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A Novel Biomimetic Hydrodynamic Wake Tracking Principle Inspired By Harbour Seal (*Phoca Vitulina*) Vibrissae

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ABSTRACT

The current study focuses on the underwater navigation inspired by harbour seals (*Phoca Vitulina*) vibrissae. Biological studies have illustrated the underwater sensing and tracking ability of harbour seals. These studies have exhibited the ability of the seals' vibrissae to track minute water velocities (up to $243 \mu\text{m/s}$) including tests in pools with background (water) noises. This depicts the robustness of the underwater sensing principle of the harbour seals' vibrissae. The evaluation of the underwater sensing principle of the seal vibrissae is performed through experiments and numerical simulations. The experiments are conducted in CENSAM water tank for measurements of forces on the vibrissa and its setups. Numerical simulations are utilised to provide insights into the physics of the flow around the structures. From these studies, it is found that the geometry of the vibrissavibrissae reduces Vortex-Induced Forces (VIF) and Vortex-Induced Vibration (VIV). Studies on the geometrical features of the vibrissa revealed that the existence of dual-axial undulations significantly reduces VIF and VIV while its drag coefficient is found to be similar to elliptical cylinder. For freely vibrating in crossflow and freely vibrating in streamwise and crossflow simulations, halved dominant crossflow response frequency is observed for the vibrissa as compared to that of circular and elliptical cylinders. From analysis of the wake, significant deformations of the vortex tubes and crossing of vortical braids from one plane to another (parallel to streamwise direction) are observed. These resulted in lower dominant response frequency of the vibrissa. It is also found that the spatial delay in vortex shedding dominates the desynchronization in the spanwise vortex shedding in reducing the amplitude of VIV. Experiments and simulations are also performed for sensory structure (i.e. vibrissa, elliptical cylinder and circular cylinder) trailing an equivalent diameter circular cylinder. For zero relative velocity between the generator and the sensor, it is found that the vibrissa and elliptical cylinder responded with significantly larger amplitude of oscillation and similar frequency of oscillation to the vortex shedding frequency of the generator. However, for non-zero relative velocity cases, the amplitude of oscillation is found to significantly decrease due to reduction in the duration of exposure of the sensory structure to the vortex tubes (perturbations). For approaching sensory structures, limiting approach speed is found to exist due to the greater modification of the vortices in the wake of the target at higher approach velocity (due to higher pressure build-up in the leading edge of the structure).

I would like to dedicate this thesis to my family and Stephanie
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Contents

1	Introduction	1
1.1	Background	1
1.2	Objectives	2
1.3	Structure of The Thesis	3
2	Literature Review	7
3	Properties of Harbour Seal Whiskers	43
3.1	Mechanical Properties	43
3.2	Results	51
3.3	Chemical Properties of Harbour Seal Vibrissae	62
3.4	Discussion	64
4	Experiments on VIV Reduction Properties of Harbour Seal Vibrissae	69
4.1	Hypothesis	70
4.2	Experimental Setup	72
4.3	Towing Experiments of A Vibrissa Model-Sensor Trailing An Upstream Wake Gen- erator	89
4.4	Results And Discussions	91
5	Methodology of Numerical Simulation	109
5.1	Cartesian-grid based solver for problems in fluid-structure interaction	110
5.2	Simulation domain and setup	114
5.3	Grid design and validations	125
5.4	Analysis method of forces and responses of the structures	138
6	VIV Reducing Properties of Harbour Seal Vibrissae Geometry	147
6.1	Simulations of Stationary Geometry	150
6.2	Simulations of 1D Vibration Geometry	191
6.3	Simulations of 2D Vibration Geometry	236
6.4	Response Types of 1D and 2D Freely Vibrating Geometries	252
7	Force And Motion Responses of Whisker At Angle of Attack	259
7.1	Stationary Simulations	260
7.2	Freely Vibrating Simulations (1D & 2D Vibrating)	274
8	Sensor-<i>Kármán</i> Street Interaction at Zero Relative Velocity	295
8.1	Stationary sensory structure	296
8.2	Freely vibrating in crossflow sensory structure	311

9	Sensor-Karman(St) Interaction at Relative Velocity	333
9.1	Equivalently-sized wake-generator and trailing sensor structure	334
9.2	Effects of CDR and relative velocity on the response of trailing structure	349
10	Conclusion & Future Work	353
A	Dynamic Scaling of Experimental Setup	365
B	ILES And QUICK Scheme	371
B.1	1D Model Problem on Application of QUICK Scheme	373
C	Analysis of the coherent structures in the wakes of vibrissa-like geometries	375
C.1	Analysis of wake topology of geometry 4	375
C.2	Analysis of wake topology of geometry 5	378
C.3	Analysis of wake topology of geometry 9	381
D	Onset of Instability in The Wake of a Circular Cylinder	383
E	Instability of The Wake of A Freely Vibrating Structures	391
E.1	Freely Vibrating in Crossflow New Vibrissa Geometry	391
E.2	Freely Vibrating in Streamwise and Crossflow Vibrissa-Like Geometry with No Major Axis Undulations and Axis Ratio of 0.44	397
F	Wake Topology In Wake-Induced Vibration (WIV) Structures	401
F.1	WIV on Trailing Circular Cylinder	401
F.2	WIV on Trailing Elliptical Cylinder	406
G	List of Awards and Publications	411
G.1	List of Awards and Honorary Appointments	411
G.2	List of Journal Publications	411
G.3	List of Invited Presentations	412
G.4	List of Conference Publications (Presentations for APS)	412

List of Figures

1.1	Harbour seal and its vibrissae on the muzzle in New England Aquarium, Boston, Massachusetts, USA (Copyright: Heather Beem [MIT Towing Tank])	2
2.1	Harbour seal vibrissae from New England Aquarium, Boston, Massachusetts, USA: Top - Top view of full length and segmented vibrissae; Middle - Top view on the major axis undulations of a vibrissa; Bottom - Side view on the minor axis undulations of a vibrissa.	9
2.2	Ideal vibrissa model and definitions of its geometrical parameters. From left to right : top view of the minor axis of the vibrissa, top view of the major axis of the vibrissa, and isometric view of the vibrissa. [40].	10
2.3	Wake-tracking behaviour of trailing harbour seal while tracking a wake generated by a fellow harbour seal. A) Linear pattern : the wake-tracking seal adheres almost-perfectly to the wake-generated by the fellow harbour seal; B) Undulatory pattern : wake-tracking seal was observed to criss-cross the wake generated by the fellow harbour seal, however, the amplitude of undulation was found not to exceed one body length [80].	11
2.4	Search pattern of wake-tracking harbour seal at various initial angular orientation to the wake generated by a fellow harbour seal. For explanation on linear pattern and undulatory pattern, please refer to Fig. 2.3.) [80].	11
2.5	Wake-generator paddles for experiments on the shape discrimination through wake analysis by harbour seals. From left to right : flat paddle, undulated paddle, circular paddle and triangular paddle. The height of the paddles were 30 cm [101].	12
2.6	Flow visualisations of flow around (Top) vertical flat plate, (Middle) circular cylinder, (Bottom) triangular cylinder [32, 104, 113].	13
2.7	Shape-discrimination experimental results on harbour seal. The wake-generators were flat paddle, undulated paddle, circular paddle and triangular paddle [101] (see Figure 2.5 for shape illustrations).	14
2.8	Strouhal number (St) of flat plate (top left), circular cylinder (top right) and triangular cylinder (bottom) as a function of Reynolds number (Re) [8, 73, 65].	15
2.9	Size-shape discrimination experiments on harbour seal between flat paddle and other paddles. The width of the paddles were varied randomly between 2,4,5 and 7 cm [101].	16
2.10	Success rates of wake-tracking by harbour seal for wakes generated by bursting (solid line) and gliding (dashed line) miniature submarine at various time delay (between the start of the submarine and start of the seal). Numbers on top (for burst) and on bottom (for glide) shows the number of trials for each time delay and types of wakes [100].	16
2.11	Snapshot from wake-tracking experiment on the seal for burst (left) and glide (right) wakes of a miniature submarine after 10 and 20 seconds, respectively. Magenta dots - path of seal, blue dots - burst phase of submarine, yellow - glide phase of submarine [100].	17

