the dimensionless bending stiffness El^* . A distinct load-shifting phenomenon is observed – characterized by a reduction in peak force amplitude and an elevation of the postpeak force trough, contrasting with the ‘peak-valley’ behaviour typical of rigid plates. Based on El^* , reconfigurable plates are classified into three regimes: extra-flexible ($El^* < 2.28 \times 10^{-3}$), flexible ($2.28 \times 10^{-3} \leq El^* \leq 0.143$) and rigid ($El^* > 0.143$). Notably, only plates within the flexible regime exhibit the load-shifting phenomenon. Flow visualizations show that the flexible plates, due to their shape reconfiguration, produce flow fields with two distinct features: initially, the formation of three-dimensional, non-axisymmetric vortex rings; subsequently, vortex breakdown occurs due to instability. By applying the vorticity moment theorem, force generation is accurately estimated from the flow field. Using a vortex-based low-order force model, the radial distribution of vorticity is identified as the key mechanism underlying the load-shifting phenomenon. This finding suggests that biological morphing structures in real propulsion scenarios can reduce force fluctuations without compromising average thrust by ‘load-shifting’, offering insights into efficient propulsion strategies.

Key words: propulsion, vortex dynamics, flow-structure interactions

1. Introduction

The ubiquity of unsteady phenomena in nature is well-established, particularly in the context of the accelerating motion of organisms within fluid environments. These organisms, such as flyers and swimmers, exhibit behaviours characterized by the acceleration and morphing of wings, fins and bodies (Mountcastle & Combes 2013; Lucas *et al.* 2014; Chin & Lentink 2016; Triantafyllou, Weymouth & Miao 2016) and, particularly, morphing provides additional benefits. For example, insects enhance their flapping efficiency through the passive deformation of their wings (Ho *et al.* 2003; Kang *et al.* 2011; Kang & Shyy 2013), birds coordinate flight efficiency (Johansson, Wolf & Hedenström 2010; Taylor *et al.* 2012) and disturbance resistance (Reynolds, Thomas & Taylor 2014; Harvey *et al.* 2019; Cheney *et al.* 2020) through active wing morphing, and marine swimmers utilize fin flexibility to enhance propulsion efficiency or to gain high acceleration during manoeuvres (Triantafyllou *et al.* 2016; Hang *et al.* 2022; Jimenez *et al.* 2023). Direct simulations or experiments provide detailed insights into these phenomena; however, due to the inherent complexity of modelling, the generality of the derived fluid dynamic laws remains limited (Ho *et al.* 2003; Kang *et al.* 2011; Ramanarivo, Godoy-Diana & Thiria 2011; Kang & Shyy 2013). Therefore, researchers tend to abstract moving appendages as plates in model problems to reveal universal fluid dynamic laws (Kim & Gharib 2011*b*; Fernando & Rival 2016*b*; Reijtenbagh *et al.* 2023*a*; Li *et al.* 2024).

Considering that any unsteady motion (neglecting rotation) can be approximated by superposing small intervals of uniformly accelerated motions (Kaiser, Kriegseis & Rival 2020; Li *et al.* 2022) and reconfigurable plates abstract the geometry of wings and fins (Kim & Gharib 2011*a,b*), accelerating reconfigurable circular plates, which incorporate both acceleration and shape deformation, are adopted as a fundamental model to investigate the unsteady fluid-dynamic forces and flow field for biological propulsion and disturbance resistance.

The influence of acceleration on the force evolution of rigid circular plates is first discussed, which has been extensively investigated (Fernando & Rival 2016*b*; Reijtenbagh *et al.* 2023*a*; Li *et al.* 2024). The force evolution exhibits a distinct ‘peak-and-valley’ pattern: during the acceleration process, the force increases with velocity, peaks at the end of acceleration and then gradually decreases to a valley as acceleration ceases, before slowly rising until stabilizing. The instantaneous force decreasing to a minimum before increasing again to reach a steady value is known as the ‘drag trough’ phenomenon (Fernando & Rival 2016*a*). Two dynamic parameters govern the force scaling for such plates: the Reynolds number, $Re = \rho_f U_c D / \mu$, and the acceleration number, $a^* = a D / U_c^2$, where D , a and U_c are the characteristic length, acceleration and velocity of the translating plate, respectively, and ρ_f and μ are the density and dynamic viscosity of the fluid. For rigid circular plates, the force evolution exhibits Reynolds number independence within the range of $50\,000 \leq Re \leq 350\,000$ when the dimensionless travel distance is less than 6 (Fernando & Rival 2016*b*). Additionally, an a^* dependence exists, indicating that for the same a^* conditions, the instantaneous force evolution follows a consistent pattern regardless of variations in actual characteristic acceleration a and velocity U_c (Fernando & Rival 2016*b*; Li *et al.* 2022).

Two perspectives are commonly used to explain the unsteady mechanisms of force generation during the acceleration of a rigid flat plate from rest: the quasisteady view (Fernando & Rival 2016*a,b*; Reijtenbagh *et al.* 2023*a*; Liu & Sun 2024; Li *et al.* 2024) and the impulse-based view (Kim & Gharib 2011*a*; Li *et al.* 2022). The quasisteady view, which is primarily focused on predictive modelling, decomposes the instantaneous force into a steady-state force, added-mass force and history force (Stokes 1851; Basset

1888; Oseen 1927; Odar & Hamilton 1964). While powerful, accurately modelling these terms in viscous, separated flows is challenging, as simple potential-flow models often fail (Grift *et al.* 2019; Fernando, Weymouth & Rival 2020; Li *et al.* 2022). This has led to an ongoing debate about the best approach: some researchers propose a variable added-mass coefficient to account for real fluid effects (Odar & Hamilton 1964; McPhaden & Rival 2018; Galler, Weymouth & Rival 2021), while others introduce a complex history force term to capture the history of the wake, with recent work proposing new scaling laws based on this concept (Reijtenbagh *et al.* 2023a; Li *et al.* 2024). In contrast, the impulse-based perspective offers a more direct link to the physical mechanisms by focusing on the evolution of the vorticity field, particularly through the vorticity moment theorem (Wu 1981; Graham *et al.* 2017; Kaiser *et al.* 2020). This approach posits that the force is governed by the growth of vorticity, which can be conceptually separated into added-mass vorticity and the shed vorticity that forms the wake (Corkery 2019). The dynamics of this shed vorticity, typically modelled as a growing vortex ring, are considered central to explaining the force history. For instance, the phenomenon of Reynolds number independence has been linked to the timing of vortex ring pinch-off (Fernando & Rival 2016b), and recent models have successfully predicted instantaneous forces by tracking the details of vortex ring growth (Li *et al.* 2022; Reijtenbagh *et al.* 2023a). In essence, while the quasisteady framework provides a structure for force prediction, the impulse-based view excels at providing a mechanistic explanation rooted in the observable dynamics of the wake.

Another key characteristic of accelerating reconfigurable circular plates is reconfiguration. As a standard model for morphing problems, the study of reconfigurable flat plates has previously focused primarily on steady-state conditions, where the free stream velocity is kept constant. A well-known initial study in this area is Vogel's work (Vogel 1989), in which he measured the drag of leaves in a steady wind field as a function of wind speed and found that, contrary to the commonly assumed quadratic force-speed relationship, deformable leaves exhibit a power law with an exponent of 4/3 in the relationship between force and wind speed. This phenomenon inspired a series of subsequent studies, particularly scaling analyses of the exponent (De Langre 2008). Notable works include Alben *et al.* (2002, 2004), who used soap film experiments in two-dimensional (2-D) flow fields to measure one-dimensional (1-D) fibres, modelled based on beam theory, and defined a dimensionless length that physically represents the ratio of fluid dynamics to elastic force. Schouveiler & Boudaoud (2006) investigated the plate reconfiguration based on cone shapes, Gosselin *et al.* (2010) analysed rectangular plates and, more recently, Baskaran, Hutin & Mulleners (2023) studied bioinspired flexible disks, each discovering similar power law phenomena. However, in all these configurations, force decreases compared with rigid plates due to deformation-induced area reduction and streamlining (Gosselin *et al.* 2010; Baskaran *et al.* 2023).

Thus, an important question arises: Is the force exerted by an accelerating reconfigurable circular plate smaller than that of a rigid plate? If so, how can previous studies justify the benefits of flexibility in enhancing propulsion and manoeuvrability? If not, what is the underlying flow mechanism responsible for the force enhancement? Limited studies in this area include Kim & Gharib (2011b), who observed that a flexible plate generates lower forces after acceleration compared with a rigid plate but larger forces after the initial peak, attributing this behaviour to the slow development of the vortex structure. And, Joshi & Bhattacharya (2022) investigated the unsteady force response of an accelerating flat plate with controlled spanwise bending and found that unsteady drag increased when the plate bent towards the incoming flow, as the growth of circulation was faster and the tip vortices remained closer to the plate during the bend-down motion. Nevertheless, while the hydrodynamics of rigid accelerating bodies (Lorite-Díez *et al.* 2018; Reijtenbagh

et al. 2023a; Liu & Sun 2024) and flexible bodies in steady flow (Vogel 1989; Alben, Shelley & Zhang 2002; Schouveiler & Eloy 2013; Baskaran *et al.* 2023) are increasingly understood, a systematic understanding of the governing dimensionless parameters and the key flow mechanisms for passive reconfiguration, accelerating bodies remains elusive. This motivates a deeper investigation, particularly one that leverages an impulse-based view to connect the force evolution directly to the underlying vortex dynamics.

The purpose of this study is thus to investigate the temporal evolution of force to analyse dimensionless number dependence and explore the underlying mechanism using a vortex-based low-order force model within the Reynolds number range $3.12 \times 10^4 \leq Re \leq 10.8 \times 10^4$ and bending stiffness range $4.19 \times 10^{-4} \leq EI \leq 26.8 \text{ N m}$ (defined in § 2.2). A high-resolution force transducer and time-resolved particle image velocimetry (PIV) are the primary research methods. The article is organized as follows: § 2 introduces the experimental methods; § 3 presents the force evolution laws; § 4 discusses the flow patterns; § 5 examines the mechanisms behind the force evolution. Concluding remarks are provided in § 6.

2. Methods

2.1. Experimental set-up

The experiments were conducted in a towing tank with a cross-sectional area of $1 \text{ m} \times 1.3 \text{ m}$ and a towing length of 2.8 m at Shanghai Jiao Tong University, as shown in figure 1 and also referenced in Li *et al.* (2022, 2024). The tank was filled with water ($\rho_f = 1.0 \times 10^3 \text{ kg m}^{-3}$) to a depth of 1.2 m . The plate with $D = 0.10 \text{ m}$ was towed along the x -axis by mounting it perpendicular to a horizontal cylindrical rod of the L-shaped sting, positioned $5D$ away from the free surface to minimize its influence (Grift *et al.* 2019; Joshi & Bhattacharya 2022). An aluminium bar is attached to the plate through its centre to control the bending of the reconfigurable plates, resulting in symmetric deformation about the midplane. This set-up preserves left–right symmetry, mimicking the bilateral coordination seen in bird or insect wings and fish fins. The vertical strut of the sting has a streamlined, $0.3D \times 0.05D$ rectangular cross-section with $0.02D$ rounded corners to minimize its wake influence. The horizontal rod, with a nominal length of $4D$, has a square cross-section with a width of $0.1D$ and features $0.02D$ rounded edges to further reduce flow disturbance. The horizontal rod length was carefully selected to avoid both short and excessive lengths: a short rod length would cause the wake from the vertical part of the L-shaped sting to interfere with the vortex ring formation, while a long rod, acting as a cantilever beam, would result in strong mechanical vibrations. Further details on the influence of the L-shaped sting on the flow and force measurements are provided in Appendix A1. The motion of the plate is driven by a closed-loop stepper motor system, which consists of a stepper motor (Times Brilliant 110HCY220AL3S-TK0) and a servo driver (Just Motion Control 3HSS2208H-110). The blockage ratios for the plate (0.8 %) and the L-shaped sting (0.01 %) were sufficiently small to minimize wall effects (Gosselin *et al.* 2010; Fernando & Rival 2016a).

2.2. Experimental parameters

The primary features of the accelerating reconfigurable circular plates lie in their acceleration, reconfiguration and circular geometry. Correspondingly, the key parameters include the plate's kinematics, flexibility and shape. To simplify the problem while maintaining the focus of this study, the plate shape was fixed as a circular plate, while the

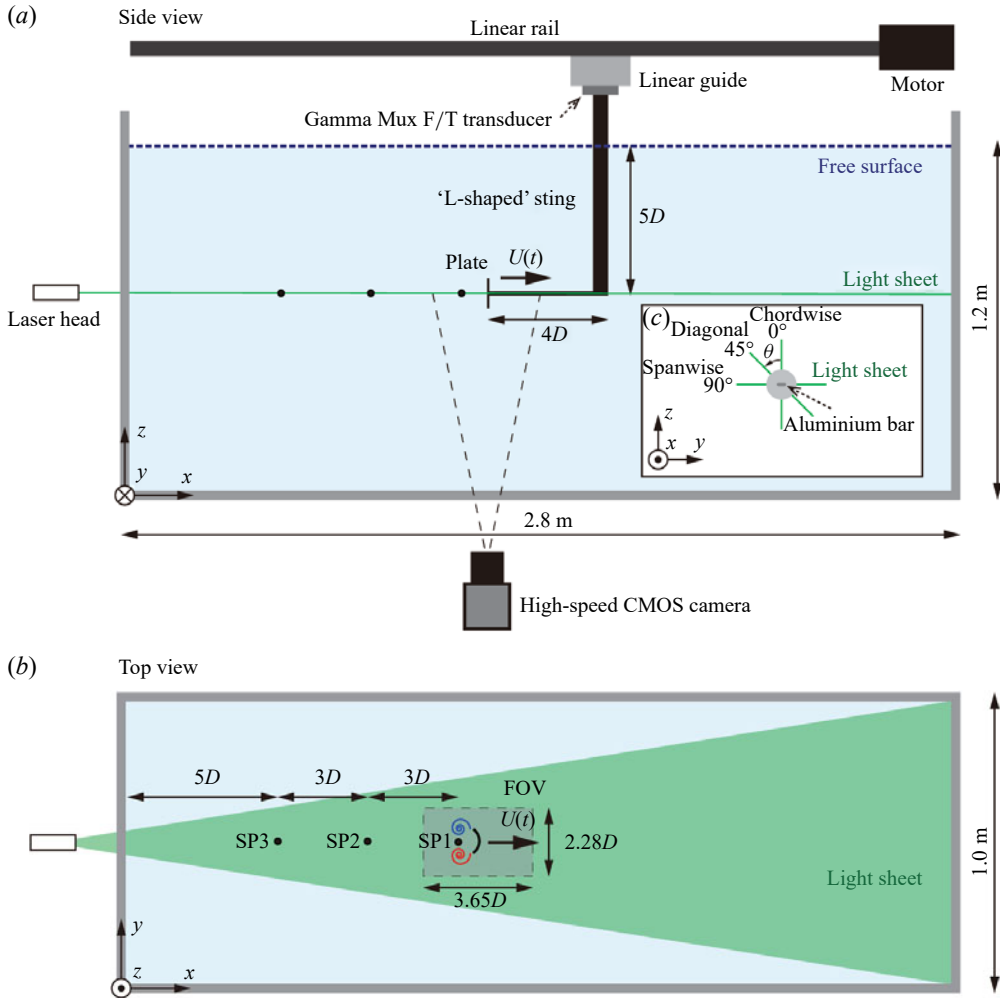


Figure 1. Schematic representation of the experimental configuration. (a) Side view of the set-up, with the L-shaped sting holding the plate, moving from left to right at the set velocity $U(t)$. The complementary metal-oxide-semiconductor (CMOS) cameras are positioned at the bottom, and the horizontal laser light sheet is illuminated from the left-hand side. (b) Top view illustrating the field of view (FOV) of $3.65D \times 2.28D$ for PIV, see later in § 2.4. To measure the flow field structure corresponding to different formation times, the starting point (SP) of the plate's motion is varied, with a spacing of $3D$ between SPs, and SP3 located $5D$ from the wall. (c) A zoomed-in view of PIV. An aluminium bar is attached to the plate passing through its centre, controlling the bending. The bending direction is defined as the chordwise direction ($\theta = 0^\circ$), the perpendicular direction as the spanwise direction ($\theta = 90^\circ$) and the intermediate direction as the diagonal direction ($\theta = 45^\circ$).

kinematic and flexibility parameters were systematically varied. This section introduces the configurations of kinematic settings and flexibility parameters.

Each plate was rectilinearly accelerated from rest at a constant acceleration a to a final towing velocity U_c . The corresponding dimensionless kinematic parameters are the acceleration number, a^* , and the Reynolds number, Re . The kinematic set-ups in previous studies are summarized in table 1. For this study, considering the dynamic viscosity of water, $\mu = 1.002 \times 10^{-3} \text{ Pa s}$ and the characteristic velocity U_c as the maximum translating velocity of the plate, we kept $a^* = 0.25$ constant and varied Re from 3.12×10^4

Author	Shape	Stiffness	h (mm)	a^*	$Re \times 10^4$
Fernando & Rival (2016a)	Elliptical, rectangular	Rigid	3	1.0	4
Fernando & Rival (2016b)	Elliptical	Rigid	–	1.0	5–35
Corkery (2019)	2-D wing	Rigid	–	0.167–0.5	2
Grift <i>et al.</i> (2019)	Rectangular	Rigid	5	0.911–3.64	4–8
Fernando <i>et al.</i> (2020)	Circular	Rigid	3	0.333–4	12.5
Li <i>et al.</i> (2022)	Circular	Rigid	5	0.5–2.0	4–8
Reijtenbagh <i>et al.</i> (2023a)	Rectangular	Rigid	–	0.121–1.62	3–7.5
Kim & Gharib (2011b)	Rectangular	Flexible	0.25~1.52	3.2	0.2
Current study	Circular	Flexible	0.125~5.00	0.25	3.12–10.8

Table 1. Summary of previous investigations into the accelerating plates. Note that the shorter side of the rectangle was used as the characteristic length in Kim & Gharib (2011b), whereas the longer side was used in Reynolds *et al.* (2014) and Grift *et al.* (2019).

to 10.8×10^4 . To facilitate the analysis, we introduced the dimensionless travel distance, also known as the formation time (Gharib, Rambod & Shariff 1998; Yang, Jia & Yin 2012), defined as $t^* = (1/D) \int_0^t U(t) dt$, where $U(t)$ is the instantaneous velocity. Formation time has been verified as a suitable parameter for describing the evolution of forces and flow fields during the acceleration phase (Kim & Gharib 2011b; Fernando & Rival 2016b; Li *et al.* 2022; Reijtenbagh *et al.* 2023a). To provide a temporal reference for the acceleration phase, we define the dimensionless acceleration time, t_a^* , as the formation number at the moment acceleration ceases (t_a). For the constant acceleration profile used in this study, t_a^* simplifies to a direct function of the acceleration number, $a^* = aD/U_c^2$:

$$t_a^* = \frac{at_a^2}{2D} = \frac{U_c^2}{2aD} = \frac{1}{2a^*}. \quad (2.1)$$

This value is reported in the captions of the relevant figures to denote the duration of the acceleration phase.

The actual kinematics were verified using two independent methods. First, a nine-axis offline accelerometer data logger (WT901SDCL) mounted on the linear guide was employed, and velocity and displacement profiles were obtained by integrating the recorded acceleration data. Second, a high-speed CMOS camera captured the positions of the plate and the L-shaped sting connection point (i.e. the centre of the circular plate) at various time instants, and differential analysis of these positions was conducted to derive the velocity. Each method was repeated 10 times to ensure reliability. A comparative analysis of the conformity between theoretical and actual velocity profiles under two distinct conditions, $a^* = 0.25$, $Re = 4 \times 10^4$ and $a^* = 2.0$, $Re = 8 \times 10^4$, is shown in figure 2. For measurements based on image data, displacement serves as the raw data, while velocity is derived through differentiation. However, acceleration is not further calculated by differentiation due to the typically unacceptable errors introduced by double differentiation, where Rick Chartrand's algorithm for numerical differentiation of noisy data will be considered in future research (Chartrand 2011). The uncertainty associated with the instantaneous velocity is 1.2 % for the accelerometer and 4.6 % for image acquisition under $a^* = 0.25$, $Re = 4 \times 10^4$, and 0.7 % and 5.3 %, respectively, under $a^* = 1.0$, $Re = 8 \times 10^4$ during the constant velocity stage. The larger error in the image-based measurements is attributed to more pronounced oscillations of the plate mounted at the bottom. In contrast, the accelerometer, mounted on the linear guider (as shown in figure 1), is unaffected by oscillations of the cantilever beam caused by the L-shaped sting.

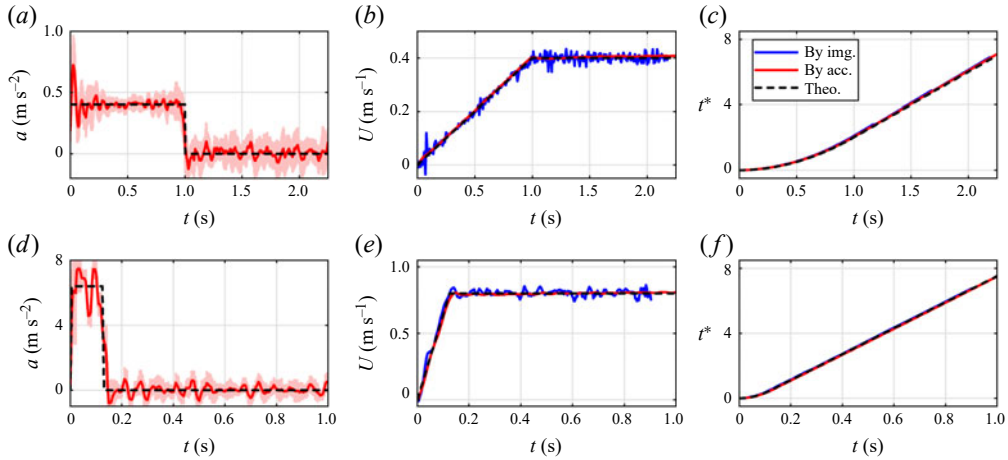


Figure 2. The kinematics validation under the conditions of $a^* = 0.25$, $Re = 4 \times 10^4$ in (a–c) and $a^* = 1.0$, $Re = 8 \times 10^4$ in (d–f) is presented. The acceleration calibration is shown in (a) and (d), the velocity calibration in (b) and (e), and the displacement calibration in (c) and (f). The black dashed lines represent the theoretical values, the red solid lines are based on accelerometer data and the blue solid lines are based on image data.