2.12	Sensitivity of harbour seal vibrissae at various frequency of perturbations. Left : amplitude threshold of the harbour seal vibrissa sensing at frequency range (100 to 2500 Hz), Mills & Renouf observed that below the amplitude threshold at a given frequency, the seal was incapable of signifying the absence or presence of the perturbation. Right : the percentage of rapidly adapting (RA) fibres responding to perturbations at various frequencies [69, 29].	18
2.13	Two-dimensional (XY-plane) flow visualization of the z-vorticity field in the wake of a swimming Giant Danio. Green - irrotational flow field, blue - counterclockwise rotation, red - clockwise rotation [88].	18
2.14	Vortex shedding of a tapered circular cylinder. Left : smoke visualisations of the vortex shedding on a tapered circular cylinder at maximum $Re = 140$; Right : a sketch of the occurrence of vortex split between the patches of the vortex shedding on the tapered circular cylinder [76, 83].	20
2.15	Comparison of the coherence of vortical structures in the von <i>Kármán</i> street of the circular cylinder and tapered circular cylinder [33].	21
2.16	Vortex shedding cells at the near-wall region of a circular cylinder with three cells. From left to right (far from wall to near wall): high frequency, medium frequency, and low frequency, respectively [53].	21
2.17	The z-vorticity field of a flow past a helical strakes fitted circular cylinder. Left : reduced velocity ($U^* = U/fD = 3$); Middle : $U^* = 6.0$; Right : $U^* = 9.0$ [54].	23
2.18	Visualisations of flow around straight circular cylinder (Top) and flow around curved circular cylinder (Bottom). Top left : flow around straight circular cylinder at $Re = 200$; Top right : flow around straight circular cylinder at $Re = 275$; Bottom left : flow around curved circular cylinder at $Re = 100$; Bottom right : flow around curved circular cylinder at $Re = 500$ [104, 68].	24
2.19	Circular cylinder with mild tapering in the mid-section, D was the characteristic diameter on the untapered section of the circular cylinder [74].	25
2.20	Globally locked-in vortex shedding on the mildly-tapered cylinder (see figure 2.19). The mid-section (smallest diameter) of the cylinder span was observed to lead the vortex shedding of the mildly-tapered cylinder [74].	26
2.21	Globally locked-in vortex shedding on the mildly-tapered cylinder (see figure 2.19). The mid-section (smallest diameter) of the cylinder span was observed to lead the vortex shedding of the mildly-tapered cylinder [74].	26
2.22	Vortex shedding modes of a wavy-trailing-edged half-ellipse. The shaded areas represented the positions of vortex splits (or dislocations). f_1 was the first vortex shedding frequency and f_2 was the second vortex shedding frequency, where $f_1 \neq f_2$ [85].	27
2.23	Variation of drag coefficients (C_D) as a function of wave steepness, w/L , w was two times the amplitude of the waves and L was the wavelength of the waves. Triangle mark - thin plates with wavy leading edge; square mark - rectangular cylinder with wavy leading edge [7].	29
2.24	Mean drag coefficients (C_D) and root mean square lift coefficients ($C_{L,RMS}$) as functions of wave steepness, W/λ , and wavelength, λ/D for small wavelengths ($\lambda \leq 5.6$), where λ was the wavelength of the undulations and W was twice the amplitude of the undulations on the wavy square cylinder, and D was the width of the square section perpendicular to the flow [24].	29

2.25	Mean drag coefficients (C_D) and root mean square lift coefficients ($C_{L,RMS}$) as functions of wave steepness, W/λ , and wavelength, λ/D for large wavelengths ($\lambda \geq 5.6$), where λ was the wavelength of the undulations and W was twice the amplitude of the undulations on the wavy square cylinder, and D was the width of the square section perpendicular to the flow [24].	30
2.26	Comparison of flow velocities in the wake of the wavy cylinder at $Re = 3,000$. U was defined as the local flow velocity, U_∞ was the freestream flow velocity, x was the perpendicular distance to the axis of the cylinder and d was the mean diameter of the wavy cylinder [62].	31
2.27	Visualisation of streamline and separation points for flow around a wavy cylinder. U_∞ was the freestream flow velocity, a was the amplitude of the undulations, λ was the wavelength of the vibrissa and D_m was the mean diameter of the undulatory cylinder [62].	32
2.28	Separation lines of flow around circular cylinder (left), wavy cylinder with small wave amplitude (middle) and wavy cylinder with large wave amplitude (right). See figure 2.27 for variables definition [59].	32
2.29	Spanwise view on the time evolution of the wakes of a wavy cylinder at $Re = 600$. The wavy cylinder was indicated by the wavy white lines on the left of each figure. U_∞ was the free stream flow velocity, t was the time elapsed in second, and T was the shedding period of the vortices in second [62].	33
2.30	Spanwise view of the mode B vortex shedding in the wake a circular cylinder. Mushroom-like structures were observed along the span of the cylinder. The triple arrows indicate d the direction of freestream flow, and L indicated the length of the circular cylinder [104].	35
2.31	Flow visualisations of the wakes of a wavy cylinder at nodal planes for various Reynolds Number. The Reynolds Number was defined based on the local Re , which was dependent on the local diameter of the wavy cylinder [62].	36
2.32	Flow visualisations of the wakes of a wavy cylinder at saddle planes for various Reynolds Number. The Reynolds Number was defined based on the local Re , which was dependent on the local diameter of the wavy cylinder [62].	37
2.33	Variability of the : a) mean drag coefficient(C_D), b) root-mean-square lift coefficient ($C_{L,RMS}$), and c) Strouhal number of the wavy cylinder at various wavelength (λ) and wave amplitude (a) combinations for $Re = 3,000$, where D_m was the mean diameter of the wavy cylinder [59].	38
2.34	Vortex formation length of circular and wavy cylinder for various wavelength and wave amplitude configurations at various spanwise planes (saddle and nodal planes) for $Re = 3,000$. See figure 2.33 for variables definition [59].	39
2.35	Variability of the mean drag coefficient(C_D) [left] and root-mean-square lift coefficient ($C_{L,RMS}$) [right] of the wavy cylinder at various wavelength (λ) and wave amplitude (a) combinations for $Re = 100$. See figure 2.33 for variables definition [60].	39
2.36	Regimes of flow around a wavy cylinder as functions of wave amplitude (a/D_m) and wavelength (λ/D_m) due to variability of the wakes and force response of the wavy cylinder at $Re = 100$. See figure 2.33 for variables definition [60].	40
2.37	Definitions of the flow regimes around a wavy cylinder due to various characteristics of wakes and force responses. See figure 2.33 for variables definition [60].	41
3.1	Variation of normalized minor and major diameter of the short, medium, and long vibrissae along their normalized lengths.	45
3.2	Modulus of Elasticity of harbour seal vibrissa subjected to axial strain of 0.001% and 0.005%.	47

3.3	Variations of modulus of elasticity for vibrissae segments with duration of immersion in saline water. First segment represents the vibrissa segment closest to the muzzle of the seal, while fourth segment represents the free end (tip) of the vibrissa. Second and third vibrissa segments are sequentially located between first and fourth vibrissa segments.	48
3.4	The overall experimental setup for determining the natural frequency of the harbour seal vibrissae. Setup: Light, high-speed camera, power amplifier, mini-shaker and vibrissa, waveform generator, oscilloscope. The field of view of camera is on the plane of vibration of the vibrissae.	50
3.5	Modulus of Elasticity of vibrissa segments immersed in dry, water, saline water and HBSS at varying frequency of axial force applications.	52
3.6	Damping of vibrissa segments immersed in dry, water, saline water and HBSS at varying frequency of axial force applications.	53
3.7	Average Modulus of Elasticity of dry (blue) and saline water-immersed (red) vibrissae segments. See figure 3.3 for information on spatial positions of vibrissa segments.	55
3.8	Average damping of dry (blue) and saline water-immersed (red) vibrissae segments. See figure 3.3 for information on spatial positioning of vibrissa segments.	56
3.9	(Left) Cross-sectional view of harbour seal vibrissa with Scanning Electron Microscope (SEM) at 70x magnification, (Right) Close up view on the hole in the vibrissa at 1000x magnification.	58
3.10	Variation on natural frequencies of long vibrissae with modulus of elasticity and inner cavity.	60
3.11	The variation of the bending stiffness (EI) of the vibrissae along their lengths. Each vibrissa segment is plotted with the variations of their bending stiffness at varying frequencies (from 1 Hz to 150 Hz).	61
3.12	Comparison between the tip displacements for different modes of vibrations at resonance of the seal vibrissa and rat vibrissa. (Top Left) First mode resonance of the long harbour seal vibrissa, (Top Middle) Second mode resonance of the long harbour seal vibrissa, (Top Right) Third mode resonance of the long harbour seal vibrissa. (Middle Left) First mode resonance of the medium length harbour seal vibrissa, (Middle Right) Second mode of resonance of the medium length harbour seal vibrissa.	62
4.1	Interaction between vibrissa and dipole sheet strength for caudal swimming fish. A and f are the amplitude and frequency of oscillation of the vibrissa due to vibrissa-fish wakes interaction [107].	71
4.2	Detailed view on the boundary condition at the base of the vibrissae [26].	72
4.3	Simplified sketch of the vibrissa towing experimental setup.	74
4.4	Details of parts and mountings of vibrissa-sensor setup consisting of half-muzzle, holding material, vibrissa model, bend sensors and vibrissa cap cap: a) Overview of vibrissa-sensor setup on the 3D-printed half-muzzle of the seal; b) Detailed view of the layout of the four bend sensors setup with the vibrissa cap passively connecting the four bend sensors; c) Overview of the mounting of the vibrissa model on the vibrissa cap, where the four bend sensors are attached to; d) Close-up view of the attachment at the coin-end of the vibrissa model to the vibrissa cap and the attachment of bend sensors on the vibrissa cap; e) Close-up view of the uncapped coin edge at the base of the vibrissa model with holding materials located near the base of the vibrissa model on the 3D-printed half-muzzle of the seal.	78
4.5	Calibration setup of bend sensors with actuation by linear-stage.	80
4.6	Raw signals of bend sensors obtained during calibration setup with linear-stage.	81

4.7	Filtered signals of bend sensors obtained during calibration setup with linear-stage. Butterworth filter is used to filter the raw signals with passband of 20 Hz and stopband of 40 Hz. The passband ripple of the filter is limited to 1 dB with stopband-attenuation of 80 dB (refer to figure 4.6 for raw signals).	82
4.8	Calibration plot for bend sensors with cubic polynomial fitting [Cubic equation is presented in the figure]. Green cross - data points from steady filtered signal (refer to figure 4.7; magenta line - cubic polynomial fitting curve.	83
4.9	Fundamental vibrissa models for evaluating the geometrical effects on the fluid-structure interaction in the towing vibrissa model experiments. A) straight circular cylinder, b) elliptical curved cylinder, c) straight vibrissa, d) curved (true) vibrissa. .	84
4.10	Holding materials in towing vibrissa experimental setup as the model of the muscle at the base of the harbour seal vibrissae. From left to right: soft, medium and hard (pink) holding materials. a) Top side of holding material, b) bottom side of holding material (not visible in figure 4.4e	85
4.11	Variations of modulus of elasticity of the holding materials (soft, medium, and hard) at various frequency of force applications for DMA experiment with compression plates.	87
4.12	Vibrissa model towing experimental setup: a) side view of experimental setup, b) close-up view on the mounting of the vibrissa model-sensor setup on linear stage. . .	88
4.13	Sketch of the vibrissa model-sensor trailing a wake-generator circular cylinder experimental setup with indicators of streamwise and transverse positioning with respect to the structures. It should be noted that the transverse position of the vibrissa model is adjustable through the use of extensor.	90
4.14	Comparisons of the experimental data on frequency response ratio for towing cantilevered uniform circular cylinder ($m^* = 1.2$, where m^* is mass ratio, as defined in equation 4.13) with the three different holding materials to the previous study ($m^*=2.4$) [51]. f^* is the frequency response ratio, defined in equation 4.11 and U^* is the reduced velocity, defined by equation 4.12	91
4.15	Schematic diagram of beam for formulation of the Euler-Lagrange beam equation . .	93
4.16	Comparison between the present experimental VIV data of the circular cylinder model with holding material at its base ($m^*=1.04$) and the previous study of the towing of a uniform circular cylinder ($m^*=1.2$) [52].	97
4.17	Normalised response frequency of four of the vibrissa models to VIV at three reduced velocities for each vibrissa model (corresponding to towing speed of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material and blue - circular cylinder, red - curved elliptical cylinder, green - straight vibrissa, magenta - curved vibrissa.	98
4.18	Amplitude Ratio of VIV of the vibrissa models for the three reduced velocities of each vibrissa models (corresponding to towing velocities of 100 mm/s, 200 mm/s, 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material and blue - circular cylinder, red - curved elliptical cylinder, green - straight vibrissa, magenta - curved vibrissa.	99
4.19	Ratio of Amplitudes of VIV of the vibrissa models trailing a ten times larger circular cylinder traveling at equivalent nominal speeds for the three reduced velocities of each vibrissa models (corresponding to towing velocities of 100 mm/s, 200 mm/s, 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material and blue - circular cylinder, red - curved elliptical cylinder, green - straight vibrissa, magenta - curved vibrissa.	102

4.20	Normalised dominant response frequency of the towed circular cylinder trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) -medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.	105
4.21	Normalised dominant response frequency of the towed curved elliptical cylinder trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) -medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.	106
4.22	Normalised dominant response frequency of the towed straight vibrissa model trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) -medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.	107
4.23	Normalised dominant response frequency of the towed curved vibrissa model trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) -medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.	108
5.1	Two-domain problem definition in Boundary Data Immersion Method (BDIM), Ω_b : solid domain, Ω_f : fluid domain, Ω_s : solid to fluid interfacial domain, 2ϵ : the width of the solid-fluid interfacial region.	111
5.2	Behaviour of the kernel integrals, δ_ϵ for a 1D case. For points $x \leq -\epsilon$ within Ω_b , the author has $\delta_\epsilon^B = 1$ and $\delta_\epsilon^F, \delta_\epsilon^S = 0$. Similarly, for $x > \epsilon$, the author has $\delta_\epsilon^F = 1$ and $\delta_\epsilon^B, \delta_\epsilon^S = 0$. Within the smoothing region, points less than ϵ from σ_s , $\delta_\epsilon^F, \delta_\epsilon^B, \epsilon\delta_\epsilon^S = O(1)$	113
5.3	Simulation domain for stationary single-body simulations. The freestream flow is parallel to the major axis of the structure. The crossflow width on each side of the domain is 4.5D, where D is the characteristic diameter of the structure (defined as the diameter of the structure that is perpendicular to the freestream flow). The lengths of the domain upstream and downstream of the body are 4.0D and 10.0D, respectively. The height of the simulation domain is 12.2D.	115
5.4	Top view of mass-spring-dashpot system for implementation of free motion on a structure (m) where k is the spring constant, and B is the damping coefficient. However, as the simulated body is freely vibrating, the damping coefficient (B) is zero. The spring and dashpot system is applied at the centre of mass of the structure, which prevents the introduction of moment on the body.	119

5.5	Simulation domain for tandem structures with no relative velocity between the structures. The freestream flow is parallel to the major axis of the structure. The crossflow width on each side of the domain is $5.5D_1$, where D_1 and D_2 are the characteristic diameters of the upstream and downstream structure, respectively (defined as the diameter of the structure that is perpendicular to the freestream flow). The lengths of the domain are divided into three sections, with the most upstream section at $4.0D_1$ (from the center of the upstream structure) and most downstream section at $4.5D_1$ (from the trailing edge of the trailing structure). A constant gap of $4D_1$ is maintained between the trailing edge of the upstream structure and the leading edge of the trailing structure. The height of the simulation domain is $12.2D_2$	120
5.6	Simulation domain for tandem structures with no relative velocity between the structures. The freestream flow is parallel to the major axis of the structure. The crossflow width on each side of the domain is $5.5D_1$, where D_1 and D_2 are the characteristic diameters of the upstream and downstream structure, respectively (defined as the diameter of the structure that is perpendicular to the freestream flow). The lengths of the domain are divided into three sections, with the most upstream section at $4.0D_1$ (from the center of the upstream structure) and most downstream section at $4.5D_1$ (from the trailing edge of the trailing structure). A constant gap of $4D_1$ is maintained between the trailing edge of the upstream structure and the leading edge of the trailing structure. The height of the simulation domain is $12.2D_2$	124
5.7	Mesh of the simulation domain for single body simulation. Grid sizes in streamwise, crossflow and spanwise directions are 0.05, 0.05, and $0.1D$, respectively (refer to table 5.1 for grid refinement). Fine-grid area can be identified through the region of darker coloration.	126
5.8	Mesh of the simulation domain for approaching and distancing vibrissa to the wake-generator circular cylinder simulation. Grid sizes in streamwise, crossflow and spanwise directions are 0.05, 0.05, and $0.1 D_2$, where D_2 is the characteristic diameter of the trailing structure ($D_2 = D_1 = D$ for two tandem structures of equivalent characteristic diameter). Fine grid area can be identified through the regions of darker coloration.	128
5.9	Isometric view (Top) and Top View (Bottom) of coherent structure (vortex cores) in the steady wake of a freely vibrating in transverse direction circular cylinder at $Re = 500$ for grid sizes based on (Left) Schematic 1 for the top three-quarter of the structure and Schematic 2 for the rest and (Right) Schematic 2 for the entire spanwise length of the structure (refer to table 5.1 for details on the schematic).	129
5.10	Time series of the force responses of a circular cylinder (Left) and an old vibrissa (Right) in flow around the structures at $Re = 500$ (separate simulation for each structure) up to simulation time, $t^* = 3000$. Each structure is stationary relative to the freestream flow. Top: Drag Coefficients (C_D); Bottom: Lift Coefficients (C_L).	133
5.11	Time series of the force responses of a circular cylinder (Left) and an old vibrissa (Right) in flow around the structures at $Re = 500$ (separate simulation for each structure) up to simulation time, $t^* = 500$. Each structure is stationary relative to the freestream flow. Top: Drag Coefficients (C_D); Bottom: Lift Coefficients (C_L).	134
5.12	Motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow at $Re = 500$ and $U^* = 5$. Left : up to $t^* = 1900$; Right : up to $t^* = 500$	134
5.13	Motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow at $Re = 500$ and $m^* = 1.04$ for various reduced velocity. Top: Steady-amplitude ratio (A/D); Bottom: dominant response frequency ratio [13].	135