Polycarbonate sheets are utilized to fabricate plates with varying degrees of flexibility. These sheets have a density of $\rho_m = 1.194 \times 10^3 \text{ kg m}^{-3}$, a Young's modulus $E = 2220 \text{ MPa}$, and a Poisson's ratio $\hat{\nu} = 0.370$. The plates differ in thickness h , and assuming 1-D deformation of a beam-type plate, the bending stiffness is calculated using the formula (Landau *et al.* 2012)

$$EI = \frac{Eh^3}{12(1 - \hat{\nu}^2)}. \quad (2.2)$$

The flexibility parameters are summarized in table 2. An uncertainty analysis for the bending stiffness, EI , was conducted. The uncertainty arises primarily from Young's Modulus (E) and plate thickness (h) (2.2). A conservative relative uncertainty of $\sigma_E/E = 5.0\%$ was adopted for E to account for batch-to-batch variations in the commercial-grade polycarbonate (Callister & Rethwisch 1999). To quantify thickness variability, each plate was measured at 16 points in a radial and azimuthal pattern using a thickness gauge (SANLIANG 325-308, 0.001 mm resolution). The results (table 2) show excellent material uniformity, with thickness variability (σ_h/h_{avg}) below 2%. Subsequent calculations used the measured mean thickness, h_{avg} , for improved accuracy. The total relative uncertainty in EI was estimated via error propagation (Taylor 2022):

$$\left(\frac{\sigma_{EI}}{EI}\right)^2 \approx \left(\frac{\sigma_E}{E}\right)^2 + \left(3\frac{\sigma_h}{h_{avg}}\right)^2. \quad (2.3)$$

This yields a maximum uncertainty of approximately 7.1% for the thinnest plate, which has the largest thickness variability (1.7%). The uncertainty from Young's modulus (5.0%) is a primary contributor, being comparable to or greater than the contribution from thickness ($0.23\% \leq 3\sigma_h/h_{avg} \leq 5.1\%$). Henceforth, plates are referred to by their nominal thickness, h_{nom} (e.g. 'the 0.30 mm plate'), for brevity. All calculations, however, use the measured mean thickness, h_{avg} , from table 2.

The flexible plate undergoes deformation due to its inability to resist the hydrodynamic forces during translation. To track this deformation, the chordwise plane crossing the

#	h_{nom} (mm)	h_{avg} (mm)	σ_h/h_{avg}	EI (Nm)	σ_{EI}/EI
1	0.125	0.1275	1.7 %	4.44×10^{-4}	7.1 %
2	0.175	0.1748	1.2 %	1.14×10^{-3}	6.2 %
3	0.30	0.2984	0.84 %	5.70×10^{-3}	5.6 %
4	0.40	0.3981	0.77 %	1.35×10^{-2}	5.5 %
5	0.50	0.4998	0.64 %	2.68×10^{-2}	5.4 %
6	0.60	0.5939	0.72 %	4.49×10^{-2}	5.4 %
7	0.70	0.7077	0.49 %	7.60×10^{-2}	5.2 %
8	0.80	0.8003	0.30 %	0.110	5.1 %
9	1.00	1.0048	0.20 %	0.217	5.0 %
10	2.00	1.9916	0.20 %	1.47	5.0 %
11	3.00	2.9889	0.15 %	4.65	5.0 %
12	4.00	3.9719	0.24 %	10.6	5.1 %
13	5.00	4.9486	0.078 %	21.5	5.0 %

Table 2. Flexibility parameters of the polycarbonate plates. Here h_{nom} is the nominal thickness, while the mean thickness (h_{avg}) and its relative variability (σ_h/h_{avg}) were determined from 16-point measurements. The bending stiffness (EI) was calculated using the measured h_{avg} . The total relative uncertainty shown in the final column, σ_{EI}/EI , results from propagating the uncertainties from two sources: the estimated uncertainty in Young's Modulus and the measured variability in thickness, as detailed in (2.3).

centre of the circular plate was illuminated by the laser sheet. The deformation was recorded using the high-speed CMOS camera. A threshold-based image segmentation method (Suzuki *et al.* 1985) was employed to extract the plate deformation. Due to the deformation, the projected area of the plate is given by

$$S_p = 2 \int_0^{D/2} 2 \sqrt{\left(\frac{D}{2}\right)^2 - l^2} \cos \theta \, dl, \quad (2.4)$$

where l is the curvilinear coordinate along the radius from the centre towards the tip in the chordwise plane and can be expressed as

$$l = \int_0^l dl = \int_0^z \frac{dl}{dz} dz = \int_0^z \sqrt{k^2 + 1} \, dz, \quad (2.5)$$

and θ is the angle between the plate and the vertical z axis. The cosine of the angle is defined as

$$\cos \theta = \frac{1}{\sqrt{k^2 + 1}}, \quad (2.6)$$

where k is the slope of the curvilinear line. The dimensionless projected area is defined as

$$S_p^* = \frac{S_p}{S}, \quad (2.7)$$

where $S = \pi D^2/4$ is considered the wetted area of the plate.

Furthermore, the velocities of the plate centre U_{centre} and tip U_{tip} can be determined based on the displacement differences of the corresponding points at successive time frames.

As a representative example, figure 3 shows the deformation of a flexible plate ($h_{nom} = 0.30$ mm at $a^* = 0.25$ and $Re = 7 \times 10^4$). The plate exhibits a symmetric, C-shaped bending profile in the chordwise direction (figure 3a), which contrasts with the multifold patterns reported by Schouveiler & Eloy (2013). This confirms that the central

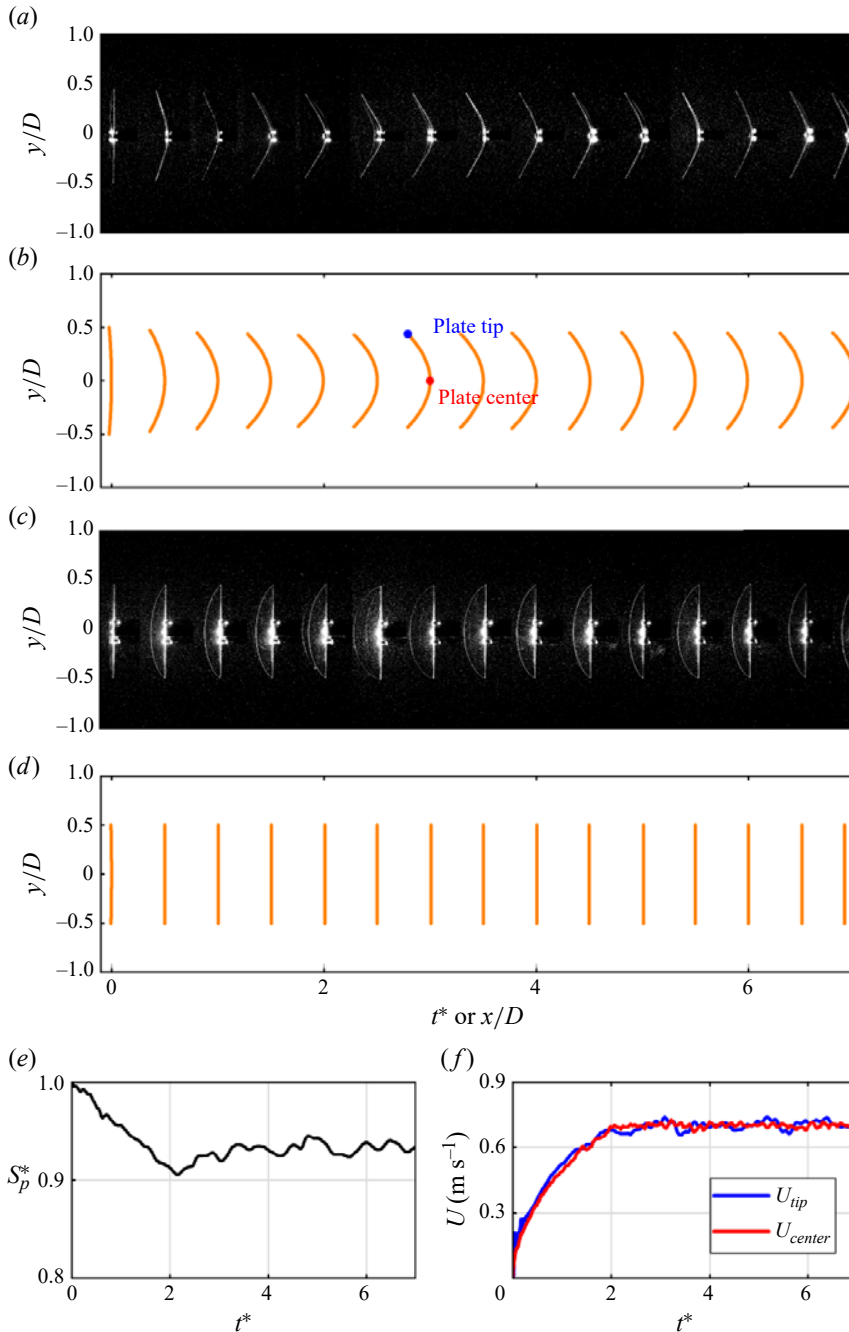


Figure 3. Shape tracing and kinematics of the flexible plate ($h_{nom} = 0.30$ mm) for $a^* = 0.25$ ($t_a^* = 2.0$) and $Re = 7 \times 10^4$. (a,c) Raw images of the plate deformation in the chordwise and spanwise planes, respectively. In these images, the thick bright line is the plate's centreline directly illuminated by the laser, while the thinner lines are the illuminated contours of the plate. The non-smooth appearance of the contour is an artefact of the multi-sub-FOV method used for image acquisition (§ 2.4), but it does not affect the accurate extraction of the centreline profile. (b,d) Corresponding traced profiles, showing a C-shaped deformation chordwise while the spanwise profile remains straight. (e) Evolution of the dimensionless projected area, S_p^* , with formation time. (f) Instantaneous velocities of the plate centre (U_{center}) and tip (U_{tip}).

Nomenclature	R-3	R-4	R-5	R-9	F-3	F-7	F-10	EF-6	EF-9
$Re \times 10^4$	3.18	4.18	5.27	9.00	3.12	7.00	10.8	6.00	9.94
h_{nom} (mm)	0.50	0.60	0.70	1.00	0.175	0.30	0.40	0.125	0.175
EI^*		2.66×10^{-1}			1.16×10^{-2}			1.19×10^{-3}	

Table 3. Nomenclature used in the study. ‘R-’ denotes rigid plates with $EI^* = 2.66 \times 10^{-1}$, ‘F-’ represents flexible plates with $EI^* = 1.16 \times 10^{-2}$ and ‘EF-’ indicates extra-flexible plates with $EI^* = 1.19 \times 10^{-3}$. The number following each letter specifies the corresponding Reynolds number. All cases were conducted under the conditions of $a^* = 0.25$ ($t_a^* = 2.0$).

support constrains deformation to the intended beam-like mode (§ 2.1). The quantified profiles further show a clear C-shape chordwise while the spanwise section remains straight, confirming the suppression of multifold deformation (figure 3*b,d*). Based on the traced geometries, the dimensionless projected area S_p^* (figure 3*e*), as well as the velocities of the plate centre and tip (figure 3*f*), are calculated. To assess the run-to-run variability of the deformation, the shape tracking experiment was repeated five times. For this representative case, the run-to-run relative standard deviation of the projected area (S_p^*) is less than 0.01 %, while for the centre and tip velocities it is 0.8 % and 1.9 %, respectively. The low variability in S_p^* is expected, as its integral calculation smooths noise from the position data. Conversely, the higher velocity variability results from numerical differentiation, which is known to amplify noise. The slightly greater variation at the tip is attributed to its unconstrained free-end condition. These low deviations confirm the high repeatability of the measurements. Across all cases considered (table 3), the maximum relative standard deviation for S_p^* did not exceed 0.13 %. This quantitative evidence of low standard deviation confirms that the plate’s motion is repeatable, dominated by large-scale fluid–structure reconfiguration rather than random vibrations.

2.3. Force data acquisition

The hydrodynamic force measurement scheme was adapted from the work of Reynolds *et al.* (2014), Grift *et al.* (2019) and Fernando & Rival (2016*a*). A force/torque transducer (ATI Gamma F/T Mux SI-130-10) was installed between the linear guide and the L-shaped sting, as shown in figure 1. The Gamma F/T Mux force transducer has a force range of ± 130 N with a resolution of 0.025 N. Data were recorded at a sampling frequency of 7000 Hz in this study. Under static conditions, the natural frequency of the system was approximately 48.7 ± 3.3 Hz in air and 46.7 ± 3.3 Hz in water. Under dynamic conditions as listed in table 1, the natural frequency was approximately 50.3 ± 1.4 Hz in air and 48.0 ± 1.1 Hz in water. Based on the kinematics of the plate in this study as mentioned in § 2.2, the time interval corresponding to the plate’s acceleration is [0.0625 s, 1.0 s] or [1 Hz, 16 Hz], different from the natural frequency to avoid resonance. A low-pass filter with a 40 Hz cutoff frequency was first applied to remove high-frequency noise. Subsequently, a second-order Savitzky–Golay filter (Savitzky & Golay 1964) with a filter window width of $\Delta t^* = 0.1$ was used to reduce noise. This window corresponds to an absolute time range of [0.015 s, 0.055 s], allowing for 6–20 filter windows during the acceleration phase to retain as much instantaneous force information as possible. The sensitivity of the filter window is quantified in Appendix A2.

In actual measurements, the force with the plate attached in the x -direction F_x was first measured and filtered as exemplified in figure 4(*a,e*). Subsequently, the force without the flexible plate in the x -direction F_{x0} was measured and filtered as shown in figure 4(*b,f*).

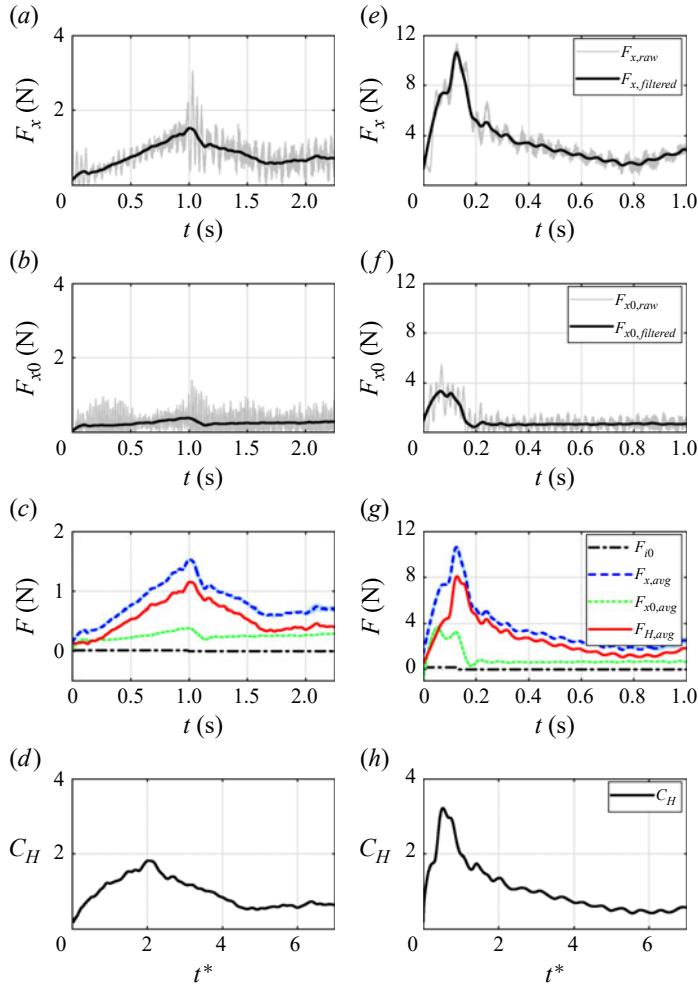


Figure 4. Force data acquisition for a 3.00 mm-thick plate was conducted under the conditions of $a^* = 0.25$ ($t_a^* = 2.0$), $Re = 4 \times 10^4$ in panels (a–d), and under the conditions of $a^* = 1.0$ ($t_a^* = 0.5$), $Re = 8 \times 10^4$ in panels (e–h). Panels (a) and (e) illustrate the raw force $F_{x,raw}$ (grey thin solid lines) obtained from a single measurement on the plate, along with its filtered results $F_{x,filtered}$ (black thin solid lines) as a function of time t . The force measurements without the plate are depicted in panels (b) and (f). The relative standard deviations of $F_{x,avg}$ and $F_{x0,avg}$ are 1.0 % and 1.3 %, respectively, relative to the maximum value in panel (c), and 0.9 % and 0.4 % in panel (g) over 10 measurements, indicating excellent repeatability of the data. The resultant hydrodynamic forces F_H (red solid line) in panels (c) and (g) are obtained by subtracting $F_{x,avg}$ (blue dashed line) from $F_{x0,avg}$ (green dotted line) and the inertial force F_{i0} (black dashed–dotted line, accounts for less than 0.1 % of F_H) in both panels (c) and (g). Lastly, the hydrodynamic force coefficients C_H are plotted against the dimensionless time t^* in panels (d) and (h).

Considering the inertial force $F_i(t)$ due to the acceleration of the plate itself, the hydrodynamic force can be expressed as

$$F_H(t) = F_x(t) - F_{x0}(t) - F_i(t), \quad (2.8)$$

where, due to the bending deformation, the acceleration at different parts of the plate varies. Specifically, since the flexible deformation is directed backwards, during the acceleration phase, the acceleration of each part of the plate is less than that of the plate

centre, i.e. $\partial U_i / \partial t \leq a$. Therefore, the actual inertial force of the flexible plate is less than its inertial force in the undeformed state, i.e.

$$F_i(t) = 2 \sum_{i=0}^N m_i \frac{\partial U_i}{\partial t} \leq F_{i0}(t) = 2 \sum_{i=0}^N m_i a(t) = \rho_m S h a(t), \quad (2.9)$$

where, the deformation of different segments with mass m_i of the plate was obtained, allowing us to determine the displacement x_i , velocity U_i and acceleration $\partial U_i / \partial t$ for each segment; $i = 0$ is the segment through the centre of the plate and $i = N$ is the segment of the tip of the plate. As shown in figure 4(c,g), $F_{i0}(t)$ accounts for less than 0.1 % of the hydrodynamic force. Thus, $F_{i0}(t)$ can be neglected and the calculation of $F_H(t)$ can be approximated as

$$F_H(t) \approx F_x(t) - F_{x0}(t). \quad (2.10)$$

The resulting hydrodynamic force is shown in figure 4(c,g). All data sets were averaged over 10 runs. The relative standard deviation of the hydrodynamic force for each plate is less than 1.3 %. Based on the hydrodynamic force, the classical hydrodynamic force coefficient is defined as

$$C_H = \frac{F_H}{\frac{1}{2} \rho_f U_c^2 S}. \quad (2.11)$$

The resulting force coefficient C_H is shown in figure 4(d,h). The uncertainty in the hydrodynamic force coefficient, C_H , was quantified using error propagation. The worst-case scenario was selected based on the physical acceleration ($a = a^* U_c^2 / D$), which is the primary driver of vibrational noise in the measurements. The high-acceleration kinematics validation case ($a^* = 1.0$, $Re = 8 \times 10^4$), with $a \approx 6.43 \text{ m s}^{-2}$, was therefore chosen as the most conservative benchmark. Assuming uncorrelated errors, the uncertainty in C_H is propagated from the standard deviations of the measured forces (σ_{F_x} and $\sigma_{F_{x0}}$) as

$$\sigma_{C_H} = \frac{\sqrt{\sigma_{F_x}^2 + \sigma_{F_{x0}}^2}}{\frac{1}{2} \rho_f U_c^2 S}. \quad (2.12)$$

Using data from this high-acceleration benchmark case (figure 4), the propagated uncertainty was estimated to be approximately 2.9 %, which confirms the high precision of the force measurements.

Further, the instantaneous force coefficient normalized by the projected area can be defined as

$$C_H^* = \frac{C_H}{S_p^*}. \quad (2.13)$$

The C_H of the 3.00 mm-thick plate at $a^* = 0.50$, 1.0 and 2.0, and $Re = 8 \times 10^4$ was used to compare with results from the previous literature, as shown in figure 5. Within $t^* \in [0, 7]$, the deviation between the force measurement results of this study and Li *et al.* (2022) is 1.1 %, and Fernando *et al.* (2020) is 4.6 %.

2.4. Flow field measurement

Two-dimensional, two-velocity-component (2D2C) time-resolved planar PIV was employed to quantitatively capture the flow fields. The 2D2C PIV configuration was deemed

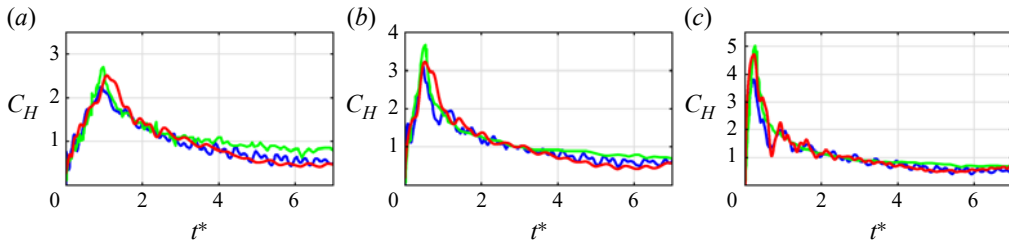


Figure 5. The force data acquisition was validated by comparing the results with previously published data for a rigid plate ($h_{nom} = 3.00$ mm): (a) $a^* = 0.5$ ($t_a^* = 1.0$), $Re = 8 \times 10^4$; (b) $a^* = 1.0$ ($t_a^* = 0.5$), $Re = 8 \times 10^4$; (c) $a^* = 2.0$ ($t_a^* = 0.25$), $Re = 8 \times 10^4$. The green solid lines are adopted from the Fernando *et al.* (2020), blue solid lines are adopted from Li *et al.* (2022) and the red solid lines are the force coefficient measured by current set-up. It is important to note that the definition of the drag coefficient in Fernando *et al.* (2020) differs from the definition of the hydrodynamic coefficient used in this study by a factor of $\pi/8$, and to ensure a fair comparison, the data from Fernando *et al.* (2020) have been adjusted accordingly.

sufficient for this study's primary goal: to uncover the physical mechanisms behind the load-shifting phenomenon. While vortex rings are inherently three-dimensional (3-D), a multislice measurement strategy was implemented to capture the critical wake features. This planar approach is directly compatible with the vorticity moment theorem used for force estimation, which requires only 2-D velocity and vorticity data (§ 5.1).

Multislice measurements were conducted in three representative cross-sectional planes relative to the plate: the chordwise ($\theta = 0^\circ$), diagonal ($\theta = 45^\circ$) and spanwise ($\theta = 90^\circ$) sections. In each case, the laser sheet was projected from the left-hand side of the water tank across the plate's centre, as illustrated in figure 1. A high-speed CMOS camera (Phantom VEO-640) positioned perpendicular to the laser sheet recorded the particle field images. To capture the flow fields of different cross-sections, the plate's orientation was altered while the laser and camera positions remained fixed. This was accomplished using three separate horizontal rods, each fabricated with a mounting slot milled at a distinct angle (0° , 45° or 90°). Appendix A3 validates its effectiveness for consistently characterizing the different cross-sections of the wake. Hollow glass beads (spherical, density $\rho_g = 1.03\text{--}1.05$ g cm $^{-3}$, diameter $d_g = 20\text{--}60$ μ m) served as tracer particles, illuminated by a double-pulsed 527 nm Nd-YLF laser (Beamtech Vlite-Hi-527, ≥ 30 mJ at 1 kHz).