5.14	Motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow for various reduced velocity, Reynolds Number and mass ratio. Top: Steady-amplitude ratio (A/D); Bottom: dominant response frequency ratio. The first, second and third signs in the legend represents our simulation (first and second signs) results and experimental results (third sign, conducted by Heather Rachel Beem, MIT Tow Tank), respectively.	137
5.15	Comparison of the topology of vortical structures for a mixing layer with: a) iso-pressure surface based on minimum pressure identification, b) iso-surface of $\lambda_2 = 0$ criterion [48].	139
5.16	Stable von <i>Kármán</i> street in the wake of a stationary circular cylinder trailed by another circular cylinder (far right) identified at $Re = 500$ with λ_2 criterion.	141
5.17	Variation of simulation time step with simulation time as a function of Courant-Friedrich-Levy number [21]. This simulation is performed on the flow around stationary circular cylinder at $Re = 500$	144
5.18	Phase plot of the lift coefficient of a stationary circular cylinder in a freestream flow at $Re = 500$. From black to bright red marker $\sim$ from initial simulation time to the end of simulation time.	145
5.19	Poincare map of the lift coefficient of a stationary circular cylinder in a freestream flow at $Re = 500$. From black to bright red $\sim$ from initial simulation time to the end of simulation time.	145
6.1	Simulated geometries for studies on the effects of geometrical features on the VIV reduction properties of vibrissa-like geometries. Black-line: the profile of the vibrissa-like geometry in major axis (straight or undulatory), Red-line: the profile of the vibrissa like geometry in minor axis (straight or undulatory). For reference comparison, (I) circular cylinder and (II) elliptical cylinder are simulated. Vibrissa-like geometries with all-but-one feature of the harbour seal vibrissa: (III) vibrissa-like geometry with no major axis undulations (Axis Ratio = $D_{min}/D_{maj} = 0.44$, where D_{min} and D_{maj} are average minor and major diameters of the vibrissa, respectively), (IV) vibrissa-like geometry with no major axis undulations (Axis Ratio = 0.54), (V) vibrissa-like geometry with no minor axis undulations, (VI) vibrissa-like geometry with no offset angle, (VII) vibrissa-like geometry with constant offset angle of 15.27° , (VIII) vibrissa-like geometry with constant offset angle of 17.6° . True vibrissa geometry: (IX) Old vibrissa geometry, (X) new vibrissa geometry.	149
6.2	Quantitative measures of the force coefficients to their corresponding geometries for simulations of flow around stationary geometries at $Re = 500$ (for legend on geometries, refer to figure ?? where the number for each geometry correspond to its Roman numerals-equivalent in figure ??). (Top Left) Mean Drag Coefficient ($\bar{C}_D$) of the simulated geometries, (Top Right) Normalised standard deviation (relative to RMS value, not mean, of the lift coefficient) to RMS value (ϕ_{rms}) for the simulated geometries, (Bottom Left) RMS Lift Coefficient ($C_{L,RMS}$) for simulated geometries, (Bottom Right) Zoomed-in plot on the $C_{L,RMS}$ for geometry 6 to 10.	151
6.3	Top view of the pressure coefficient (refer to equation 6.1) and streamwise velocity profiles of a stationary circular cylinder (Top) and elliptical cylinder (Bottom) in a freestream flow at $Re = 500$ during their steady period of response. Left (top and bottom): Pressure profile ; Right (top and bottom) : Streamwise velocity profile. Note: 'p' in the legend indicates pressure coefficient and 'u' indicates the normalised streamwise flow velocity (freestream flow velocity, $U = 1$).	152

- 6.4 Top view of the pressure coefficient (refer to equation 6.1) profiles of a vibrissa-like geometry with no minor axis undulations in a freestream flow at $Re = 500$ during their steady period of response. As there are offset angle between the leading edge and trailing edge of the major axis, four planes are presented. Top left : plane of widest cross section (node) at the peak of the leading edge ; Top right : plane of widest cross section (node) at the peak of the trailing edge ;Bottom left : plane of narrowest cross section (saddle) at the trough of the leading edge ;Bottom right : plane of cross section (node) at the trough of the leading edge. Note: 'p' in the legend indicates pressure coefficient. 154
- 6.5 Top view of the pressure coefficient (refer to equation 6.1) profiles of a vibrissa-like geometry with no major axis undulations and axis ratio of 0.54 in a freestream flow at $Re = 500$ during their steady period of response. Left : Nodal plane (widest cross section) ; Right : Saddle plane (narrowest cross section). Note: 'p' in the legend indicates pressure coefficient. 155
- 6.6 Spanwise view of the pressure coefficient (refer to equation 6.1) profiles of a geometry 5 (left) and geometry 4 (right), refer to figure 6.1, in the plane of symmetry (through the centroid of the geometries) for a freestream flow at $Re = 500$ during their steady period of response. Note: 'p' in the legend indicates pressure coefficient. 156
- 6.7 in the plane of symmetry (through the centroid of the geometries) for a freestream flow at $Re = 500$ during their steady period of response. Note: 'p' in the legend indicates pressure coefficient. 157
- 6.8 Isometric view of the spanwise flow velocity (w) of the flow around a stationary elliptical cylinder (Top) and new vibrissa geometry (Bottom) at $Re = 500$. The plane perpendicular to y-axis is located about the centroid of the geometries and the plane perpendicular to the x-axis is chosen to reflect the location of the largest spanwise flow velocity. Note: 'w' in the legend indicates normalised spanwise velocity, defined in equation 6.2. 159
- 6.9 Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.0 (at trailing edge of vibrissa), 2.4, 3.45, 4.15, 4.85, 5.9 ; bottom-ow from left to bottom right : $x/D = 6.25, 6.6, 7.3, 8.35, 9.05, 10.1$, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry). 160
- 6.10 Isometric view of the spanwise flow velocity (w) of the flow around a stationary old vibrissa geometry at $Re = 500$. The plane perpendicular to y-axis is located about the centroid of the geometries and the plane perpendicular to the x-axis is chosen to reflect the location of the largest spanwise flow velocity. Note: 'w' in the legend indicates normalised spanwise velocity, defined in equation 6.2. 161

6.11	Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.14 (near trailing edge of old vibrissa model), 1.52, 1.9, 3.04, 3.8 ; middle-row from left to right : $x/D = 4.56, 5.7, 6.46, 6.84, 7.22, 7.98$; bottom-row from left to right : 8.74, 9.12, 9.88, 10.64, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).	162
6.12	Lowest-pressure planes (perpendicular to the freestream flow) in the wake of new vibrissa geometry (left) and old vibrissa geometry (right) for flow around the respective geometries at $Re = 500$	163
6.13	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a circular cylinder and elliptical cylinder at $Re = 500$ during steady period of force response. Top and bottom left : Isometric and top views of vortex cores for flow around a circular cylinder, respectively ; Top and bottom right : Isometric and top views of vortex cores for flow around an elliptical cylinder.	165
6.14	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a vibrissa-like geometry with no major axis undulations of axis ratio 0.44 (geometry 3) and a vibrissa-like geometry with no major axis undulations of axis ratio 0.54 (geometry 4) at $Re = 500$ during steady period of force response. Top and bottom left : Isometric and top views of vortex cores for flow around a geometry 3, respectively ; Top and bottom right : Isometric and top views of vortex cores for flow around a geometry 4.	167
6.15	Side view of the coherent structures (identification based on λ_2 criterion) of flow around geometry 3 (left) and 4 (right) at $Re = 500$ during their steady period of force response.	168
6.16	Pressure coefficient profiles of flow around a vibrissa-like geometry with no major axis undulation for axis ratio of 0.44 (geometry 3) at the nodal plane (left) and saddle plane(right) of the geometry for freestream flow at $Re = 500$. Top figures : geometry 3 ; bottom figures : geometry 4 - reprint from figure 6.5 for easier comparison.	168
6.17	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a vibrissa-like geometry with no offset angle (geometry 6) and a vibrissa-like geometry with constant offset angle of 17.6° (geometry 8) at $Re = 500$ during steady period of force response. Top, middle and bottom left : Isometric, side and top views of vortex cores for flow around a geometry 6, respectively ; Top, middle and bottom right : Isometric, side and top views of vortex cores for flow around a geometry 8.	170
6.18	Pressure profiles of flow around geometry 6 (vibrissa-like geometry with no offset angle) at the nodal (left) and saddle planes (right) for $Re = 500$	171
6.19	Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.09 (near trailing edge of old vibrissa model), 1.36, 2.17, 2.99, 3.53, 4.07 ; middle-row from left to right : $x/D = 4.62, 4.89, 5.43, 5.97, 6.24, 6.52, 7.33$; bottom-row from left to right : 7.87, 8.42, 8.69, 9.23, 9.50, 10.32, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).	172

6.20	Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.09 (near trailing edge of old vibrissa model), 1.36, 1.63, 1.90, 2.99, 3.80, 4.07 ; middle-row from left to right : $x/D = 4.34, 4.62, 5.70, 5.97, 6.24, 6.79, 7.33$; bottom-row from left to right : 7.87, 8.15, 8.42, 8.96, 9.50, 10.05, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).	173
6.21	Depiction of vortex cores (identification based on λ_2 criterion) of flow around the old vibrissa geometry (geometry 9) at $Re = 500$ during steady period of force response. Top left : Isometric view, top right : side view, bottom : top view.	175
6.22	Depiction of vortex cores (identification based on λ_2 criterion) of flow around the new vibrissa geometry (geometry 10) at $Re = 500$ during steady period of force response. Top left : Isometric view, top right : side view, bottom : top view.	176
6.23	Pressure profiles of flow around the old vibrissa geometry at $Re = 500$ during its steady period of oscillation. Top left : top view of nodal plane at the peak of leading edge; top right : top view of nodal plane at the peak of trailing edge ; middle left : top view of saddle plane at the trough of leading edge ; middle right : top view of saddle plane at the trough of trailing edge ; bottom : side view of the plane of symmetry about the centroid of the geometry.	177
6.24	Isometric view of the iso-surface of the crossflow velocity of the old (left) and new (right) vibrissa geometries during their steady period of oscillation for flow at $Re = 500$	179
6.25	Direction of rotation of vortical braids based on tangential velocity vector on the plane-cut perpendicular to the streamwise (x -axis) in the wake of a circular cylinder. Grey areas in top right and bottom figures indicated the vortex cores based on identification with λ_2 criterion and arrows indicated the tangential velocity vectors. The planes are lightly transparent, as such, the light grey region in the background of top right and bottom figures represented the circular cylinder (rear-view). Top left : Isometric view of coherent structures in the wake of the circular cylinder, the vertical line indicates the location of plane-cut (perpendicular to x -axis) and the horizontal lines acted as the separator of the the spanwise zone (top, middle and left) ; top right : vortical braids and tangential velocity vector in the 'top' zone of the plane-cut; bottom left : vortical braids and tangential velocity vector in the 'middle' zone of the plane-cut ; bottom right : vortical braids and tangential velocity vector in the 'bottom' zone of the plane-cut.	180
6.26	Illustration of the vortex cores (based on λ_2 criterion) of the new vibrissa geometry in a freestream flow at $Re = 500$ during its steady period of response ($t^* = 219.1$). Left: rear-view vortex cores ; Middle : Isometric view of the vortex cores and the location of plane-cut (perpendicular to streamwise) ; Right : view of the contours in the plane cut (orange region - vortex cores).	182

6.27	Enlarged view of the first complete bulk of vortical braids in figure 6.26 from the plane-cut (perpendicular to streamwise) through the midst of the vortical braids. The planes are lightly transparent with the new vibrissa geometry lightly visible in the background (located in the middle of each figure), rear-view. Legend: orange area - vortex cores, black arrows - tangential velocity vectors. Top left : plot of a single bulk of vortical braids in the plane-cut ; top right : plot of vortex cores and tangential velocity vector in the central region of bulk of the vortical braids ; bottom left : top half plot of the vortex cores and tangential velocity vector in the top half of the bulk of the vortical braids ; bottom right : bottom half plot of the vortex cores and tangential velocity vector in the top half of the bulk of the vortical braids.	183
6.28	Sketch of the polarities of the vortical braids observed in a single period of undulation of the new vibrissa geometry. Legend : orange area - vortex cores.	184
6.29	Depiction of coherent structures in the wake of geometry 7 (vibrissa-like geometry with constant offset angle 15.6°) at simulation time, $t^* = 221$, at $x/D = 6.67$. top right : side view of the vortex cores and the location of plane-cut (perpendicular to streamwise) ; top right : overview of the vortex cores and tangential velocity vector in the plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.	186
6.30	Depiction of coherent structures in the wake of geometry 7 (vibrissa-like geometry with constant offset angle 15.6°) at simulation time, $t^* = 219$, at $x/D = 4.9$ (left) [and 5.7 - (right), details in figure 6.31)], where x is measured from the centroid of the geometry . top right : side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.	187
6.31	Depiction of coherent structures in the wake of geometry 7 (vibrissa-like geometry with constant offset angle 15.6°) at simulation time, $t^* = 219$, at $x/D = 5.7$ (refer to top left figure in figure 6.30), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores and tangential velocity vector in the right plane-cut ; top right : enlarged view of the top third of the vortex cores and tangential velocity vector in the right plane-cut ; bottom left : enlarged view of the middle third of the vortex cores and tangential velocity vector in the right plane-cut ; bottom right : enlarged view of the lower third of the vortex cores and tangential velocity vector in the right plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.	188
6.32	Power Spectral Density (PSD) plot of the lift force (blue – viscous, red – pressure and green – total) on a stationary circular cylinder at $Re = 500$	190
6.33	Sketch of the polarities of the vortical braids observed in a single period of undulation of the new vibrissa geometry. Legend : orange area - vortex cores.	190
6.34	VIV response of geometry 2 to 10 near their resonance condition (such condition is achieved through equivalence of the natural frequency of the system to the frequency of vortex shedding of the corresponding geometries in stationary simulations at similar $Re = 500$). Top : amplitude ratio of VIV ; middle : dominant response frequency ratio (f_r/f_n) ; bottom : ratio of the dominant frequency response to the natural frequency of system.	196