To maximize spatial resolution, theoretically, the FOV of over $7D$ was divided into three sub-FOVs along the x -axis: sub-FOV1 ($-1D$ to $2D$), sub-FOV2 ($2D$ to $5D$) and sub-FOV3 ($5D$ to $8D$), where each sub-FOV was designed to have a theoretical size of $3D \times 2D$. The sub-FOVs were captured by fixing the camera and laser while altering the plate's SP: SP1 for sub-FOV1, SP2 for sub-FOV2 and SP3 for sub-FOV3. Each sub-FOV was measured five times, with a 60 s interval between runs to avoid flow interference from prior tests. The distance between SP3 and the wall was $5D$, sufficient to minimize wall effects. This distance was chosen also based on the laser beam waist, which was approximately 1 m from the laser head. As the deviation from this distance increases, the laser sheet thickness also increases, potentially introducing larger PIV errors. In this experiment, the distance between the centre of captured FOV to the laser head was 1.35 m, ensuring good imaging results while maintaining acceptable laser sheet thickness.

In practice, each sub-FOV spanned $3.65D \times 2.28D$ by recorded with 2560×1600 pixels (figure 1) and the adjacent windows overlapped and were stitched using interpolation (Li *et al.* 2022). The sampling frequency for PIV data was 800 Hz, ensuring

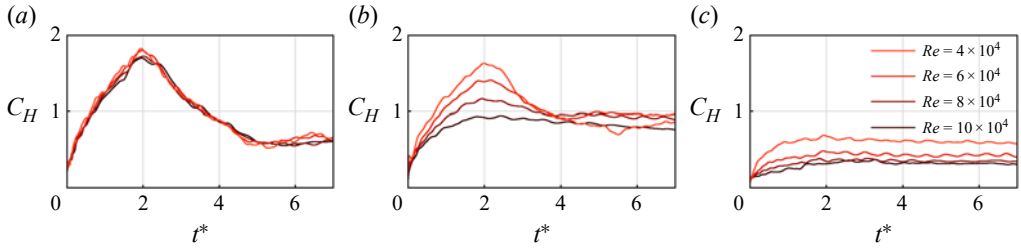


Figure 6. Effect of Reynolds number Re on transient force evolution for plates with $a^* = 0.25$ ($t_a^* = 2.0$) and varying bending stiffness: (a) $h = 2.00$ mm, $EI = 1.72$ Nm; (b) $h = 0.30$ mm, $EI = 5.79 \times 10^{-3}$ Nm; (c) $h = 0.125$ mm, $EI = 4.19 \times 10^{-4}$ Nm.

high temporal resolution. To ensure an optimal time delay (Δt) between frames, the maximum particle displacement was restricted to $0.25S_{IS}$, where S_{IS} is the size of the interrogation spot. Using characteristic velocity U_c , Δt was computed as $\Delta t = 0.25S_{IS}/2U_c$, considering the maximum flow velocity magnitude to be approximately twice the plate's translation velocity (Li *et al.* 2022).

Image processing was performed using TSI Insight 4G software, following PIV analysis guidelines (Keane & Adrian 1989). Background subtraction and light intensity normalization (10×10 pixel window) were applied to mitigate measurement bias caused by particle brightness and laser intensity variations. The multipass method was applied with interrogation spot sizes of 64×64 pixels (initial pass) and 32×32 pixels (final pass), with a 50 % overlap. This yielded 159×99 velocity vectors per sub-FOV with a vector spacing of 2.31 mm. Vorticity fields were calculated using second-order finite differences on velocity data from eight neighbouring points.

Uncertainty in flow measurements was estimated using the peak ratio method (Charonko & Vlachos 2013), incorporated in the Insight 4G software. This global method accounts for errors such as particle density, pixel displacement and preprocessing. For this study, velocity field uncertainty across all runs was less than 1.2 %.

To ensure statistical robustness, five independent runs were conducted for each representative case. The run-to-run variability of the derived flow metrics (discussed in §§ 4 and 5) proved to be minimal, with relative standard deviations for key integral quantities remaining below 4.9 %. Consequently, $\pm 1\sigma$ confidence intervals are included in the subsequent time-evolution plots to demonstrate the deterministic nature of the observed flow features. Furthermore, to calculate time derivatives for force estimation (§ 5), the same second-order Savitzky–Golay filter ($\Delta t^* = 0.1$, consistent with the force filter in Appendix A2) was applied to the PIV-derived time series.

3. Force

3.1. Force evolution across different Re and EI

The transient hydrodynamic force coefficients for circular plates with three orders of magnitude in bending stiffness, $EI = 1.72$ Nm, 5.79×10^{-3} Nm and 4.19×10^{-4} Nm, under varying Reynolds numbers Re are shown in figure 6.

For the plate with bending stiffness $EI = 1.72$ Nm (corresponding to a thickness of $h = 2.00$ mm), which behaves as a rigid plate with negligible deformation, the force curves collapse well across different Reynolds numbers, confirming a clear Reynolds number independence (Fernando & Rival 2016b; Li *et al.* 2022). Furthermore, the force evolution exhibits a distinct peak-and-valley pattern (figure 6a). During acceleration, the force

increases rapidly with t^* , reaching a peak at $t^* = 2$, followed by a decrease to a minimum at $t^* \approx 5.5$ – the so-called ‘drag trough’ (Fernando & Rival 2016a) – before rising again.

As the bending stiffness decreases to $EI = 5.79 \times 10^{-3}$ Nm (corresponding to a plate thickness of $h = 0.30$ mm), the force response exhibits significant changes with increasing Reynolds number, leading to a breakdown of Reynolds number independence (figure 6b). At $Re = 4 \times 10^4$, the characteristic peak-and-valley pattern persists, with a distinct force peak at $t^* = 2$ and a trough at $t^* \approx 5.5$. However, compared with the rigid plate, the peak force is reduced, while the trough force is elevated. As Re increases, the force peak gradually flattens, and the trough becomes less pronounced. At $Re = 10 \times 10^4$, the peak-and-valley pattern disappears entirely, resulting in a smooth and uniform force evolution with no discernible extrema. Notably, the force coefficient at $t^* \approx 5.5$ initially increases with Re before declining at higher values. These results indicate that, as the bending stiffness decreases, the unsteady force dynamics lose their Reynolds number independence, underscoring the coupled effects of flexibility and Reynolds number on force evolution.

For plates with further reduced bending stiffness to $EI = 4.19 \times 10^{-4}$ Nm (corresponding to a thickness of $h = 0.125$ mm), the force curves remain non-collapsing across Reynolds numbers but exhibit consistent trends: a rapid rise during acceleration followed by stabilization with negligible fluctuations. The peak-and-valley pattern disappears entirely.

Overall, these results emphasize the coupled effects of Reynolds number and flexibility on unsteady force dynamics, causing the force evolution on reconfigurable plates to deviate from the Reynolds number independence and peak-and-valley pattern observed in rigid plates.

3.2. Non-dimensional bending stiffness EI^* independence of force evolution

The breakdown of Reynolds number scaling stems from its primary focus on fluid dynamics, whereas the reconfigurable plate in this study undergoes deformation that modifies the surrounding flow, resulting in a coupled fluid–structure interaction. To characterize this interaction, a non-dimensional bending stiffness, EI^* , is employed as

$$EI^* = \frac{EI}{\rho_f U_c^2 D^3}. \quad (3.1)$$

The physical significance of EI^* has been interpreted differently across various studies. From an energy perspective, Schouveiler & Boudaoud (2006) defined the elastohydrodynamical number, expressed as $1/EI^*$, to represent the ratio of flow-induced potential energy to elastic bending energy. Similarly, Alben *et al.* (2002, 2004) introduced a parameter equivalent to $\sqrt{1/EI^*}$, balancing fluid kinetic energy with elastic potential energy. Alternatively, from a force perspective, De Langre (2008) and Gosselin *et al.* (2010) considered the Cauchy number, Ca , expressed as $1/EI^*$, as the ratio of fluid dynamic pressure to plate bending rigidity. In the present study, EI^* is adopted from the perspective of wing flapping aerodynamics (Shyy *et al.* 2010; Kang *et al.* 2011), characterizing the balance between elastic bending forces and aerodynamic forces. The derivation of EI^* can be found in the Supplementary material.

Three orders of magnitude of EI^* were selected as representative cases (table 3). The corresponding deformation and force data are shown in figure 7. From figure 7, despite variations in EI and Re , the deformation (S_p^* defined in (2.7)) and force evolution (C_H defined in (2.11) and C_H^* defined in (2.13)) collapse onto a single curve as a function of formation time, with only limited deviations under the same EI^* condition.

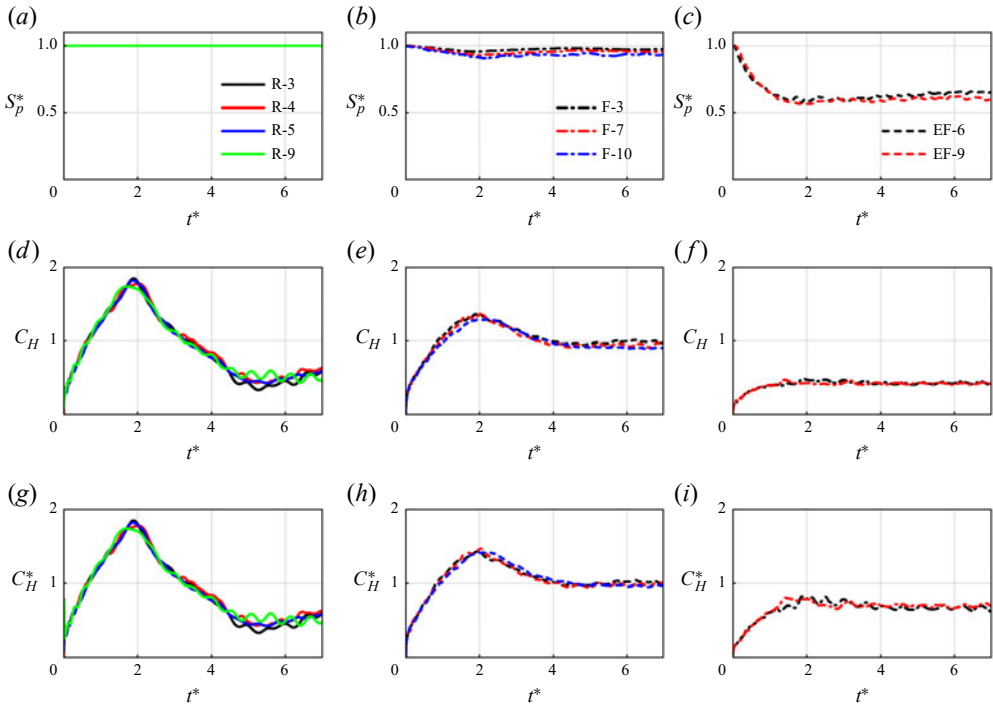


Figure 7. Instantaneous S_p^* (a–c), C_H (d–f) and C_H^* (g–i) for three orders of magnitude of EI^* ; legend details can be found in [table 3](#).

The deformation behaviour under different EI^* values is presented in [figure 7\(a,b\)](#). For $EI^* = 2.66 \times 10^{-1}$, no deformation is observed, and the plate maintains its projected area as a perfect circle, effectively behaving as rigid. At $EI^* = 1.16 \times 10^{-2}$, moderate deformation occurs, categorizing the plate as flexible. For $EI^* = 1.19 \times 10^{-2}$, significant deformation reduces the projected area to nearly half its original size, showing the plate as extra flexible.

As for the force evolution, from [figure 7\(d,g\)](#), when $EI^* = 2.66 \times 10^{-1}$, the instantaneous force evolution follows the so called peak-and-valley pattern as discussed in § 3.1. When EI^* decreases to 1.16×10^{-2} ([figure 7e,h](#)), the peak-and-valley characteristic weakens: the peak force decreases, the valley force increases and the force stabilizes around the steady drag coefficient after $t^* \approx 4$. Further reduction of EI^* to 1.19×10^{-2} eliminates the peak-valley pattern entirely ([figure 7f,i](#)). The instantaneous force increases during the acceleration phase and stabilizes at $t^* \approx 1.5$, earlier than the end of acceleration ($t^* = 2$). However, regardless of whether C_H or C_H^* is considered, the instantaneous force coefficient curves collapse under identical EI^* . This demonstrates that force evolution is primarily governed by EI^* and remains consistent under the same non-dimensional bending stiffness conditions.

3.3. Load-shifting phenomenon in force evolution

To further investigate the impact of different values of EI^* on the force evolution pattern, the reconfiguration number, originally defined under steady conditions by Gosselin *et al.* (2010), is extended to the unsteady accelerating process. The reconfiguration number $R(t^*)$ is defined as

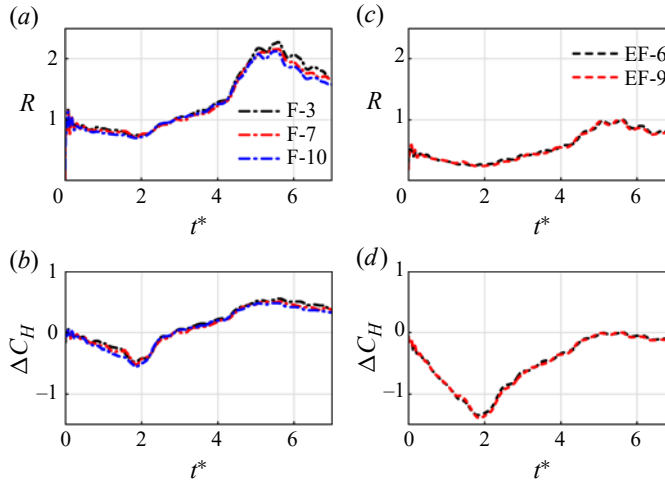


Figure 8. The reconfiguration number R and the absolute hydrodynamic force coefficient difference ΔC_H for the flexible plate (a,b) and extra-flexible plate (c,d). Legend details can be found in table 3.

$$R(t^*) = \frac{C_{H_f}(t^*)}{C_{H_r}(t^*)}, \quad (3.2)$$

where $C_{H_f}(t^*)$ and $C_{H_r}(t^*)$ represent the instantaneous hydrodynamic force coefficients of the flexible/extra-flexible and rigid plates, respectively. This normalization employs the rigid response as a dynamic baseline, thereby removing the inherent time-dependent variations of the accelerating flow and isolating the effect of reconfiguration (Gosselin *et al.* 2010). A value of $R(t^*) < 1$ indicates force reduction relative to the rigid case, and $R(t^*) > 1$ indicates enhancement.

Additionally, to clearly present the results, the absolute hydrodynamic force coefficient difference ΔC_H is defined as

$$\Delta C_H(t^*) = C_{H_f}(t^*) - C_{H_r}(t^*). \quad (3.3)$$

A negative value ($\Delta C_H < 0$) indicates force reduction, whereas a positive value ($\Delta C_H > 0$) represents force enhancement. While $R(t^*)$ is ideal for assessing the relative impact of reconfiguration, $\Delta C_H(t^*)$ provides a direct measure of the force changes.

Results for R and ΔC_H for flexible plates with $ET^* = 1.16 \times 10^{-2}$ and extra-flexible plates with $ET^* = 1.19 \times 10^{-2}$ are shown in figure 8.

As shown in figure 8(a,b), during acceleration, the force on the flexible plate is lower than that on the rigid plate ($R < 1$ and $\Delta C_H < 0$), with the difference increasing over time and peaking at $t^* = 2$ (end of acceleration). Subsequently, the difference diminishes, and the forces equalize at $t^* \approx 2.8$ ($R = 1$ and $\Delta C_H = 0$). Beyond this point, the force on the flexible plate exceeds that on the rigid plate ($R > 1$ and $\Delta C_H > 0$), with the maximum difference observed at $t^* = 5.5$. This pattern of force evolution is herein referred to as ‘load-shifting’. For the extra-flexible plate (figure 8c,d), the hydrodynamic force coefficient remains consistently lower than that of the rigid plate.

Based on these results, the classification of rigid, flexible and extra-flexible plates can be more clearly defined. A rigid plate is characterized by its force evolution following Reynolds number independence. In contrast, flexible and extra-flexible plates exhibit Reynolds-number-dependent force evolution during acceleration. A flexible plate exhibits load-shifting phenomenon, with its peak force at $t^* = 2$ being lower than that of the rigid

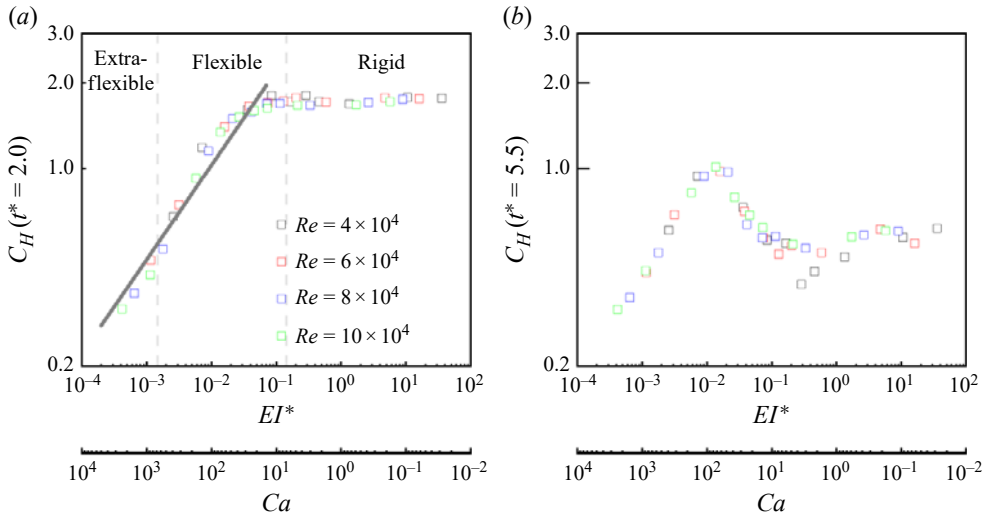


Figure 9. Instantaneous force coefficients at $t^* = 2$ (a) and $t^* = 5.5$ (b) against non-dimensional bending stiffness EI^* (or, the Cauchy number, $Ca = 1/EI^*$), for different bending stiffness EI as listed in table 2 and four Reynolds numbers $Re = 4.0, 6.0, 8.0, 10.0 \times 10^4$. The boundaries (vertical light grey dashed lines) between extra-flexibility, flexibility and rigidity are identified at two inflection points: $EI^* = 2.28 \times 10^{-3}$ and $EI^* = 0.143$. The thick grey solid line is the power-law fit from (3.10).

plate and its valley force at $t^* = 5.5$ being higher. An extra-flexible plate, however, does not exhibit load-shifting phenomenon, as its force evolution remains consistently lower than that of the rigid plate throughout the acceleration process.

Reynolds number scaling is well recognized (Fernando & Rival 2016a,b; Grift *et al.* 2019; Li *et al.* 2022, 2024), while load-shifting phenomenon is particularly relevant in biological propulsion, as it indicates that reconfiguration can reduce force variation without altering the mean force (Shyy *et al.* 2010; Kim & Gharib 2011a). From a gust alleviation perspective, reconfiguration can mitigate instantaneous high loads and reduce force fluctuations, improving biological flight stability and enhancing vegetation survival (Triantafyllou *et al.* 2016; Cheney *et al.* 2020). Extended discussions on the physical value of load-shifting can be found in the Supplementary material.

Furthermore, when EI^* is large (figure 7a,d,g), the plate behaves rigidly without deformation. Conversely, when EI^* is too small, the plate collapses entirely (figure 7c,f,i), and the instantaneous force remains consistently below the minimum force of the rigid plate (figure 8c,d). This indicates that both excessively large and small EI^* are suboptimal.

3.4. Load-shifting regime boundaries defined by EI^*

To determine the boundaries of this load-shifting phenomenon, the instantaneous forces were measured for each plate (EI ranging from 4.19×10^{-4} to 36.2 Nm) in table 2 at Reynolds numbers $Re = 4.0, 6.0, 8.0, 10.0 \times 10^4$, covering an EI^* range from approximately 10^{-4} to 10^2 . Force coefficients at $t^* = 2$ (peak) and $t^* = 5.5$ (valley) were extracted and plotted against EI^* (figure 9). Variations of the reconfiguration number R and force difference ΔC_H , referenced to the rigid plate at $t^* = 2$ and $t^* = 5.5$, are shown in figure 10. As EI^* decreases, both R and ΔC_H at $t^* = 2$ decline, whereas at $t^* = 5.5$, they increase and then decrease.

The load-shifting phenomenon is defined by a peak force reduction ($R(t^* = 2) < 1$) coupled with a valley force enhancement ($R(t^* = 5.5) > 1$). As neither the rigid ($R \approx 1$)

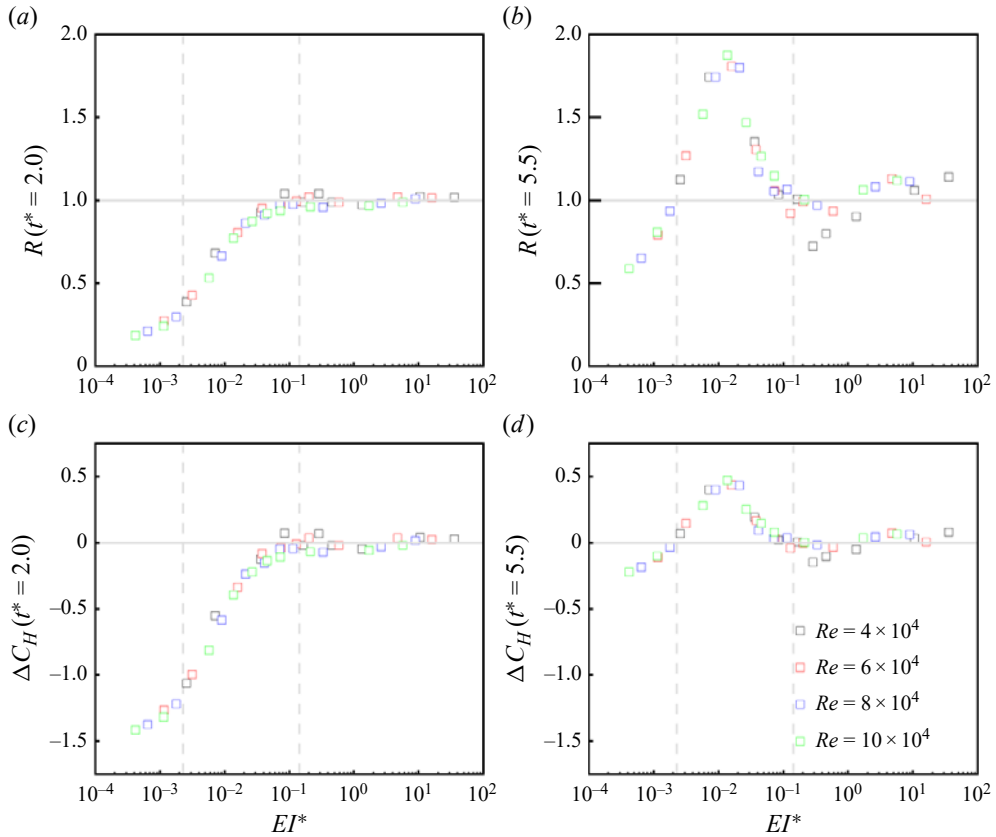


Figure 10. Absolute hydrodynamic force coefficient difference ΔC_H (a,b) as functions of EI^* . Panels (a) and (c) correspond to the results at $t^* = 2$, while panels (b) and (d) correspond to the results at $t^* = 5.5$. Here ΔC_H is referenced to the rigid plate values, as detailed in figure 9. The vertical light grey dashed lines indicate the boundaries between extra-flexible, flexible and rigid, identified at $EI^* = 2.28 \times 10^{-3}$ and $EI^* = 0.143$.

nor the extra-flexible ($R(t^* = 5.5) < 1$) plates satisfy these conditions, the load-shifting phenomenon is confined to the intermediate EI^* regime. The following analysis seeks to identify the critical transition points, $EI_{crit,1}^*$ and $EI_{crit,2}^*$, that delineate the boundaries of this flexible regime.