6.35	Arrangement of coherent structures ('P+S' arrangement) in the wake of a freely vibrating in crossflow circular cylinder (Left) during its amplitude of motion in positive Y direction at its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 5.2$ and simulation time, $t^* = 224$ and stationary circular cylinder (Right) [reprint from figure 6.13 for ease of comparison]. Top: Isometric view ; middle : side view ; bottom : top view.	198
6.36	Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder, $t^* = 224$, at $x/D = 6.2$ (refer to top left figure in figure 6.35), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores and tangential velocity vector in the plane-cut ; top right : enlarged view of the top third of the vortex cores and tangential velocity vector in the plane-cut ; bottom left : enlarged view of the middle third of the vortex cores and tangential velocity vector in the plane-cut ; bottom right : enlarged view of the lower third of the vortex cores and tangential velocity vector in the plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores.	199
6.37	Top view depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder, $t^* = 224$, in a plane-cut (perpendicular to spanwise) at $z/D = 8.7$ (refer to top left figure in figure 6.35), where z is measured from the base of the geometry. Legend: cyan area denotes cross-sectional area of vortex cores.	200
6.38	Arrangement of coherent structures in the wake of a freely vibrating in crossflow elliptical cylinder (Left) during its amplitude of motion in negative Y direction of its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 4.81$ and simulation time, $t^* = 274$ and stationary elliptical cylinder (Right) [reprint from figure 6.13 for ease of comparison]. Top: Isometric view ; bottom : top view.	202
6.39	Pressure profiles in the wake of a freely vibrating in crossflow circular cylinder (left) and elliptical cylinder (right) at $U^* = 5.2$ and $U^* = 4.81$. Top : freely vibrating body ; bottom : stationary body (reprint from figure 6.3).	203
6.40	Arrangement of coherent structures in the wake of a freely vibrating in crossflow geometry 4 (Left) during its amplitude of motion in negative Y direction at its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 4.76$ and simulation time, $t^* = 185$ and stationary circular cylinder (Right) [reprint from figures 6.14, 6.15 for ease of comparison]. Top: Isometric view ; middle : side view ; bottom : top view. . .	205
6.41	Side view of the alternate arrangement of coherent structures in the wake of a freely vibrating in crossflow geometry 4 during its amplitude of motion in negative Y direction at its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 4.76$ and simulation time, $t^* = 251$	206
6.42	Pressure profiles in the wake of freely vibrating in crossflow geometry 4 at $t^* = 185$ (left), 216 (right) for $Re = 500$. top right : side view about the plane of symmetry through the centroid of geometry 4 at neutral position (perpendicular to crossflow) for $t^* = 185$; top right : side view about the plane of symmetry through the centroid of geometry 4 at neutral position for $t^* = 251$; middle left : top view of the pressure profiles in the nodal plane of geometry at $t^* = 185$; middle right : top view of the pressure profiles in the nodal plane of geometry at $t^* = 251$; bottom left : top view of the pressure profiles in the saddle plane of geometry at $t^* = 185$; bottom right : top view of the pressure profiles in the saddle plane of geometry at $t^* = 251$	207

6.43	Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 4 in the plane-cut (perpendicular to streamwise) for U-shaped vortical braids arrangement (top), refer to figure 6.40, and ribbed-U-shaped vortical braids arrangement (bottom), refer to figure 6.41. Top left: cross-section of vortex cores at a plane-cut $x/D = 6.3$ at $t^* = 185$; top right: enlarged view of one period of U-shaped vortical braids in the plane-cut $x/D = 6.3$ at $t^* = 185$; bottom left: cross-section of vortex cores at a plane-cut $x/D = 7.1$ at $t^* = 251$; bottom right: enlarged view of two periods of U-shaped vortical braids in the plane-cut $x/D = 7.1$ at $t^* = 251$. Legend: orange areas denote vortex cores.	209
6.44	Depiction of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 5 (vibrissa-like geometry with no minor axis undulations) at $Re = 500$, $t^* = 200$, and $U^* = 5.22$. Top left : isometric view ; top right : side view ; bottom : top view.	211
6.45	Pressure profiles in the wake of freely vibrating in crossflow geometry 5 at $t^* = 199$ (top left figure only) , 200 for $Re = 500$. top right : side view about the plane of symmetry through the centroid of geometry 5 at neutral position (perpendicular to crossflow) for $t^* = 199$; top right : side view about the plane of symmetry through the centroid of geometry 5 at neutral position ; middle left : top view of the pressure profiles in the nodal plane at the peak of leading edge ; middle right : top view of the pressure profiles in the nodal plane at the peak of trailing edge ; bottom left : top view of the pressure profiles in the saddle plane at the valley of leading edge ; bottom right : top view of the pressure profiles in the saddle plane at the valley of trailing edge.	212
6.46	Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 5 in the plane-cut (perpendicular to streamwise) at $x/D = 6.8$, $t^* = 200$ and $U^* = 5.22$. Top left: spanwise overview ; top right: enlarged view of the first one and a half period of undulations ; bottom left: enlarged view of the subsequent 1.25 period of undulations ; bottom right: enlarged view of the final 1.25 period of undulations. Legend: orange area denotes vortex cores.	214
6.47	Perspective view [45° rotation from streamwise direction about Z-axis (spanwise)] of geometry 5 and the corresponding plane-cut at $x/D = 6.8$ (refer to figure 6.46) depicting the cross-section of the vortex cores and tangential velocity vector of the plane.	215
6.48	Depiction of the side view of vortex cores (based on λ_2 criterion) in the wake of various geometries. Top left : geometry 6 (vibrissa-like geometry with no offset angle) at $t^* = 187$ and $U^* = 5.23$; top right : geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) at $t^* = 213$ and $U^* = 5.78$; bottom : geometry 8 (vibrissa-like geometry with constant offset angle of 17.6°) at $t^* = 211$ and $U^* = 5.14$.	217
6.49	Pressure profiles in the wake of freely vibrating in crossflow geometry 7 at $t^* = 213$ for $Re = 500$ and $U^* = 5.23$. Top left : top view of the pressure profiles in the nodal plane at the peak of leading edge ; top right : top view of the pressure profiles in the nodal plane at the peak of trailing edge ; middle left : top view of the pressure profiles in the saddle plane at the valley of leading edge ; middle right : top view of the pressure profiles in the saddle plane at the valley of trailing edge ; bottom : side view of the pressure profiles about the plane of symmetry of the body at neutral position (perpendicular to crossflow direction).	219
6.50	Spanwise flow velocity profile for various freely vibrating geometries at freestream flow of $Re = 500$. Left : geometry 4 at $t^* = 185$, $U^* = 4.76$, and $x/D = 3.8$; middle : geometry 5 $t^* = 200$, $U^* = 5.22$, and $x/D = 5.43$; right : geometry 7 at $t^* = 213$, $U^* = 5.78$, and $x/D = 5.97$	222

- 6.51 Rear view of the spanwise flow velocity of the flow around a freely vibrating in crossflow geometry 5 at $Re = 500$, $t^* = 200$, and $U^* = 5.22$ for various spatial positions. All plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry (grey). Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 3.2, 3.8, 4.6, 5.2, 5.6, 6.0, 6.4 ; bottom-row from left to right : 7.2, 7.8, 8.2, 8.6, 9.0, 9.6, where D is the characteristic diameter in the simulation (minor diameter of vibrissa-like geometry). . 223
- 6.52 Rear view of the spanwise flow velocity of the flow around a freely vibrating in crossflow geometry 7 at $Re = 500$, $t^* = 213$, and $U^* = 5.78$ for various spatial positions. All plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry (grey). Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.8, 2.4, 3.2, 4.0, 4.4, 4.8 ; middle-row from left to right : $x/D = 5.4, 6.0, 6.4, 7.0, 7.4, 7.6, 7.8$; bottom-row from left to right : 8.2, 8.4, 8.8, 9.6, 10.0, 10.4, where D is the characteristic diameter in the simulation (minor diameter of vibrissa-like geometry). 224
- 6.53 Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 7 in the plane-cut (perpendicular to streamwise) at $x/D = 7.1, t^* = 213$ and $U^* = 5.78$. Top left: spanwise overview ; top right: enlarged view of the first one and a quarter period of undulations ; bottom left: enlarged view of the subsequent 1.5 period of undulations ; bottom right: enlarged view of the final 1.25 period of undulations. Legend: orange area denotes vortex cores. 225
- 6.54 Side view of the vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 7 at $Re = 500, t^* = 265$ and $U^* = 5.78$ 226
- 6.55 Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 7 in the plane-cut (perpendicular to streamwise) at $x/D = 7.3, t^* = 265$ and $U^* = 5.78$. Top left: spanwise overview ; top right: enlarged view of the first one and a half period of undulations ; bottom left: enlarged view of the subsequent 1.25 period of undulations ; bottom right: enlarged view of the final 1.25 period of undulations. Legend: orange area denotes vortex cores. 227
- 6.56 Depiction of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 9 (old vibrissa geometry) at $Re = 500, t^* = 171$, and $U^* = 4.73$. Top left : isometric view ; top right : side view ; bottom : top view. 228
- 6.57 Depiction of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 10 (new vibrissa geometry, left) at $Re = 500, t^* = 194$, and $U^* = 5.97$ and stationary new vibrissa geometry (right, reprint from figure 6.22). Top left : isometric view ; top right : side view ; bottom : top view. 230
- 6.58 Translucent iso-surface of the pressure profiles in the wake of a freely vibrating geometry 10 at $Re = 500, t^* = 194$, and $U^* = 5.97$. Top left : isometric view ; top right : side view ; bottom : top view. 231
- 6.59 Rear view of the spanwise flow velocity of the flow around a freely vibrating in crossflow geometry 10 at $Re = 500, t^* = 194$, and $U^* = 5.97$ for various spatial positions. All plots share similar colouring legend as the bottom right-most figure. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 2.6, 3.8, 4.6, 5.0, 5.6, 6.0 ; bottom-row from left to right : $x/D = 6.6, 7.2, 8.2, 8.6, 9.4, 10.2$, where D is the characteristic diameter in the simulation (minor diameter of vibrissa-like geometry). . 233