The flexible-to-rigid transition ($EI_{crit,2}^*$). This occurs when the plate's elastic restoring moment can no longer balance the peak hydrodynamic bending moment, marking the onset of deformation. Since the plate undergoes symmetric beam-like bending (figure 3), the peak hydrodynamic moment can be written as

$$M_{hydro,peak} = F_{H,half} \times L_{arm}, \quad (3.4)$$

where $F_{H,half} = F_{H,peak}/2$ is the hydrodynamic force on half of the plate, and $L_{arm} = k_p(D/2)$ is the effective moment arm with k_p denoting a pressure distribution factor (Batchelor 2000).

The elastic resisting moment of the deforming half-plate, M_{resist} , follows from plate theory (Landau *et al.* 2012). The moment per unit length scales as $M_l \sim EI/D$, and integrating over the deforming radius ($D/2$) gives

$$M_{resist} \approx M_l \times D/2 \approx EI/2. \quad (3.5)$$

Equating $M_{hydro,peak} \approx M_{resist}$ yields the predicted critical stiffness:

$$EI_{crit,2}^* = \frac{EI}{\rho_f U_c^2 D^3} \approx \frac{\pi}{16} k_p C_{H,rigid,peak}. \quad (3.6)$$

Here, $C_{H,rigid,peak}$ denotes the peak force coefficient of a rigid plate, which is primarily governed by a^* (Li *et al.* 2022, 2024), Re (Liu & Sun 2024) and the plate geometry (Fernando & Rival 2016a,b; Reijtenbagh *et al.* 2023a), while additional factors may also contribute. In this study for $a^* = 0.25$, $Re \sim 10^4$ and circular plates, $C_{H,rigid,peak} \approx 1.73$ (figure 9). And, assuming a uniform pressure distribution gives $k_p \approx 0.42$. Substituting into (3.6) yields

$$EI_{crit,2}^* \approx 0.143. \quad (3.7)$$

This theoretical prediction (≈ 0.143) agrees reasonably with the value reported by Baskaran *et al.* (2023) (≈ 0.111). The slightly larger value in the present study can be attributed to two main factors. First, the accelerating flow conditions here differ from the steady flow in Baskaran *et al.* (2023); under unsteady conditions, added-mass effects amplify the peak force (Li *et al.* 2022, 2024), thus requiring greater bending stiffness to initiate deformation and shifting the transition to a larger EI^* . Second, the discrepancy may also arise from the assumption of a uniform pressure distribution in our calculation. In reality, as the plate tips bend to align with the flow, the local pressure likely decreases (Gosselin *et al.* 2010), shifting the centre of pressure inward and resulting in a smaller actual value for the pressure distribution factor, k_p .

The extra-flexible to flexible transition ($EI_{crit,1}^*$). Once $EI_{crit,2}^*$ is established, the forces in the extra-flexible/flexible range can be analysed with the $1/3$ power law (Alben *et al.* 2002, 2004; Gosselin *et al.* 2010). The distinction between extra-flexible and flexible regimes in this study is based on deviations of the force coefficients from those of a rigid plate, particularly at the force valley. From a fluid–structure interaction perspective, both regimes involve significant deformation and can therefore be described within the Vogel exponent framework (Vogel 1989), which underlies the $1/3$ power law. In these regimes, the drag force scales with velocity as $F \propto U^{1+\mathcal{V}}$, with Vogel exponent $\mathcal{V}$.

Following Gosselin *et al.* (2010) and our experimental observations in small EI^* regime (figures 3 and 7), two features are crucial. First, large deformations align the plate edges with the flow, minimizing their contribution to drag; only the central region near the clamped edge generates appreciable drag. Thus, the chordwise dimension D becomes irrelevant as a characteristic length. Second, the plate undergoes quasi-2-D symmetric beam-like bending without 3-D folding, allowing the drag to be approximated as a 2-D force per spanwise length F/D , analogous to flexible fibres in soap films (Alben *et al.* 2002, 2004).

Accordingly, the governing physical quantities are

$$EI \text{ [Nm]}, \quad U_c \text{ [m s}^{-1}\text{]}, \quad \rho_f \text{ [kg m}^{-3}\text{]}, \quad F/D \text{ [N m}^{-1}\text{]} \quad (3.8)$$

.Dimensional analysis then yields

$$F \sim D \rho_f^{2/3} U_c^{4/3} EI^{1/3}. \quad (3.9)$$

This corresponds to a Vogel exponent of $\mathcal{V} = -2/3$, consistent with the $1/3$ power law proposed in Alben *et al.* (2002, 2004), where beam bending stiffness (Beer, Johnston & DeWolf 1999) (with units N m^2) is used. Despite the unit differences, both studies conform to the same scaling exponent. This relationship can be further simplified in

non-dimensional form as

$$C_H = A \times (EI^*)^{1/3}, \quad (3.10)$$

where A is derived directly from the values of $C_{H,rigid,peak}$. Since $C_{H,rigid,peak}$ is influenced by a^* (Li *et al.* 2022, 2024), Re (Liu & Sun 2024) and plate geometry (Fernando & Rival 2016a,b; Reijtenbagh *et al.* 2023a), A can also be expected to depend on these physical parameters. In this study, using data from figure 9, we obtain $A = 4.56$, with a regression fit of $R^2 = 0.928$. The fit remains below 0.95 because, near $EI^* \lesssim EI_{crit,2}^*$, the deformation is insufficient and the plate tip still contributes to the pressure distribution, violating the model assumptions. Hence, (3.10) is valid primarily below $EI_{crit,2}^*$ and is not used to back-calculate $EI_{crit,2}^*$. Nevertheless, when $EI^* \ll EI_{crit,2}^*$, the scaling holds. The intersection of (3.10) with the rigid trough coefficient $C_{H,rigid,trough} \approx 0.60$ gives

$$EI_{crit,1}^* = (0.60/4.56)^3 \approx 2.28 \times 10^{-3}. \quad (3.11)$$

In summary, based on moment balance and the 1/3 power law, the onset of both rigid and extra-flexible regimes is successfully predicted. The load-shifting phenomenon is confined to $2.28 \times 10^{-3} \leq EI^* \leq 0.143$, as indicated by the grey dashed lines in figures 9 and 10, where plates in the intermediate flexible regime exhibit load-shifting phenomenon.

4. Flow pattern

4.1. Temporal evolution of flow fields

To investigate the temporal evolution of the flow field associated with load-shifting phenomenon, plate ‘F-7’, is taken as a representative case. The non-dimensional vorticity is used to characterize the flow field,

$$\omega^* = \frac{\omega D}{U_c}, \quad (4.1)$$

where ω is the vorticity.

Figure 11 illustrates the vortex evolution in the chordwise cross-section ($\theta = 0^\circ$). The vorticity supply for vortex ring growth originates from the shear layer at the plate’s edge. As the plate moves, the fluid rolling past its edge forms a vortex ring. During the acceleration phase ($t^* \leq 2$), the vortex ring grows continuously. After acceleration ceases, during the constant-speed phase, the vortex ring continues to develop. At $t^* = 3$, the vortex ring begins to break down as vorticity is axially stretched, with portions of vorticity lagging behind the plate’s motion. At $t^* = 4$, the concentrated vorticity region undergoes significant breakdown into smaller patches, some of which are transported farther downstream. At $t^* = 5$, the breakdown intensifies, and vorticity transport in the wake becomes more pronounced. By $t^* = 6$, the vortex ring has completely broken down, and the vorticity fragments become highly dispersed, transitioning from an initial clustered distribution to a dispersed distribution.

In the diagonal ($\theta = 45^\circ$) (figure 12) and spanwise ($\theta = 90^\circ$) (figure 13) cross-sections, vorticity initially rolls up from the edges, forming a distinct vorticity layer that later rolls inward to create a concentrated vorticity region near the central axis. By $t^* = 3$, the vortex begins to break down, and this process becomes more pronounced as t^* increases. At $t^* = 6$, the vortex ring has completely broken down, and the concentrated vorticity region transitions into a dispersed state.

Another notable feature of the wake flow behind the flexible plate is its non-axisymmetry. The plate undergoes obvious bending deformation in the chordwise (figure 11) and diagonal cross-sections (figure 12), while limited deformation is observed

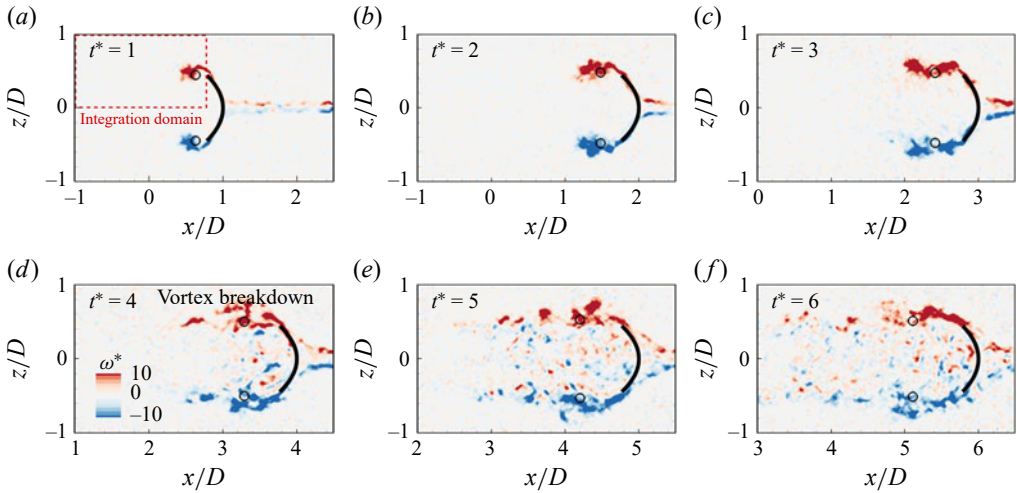


Figure 11. Evolution of the non-dimensional vorticity ω^* over non-dimensional time t^* for the flexible plate 'F-7' in the chordwise cross-section ($\theta = 0^\circ$). The thick black line indicates the deformation and position of the plate. The plate thickness is exaggerated for clarity. The dashed rectangle in panel (a) illustrates the representative integration domain used for calculating circulation (4.2) and other integral quantities ((4.3), (4.4) and (5.14)). The open circles in each panel indicate the locations corresponding to r_v^* (4.3) and z_v^* (4.4) within the respective measurement cross-sections.

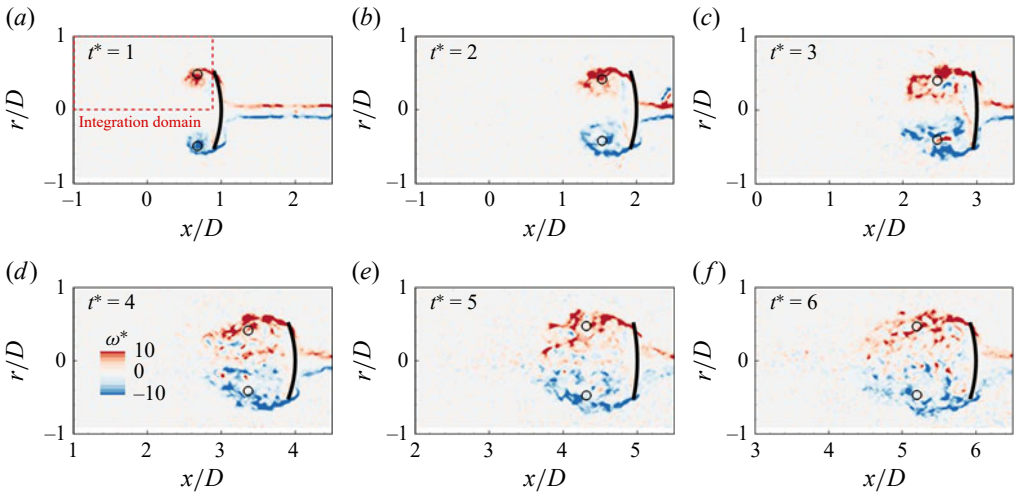


Figure 12. Evolution of the non-dimensional vorticity ω^* over non-dimensional time t^* for the flexible plate 'F-7' in the diagonal cross-section ($\theta = 45^\circ$). Annotations are consistent with those in figure 11.

in the spanwise cross-section (figure 13). Due to the deformation, before breakdown occurs (approximately $t^* < 3$), the evolution of the vorticity distribution differs significantly among the three cross-sections. In the spanwise cross-section, the vorticity distribution exhibits a distinct vortex structure, whereas in the diagonal and chordwise cross-sections, apart from the vortex, a clear vorticity feeding layer is present. Axially, the vortex in the spanwise cross-section lags behind those in the other two cross-sections. Radially, the vorticity in the chordwise cross-section remains roughly parallel to the plate tip, whereas in the diagonal and spanwise cross-sections, it is more concentrated near the symmetry

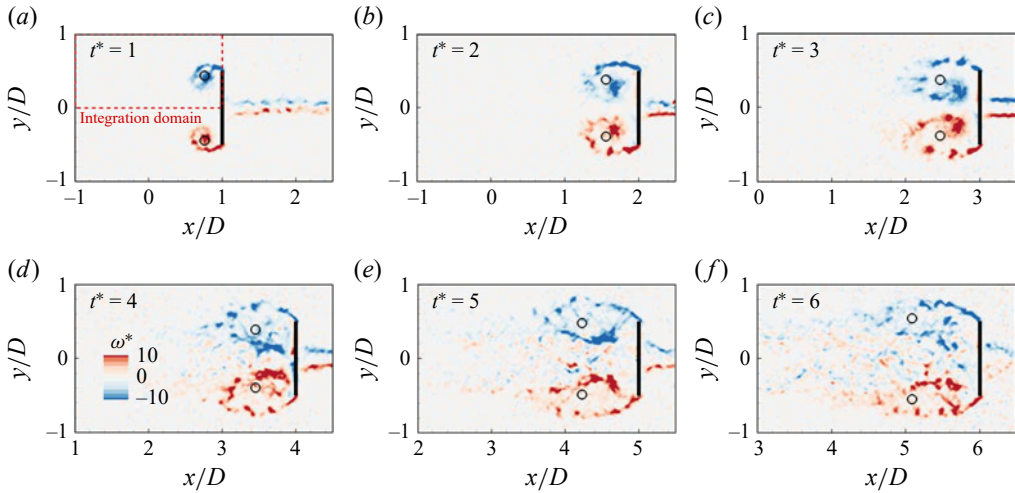


Figure 13. Evolution of the non-dimensional vorticity ω^* over non-dimensional time t^* for the flexible plate ‘F-7’ in the spanwise cross-section ($\theta = 90^\circ$). Annotations are consistent with those in figure 11.

axis. After breakdown, this non-axisymmetry gradually diminishes. Quantitative results related to this non-axisymmetry will be discussed in § 4.2.

Furthermore, a comparison between the flow evolution of extra-flexible, flexible and rigid plates highlights the two key features that characterize the wake of a flexible plate: non-axisymmetry and subsequent vortex breakdown, as detailed in the Supplementary materials. These features are robust; for a given dimensionless stiffness ($El^* = 1.16 \times 10^{-2}$), they persist across all tested Reynolds numbers, underscoring that El^* governs the fundamental flow topology.

4.2. Circulation and vorticity distribution

4.2.1. Circulation

To quantify the strength of the shed vorticity, the non-dimensional total circulation, Γ_v^* , is computed by integrating the vorticity, ω , over the wake region upstream of the plate tip (z'_{tip}) as follows:

$$\Gamma_v^* = \frac{1}{U_c D} \int_0^D \int_{-D}^{z'_{tip}} \omega \, dz' \, dr'. \quad (4.2)$$

The integral is formulated in a cylindrical coordinate system for consistency across the different 2-D measurement planes, where z' is the streamwise axial coordinate and r' is the radial coordinate. The integration domain is illustrated for a representative case in the flow fields shown in figures 11, 12 and 13. This circulation is commonly referred to as the total circulation, in contrast to vortex circulation (Gharib *et al.* 1998; Fernando & Rival 2016b). Due to vortex breakdown, defining the boundary of individual vortex rings for calculating their circulation is challenging. Thus, the total circulation is used to evaluate the overall vorticity strength in the wake. The computed results are shown in figure 14.

The growth of circulation in different cross-sections suggests that, despite differences in non-axisymmetry, the variations in circulation across the different cross-sections of the flexible plate ‘F7’ remain limited. The circulation growth curves in the chordwise and spanwise cross-sections are nearly identical, while the diagonal cross-section exhibits slightly lower values. Similarly, for the rigid plate ‘R5’, where the axisymmetric vortex

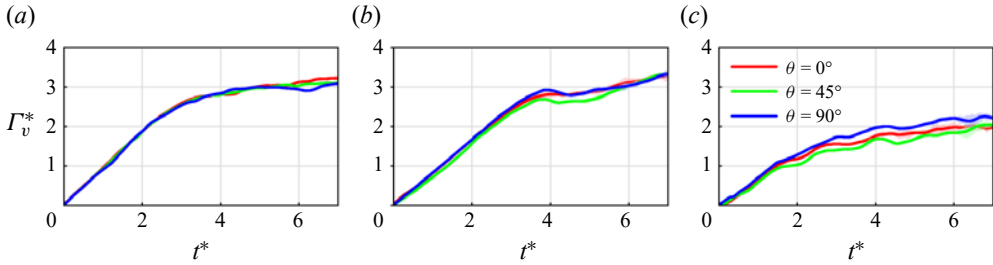


Figure 14. Comparison of the non-dimensional circulation Γ_v^* over non-dimensional time t^* in the chordwise (red line), diagonal (green line) and spanwise (blue line) cross-sections for plates with three different non-dimensional bending stiffness: (a) ‘R-5’, (b) ‘F-7’ and (c) ‘EF-6’.

ring is maintained, the circulation growth curves in all three cross-sections almost overlap. In contrast, the extra-flexible plate ‘EF6’ exhibits differences in circulation growth across the cross-sections due to its non-axisymmetry.

Comparing the circulation among different plates, the flexible plate ‘F7’ exhibits circulation values comparable to those of the rigid plate ‘R5’. This suggests that the total vorticity generated by the flexible plate ‘F7’ is similar to that of the rigid plate ‘R5’, which may be attributed to their similar projected shape and area, as shown in [figure 7\(a,b\)](#). Additionally, the flexible plate ‘F7’ exhibits a longer linear growth phase in circulation, extending up to approximately $t^* = 3.8$, whereas for the rigid plate ‘R5’, the linear growth phase ends at approximately $t^* = 3$. This prolonged growth phase may be related to the contribution of vortex breakdown. In contrast, the extra-flexible plate ‘EF6’ exhibits significantly lower circulation values than the other two cases. Due to its pronounced bending ([figure 7c](#)), the projected area is nearly halved, leading to a weakened vorticity supply from the plate tip.

4.2.2. Vorticity distribution

The vorticity distribution can be characterized by

$$r_v^* = \frac{1}{U_c D^2} \int_0^D \int_{-D}^{z'_{tip}} r' \omega \, dz' \, dr', \quad (4.3)$$

and

$$z_v^* = \frac{1}{U_c D^2} \int_0^D \int_{-D}^{z'_{tip}} z' \omega \, dz' \, dr' - z'_{plate}/D. \quad (4.4)$$

Here, r_v^* is defined as the radial vorticity distribution index, and z_v^* is defined as the axial vorticity distribution index. The term z'_{plate} represents the axial position of the plate centre in cylindrical coordinates, corresponding to the x -coordinate in [figure 1](#), and the subtraction accounts for the relative position of the axial vorticity distribution with respect to the plate. The integration domains are identical to those defined in (4.2). While standardly used to locate single vortex cores (Wu, Ma & Zhou 2007; Xiang *et al.* 2021), for the complex multicore wakes of flexible/extra-flexible plates (§4.1), these indices serve to characterize the global vorticity distribution, thereby quantifying the ring’s 3-D deformation. The computed results are presented in [figures 15](#) and [16](#).

The wake non-axisymmetry of ‘F7’ is quantitatively reflected in the evolution of r_v^* and z_v^* across different cross-sections. Unlike ‘R-5’, where these indices exhibit nearly identical evolution across all three cross-sections ([figures 15a](#) and [16a](#)), the flexible plate

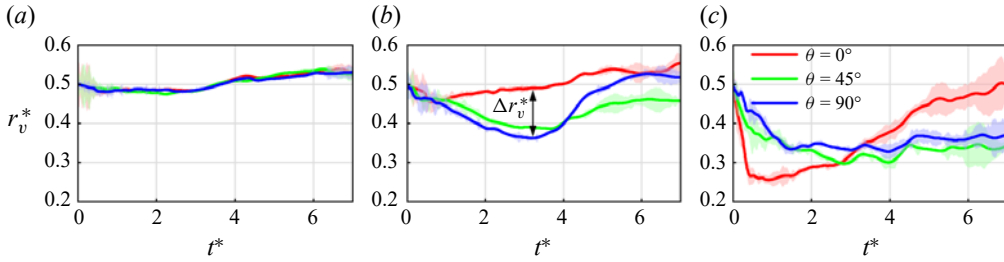


Figure 15. Comparison of the radial vorticity distribution index r_v^* over non-dimensional time t^* in the chordwise (red line), diagonal (green line) and spanwise (blue line) cross-sections for plates with three different non-dimensional bending stiffness: (a) 'R-5', (b) 'F-7' and (c) 'EF-6'. To verify that the observed radial redistribution is physical, the signal-to-noise ratio over five runs was analysed. For the flexible plate ('F-7'), the run-to-run variability ($\sigma_{r_v^*} \approx 0.01$) is an order of magnitude smaller than the physical variation during load-shifting ($\Delta r_v^* \approx 0.12$), confirming statistical robustness. Similarly, for the extra-flexible plate ('EF-6'), despite slightly higher variability due to flow complexity ($\sigma_{r_v^*} \approx 0.02$), the standard deviation remains sufficiently low to clearly distinguish the mean radial trend ($\Delta r_v^* \approx 0.26$) from measurement uncertainty.