6.60	Vortex cores (based on λ_2 criterion) in the wake of a freely vibrating new vibrissa geometry for $Re = 500$, $t^* = 194$, and $U^* = 5.97$. From left to right : side view of the vortex cores and location of plane-cuts (perpendicular to streamwise direction), $x/D = 6.6, 8.9$, and 11.8 ; cross-sectional view of vortex cores in plane-cut at $x/D = 6.6$; cross-sectional view of vortex cores in plane-cut at $x/D = 8.9$; cross-sectional view of vortex cores in plane-cut at $x/D = 11.8$	234
6.61	Vortex cores and tangential velocity vectors of the new vibrissa geometry in the plane-cut at $x/d = 8.9$ for bulk of vortical braids in the first nodal plane of minor axis (left) and second nodal plane of minor axis (right), also the mid-span of the geometry. Top : overview of bulk of vortical braids ; middle : enlarged view of the upper half of the bulk of vortical braids ; bottom enlarged view of the lower half of the bulk of the vortical braids.	235
6.62	Responses of various geometries (refer to figure 6.1) to flow around these freely vibrating in streamwise geometries at their respective near-resonance conditions $Re = 500$, $m^* = 1.04$ and $f_{nx}/f_{ny} = 2.0$. Top left : streamwise and crossflow amplitudes ; top right : streamwise and crossflow frequency responses ; bottom left : normalised streamwise and crossflow response frequencies ; bottom right : ratio of streamwise to crossflow response frequencies.	237
6.63	Vortex cores (based on λ_2 criterion) in the wake of a freely vibrating in streamwise and crossflow circular cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$. Top left : isometric ; top right : side view ; bottom : top view.	239
6.64	Vortex cores and tangential velocity vectors in the wake of a 2D vibrating circular cylinder at $Re = 500$, $U^* = 5.2$, and $t^* = 246$. Top left : cross-sectional view (perpendicular to streamwise) at $x/D = 2.7$ (across vortical braids) ; top right : enlarged view of the cross-sectional view corresponding to figure on the left ; bottom left : cross-sectional view (perpendicular to crossflow) about the plane of symmetry of the geometry at the neutral axis (across vortex tube) ; bottom right : enlarged view of the cross-sectional view corresponding to figure on the left.	240
6.65	Pressure field in the wake of a freely vibrating in streamwise and crossflow circular cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$. Top left : isometric ; top right : side view ; bottom : top view.	241
6.66	Motion trajectory of a 2D vibrating circular cylinder during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Legend: Darker green - earlier time, brighter green -later time.	241
6.67	Vortex cores (based on λ_2 criterion) in the wake of a freely vibrating in streamwise and crossflow elliptical cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$. Top left : isometric ; top right : side view ; bottom : top view.	243
6.68	Isometric view of the pressure field (left) and streamwise velocity (right) contours in the wake of a freely vibrating in streamwise and crossflow elliptical cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$	243
6.69	Motion trajectory of a 2D vibrating elliptical cylinder during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Legend: Darker green - earlier time, brighter green -later time.	244
6.70	Arrangement of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) at $Re = 500$, $U_y^* = 4.8$, and $t^* = 234$ (top), 246 (middle), and 284 (bottom). Left: isometric view ; right : side view.	246

6.71	Added mass coefficients and force coefficients in phase with velocities for 1D freely vibrating (crossflow) and 2D freely vibrating (streamwise and crossflow) geometries. Legend: blue cross - added mass coefficient of crossflow motion for 2D freely vibrating geometries ; red square - added mass coefficient of streamwise motion for 2D freely vibrating geometries ; black diamond - lift coefficient in phase with velocity of crossflow motion for 2D freely vibrating geometries ; green inverted triangle - lift coefficient in phase with velocity of streamwise motion for 2D freely vibrating geometries ; teal triangle - added mass coefficient of crossflow motion for 1D freely vibrating geometries ; dark yellow circle - lift coefficient in phase with velocity of crossflow motion for 1D freely vibrating geometries.	248
6.72	Motion trajectory of a 2D vibrating geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44 -left) and geometry 4 (vibrissa-like geometry with no minor axis undulations and axis ratio of 0.54 - right) during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Legend: Darker green - earlier time, brighter green -later time.	249
6.73	Motion trajectory of a 2D vibrating geometry 5, 6, 7, 9 and 10 during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Top left: isometric view of motion trajectory of geometry 5 (vibrissa-like geometry with no minor axis undulations) ; top right : top view of corresponding figure on the left ; middle left : isometric view of the motion trajectory of geometry 6 (vibrissa-like geometry with no offset angle) ; middle right : top view of the motion trajectory of geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) ; bottom left : top view of the motion trajectory of geometry 9 (old vibrissa geometry) ; bottom right : top view of the motion trajectory of geometry 10 (new vibrissa geometry). Legend: Darker green - earlier time, brighter green -later time. Note : features of some motion trajectory is better visualised with isometric view, instead of top view (standard). . .	251
6.74	Phase plot (top) and Poincare map (bottom) of the crossflow motion of freely vibrating in crossflow geometries (refer to figure 6.1 for details of each geometry).	254
6.75	Phase plot (top) and Poincare map (bottom) of the crossflow motion of freely vibrating in streamwise and crossflow geometries (refer to figure 6.1 for details of each geometry).	256
6.76	Phase plot (top) and Poincare map (bottom) of the streamwise motion of freely vibrating in streamwise and crossflow geometries (refer to figure 6.1 for details of each geometry).	258
7.1	Simulation domain for vibrissa geometry at 0° (Top), 45° , (Middle) and 90° (Bottom) angle of attack. By defining the centroid of the geometry in XY plane as point of origin (O), the upstream and downstream lengths (x-axis) of the domain measured from O are 4D and 10D (4.5D and 17D for new vibrissa geometry for longer region of wake visualisation, providing better aspect ratio), respectively, where D is the characteristic diameter of the structure in the simulation. The width of the simulation domain (y-axis) is symmetrical about the x-axis passing through O at 4.5D. The height of the simulation domain is 12.2D (27.1D for new vibrissa geometry).	262
7.2	Time history of Drag and Lift Coefficient of new and old vibrissa geometry. Top left $-C_D$ of new vibrissa geometry ; top right $-C_L$ of new vibrissa geometry ; bottom left $-C_D$ of old vibrissa geometry; bottom right $-C_L$ of old vibrissa geometry. . . .	263
7.3	Power Spectral Density (PSD) plot of the lift coefficient on the new vibrissa geometry (left) and old vibrissa geometry (right). The peak frequencies of the lift coefficients of the new and old vibrissa geometries are 0.1673 and 0.2112, respectively.	264