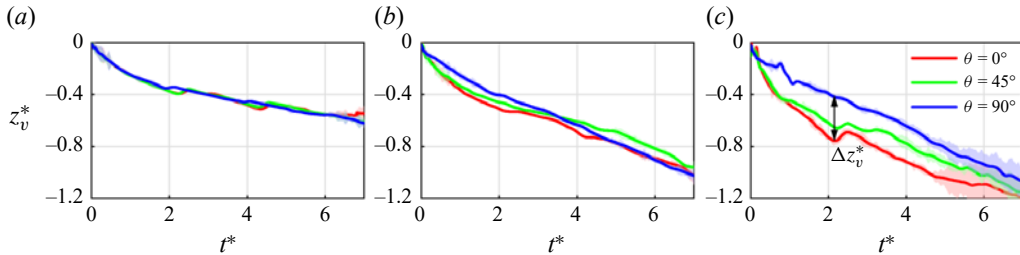


Figure 16. Comparison of the axial vorticity distribution index z_v^* over non-dimensional time t^* in the chordwise (red line), diagonal (green line) and spanwise (blue line) cross-sections for plates with three different non-dimensional bending stiffness: (a) 'R-5', (b) 'F-7' and (c) 'EF-6'.

shows distinct variations among them. As shown in figure 15(b), in the chordwise cross-section, r_v^* remains nearly constant, whereas in the diagonal and spanwise cross-sections, it first decreases and then increases. The maximum difference Δr_v^* occurs between the chordwise and spanwise cross-sections, reaching approximately 0.12 around $t^* = 3$. A similar trend is observed in the axial vorticity index (figure 16b), where the largest discrepancy Δz_v^* appears around $t^* = 1.5$ between the chordwise and spanwise cross-sections. These variations highlight the non-axisymmetry in the wake of the flexible plate.

Compared with the flexible plate, the extra-flexible plate 'EF6' exhibits even stronger non-axisymmetry, with Δr_v^* exceeding 0.15 and Δz_v^* exceeding 0.32 across different cross-sections. As shown in figure 15, in the chordwise cross-section, the radial vorticity index r_v^* rapidly decreases to approximately $r_v^* = 0.26$ after the plate's initial acceleration phase, followed by a secondary increase due to outward vorticity diffusion (as shown in the Supplementary materials), reaching $r_v^* \approx 0.5$ at $t^* = 7$. In contrast, in the diagonal and spanwise cross-sections, r_v^* decreases after acceleration and remains around 0.3. For the axial vorticity distribution (figure 16), the maximum deviation across chordwise and spanwise cross-section reaches 0.32 at approximately $t^* = 2$, significantly exceeding the values observed for the flexible plate and indicating stronger wake non-axisymmetry.

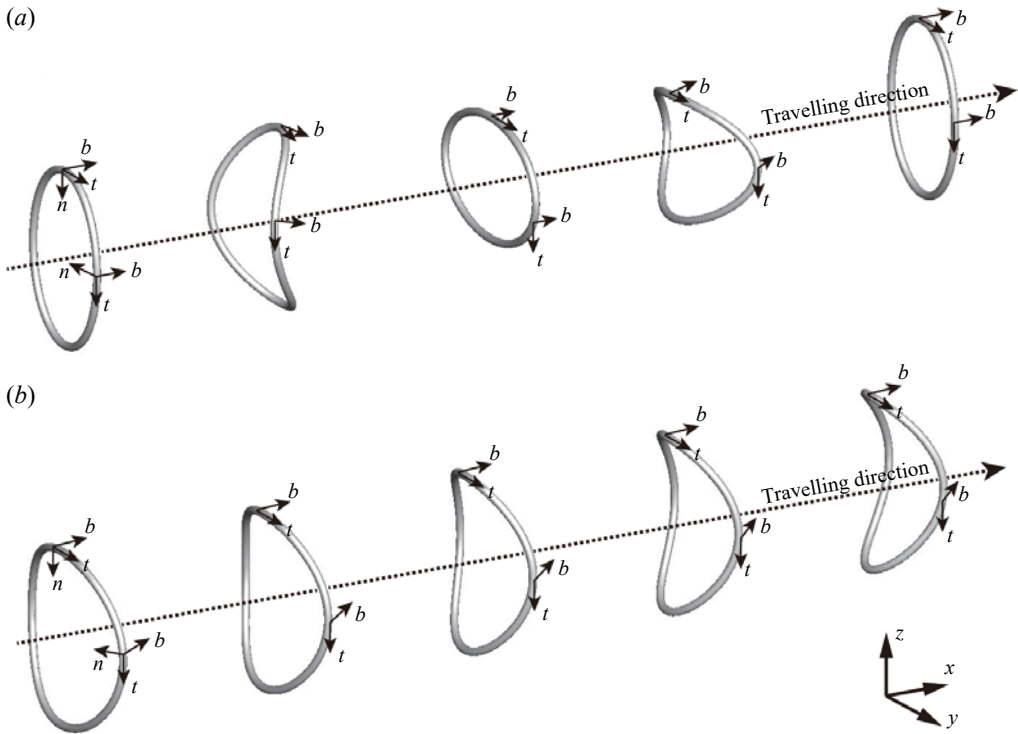


Figure 17. Schematic of the evolution of the jet elliptic vortex ring (a) and the vortex ring behind the flexible plate (b). The schematic in panel (b) is a data-driven visualization of the vortex ring's 3-D evolution. Its shape and radial contraction are quantitatively supported by the index in [figure 15](#), while its axial bending and lag are supported by the index in [figure 16](#).

Beyond non-axisymmetry, the initial decrease and subsequent increase in the radial vorticity index for the flexible plates warrant further attention. This flow phenomenon is particularly relevant for the load-shifting force phenomenon of the flexible plate, given its circulation is comparable to that of the rigid plate ([figure 14](#)). The redistribution of vorticity in the radial direction plays a crucial role in determining the force contribution of the wake, as further discussed in § 5.3.

Additionally, the absolute slope of the axial vorticity index curve is significantly larger for the flexible plate than for the rigid plate ([figure 16](#)), indicating enhanced downstream vorticity transport. This may be attributed to vortex breakdown, which facilitates vorticity propagation downstream, unlike the rigid plate, where a single coherent vortex structure persists. For the extra-flexible plate, vortex shedding resembles aerofoil wake dynamics, resulting in even stronger downstream vorticity transport.

4.3. Vortex ring non-axisymmetric evolution and vortex breakdown

Based on the previous discussion, driven by the reconfiguration of the plate, the evolution of the wake vortices behind the flexible plate can be divided into three stages.

The first stage involves the growth and evolution of a single non-axisymmetric vortex ring, as illustrated in [figure 17\(b\)](#). In this phase, the vorticity distribution index can serve as an indicator of the vortex core position. Morphologically, this vortex ring exhibits distinct characteristics. Radially, the vortex ring radius in the chordwise cross-section exceeds that in the spanwise cross-section, forming an elliptical shape when projected onto the yz -plane, with its major axis in the chordwise cross-section and minor axis in the spanwise

cross-section, as indicated in figure 15(b). Axially, the vortex core in the chordwise cross-section lags behind that in the spanwise cross-section, leading to a 3-D deformation that mirrors the flexible plate's shape, as quantified by figure 16(b). As t^* increases, the major axis radius remains relatively constant, whereas the minor axis radius gradually decreases, resulting in a continuous increase in the vortex ring's aspect ratio. Additionally, the axial displacement Δz_r^* between the vortex cores in the spanwise and chordwise cross-sections grows, indicating a progressive enhancement of the vortex ring's streamwise curvature.

The second stage is the transition phase. During this period, the single vortex ring structure begins to break down (§ 4.1). In the chordwise cross-section, a second shedding vortex emerges and propagates downstream. Meanwhile, in the spanwise cross-section, the vortex cores approach each other, leading to the annihilation of opposite-signed vorticity, which amplifies disturbances and eventually triggers vortex breakdown.

The third stage is the vortex breakdown phase. At this point, the vortex transitions from a clustered distribution to a dispersed distribution (§ 4.1). This dispersed distribution enhances fluid momentum and energy exchange while also promoting vorticity diffusion. Consequently, in the diagonal and spanwise cross-sections, the vorticity, which was previously concentrated near the symmetry axis due to vortex induction, spreads outward into low-vorticity regions. This redistribution leads to an increase in the radial vorticity index r_v^* for $t^* > 3$, by which time vortex breakdown has already occurred. Note that in this study, 'vortex breakdown' denotes the rapid disintegration of a coherent vortex ring into a dispersed turbulent state. This usage emphasizes the abrupt loss of vortex coherence, distinct from the classical bubble-type or spiral breakdown observed in strongly swirling flows (Leibovich 1978; Lucca-Negro & O'Doherty 2001).

Having outlined the temporal evolution, the following analysis deconstructs the two most critical features of this process: the non-axisymmetric evolution of the vortex ring and its subsequent breakdown. The former is analysed through the principle of vortex self-induction (§ 4.3.1), whereas the latter is characterized using objective, statistical metrics of flow instability (§ 4.3.2).

4.3.1. Vortex ring non-axisymmetric evolution

Given that the projection area of the flexible plate changes slightly (figure 7b), where the dimensionless projected area remains nearly 1, the observed non-axisymmetric vortex ring growth is thus primarily attributed to the 3-D reconfiguration of the plate rather than variations in its projected area. As the plate deforms into a 3-D shape, the shed vortex ring inherits this curvature. This process can be better understood using the Biot–Savart law. According to the work of Wu (2021), the self-induced velocity of a vortex filament in the binormal direction $\mathbf{b}$ is given by

$$\mathbf{v}_b = \frac{\Gamma_f \kappa_f}{4\pi} \mathbf{b} \ln \frac{L_f}{\epsilon_f}, \quad (4.5)$$

where Γ_f , κ_f , L_f and ϵ_f represent the circulation, curvature, length and vortex core radius of the filament segment, respectively. For a rigid plate, the vortex properties are uniform along the azimuthal direction. Consequently, during its evolution, the vortex ring retains a circular and axisymmetric shape as shown in the Supplementary materials.

In contrast, for flexible plates, plate bending induces 3-D features in the vortex ring. As illustrated in figure 17(b), the vortex ring bends downstream following the plate's curvature. According to (4.5), the binormal direction $\mathbf{b}$ in the chordwise cross-section points away from the vortex ring symmetry axis, whereas in the spanwise cross-section, it points towards the symmetry axis. Consequently, under the self-induced effects of curved

vortex filaments, the chordwise-axis segments expand outward while the spanwise-axis segments contract inward. Moreover, throughout the evolution, the binormal direction $\mathbf{b}$ in both the chordwise and spanwise cross-sections remains relatively unchanged, causing a sustained outward motion of the vortex filaments in the chordwise cross-section and an inward contraction in the spanwise cross-section.

A comparison with non-axisymmetric jet vortex rings provides further insight into this process. Unlike plate-induced vortex rings, non-axisymmetric jet vortex rings exhibit the well-known axis-switching phenomenon (Cheng, Lou & Lim 2016; Fernando & Rival 2016a), where the major and minor axes periodically exchange, as shown in figure 17(a). The binormal direction $\mathbf{b}$ continuously alternates its orientation relative to the symmetry axis, causing different regions of the filament to expand/contract and accelerate/decelerate in succession. The vortex filament segments with the highest curvature near the major axis propagate faster than the rest of the ring. This forward propagation reduces their curvature while increasing that of the slower-moving regions, eventually leading to a role reversal between the major and minor axes. This process repeats cyclically. In contrast, for flexible plates, axis-switching does not occur – the vortex ring radius in the spanwise plane remains consistently smaller than in the chordwise plane.

4.3.2. Vortex breakdown

To understand the mechanism behind the vortex breakdown observed in the flexible plate's wake, we begin from the premise that the breakdown is triggered by the growth of a 3-D fluid instability (Saffman 1995; Wu *et al.* 2007). A primary mechanism for the amplification of such instabilities in a vortex is the intense, localized stretching of vortex lines (Adrian, Meinhart & Tomkins 2000; Kundu *et al.* 2024). For an incompressible flow, the intensity of out-of-plane vortex-line stretching can be directly measured from 2-D planar PIV data via the in-plane divergence (Kaiser *et al.* 2020). We therefore utilize the normalized cylindrical divergence, ζ , to evaluate the local intensity of this stretching relative to the in-plane fluid deformation (shear and strain) (Reijtenbagh *et al.* 2023b). A detailed derivation and definition of ζ is provided in Appendix B1.

The instantaneous fields of normalized divergence ζ and velocity vectors are compared in the chordwise cross-section ($\theta = 0^\circ$) in figure 18. For the rigid plate 'R-5' (figure 18a–c), the ζ field remains weak and the corresponding velocity vectors illustrate a smooth, stable flow pattern, confirming the wake's axisymmetry. In contrast, the flexible plate 'F-7' (figure 18d–f) exhibits a dramatic evolution. At $t^* = 4$, strong, spatially organized regions of positive and negative ζ emerge, coinciding with the onset of vortex breakdown. The velocity vectors in these regions reveal local sources (diverging vectors), which are unambiguous kinematic signatures of the out-of-plane fluid motion associated with intense vortex stretching. For the extra-flexible plate 'EF-6' (figure 18g–i), the wake shows 3-D shear-layer characteristics from the outset; its ζ field is strong early on but lacks the organized structures that signify the single catastrophic breakdown event seen in the flexible case.

Although the wake is non-axisymmetric, these qualitative characteristics are consistent across the different measurement planes. The flow fields in the diagonal ($\theta = 45^\circ$, figure 19) and spanwise ($\theta = 90^\circ$, figure 20) cross-sections likewise show that the rigid plate remains stable, the flexible plate undergoes a distinct transition to a 3-D breakdown and the extra-flexible plate is inherently 3-D throughout its evolution.

To statistically quantify the overall evolution of these 3-D effects, the root-mean-square (r.m.s.) of the ζ field, ζ_{rms} , is calculated over the physically relevant regions of the wake (see Appendix B2), as shown in figure 21. The ζ_{rms} for the rigid plate remains at a low, baseline level, indicating persistent flow stability. A slight increase is observed around

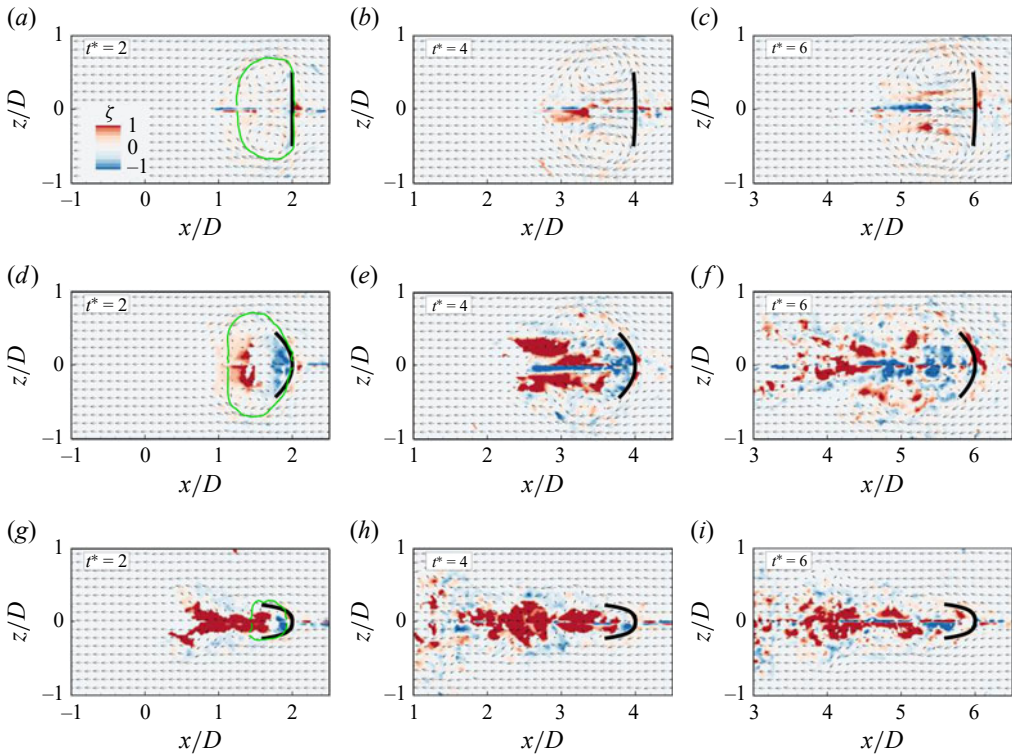


Figure 18. Instantaneous fields of the normalized divergence ζ (colour contours) and velocity vectors in the chordwise ($\theta = 0^\circ$) cross-section for plates with three different non-dimensional bending stiffness: (a–c) for ‘R-5’, (d–f) for ‘F-7’ and (g–i) for ‘EF-6’. The velocity vectors represent the in-plane components ($u'_{z'}, u'_{r'}$) in the moving frame of reference. Here $u'_{z'} = u_{z'} - U(t)$ is the axial velocity component, where $u_{z'}$ is the instantaneous axial flow velocity in the earth frame and $U(t)$ is the instantaneous velocity of the plate. The area enclosed by the green line denotes the region where $\psi_0 < 0$ within the mask $\mathcal{H}_{mask}$ (Appendix B.2).

$t^* \approx 5$, which may be attributed to vortex ring transition to turbulence in the later stage of the plate motion (Li *et al.* 2022; Reijtenbagh *et al.* 2023b). In contrast, the flexible plate’s ζ_{rms} exhibits a distinct change in slope and a rapid increase starting around $t^* \approx 3$, providing a clear quantitative signature for the transition of the wake from a quasi-2-D state to a fully 3-D, fragmented state. The ζ_{rms} for the extra-flexible plate is moderately elevated from the beginning and grows continuously, consistent with its non-transient, 3-D wake. These results lead to the conclusion that the plate’s passive reconfiguration is the root cause of the vortex breakdown. For the flexible plate, plate reconfiguration induces a non-axisymmetric vortex ring (§ 4.3.1) that, under its own self-induced velocity, develops localized regions of intense vortex stretching (high $|\zeta|$), ultimately triggering the catastrophic vortex breakdown. The rigid plate avoids this instability due to its axisymmetry, while the extra-flexible plate’s wake is inherently 3-D and does not undergo a distinct breakdown of a coherent structure.

5. Mechanism for the load-shifting phenomenon

5.1. Force estimation based on the vorticity moment theorem

As discussed in § 1, while the classical quasisteady approach, which decomposes the force into quasisteady, added-mass and history components, is powerful, its history force

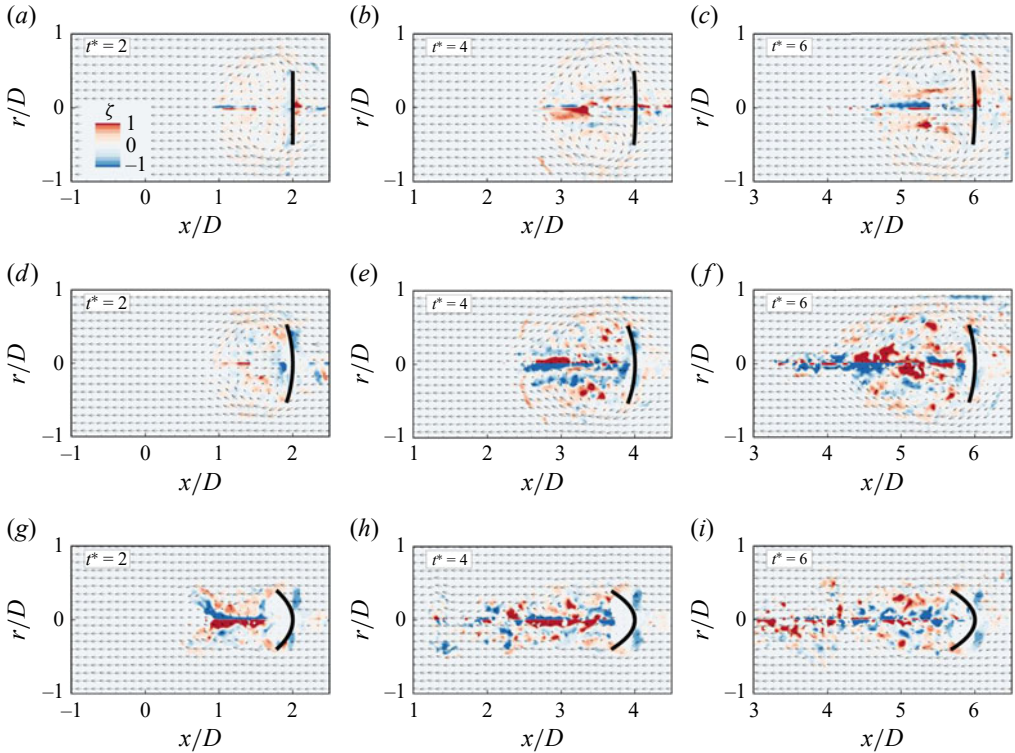


Figure 19. Instantaneous fields of the normalized divergence ζ (colour contours) and velocity vectors in the diagonal ($\theta = 45^\circ$) cross-section for plates with three different non-dimensional bending stiffness: (a–c) for ‘R-5’, (d–f) for ‘F-7’ and (g–i) for ‘EF-6’. Annotations are consistent with those in figure 18.

term is challenging to interpret physically for the complex, large-scale separated flows (Fernando & Rival 2016a,b; Li *et al.* 2022; Liu & Sun 2024). Therefore, to achieve this study’s primary objective of directly linking force generation to the observed wake dynamics, the impulse-based framework is adopted. Rooted in the vorticity moment theorem (Wu 1981), this approach is ideal as it directly connects the hydrodynamic force to the evolution of the vorticity field (Kim & Gharib 2011a,b; Kaiser *et al.* 2020; Li *et al.* 2022).

According to the vorticity moment theorem (Wu 1981), for a 3-D incompressible and viscous flow in an infinite fluid field, which is at rest at infinity, the force on the body can be decomposed into two parts,

$$\mathbf{F} = \mathbf{F}_I + \mathbf{F}_b, \quad (5.1)$$

where $\mathbf{F}_I$ represents the impulsive force and $\mathbf{F}_b$ represents the buoyancy force.

The second term $\mathbf{F}_b$ in (5.1) is first discussed and defined as

$$\mathbf{F}_b = \rho_f \frac{d}{dt} \int_{V_b} \mathbf{u} dV, \quad (5.2)$$

where $\mathbf{u}$ is the velocity of the object, and V_b is the volume occupied by the body. According to the kinematic set-up, the plate translates in one direction, and this study focuses on the force in the x -direction. Thus, $\mathbf{F}_b$ can be rewritten as F_b , representing the force in the x -direction only. As reference in figure 3, the velocity at the tip and the centre of the plate is nearly identical, suggesting that the velocity at all points on the plate can be considered

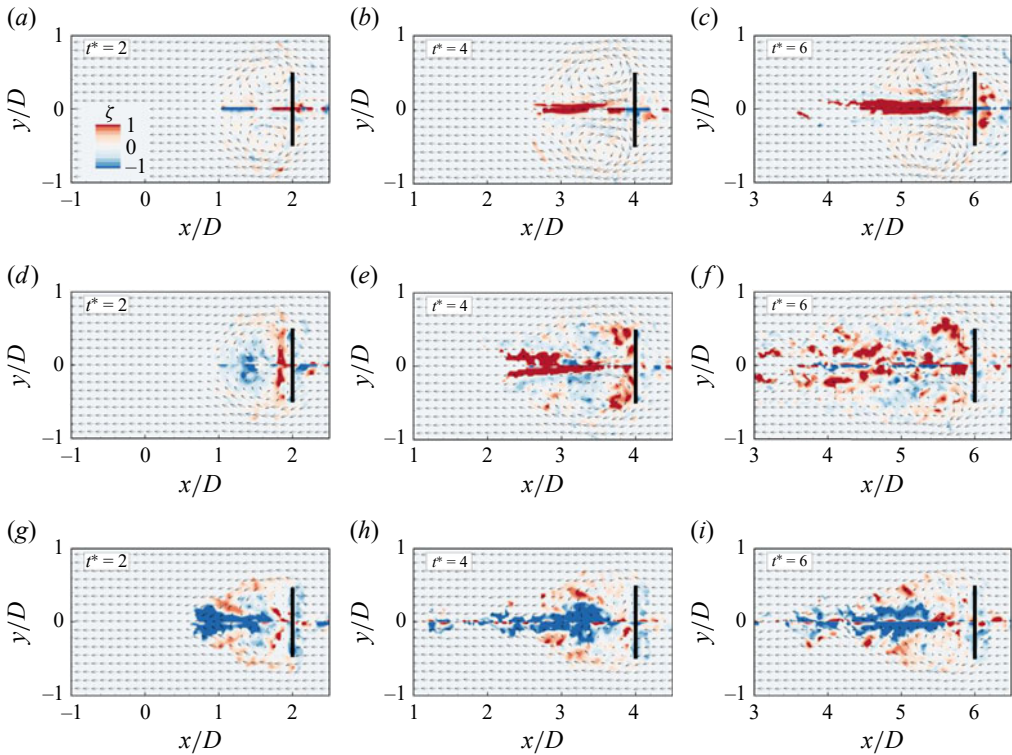


Figure 20. Instantaneous fields of the normalized divergence ζ (colour contours) and velocity vectors in the spanwise ($\theta = 90^\circ$) cross-section for plates with three different non-dimensional bending stiffness: (a–c) for ‘R-5’, (d–f) for ‘F-7’ and (g–i) for ‘EF-6’. Annotations are consistent with those in figure 18.

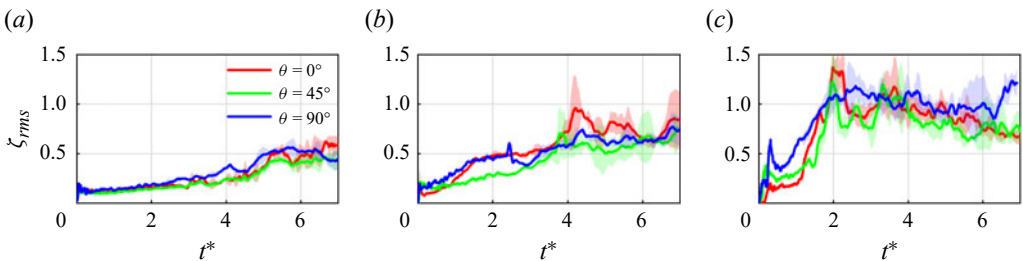


Figure 21. Comparison of the r.m.s. of the normalized divergence ζ_{rms} over non-dimensional time t^* in the chordwise (red line), diagonal (green line) and spanwise (blue line) cross-sections for plates with three different non-dimensional bending stiffness: (a) ‘R-5’, (b) ‘F-7’ and (c) ‘EF-6’.

uniform. And although the shape of the plate changes, its volume remains constant, so $V_b = Sh$, where S is the area of the plate and h is its thickness. The force F_b can then be expressed as

$$F_b = \rho_f \frac{dU(t)}{dt} V_b = \rho_f a(t) Sh, \quad (5.3)$$

and the corresponding force coefficient is

$$C_b = \frac{F_b}{\frac{1}{2} \rho U_c^2 S}. \quad (5.4)$$

It is noteworthy that C_b only exists during the acceleration phase. In the steady translation phase, where there is no acceleration, F_b becomes zero. For the acceleration phase, substituting further, the final expression can be derived as

$$C_b = 2a^* \frac{h}{D}. \quad (5.5)$$

In this study, $a^* = 0.25$ is a constant, and the maximum value of h is 3.00 mm. Consequently, $C_b \leq 0.015$.

The first term F_I in (5.1) is calculated as

$$\mathbf{F}_I = -\frac{\rho_f}{2} \frac{d\mathbf{I}}{dt}, \quad (5.6)$$

where,

$$\mathbf{I} = \int_{V_\infty} \mathbf{x} \times \boldsymbol{\omega} dV, \quad (5.7)$$

is the first moment of the combined fluid-body vorticity field. Here, $\mathbf{x}$ is the position vector, $\boldsymbol{\omega}$ is the vorticity and $V_\infty = V_f + V_b$ is the volume occupied by the fluid and the body. For the translating plate without rotation in this study, the integration of V_b vanishes (Graham 2019), leading to

$$\mathbf{I} = \int_{V_f} \mathbf{x} \times \boldsymbol{\omega} dV. \quad (5.8)$$

As a widely accepted theory, any object moving in the real viscous fluid can be represented by a vortex sheet located on its surface, which models the viscous boundary layer (Saffman 1995). Referring to Corkery (2019) and Gehlert (2021), the complex viscous flow can be represented as the vortex sheet in the potential flow by superposing two independent components. One contribution arises from the inviscid flow around the plate during translation at an angle of incidence of 90° , while the other contribution comes from the shed vorticity behind the plate. The former generates an added mass vortex sheet, while the latter induces a circulatory vortex sheet. Thus, the decomposition of (5.8) aligns with the physical interpretation of the vortex sheet,

$$\mathbf{I} = \mathbf{I}_a + \mathbf{I}_v, \quad (5.9)$$

where, as this study focuses on the force in the x -direction, $\mathbf{I}$ can be rewritten as I , representing the force in the x -direction only; I_a corresponds to the impulse due to the vortex sheet associated with the plate translation, and I_v corresponds to the impulse due to shed vorticity.

According to the results of Corkery, Babinsky & Graham (2019), I_a has the same form in both potential and viscous fluids. Based on this consistency, the added mass model in potential flow can be directly utilized,

$$C_a \text{ in viscous flow} = C_a \text{ in potential flow}. \quad (5.10)$$

Referring to Brennen (1982), this paper approximates the added mass force of the reconfigurable plate using the added mass force of an elliptical plate. The specific calculation of C_a is detailed in Appendix C.

The calculation of I_v requires consideration of the shed vorticity in the fluid. In cylindrical coordinates,

$$I_v = \frac{\rho}{2} \int_{V_f} r' \omega_\theta dV, \quad (5.11)$$

where ω_θ is the vorticity in the azimuthal direction, and V_f represents the wake flow where vorticity is shed. Expanding the integration in cylindrical coordinates,

$$I_v = \frac{\rho}{2} \frac{d}{dt} \int_{z'_{-\infty}}^{z'_{+\infty}} \int_0^{2\pi} \int_0^{r'_{\infty}} r'^2 \omega_\theta dr' d\theta dz', \quad (5.12)$$

where z' represents the axial direction in cylindrical coordinates, corresponding to the x -axis in the Cartesian coordinate system (see [figure 1](#)). For axisymmetric vortex rings, where ω_θ is constant along the azimuth, integrating over θ yields

$$I_v = \rho\pi \frac{d}{dt} \int_{z'_{-\infty}}^{z'_{+\infty}} \int_0^{r'_{\infty}} r'^2 \omega_\theta dr' dz'. \quad (5.13)$$

In this experiment, due to the non-axisymmetric evolution of vortex rings, the assumption that the circulation ω_θ remains constant along the circumferential direction does not hold. In practice, the impulse can be calculated by averaging the data across multiple cross-sections passing through the centre, as follows:

$$I_v \approx \frac{1}{N} \sum_{i=1}^N \rho\pi \frac{d}{dt} \int_{z'_{-\infty}}^{z'_{+\infty}} \int_0^{r'_{\infty}} r'^2 \omega_{\theta,i} dr' dz', \quad (5.14)$$

where i represents the i th cross-section, N is the total number of cross-sections, and a higher number of cross-sections improves the accuracy of the impulse calculation. In this study, the global impulse is estimated by averaging data from three representative cross-sections (chordwise, diagonal and spanwise). For a sensitivity analysis regarding the selection of measurement planes, please refer to [Appendix A4](#). The integration domain in each cross-section is identical to that defined in (4.2). Based on this, the force coefficient associated with the shed vorticity can be calculated as

$$C_v = \frac{1}{q_\infty S} \frac{dI_v}{dt}. \quad (5.15)$$

In summary, the total force coefficient can be expressed as

$$C_{H,impulse} = C_v + C_a + C_b. \quad (5.16)$$

As shown in [figure 22](#), the impulse-based calculation ($C_{H,impulse}$) exhibits strong quantitative agreement with the transducer measurement ($C_{H,transducer}$). For the rigid and flexible plates, the Pearson correlation coefficient (PCC) exceeds 0.94, confirming that the $C_{H,impulse}$ accurately captures the temporal evolution trends. Crucially, the relative r.m.s. (RRMSE) within the peak force remains below 6% for these cases. This high fidelity ensures that the load-shifting characteristics – specifically the reduced peak at $t^* = 2$ and the elevated valley at $t^* = 5.5$ for the flexible plate – are physically resolved. Notably, even for the extra-flexible plate where 3-D asymmetry is intensified, the peak load is captured with a relative error of less than 8%. This statistical robustness validates the vorticity moment theorem as a reliable framework for force analysis.

[Figure 23](#) further illustrates the force decomposition results based on vorticity moment theorem. As shown in the figure, C_v dominates the overall contribution, regardless of whether the plate is rigid, flexible or extra-flexible. Given the strong correlation between C_v and the growth and evolution of the vortex ring, these results indicate that, under the parameter set-up in this study, the vorticity evolution serves as the key mechanism driving the force response.

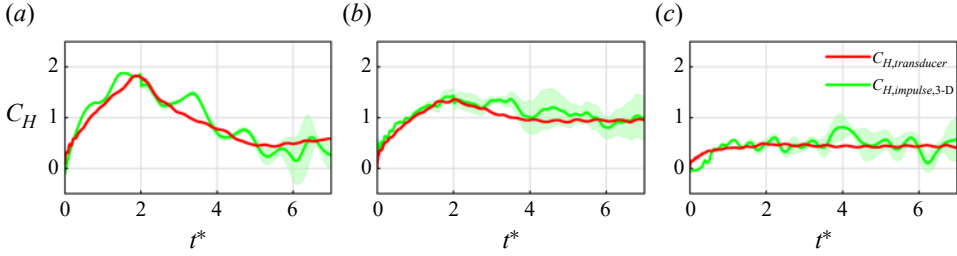


Figure 22. Comparison between $C_{H,transducer}$ measured by the force transducer (red line) and $C_{H,impulse}$ based on the vorticity moment theorem (green line) for three representative cases: (a) 'R-5', (b) 'F-7' and (c) 'EF-6'.

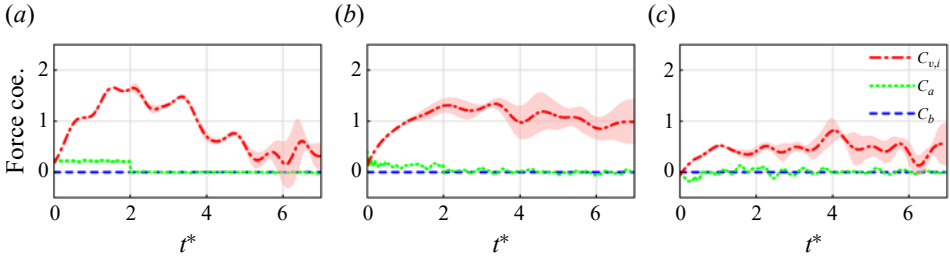


Figure 23. Force decomposition based on the vorticity moment theorem, showing $C_{v,i}$ (dash-dotted line), C_a (dotted line) and C_b (dashed line) for three different non-dimensional bending stiffness values: (a) 'R-5', (b) 'F-7' and (c) 'EF-6'. The force decomposition corresponds to $C_{H,impulse}$ in figure 22.

5.2. Low-order force model for impulse force from shed vorticity

To analyse the dominant force from shed vorticity, we employ a low-order model that treats the wake as a thin, closed vortex loop (Wu *et al.* 2007; Kim & Gharib 2011b; Li *et al.* 2022). This allows the impulse of the shed vorticity (I_v) to be rewritten as

$$I_v \approx \rho \Gamma_v S_v, \quad (5.17)$$

where the circulation Γ_v and the vortex loop area S_v are calculated as

$$\Gamma_v = \frac{1}{N} \sum_{i=1}^N \Gamma_i, \quad (5.18)$$

and

$$S_v = \frac{1}{N} \pi \sum_{i=1}^N r_i^2, \quad (5.19)$$

where, Γ_i and r_i are the circulation and radial vorticity distribution in the i th cross-section, respectively. Similar to the calculations in (5.14), increasing the number of selected cross-sections improves the accuracy of the results. In this study, flow field data from chordwise, diagonal and spanwise cross-sections, are used to compute the results. The normalized values of Γ_i and r_i for these three cross-sections are shown in figures 14 and 15. By differentiating (5.17), the force coefficient associated with the time derivative of circulation is expressed as

$$C_{v,\Gamma_v} = \frac{1}{\frac{1}{2} U_c^2 S} S_v \frac{d\Gamma_v}{dt}, \quad (5.20)$$

and the force coefficient associated with the time derivative of vortex loop area is

$$C_{v,S_v} = \frac{1}{\frac{1}{2}U_c^2 S} \Gamma_v \frac{dS_v}{dt}. \quad (5.21)$$

Based on these, the low-order force model for impulse force from shed vorticity can be expressed as

$$C_{v,m} = C_{v,\Gamma_v} + C_{v,S_v}, \quad (5.22)$$

where $C_{v,m}$ represents the force coefficient obtained from the low-order model. The predicted results of $C_{v,m}$ are presented in [figure 24\(a–c\)](#). Quantitative analysis reveals that for the rigid and flexible plates, $C_{v,m}$ exhibits strong alignment with $C_{v,i}$, characterized by PCC > 0.75 and RRMSE < 20 %. This statistical agreement confirms that the low-order model effectively captures the key force characteristics in regimes governed by coherent vortex dynamics. More error analysis can be found in [Appendix D](#).

[Figure 24\(d–f\)](#) further shows the decomposition results of $C_{v,m}$ for different EI^* plates.

For rigid plate ([figure 24d](#)), the contribution of C_{v,Γ_v} is dominant, while that of C_{v,S_v} is secondary. The contribution from C_{v,S_v} , which is proportional to the rate of change of the vortex area (dS_v/dt) as defined in (5.21), remains negligible because the radial vorticity distribution stays nearly constant during the motion ([figure 15](#)), leading to an area change rate close to zero. In contrast, C_{v,Γ_v} governs the overall force evolution, producing the characteristic peak around $t^* = 2$ and the subsequent trough near $t^* \approx 5.5$. This is because C_{v,Γ_v} is proportional to the circulation growth rate ($d\Gamma_v/dt$) as shown in (5.20), and the circulation undergoes a clear and significant increase over time ([figure 14](#)). Therefore, for the rigid plate, the force variation is almost entirely driven by the evolution of vortex circulation – an observation consistent with previous studies (Wu *et al.* 2007; Kim & Gharib 2011b; Li *et al.* 2022).

For the flexible plate ([figure 24e](#)), the contribution of C_{v,Γ_v} decreases during the acceleration phase, while C_{v,S_v} exhibits a negative value in this phase and significantly increases after the acceleration ends. These changes lead to the load-shifting phenomenon observed in the force evolution of the flexible plate, which will be further discussed in § 5.3.

The absence of load-shifting phenomenon for extra-flexible plates can be well explained using [figure 24\(f\)](#). First, the contributions of C_{v,Γ_v} and C_{v,S_v} for the extra-flexible plate are relatively small, leading to a smaller overall force. Second, C_{v,Γ_v} and C_{v,S_v} complement each other, resulting in a relatively stable force evolution without distinct peaks and troughs.

5.3. Mechanism for the load-shifting phenomenon

The low-order force model accurately predicts the impulse force generated by shed vorticity. By decomposing the impulse force into its constituent components ((5.20) and (5.21)) and synthesizing these with the evolution of Γ_v^* ([figure 14](#)) and r_v^* ([figure 15](#)), the model provides a mechanistic explanation for the load-shifting phenomenon.

The load-shifting phenomenon is characterized by two distinct features when comparing the flexible plate to the rigid one: a reduced force peak at the end of acceleration ($t^* = 2$) and an elevated force valley in the postacceleration phase ($t^* \approx 5.5$) ([figures 7 and 8](#)). The analysis begins with the full decomposition of the total hydrodynamic force into components from shed vorticity (C_v), added mass (C_a) and buoyancy (C_b) (5.16). As shown in [figure 23](#), the force from shed vorticity, C_v , is the overwhelmingly dominant component. Therefore, to understand the mechanism, the analysis must focus on the

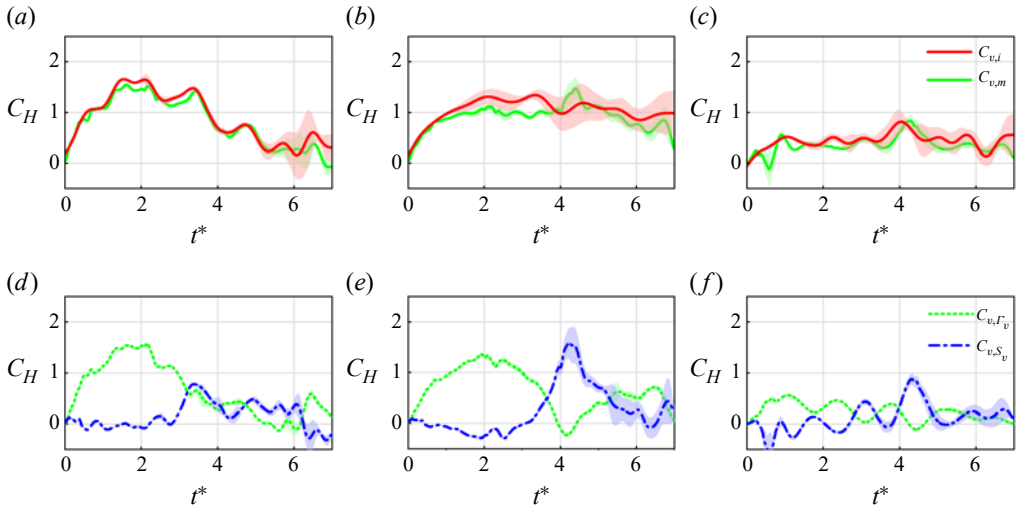


Figure 24. Comparison between the impulse-based coefficient $C_{v,i}$ (red solid line) and the model-based coefficient $C_{v,m}$ (green solid line) for three representative cases with varying non-dimensional bending stiffness: (a,d) ‘R-5’, (b,e) ‘F-7’ and (c,f) ‘EF-6’. The values of $C_{v,i}$ shown here correspond to those presented in figure 23. Panels (d)–(f) further decompose $C_{v,m}$ into its two components: C_{v,Γ_v} (green dotted line) and C_{v,S_v} (blue dash-dotted line). Quantitative metrics highlight the regime transition for panels (a)–(c): the rigid case shows an excellent fit (PCC = 0.98, RRMSE < 10 %); the flexible case maintains reasonable agreement (PCC = 0.75, RRMSE < 20 %); whereas the extra-flexible case exhibits deviation (PCC = 0.68, RRMSE < 36 %).

composition of C_v . To this end, the low-order model is employed (5.22) to further decompose C_v into two terms: C_{v,Γ_v} (5.20) and C_{v,S_v} (5.21).

First, the mechanism behind the reduced peak force of the flexible plate is examined. At the force peak ($t^* = 2$), figure 24 shows that both C_{v,Γ_v} and C_{v,S_v} are smaller than those of the rigid plate, with C_{v,S_v} even exhibiting a negative value. According to (5.20), the force component C_{v,Γ_v} is governed by the product of the vortex loop area S_v and the circulation growth rate $d\Gamma_v^*/dt$. Although the flexible plate has a slightly higher circulation growth rate (figure 14), its C_{v,Γ_v} is smaller than that of the rigid plate (figure 24e versus figure 24d), due to a significantly smaller vortex loop area S_v , which scales with $(r_v^*)^2$ (figure 15). Similarly, the negative value of C_{v,S_v} can be explained by the definition in (5.21). With the flexible plate having larger circulation (figure 14), the observed negative C_{v,S_v} must result from a decreasing radial vorticity distribution, i.e. $dr_v^*/dt < 0$ (figure 15). Together, these results demonstrate that the smaller radial vorticity distribution r_v^* and its negative temporal change dominate the reduction in peak force. Therefore, the radial vorticity distribution is the key mechanism responsible for the reduced peak force.

Next, the mechanism behind the elevated force valley in the flexible plate is analysed. As shown in figure 24, this postpeak phase ($t^* \approx 5.5$) is dominated by the C_{v,S_v} component, which is large and positive for the flexible plate but near-zero for the rigid plate; in contrast, the C_{v,Γ_v} component is small for both cases at this time. According to (5.21), as the total circulation, Γ_v^* , is comparable for both plates (figure 14), the larger C_{v,S_v} in the flexible case must be caused by a greater vortex area growth rate (dS_v/dt). Given that the vortex area, S_v , scales with $(r_v^*)^2$, this larger dS_v/dt directly reflects an outward movement of the vortex ring ($dr_v^*/dt > 0$), a trend confirmed by the vorticity distribution data in figure 15. Thus, the radial redistribution of vorticity is also the key mechanism responsible for elevating the force in the postpeak valley.

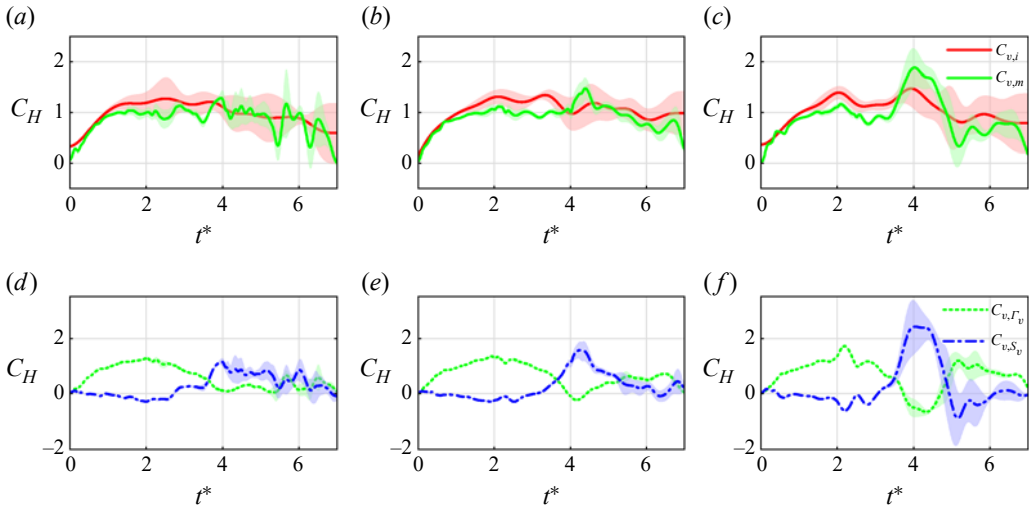


Figure 25. Comparison (a–c) between $C_{v,i}$ based on the vorticity moment theorem (red line) and $C_{v,m}$ based on the low-order model (green line), and the decomposition (d–f) of $C_{v,m}$, C_{v,Γ_v} (green dotted line) and C_{v,S_v} (blue dash-dotted line) with non-dimensional bending stiffness $EI^* = 1.18 \times 10^{-2}$ but three different Reynolds numbers: (a,d) ‘F-3’, (b,e) ‘F-7’ and (c,f) ‘F-10’.

In summary, the load-shifting phenomenon – manifested as a reduced peak force at $t^* = 2$ and an elevated force valley at $t^* = 5.5$ in the flexible plate – is primarily driven by the radial vorticity distribution (r_v^*). While the circulation Γ_v^* directly contributes to the overall force magnitude and dominates the formation of the peak at $t^* = 2$, it does not explain the difference between the flexible and rigid plates. Furthermore, based on the consistency of the vorticity dispersion length across different plates shown in [Appendix D](#), we also rule out the potential impact of vorticity dispersion on load-shifting. Combining the discussion of § 4.3, this analysis further highlights the critical role of structural reconfiguration in shaping the vortex-force interaction through radial redistribution.

Figure 25 further illustrates the force decomposition results for varying Reynolds numbers at the fixed $EI^* = 1.16 \times 10^{-2}$. Although the overall magnitude of $C_{v,m}$ remains similar, the relative contributions of C_{v,Γ_v} and C_{v,S_v} vary. As the Reynolds number increases, C_{v,Γ_v} decreases more significantly around $t^* = 5.5$, whereas C_{v,S_v} becomes more dominant. As discussed in the Supplementary materials, this trend likely stems from enhanced vortex breakdown at higher Reynolds numbers, which promotes stronger radial vorticity redistribution and further amplifies the load-shifting behaviour in flexible plates.

6. Conclusion

To understand the adaptive strategies in biological propulsion and disturbance-resistance, this study investigates the accelerating reconfigurable circular plate as a canonical model problem. The terms accelerating and reconfigurable model the unsteady motion and shape adaptation of biological propulsors, respectively, while the circular geometry simplifies the analysis without sacrificing key physical insights. Using force measurements and time-resolved PIV, the problem was investigated at a fixed acceleration number ($a^* = 0.25$). The study systematically explored a wide parameter space, including Reynolds numbers Re from 3.12×10^4 to 10.8×10^4 and a dimensional stiffness range of $EI \in [4.19 \times 10^{-4}, 26.8] \text{ Nm}$, which corresponds to non-dimensional bending stiffnesses EI^* from 4.19×10^{-4} to 36.2 ([tables 1, 2 and 3](#)).

The force evolution of an accelerating reconfigurable plate is characterized by two primary findings: its independence on the non-dimensional bending stiffness, EI^* , and the emergence of a load-shifting phenomenon. Unlike rigid plates, the force on reconfigurable plates varies with Re (figure 6) but collapses for a given EI^* , confirming dimensionless stiffness as the governing parameter (figure 7). The analysis also reveals a load-shifting phenomenon that smooths the characteristic ‘peak-valley’ force profile of a rigid body by reducing the peak and elevating the trough (figure 8). This behaviour is unique to the intermediate ‘flexible’ regime ($2.28 \times 10^{-3} \leq EI^* \leq 0.143$), offering a mechanism to soften force generation without compromising mean force (figure 9).

The flow evolution behind flexible plates is characterized by two dominant features: the formation of a non-axisymmetric vortex ring and its subsequent breakdown. The plate’s reconfiguration induces a curved, 3-D vortex ring whose evolution is governed by self-induction, resulting in a stable non-axisymmetric shape distinct from the axis-switching phenomenon of jet-generated rings (figure 17). This unique evolution further triggers the 3-D instability where self-induced motion leads to intense vortex-line stretching, culminating in the catastrophic breakdown of the coherent vortex structure (figures 18, 19 and 20).

A systematic analysis of the wake dynamics reveals that the load-shifting phenomenon is governed by the radial distribution of vorticity (r_v^*). This statement is reached through a three-step analysis. First, force decomposition via the vorticity moment theorem shows that the hydrodynamic force is overwhelmingly dominated by the impulse from shed vorticity, C_v (figure 23). Second, a low-order model decomposes this dominant component into contributions from C_{v,r_v} and C_{v,s_v} (figure 24). Third, the load-shifting phenomenon (reduced force peak and elevated force valley) is analysed from the initial contraction and subsequent expansion of the flexible plate’s radial vorticity distribution (r_v^*). While prior studies on biological propulsion have largely focused on vortex strength (Γ_v^*) to explain force generation (Kim & Gharib 2011*b*; Li *et al.* 2022), these findings highlight the critical role of vorticity’s spatial distribution in shaping the force evolution of flexible structures.

In conclusion, this study enhances our understanding of biological propulsion and disturbance-resistance based on a model problem. When high instantaneous forces are needed, organisms may stiffen appendages to maintain shape. During routine locomotion, moderate flexibility redistributes vorticity to avoid large force peaks, maintaining smooth propulsion, reducing fatigue and improving energetic efficiency (Lucas *et al.* 2014). For disturbance resistance – such as in suspension-like wing systems (Cheney *et al.* 2020) – adaptable deformation reduces peak forces and force fluctuations, minimizing vibration and improving flight stability.

Future work should expand upon the current findings by exploring a wider range of kinematic and geometric parameters. Key kinematic variables include the acceleration number (a^*), reflecting the time-varying nature of biological propulsion (Kang *et al.* 2011). Furthermore, investigating hysteresis and path-dependence – such as through start-stop or reversed acceleration sequences – would elucidate how wake history effects interact with structural reconfiguration to modulate the load-shifting. Additionally, the Reynolds number, particularly the lower regimes (10^2 – 10^3) relevant to insect flight (Liu & Sun 2024). Geometric variations, such as elliptical plates with low aspect ratios typical of insect wings, also warrant attention to generalize the understanding of load-shifting (Fernando & Rival 2016*a*).

While this study focused on a constant acceleration number ($a^* = 0.25$) to isolate the effects of reconfiguration, a preliminary investigation confirms that a^* is a critical parameter. This analysis shows that the key characteristics of the load-shifting

phenomenon persist at higher acceleration numbers ($a^* = 1.0$), demonstrating the mechanism's robustness. However, a comparison of the peak force against the $\sqrt{a^*}$ scaling law proposed for rigid plates by Reijtenbagh *et al.* (2023a) reveals a significant deviation for flexible and extra-flexible plates, even though our rigid plate data is consistent with the scaling. This deviation highlights that the fluid–structure interaction of reconfiguring plates introduces additional physics not captured by rigid-body models. A comprehensive study of the coupled parameter space of El^* and a^* is therefore a critical avenue for future research.

Although the circular plate serves as a canonical model, the fundamental mechanism identified – that passive deformation modulates unsteady forces by altering the wake's spatiotemporal evolution – is believed to be broadly applicable to many biological and engineering systems (Triantafyllou *et al.* 2016; Cheney *et al.* 2020). For geometries with lower aspect ratios, such as elliptical or rectangular plates, the interaction between the deforming structure and the wake would be more complex. We hypothesize that while load-shifting would still occur, the optimal non-dimensional stiffness (El^*) required to achieve it would likely be a function of the aspect ratio. Extending these findings to realistic 3-D wings and fins with non-uniform thickness and curvature is another important next step (Lucas *et al.* 2014). In particular, material gradients-such as the non-uniform stiffness characteristic of insect wings (Combes & Daniel 2003; Mountcastle & Combes 2013) warrant attention, as they could induce higher-order deformation modes that enable precise passive control over vortex shedding.

Finally, to fully resolve the instantaneous 3-D vortex topology, including the non-axisymmetric evolution and subsequent breakdown, future work will require advanced 3-D velocimetry techniques. Methods such as tomographic PIV or 3-D particle tracking velocimetry would provide a more complete picture of the complex fluid–structure interaction and further validate the mechanisms proposed in this study.

Supplementary material. Supplementary material is available at <https://doi.org/10.1017/jfm.2026.11165>

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Declaration of interests. The authors report no conflict of interest.

Appendix A. Validation of experimental and analytical methodologies

A.1. Influence of the support structure

The L-shaped sting is a necessary component of the experimental set-up, but its presence can potentially influence the flow field and force measurements.

The length of the horizontal rod is the key design parameter for optimizing the L-shaped sting. We evaluated the trade-off between wake interference and structural rigidity. A horizontal rod that is too short allows the vertical strut's wake to contaminate the flow (wake deficit), whereas an excessively long rod reduces stiffness, leading to potential resonance (frequency contamination). We compared rod lengths of $3D$, $4D$ and $6D$ under a representative condition ($a^* = 1.0$, $U_c = 0.400 \text{ m s}^{-1}$, $h_{nom} = 3.00 \text{ mm}$).

Figure 26 shows that the $6D$ rod exhibits distinct oscillations characteristic of a step response, indicating that its reduced stiffness lowers the natural frequency close to the motion's excitation spectrum. In contrast, the $4D$ rod maintains a smooth force profile,

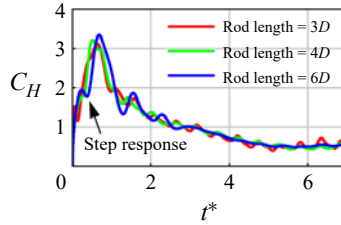


Figure 26. Evaluation of the sting's influence on force measurements. Evolution of hydrodynamic force coefficient C_H over t^* for a 2.9889 mm-thick circular plate ($a^* = 1.0$, $U_c = 0.400 \text{ m s}^{-1}$), under different horizontal rod lengths: 3D (red line), 4D (green line) and 6D (blue line).

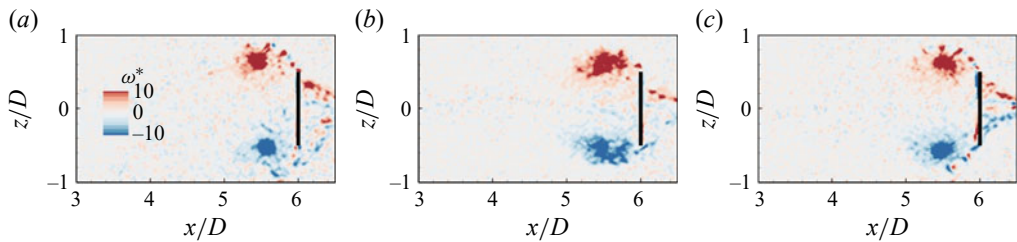


Figure 27. Evaluation of the sting's influence on the wake. Non-dimensional vorticity fields at $t^* = 6$ for a 3.00 mm-thick circular plate ($a^* = 1.0$, $U_c = 0.400 \text{ m s}^{-1}$), under different horizontal rod lengths: 3D (a), 4D (b) and 6D (c).

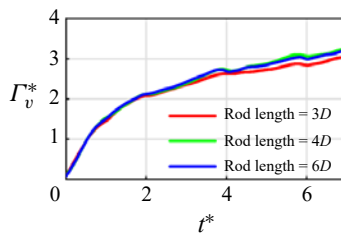


Figure 28. Evaluation of the sting's influence on vortex dynamics. Evolution of non-dimensional circulation Γ_v^* over t^* for a 3.00 mm-thick circular plate ($a^* = 1$, $U_c = 0.4 \text{ m s}^{-1}$), under different horizontal rod lengths: 3D (red line), 4D (green line) and 6D (blue line).

ensuring a sufficient frequency separation margin ($f_n \approx 48 \text{ Hz}$ versus $f_{\text{motion}} \leq 16 \text{ Hz}$, calculated in § 2.3) to preclude vibrational noise.

Regarding the flow field (figure 27), the 3D rod produces a clearly asymmetric vortex ring due to the wake deficit from the vertical strut. The 4D and 6D configurations, however, yield perfectly symmetric structures, confirming that the wake disturbance is fully dissipated at these lengths.

The influence of the horizontal rod length on the vortex dynamics is further evaluated in figure 28, which shows the evolution of the non-dimensional circulation Γ_v^* with t^* . It is observed that for $t^* > 3$, the circulation for the 3D case is slightly lower than those for the 4D and 6D cases, whereas the results for the 4D and 6D cases almost entirely overlap.

Consequently, the 4D length was adopted as the optimal configuration, effectively eliminating both wake interference and frequency contamination.

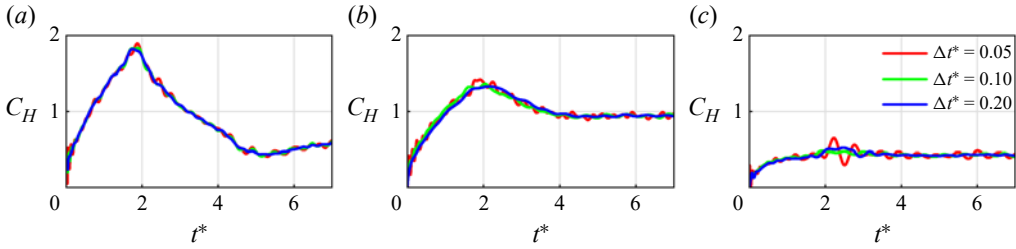


Figure 29. Sensitivity analysis of the hydrodynamic force coefficient (C_H) under different filtering window widths (Δt^*). Panels show (a) rigid (R-5), (b) flexible (F-7) and (c) extra-flexible (EF-6) cases. The red, green and blue lines correspond to $\Delta t^* = 0.05$ (less smoothing), $\Delta t^* = 0.10$ (baseline) and $\Delta t^* = 0.20$ (more smoothing), respectively. Quantitative comparison confirms the robustness of the baseline choice: for the rigid and flexible cases, the RRMSE of the peak/valley force across different windows remains below 3 % and the overall trend consistency is preserved.

A.2. Sensitivity to data processing

To ensure the robustness of our conclusions and maintain methodological consistency, we applied the same physically validated filter (second-order Savitzky–Golay, window width $\Delta t^* = 0.1$) to all time-resolved signals derived from both the force transducer (C_H) and PIV measurements (Γ_v^* , r_v^* , z_v^* , *etc.*). The choice of $\Delta t^* = 0.1$ is first validated by the excellent agreement with the literature benchmarks shown in figure 5.

To quantify the impact of this processing choice on the peak/valley metrics and the inferred regime thresholds, a sensitivity analysis was conducted by varying the filter window size ($\Delta t^* \in \{0.05, 0.10, 0.20\}$). As illustrated in figure 29, doubling or halving the window size results in minimal deviations. Quantitatively, the absolute variations in the peak force coefficient ($C_{H,peak}$) are less than 0.01, 0.03 and 0.06 for the ‘R-5’, ‘F-7’ and ‘EF-6’ cases, respectively. Similarly, variations in the valley force ($C_{H,valley}$) are limited to within 0.05. Given the reference force coefficients for the regime boundaries ($C_H \approx 1.73$ for $Er_{crit,2}^*$ and 0.60 for $Er_{crit,1}^*$, see § 3.4), the corresponding relative variations are less than 3.5 % and 8.3 %. This confirms that the inferred thresholds are robust to data processing choices.

Regarding the PIV data, vector calculation followed standard protocols, yields a velocity field uncertainty of <1.2 % (see § 2.4). For the computation of model-based force coefficients like $C_{H,impulse}$, the same filter ($\Delta t^* = 0.1$) was applied to the integral quantities prior to time differentiation. This prevents noise amplification while ensuring a rigorous, consistent comparison between modelled and measured forces.

A.3. Multirod measurement consistency

To obtain PIV measurements across multiple azimuthal planes (0° , 45° , 90°), three interchangeable horizontal rods were employed. We addressed potential errors arising from separate experimental runs through three specific controls.

First, mechanical backlash was minimized using a unidirectional positioning strategy, where the sting was retracted by 0.1 m before approaching the start position from a consistent forward direction. High-speed imaging confirmed a start-position repeatability within 1 pixel. And, phase consistency is governed by the high repeatability of the velocity profile (see § 2.2), rendering synchronization errors minimal.

Second, to assess systematic bias, we utilized the rigid plate (‘R-5’) as an axisymmetric benchmark. As quantified in the results section (see figures 14a and 15a), the wake statistics (Γ_v^* and r_v^*) measured across the three independent planes exhibit consistent

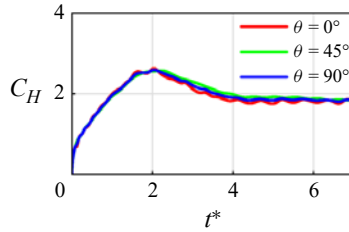


Figure 30. Validation of the multirod PIV measurement method. Evolution of the hydrodynamic force coefficient C_H over t^* for a representative flexible plate ($h_{nom} = 0.30$ mm) under the kinematic conditions of $a^* = 0.25$ ($t_a^* = 2.0$) and $U_c = 0.700$ m s⁻¹. The comparison shows nearly identical force profiles for the three rods with different mounting slot angles: 0° (red line), 45° (green line) and 90° (blue line), confirming the repeatability of the method.

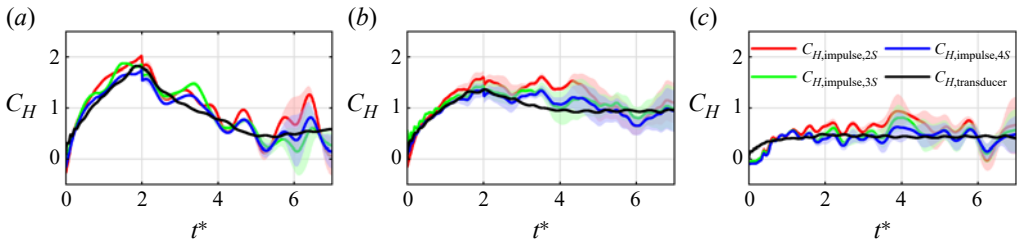


Figure 31. Sensitivity analysis of $C_{H,impulse}$ evolution over t^* using different cross-sectional slice strategies. Panels correspond to (a) ‘R-5’, (b) ‘F-7’ and (c) ‘EF-6’. Quantitative validation against transducer measurements confirms the robustness of the chosen strategy. Here $C_{H,impulse,2S}$ shows larger deviations (e.g. ‘F-7’ with RRMSE = 21.3 %). Here $C_{H,impulse,3S}$ significantly reduces error, achieving total RRMSE values of 11.6 % (‘R-5’), 10.7 % (‘F-7’) and 22.5 % (‘EF-6’), with strong correlation (PCC > 0.94 for ‘R-5’ and ‘F-7’; PCC = 0.77 for ‘EF-6’). Here $C_{H,impulse,4S}$ offers limited marginal improvement (e.g. ‘F-7’ with RRMSE = 10.4 % and PCC = 0.92), verifying that the three-slice approximation is sufficient to resolve the key force dynamics.

evolution, with maximum deviations below 0.5 % and 3.1 %. This confirms that the rod exchange introduces no significant systematic bias to the flow statistics.

Finally, consistent with the flow validation, the force evolution for a flexible plate ($h_{nom} = 0.30$ mm) measured with the three different rods (figure 30) shows negligible variation (RRMSE < 1 %).

A.4. Sensitivity to measurement plane resolution

The calculation of the impulse force, as detailed in (5.11), relies on the azimuthal integration of the vorticity moment. To quantify the sensitivity of our results to the angular resolution of the measurement planes and local non-axisymmetry, a convergence study was conducted by introducing an additional experimental dataset at an azimuthal angle of $\theta = 135^\circ$ (defined in figure 1). We evaluated three reconstruction strategies against the independent force transducer measurements: $C_{H,impulse,2S}$, average of orthogonal planes (0°, 90°); $C_{H,impulse,3S}$, baseline strategy employed in this manuscript (0°, 45°, 90°); $C_{H,impulse,4S}$, high-resolution reconstruction (0°, 45°, 90°, 135°).

The analysis reveals a clear convergence trend as shown in figure 31. While $C_{H,impulse,2S}$ exhibits higher deviation, $C_{H,impulse,3S}$ significantly improves accuracy. Crucially, extending the integration to four slices ($C_{H,impulse,4S}$) yields only marginal gains in prediction accuracy compared with $C_{H,impulse,3S}$. Therefore, balancing the prediction

accuracy with experimental efficiency, the three-slice strategy, capturing the dominant 3-D flow effects successfully and providing a robust estimate of the hydrodynamic force, is chosen.

Appendix B. Quantification of 3-D flow characteristics

B.1. Normalized divergence (ζ)

To quantify the intensity of out-of-plane vortex stretching relative to in-plane deformation, the normalized divergence, ζ , is used, as proposed by Reijtenbagh *et al.* (2023b). It is defined as the ratio of the planar divergence, D_{div} , to the Frobenius norm of the velocity gradient tensor, $\mathcal{N}$:

$$\zeta = \frac{D_{div}}{\mathcal{N}}. \quad (\text{B1})$$

Based on the continuity equation for incompressible flow ($\nabla \cdot \mathbf{u} = 0$) in cylindrical coordinates, the planar divergence D_{div} serves as a direct proxy for the azimuthal velocity gradient (Kaiser *et al.* 2020):

$$D_{div} = \frac{1}{r'} \frac{\partial(r' u_{r'})}{\partial r'} + \frac{\partial u_{z'}}{\partial z'} = -\frac{1}{r'} \frac{\partial u_{\theta}}{\partial \theta}. \quad (\text{B2})$$

The term $\partial u_{\theta}/\partial \theta$ signifies the stretching of vortex lines along the azimuthal direction, a mechanism governed by the $(\boldsymbol{\omega} \cdot \nabla)\mathbf{u}$ term in the vorticity transport equation (Wu *et al.* 2007). Consequently, ζ provides a localized, dimensionless metric for analysing 3-D instabilities, where $\mathcal{N}$ normalizes the stretching against the local shear and strain rate (Adrian *et al.* 2000).

B.2. The r.m.s. of normalized divergence (ζ_{rms})

To quantify the overall intensity of 3-D effects, the r.m.s. of the normalized divergence field, ζ_{rms} , is computed, within a specific domain of interest, $\mathcal{H}_{mask}$,

$$\zeta_{rms}(t^*) = \sqrt{\frac{1}{A_{\mathcal{H}}} \iint_{\mathcal{H}_{mask}} \zeta^2 dA}, \quad (\text{B3})$$

where $A_{\mathcal{H}}$ denotes the area of the domain. The mask $\mathcal{H}_{mask}$ is defined as the union of the primary vortex recirculation region and regions of significant vorticity: $\mathcal{H}_{mask} = \{\mathbf{x} \mid \psi_0(\mathbf{x}) < 0\} \cup \{\mathbf{x} \mid |\boldsymbol{\omega}(\mathbf{x})| > \omega_{threshold}\}$. The recirculation region is identified by the negative values of the Stokes stream function in the moving frame, ψ_0 (Akhmetov 2009), calculated as

$$\psi_0(r', z') = \int_0^{r'} \xi u'_{z'}(\xi, z') d\xi. \quad (\text{B4})$$

The vorticity threshold, $\omega_{threshold}$, is set to twice the mean absolute vorticity of the background noise, determined from baseline PIV measurements with a stationary plate, to ensure the exclusion of measurement noise.

Appendix C. Calculation of added mass coefficient C_A

The added mass coefficient C_A is calculated using the shape-tracking data to obtain the major and minor axes of the projected shape. The added mass m_a is approximated using

the formula for an elliptical plate,

$$m_a = K_b a_e^2 \frac{\pi}{6} \rho_f, \quad (\text{C1})$$

where K_b is a shape factor determined from the approximate aspect ratio of the ellipse (Brennen 1982) and a_e is the semimajor axis.

The impulse due to the vortex sheet associated with the plate translation I_a is given by

$$I_a = m_a U. \quad (\text{C2})$$

Considering the time-dependent nature of the added mass, the force can be expressed as

$$F_a = U \frac{dm_a}{dt} + m_a \frac{dU}{dt}. \quad (\text{C3})$$

The added mass coefficient C_A is then decomposed into two components,

$$C_A = C_{A1} + C_{A2}, \quad (\text{C4})$$

where

$$C_{A1} = \frac{1}{\frac{1}{2} \rho_f U_c^2 S} U \frac{dm_a}{dt}, \quad (\text{C5})$$

and

$$C_{A2} = \frac{1}{\frac{1}{2} \rho_f U_c^2 S} m_a \frac{dU}{dt}. \quad (\text{C6})$$

Based on the shape tracking data, the evolution of C_{A1} and C_{A2} with respect to the non-dimensional time t^* can be obtained. This provides insight into how the added mass effects change over time during the motion.

Appendix D. Error analysis of the low-order force model

This appendix provides a systematic analysis of the deviation between $C_{v,m}$ and $C_{v,i}$, specifically addressing the error observed in the extra-flexible regime (figure 24). The low-order force model proposed in § 5.2 relies on two fundamental kinematic assumptions: the shed vorticity remains radially compact ('thin-core'); and the wake rolls up into a single, coherent vortex ring.

For rigid and flexible plates, the model demonstrates good agreement, validating its applicability to coherent vortex structures. However, for the extra-flexible plate ('EF-6'), the model underperforms. This quantitative divergence necessitates a physical explanation to identify which assumption has been violated.

To assess the first assumption, the spatial spread of the vorticity field using the total vorticity dispersion index, L_v^* , is used, defined as the vorticity-weighted standard deviation from the centroid (z_v^* , r_v^*) (Wu 1981; Akhmetov 2009). This metric decomposes into axial and radial components as $(L_v^*)^2 = (L_{v,z'}^*)^2 + (L_{v,r'}^*)^2$. The radial dispersion index, $L_{v,r'}^*$, is the key metric for our justification, directly quantifying the 'thickness' of the vortex ring:

$$L_{v,r'}^* = \sqrt{\frac{1}{U_c D^3 \Gamma_v^*} \int_0^D \int_{-D}^{z'_{tip}} (r' - r_v)^2 \omega \, dz' \, dr'}. \quad (\text{D1})$$

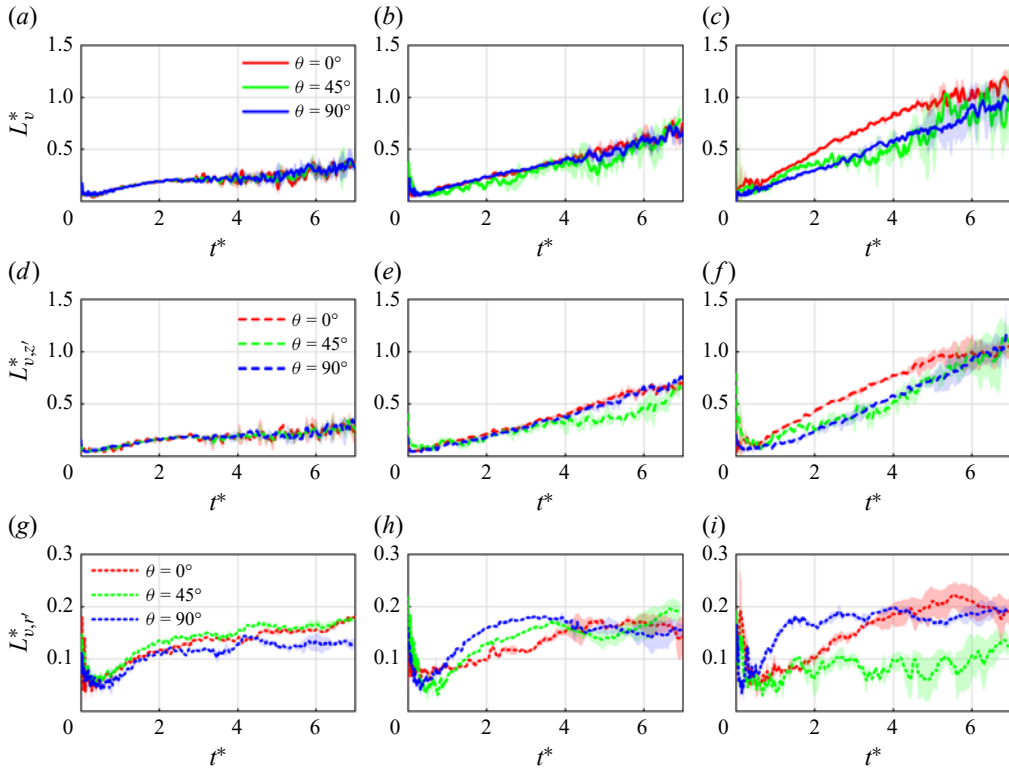


Figure 32. Evolution of the vorticity dispersion index for the (a,d,g) rigid ‘R-5’, (b,e,h) flexible ‘F-7’ and (c,f,i) extra-flexible ‘EF-6’ plates. The axial component ($L_{v,z'}^*$, dashed lines) varies significantly, indicating anisotropic dispersion. In contrast, the radial component ($L_{v,r}^*$, dotted lines) remains small (< 0.2) in all cases, supporting the ‘radially thin vortex ring’ assumption.

Correspondingly, the axial component, $L_{v,z'}^*$, quantifies the spread of vorticity in the downstream direction:

$$L_{v,z'}^* = \sqrt{\frac{1}{U_c D^3 \Gamma_v^*} \int_0^D \int_{-D}^{z'_{tip}} (z' - z_v)^2 \omega \, dz' \, dr'}. \quad (\text{D2})$$

As shown in figure 32, the axial dispersion $L_{v,z'}^*$ varies significantly and grows to large values for flexible plates, reflecting the elongation of the wake. In contrast, the radial index $L_{v,r}^*$ remains consistently low (< 0.2) across all cases, including the extra-flexible plate. This indicates that the vorticity remains radially compact and does not undergo significant diffusion, thereby confirming that the ‘thin-core’ geometric assumption holds true even when the model fails.

Since the first assumption is valid, the significant prediction error for the extra-flexible plate should arise from the violation of the second assumption. As visualized in the flow fields (in § S3 of the Supplementary materials), the wake of the extra-flexible plate does not form a coherent vortex ring but instead evolves into a continuous, fragmented shear layer. This topological breakdown fundamentally violates the kinematic basis of the single-ring model. Thus, the high error can serve as a diagnostic indicator, quantitatively confirming the transition from a vortex-ring regime to a shear-layer regime.

REFERENCES

- ADRIAN, R.J., MEINHART, C.D. & TOMKINS, C.D. 2000 Vortex organization in the outer region of the turbulent boundary layer. *J. Fluid Mech.* **422**, 1–54.
- AKHMETOV, D.G. 2009 *Vortex Rings*. Springer Science & Business Media.
- ALBEN, S., SHELLEY, M. & ZHANG, J. 2002 Drag reduction through self-similar bending of a flexible body. *Nature* **420** (6915), 479–481.
- ALBEN, S., SHELLEY, M. & ZHANG, J. 2004 How flexibility induces streamlining in a two-dimensional flow. *Phys. Fluids* **16** (5), 1694–1713.
- BASKARAN, M., HUTIN, L. & MULLENERS, K. 2023 Reconfiguring it out: how flexible structures interact with fluid flows. *Phys. Rev. Fluids* **8** (11), 110509.
- BASSET, A.B. 1888 *A Treatise On Hydrodynamics: with Numerous Examples*, vol. 2. Deighton, Bell and Company.
- BACHELOR, G.K. 2000 *An Introduction to Fluid Dynamics*. Cambridge University Press.
- BEER, F., JOHNSTON, E. & DEWOLF, J. 1999 Mechanics of materials. *Stress* **1** (10), 1–12. 5th si edition.
- BRENNEN, C.E. 1982 *A Review of Added Mass and Fluid Inertial Forces*. Naval Civil Engineering Laboratory.
- CALLISTER, W.D. & RETHWISCH, D.G. 1999 *Materials Science and Engineering: An Introduction*. Wiley New York.
- CHARONKO, J.J. & VLACHOS, P.P. 2013 Estimation of uncertainty bounds for individual particle image velocimetry measurements from cross-correlation peak ratio. *Meas. Sci. Technol.* **24** (6), 065301.
- CHARTRAND, R. 2011 Numerical differentiation of noisy, nonsmooth data. *Intl Scholarly Res. Not.* **2011** (1), 164564.
- CHENEY, J.A., STEVENSON, J.P.J., DURSTON, N.E., SONG, J., USHERWOOD, J.R., BOMPHELY, R.J. & WINDSOR, S.P. 2020 Bird wings act as a suspension system that rejects gusts. *Proc. R. Soc. Lond. B* **287** (1937), 20201748.
- CHENG, M., LOU, J. & LIM, T.T. 2016 Evolution of an elliptic vortex ring in a viscous fluid. *Phys. Fluids* **28** (3), 037104.
- CHIN, D.D. & LENTINK, D. 2016 Flapping wing aerodynamics: from insects to vertebrates. *J. Expl Biol.* **219** (7), 920–932.
- COMBES, S.A. & DANIEL, T.L. 2003 Flexural stiffness in insect wings II. Spatial distribution and dynamic wing bending. *J. Expl Biol.* **206** (17), 2989–2997.
- CORKERY, S.J., BABINSKY, H. & GRAHAM, W.R. 2019 Quantification of added-mass effects using particle image velocimetry data for a translating and rotating flat plate. *J. Fluid Mech.* **870**, 492–518.
- CORKERY, S.J. 2019 Unsteady aerodynamics of wing-gust encounters. PhD thesis, University of Cambridge, Cambridge, UK.
- DE LANGRE, E. 2008 Effects of wind on plants. *Annu. Rev. Fluid Mech.* **40** (1), 141–168.
- FERNANDO, J.N. & RIVAL, D.E. 2016a On vortex evolution in the wake of axisymmetric and non-axisymmetric low-aspect-ratio accelerating plates. *Phys. Fluids* **28** (1), 017102.
- FERNANDO, J.N. & RIVAL, D.E. 2016b Reynolds-number scaling of vortex pinch-off on low-aspect-ratio propulsors. *J. Fluid Mech.* **799**, R3.
- FERNANDO, J.N., WEYMOUTH, G.D. & RIVAL, D.E. 2020 On the limits of added-mass theory in separated flows and with varying initial conditions. *J. Fluids Struct.* **93**, 102835.
- GALLER, J.N., WEYMOUTH, G.D. & RIVAL, D.E. 2021 On the concept of energized mass: a robust framework for low-order force modeling in flow past accelerating bodies. *Phys. Fluids* **33** (5), 057103.
- GEHLERT, P. 2021 Unsteady flow and force development in the case of a circular cylinder. PhD thesis, Technical University of Darmstadt, Darmstadt, Germany.
- GHARIB, M., RAMBOD, E. & SHARIFF, K. 1998 A universal time scale for vortex ring formation. *J. Fluid Mech.* **360**, 121–140.
- GOSSELIN, F., DE LANGRE, E. & MACHADO-ALMEIDA, B.A. 2010 Drag reduction of flexible plates by reconfiguration. *J. Fluid Mech.* **650**, 319–341.
- GRAHAM, W.R. 2019 Decomposition of the forces on a body moving in an incompressible fluid. *J. Fluid Mech.* **881**, 1097–1122.
- GRAHAM, W.R., PITT FORD, C.W. & BABINSKY, H. 2017 An impulse-based approach to estimating forces in unsteady flow. *J. Fluid Mech.* **815**, 60–76.
- GRIFF, E.J., VIJAYARAGAVAN, N.B., TUMMERS, M.J. & WESTERWHEEL, J. 2019 Drag force on an accelerating submerged plate. *J. Fluid Mech.* **866**, 369–398.
- HANG, H., HEYDARI, S., COSTELLO, J.H. & KANSO, E. 2022 Active tail flexion in concert with passive hydrodynamic forces improves swimming speed and efficiency. *J. Fluid Mech.* **932**, A35.
- HARVEY, C., BALIGA, V.B., LAVOIE, P. & ALTSHULER, D.L. 2019 Wing morphing allows gulls to modulate static pitch stability during gliding. *J. R. Soc. Interface* **16** (150), 20180641.

- HO, S., NASSEF, H., PORNINSIRIRAK, N., TAI, Y.-C. & HO, C.-M. 2003 Unsteady aerodynamics and flow control for flapping wing flyers. *Prog. Aerosp. Sci.* **39** (8), 635–681.
- JIMENEZ, Y.E., LUCAS, K.N., LONG JR., J.H. & TYTELL, E.D. 2023 Flexibility is a hidden axis of biomechanical diversity in fishes. *J. Expl Biol.* **226** (Suppl_1), jeb245308.
- JOHANSSON, L.C., WOLF, M. & HEDENSTRÖM, A. 2010 A quantitative comparison of bird and bat wakes. *J. R. Soc. Interface* **7** (42), 61–66.
- JOSHI, K. & BHATTACHARYA, S. 2022 The unsteady force response of an accelerating flat plate with controlled spanwise bending. *J. Fluid Mech.* **933**, A56.
- KAISER, F., KRIEGSEIS, J. & RIVAL, D.E. 2020 The influence of edge undulation on vortex formation for low-aspect-ratio propulsors. *J. Fluid Mech.* **883**, A55.
- KANG, C.-K., AONO, H., CESNIK, C.E.S. & SHYY, W. 2011 Effects of flexibility on the aerodynamic performance of flapping wings. *J. Fluid Mech.* **689**, 32–74.
- KANG, C.-K. & SHYY, W. 2013 Scaling law and enhancement of lift generation of an insect-size hovering flexible wing. *J. R. Soc. Interface* **10** (85), 20130361.
- KEANE, R.D. & ADRIAN, R.J. 1989 Optimization of particle image velocimeters. In *International Congress on Applications of Lasers & Electro-Optics*, pp. 141–161. AIP Publishing.
- KIM, D. & GHARIB, M. 2011a Characteristics of vortex formation and thrust performance in drag-based paddling propulsion. *J. Expl Biol.* **214** (13), 2283–2291.
- KIM, D. & GHARIB, M. 2011b Flexibility effects on vortex formation of translating plates. *J. Fluid Mech.* **677**, 255–271.
- KUNDU, P.K., COHEN, I.M., DOWLING, D.R. & CAPECELATRO, J. 2024 *Fluid Mechanics*. Elsevier.
- LANDAU, L.D., PITAEVSKII, L.P., KOSEVICH, A.M. & LIFSHITZ, E.M. 2012 *Theory of Elasticity*, vol. 7, Elsevier.
- LEIBOVICH, S. 1978 The structure of vortex breakdown. *Annu. Rev. Fluid Mech.* **10**, 221–246.
- LI, Z., CHEN, L., XIANG, Y., LIU, H. & WANG, F. 2024 Two independent mechanisms with distinct laws for the generation of drag force on accelerating plates. *Phys. Fluids* **36** (10), 107163.
- LI, Z., XIANG, Y., QIN, S., LIU, H. & WANG, F. 2022 Two models and the generation mechanisms of the drag on an accelerating starting disk. *Phys. Fluids* **34** (8), 081908.
- LIU, W. & SUN, M. 2024 Drag force on an accelerating flat plate at low Reynolds numbers. *Phys. Fluids* **36** (9), 091908.
- LORITE-DÍEZ, M., JIMÉNEZ-GONZÁLEZ, J.I., GUTIÉRREZ-MONTES, C. & MARTÍNEZ-BAZÁN, C. 2018 Effects of rear cavities on the wake behind an accelerating D-shaped bluff body. *Phys. Fluids* **30** (4), 044103.
- LUCAS, K.N., JOHNSON, N., BEAULIEU, W.T., CATHCART, E., TIRRELL, G., COLIN, S.P., GEMMELL, B.J., DABIRI, J.O. & COSTELLO, J.H. 2014 Bending rules for animal propulsion. *Nat. Commun.* **5** (1), 1–7.
- LUCCA-NEGRO, O. & O'DOHERTY, T. 2001 Vortex breakdown: a review. *Prog. Energy Combust. Sci.* **27** (4), 431–481.
- MCPHADEN, C.J. & RIVAL, D.E. 2018 Unsteady force estimation using a Lagrangian drift-volume approach. *Exp. Fluids* **59**, 1–11.
- MOUNTCASTLE, A.M. & COMBES, S.A. 2013 Wing flexibility enhances load-lifting capacity in bumblebees. *Proc. R. Soc. B: Biol. Sci.* **280** (1759), 20130531.
- ODAR, F. & HAMILTON, W.S. 1964 Forces on a sphere accelerating in a viscous fluid. *J. Fluid Mech.* **18** (2), 302–314.
- OSEEN, C.W. 1927 *Neuere methoden und ergebnisse in der hydrodynamik*. Akademische Verlagsgesellschaft.
- RAMANANARIVO, S., GODOY-DIANA, R. & THIRIA, B. 2011 Rather than resonance, flapping wing flyers may play on aerodynamics to improve performance. *Proc. Natl Acad. Sci. USA* **108** (15), 5964–5969.
- REIJTENBAGH, J., TUMMERS, M.J. & WESTERWEEL, J. 2023a Drag force on a starting plate scales with the square root of acceleration. *Phys. Rev. Lett.* **130** (17), 174001.
- REIJTENBAGH, J., WESTERWEEL, J. & VAN DE WATER, W. 2023b Transition to turbulence behind a traveling plate. *Phys. Rev. Fluids* **8** (10), 104601.
- REYNOLDS, K.V., THOMAS, A.L.R. & TAYLOR, G.K. 2014 Wing tucks are a response to atmospheric turbulence in the soaring flight of the steppe eagle *Aquila nipalensis*. *J. R. Soc. Interface* **11** (101), 20140645.
- SAFFMAN, P.G. 1995 *Vortex Dynamics*. Cambridge University Press.
- SAVITZKY, A. & GOLAY, M.J.E. 1964 Smoothing and differentiation of data by simplified least squares procedures. *Anal. Chem.* **36** (8), 1627–1639.
- SCHOUVEILER, L. & BOUDAUD, A. 2006 The rolling up of sheets in a steady flow. *J. Fluid Mech.* **563**, 71–80.
- SCHOUVEILER, L. & ELOY, C. 2013 Flow-induced draping. *Phys. Rev. Lett.* **111** (6), 064301.

- SHYY, W., AONO, H., CHIMAKURTHI, S.K., TRIZILA, P., KANG, C.-K., CESNIK, C.E.S. & LIU, H. 2010 Recent progress in flapping wing aerodynamics and aeroelasticity. *Prog. Aerosp. Sci.* **46** (7), 284–327.
- STOKES, G.G. 1851 On the effect of the internal friction of fluids on the motion of pendulums. *Trans. Camb. Phil. Soc.* **9**, 8–106.
- SUZUKI, S., *et al.* 1985 Topological structural analysis of digitized binary images by border following. *Comput. Vis. Graph. Image Process.* **30** (1), 32–46.
- TAYLOR, G.K., CARRUTHERS, A.C., HUBEL, T.Y. & WALKER, S.M. 2012 Wing morphing in insects, birds and bats: mechanism and function. *Morphing Aerospace Vehicles and Structures* (ed. J. Valasek), pp. 11–40. Wiley.
- TAYLOR, J.R. 2022 *An Introduction to Error Analysis: The Study of Uncertainties in Physical Measurements*. MIT Press.
- TRIANAFYLLOU, M.S., WEYMOUTH, G.D. & MIAO, J. 2016 Biomimetic survival hydrodynamics and flow sensing. *Annu. Rev. Fluid Mech.* **48** (1), 1–24.
- VOGEL, S. 1989 Drag and reconfiguration of broad leaves in high winds. *J. Expl Bot.* **40** (8), 941–948.
- WU, J.C. 1981 Theory for aerodynamic force and moment in viscous flows. *AIAA J.* **19** (4), 432–441.
- WU, J.-Z., MA, H.-Y. & ZHOU, M.-D. 2007 *Vorticity and Vortex Dynamics*. Springer Science & Business Media.
- WU, W. 2021 *Fluid Mechanics (in Chinese)*. Peking University Press.
- XIANG, Y., LI, Z., QIN, S. & LIU, H. 2021 Formation number and pinch-off signals of disc vortex ring based on a Lagrangian analysis. *Exp. Therm. Fluid Sci.* **129**, 110452.
- YANG, A.-L., JIA, L.-B. & YIN, X.-Z. 2012 Formation process of the vortex ring generated by an impulsively started circular disc. *J. Fluid Mech.* **713**, 61–85.