7.4	Time history of Drag and Lift Coefficient of new and old vibrissa geometry at 45° angle of attack. Top left $-C_D$ of new vibrissa geometry ; top right $-C_L$ of new vibrissa geometry ; bottom left $-C_D$ of old vibrissa geometry; bottom right $-C_L$ of old vibrissa geometry.	266
7.5	Depiction of vortex cores (identification based on λ_2 criterion) of flow around stationary new vibrissa geometry at an angle of attack of 45° at $Re = 500$ and $t^* = 286$. Top left : isometric view ; top right : side view ; bottom : top view.	267
7.6	Depiction of vortex cores (identification based on λ_2 criterion) of flow around stationary new vibrissa geometry at an angle of attack of 45° at $Re = 500$ and $t^* = 286$. Top left : isometric view ; top right : side view ; bottom : top view.	268
7.7	Power Spectral Density (PSD) plot of the drag (left) and lift (right) coefficients on the new vibrissa geometry (top) and old vibrissa geometry (bottom). The peak frequencies of the drag coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively. The peak frequencies of the lift coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively.	269
7.8	Pressure field in the wake of a stationary new vibrissa geometry at an angle of attack of 45° for one-period of force oscillation. Top left : $t^* = 145$; top right : $t^* = 147$; bottom left : $t^* = 148$; bottom right : $t^* = 149$	270
7.9	Enlarged view of the corresponding drag (left) and lift coefficient (right) on the stationary new vibrissa geometry for one-period of force oscillation (corresponding to figure 7.8).	270
7.10	Time history of Drag and Lift Coefficient of new and old vibrissa geometry at 90° angle of attack. Top left $-C_D$ of new vibrissa geometry ; top right $-C_L$ of new vibrissa geometry ; bottom left $-C_D$ of old vibrissa geometry; bottom right $-C_L$ of old vibrissa geometry.	271
7.11	Depiction of vortex cores (identification based on λ_2 criterion) of flow around stationary new vibrissa geometry at an angle of attack of 90° at $Re = 500$ and $t^* = 286$. Top left : isometric view ; top right : side view ; bottom : top view.	272
7.12	Power Spectral Density (PSD) plot of the drag (left) and lift (right) coefficients on the new vibrissa geometry (top) and old vibrissa geometry (bottom). The peak frequencies of the drag coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively. The peak frequencies of the lift coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively.	273
7.13	Characteristic response of the new vibrissa geometries at an angle of attack of 0° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.	275
7.14	Characteristic response of the old vibrissa geometries at an angle of attack of 0° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.	277
7.15	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in crossflow new vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 5$ and $t^* = 194$. Top left : isometric view ; top right : side view ; bottom : top view.	278
7.16	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in streamwise and crossflow new vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 11$ (left) and 17 (right). Top : isometric view ; middle : side view ; bottom : top view.	279

7.17	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in streamwise and crossflow old vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 4$ (left) and 10 (right). Top : isometric view ; middle : side view ; bottom : top view.	281
7.18	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in crossflow old vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 8$ (left) and 17 (right). Top : isometric view ; middle : side view ; bottom : top view.	282
7.19	Characteristic response of the old vibrissa geometries at an angle of attack of 45° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.	284
7.20	Power Spectral Density (PSD) plots of the 1D vibrating and 2D vibrating new vibrissa geometry at an angle of attack of 45° for $Re = 500$ and $U^* = 4$. Top left : streamwise (2D vibrating) [0.2166, 2454, 0.4187] ; top right : crossflow (2D vibrating) [0.2025, 0.2459] ; bottom : streamwise (1D vibrating) [0.2355] . All corresponding frequencies of the peaks are indicated from left to right of the plot on the square bracket of each.	285
7.21	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a 1D vibrating (right) and 2D vibrating (left) new vibrissa geometry at an angle of attack of 45° at $Re = 500$, $U^* = 4$ during their stable period oscillation. Top : isometric view ; middle : side view ; bottom : top view.	288
7.22	Depiction of vortex cores (identification based on λ_2 criterion) of flow around a 2D vibrating (left) new vibrissa geometry at an angle of attack of 45° at $Re = 500$, $U^* = 3$ during their stable period oscillation. Top : isometric view ; middle : side view ; bottom : top view.	289
7.23	Motion trajectory of the 2D vibrating new vibrissa geometry at an angle of attack 45° and $U^* = 3, 4$ and 5.	290
7.24	Characteristic response of the old vibrissa geometries at an angle of attack of 90° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.	291
7.25	Depiction of vortex cores (identification based on λ_2 criterion) of flow around 1D vibrating (left) and 2D vibrating (right) new vibrissa geometry at an angle of attack of 90° at $Re = 500$, $U^* = 4$. Top : isometric view ; middle : side view ; bottom : top view.	293
8.1	Mean Drag coefficient (left) and mean Lift coefficient (right) on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 9600, 14500$, and 19200 [4].	298
8.2	Mean Drag coefficient (left) and mean Lift coefficient (right) on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at various Reynolds Number. Legend : blue circle - $Re = 500$; red square - $Re = 5000$; black diamond - $Re = 10,000$; magenta inverted triangle - $Re = 20,000$; green triangle - $Re = 30,000$	299
8.3	Response frequencies of the drag forcing on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at various Reynolds Number. Legend : blue circle - $Re = 500$; red square - $Re = 5000$; black diamond - $Re = 10,000$; magenta inverted triangle - $Re = 20,000$; green triangle - $Re = 30,000$	301

8.4	Response frequencies of the lift forcing on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at various Reynolds Number. Legend : blue circle - $Re = 500$; red square - $Re = 5000$; black diamond - $Re = 10,000$; magenta inverted triangle - $Re = 20,000$; green triangle - $Re = 30,000$	306
8.5	Mean Drag coefficient (left) and mean Lift coefficient (right) on the stationary sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry) in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 500$ and 5000	307
8.6	Response frequencies of the drag forcing on the stationary sensor structures (type of structure indicated by legend) in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 500$ and 5000 . Legend : blue circle - circular cylinder $Re = 500$; red square - elliptical cylinder at $Re = 500$; black diamond - old vibrissa geometry at $Re = 500$; green inverted triangle - circular cylinder at $Re = 5000$; teal triangle - elliptical cylinder at $Re = 5000$; yellow circle - old vibrissa geometry at $Re = 5000$	309
8.7	Response frequencies of the lift forcing on the stationary sensor structures (type of structure indicated by legend) in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 500$ and 5000 . Legend : blue circle - circular cylinder $Re = 500$; red square - elliptical cylinder at $Re = 500$; black diamond - old vibrissa geometry at $Re = 500$; green inverted triangle - circular cylinder at $Re = 5000$; teal triangle - elliptical cylinder at $Re = 5000$; yellow circle - old vibrissa geometry at $Re = 5000$	310
8.8	RMS amplitude of oscillation of trailing freely vibrating in crossflow sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry, new vibrissa geometry) at $Re = 500$ (based on characteristic diameter of trailing sensor structures) for inline spacing of $4D$ and no offset positioning. Natural frequency of each simulation setting is tuned to the resonance frequency of the respective sensor structure. Legend for trailing sensor structure: blue cross - circular cylinder; red square - elliptical cylinder ; black diamond - old vibrissa geometry at $U^* = 12$; green circle - new vibrissa geometry at $U^* = 5$; dark yellow inverted triangle - old vibrissa geometry at $U^* = 5$	314
8.9	Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von <i>Kármán</i> street) on the interaction between the von <i>Kármán</i> street and a trailing circular cylinder at inline spacing of $4D$ and $CDR = 1.0$. Direction of view: left to right and top to bottom (refer to dimensionless time located at the bottom of each picture, for reference).	315
8.10	Response frequency ratios of oscillation of trailing freely vibrating in crossflow sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry, new vibrissa geometry) at $Re = 500$ (based on characteristic diameter of trailing sensor structures) for inline spacing of $4D$ and no offset positioning. Natural frequency of each simulation setting is tuned to the resonance frequency of the respective sensor structure. Legend for trailing sensor structure: blue cross - circular cylinder; red square - elliptical cylinder ; black diamond - old vibrissa geometry at $U^* = 12$; green inverted triangle - new vibrissa geometry at $U^* = 5$; violet triangle - old vibrissa geometry at $U^* = 5$	318

- 8.11 Scaled response frequency ratios of the oscillation of trailing freely vibrating in cross-flow sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry, new vibrissa geometry) at $Re = 500$ (based on characteristic diameter of trailing sensor structures) for inline spacing of $4D$ and no offset positioning. Natural frequency of each simulation setting is tuned to the resonance frequency of the respective sensor structure. Legend for trailing sensor structure: blue cross - circular cylinder; red square - elliptical cylinder ; black diamond - old vibrissa geometry at $U^* = 12$; green inverted triangle - new vibrissa geometry at $U^* = 5$; violet triangle - old vibrissa geometry at $U^* = 5$ 321
- 8.12 Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von *Kármán* street) on the interaction between the von *Kármán* street and a trailing elliptical cylinder at inline spacing of $4D$ and $CDR = 1.4$. Direction of view: left to right and top to bottom (refer to dimensionless time located at the bottom of each picture, for reference). 322
- 8.13 Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von *Kármán* street) on the interaction between the von *Kármán* street and a trailing new vibrissa geometry at $U^* = 5$, inline spacing of $4D$ and $CDR = 3.0$. Direction of view: left to right and top to bottom (refer to dimensionless time located at the bottom of each picture, for reference). . . 328
- 8.14 RMS amplitude of oscillation of trailing freely vibrating in crossflow new and old vibrissa geometry at $Re = 500$ (based on characteristic diameter of trailing sensor structure) for various reduced velocities (U^*). Legend for trailing sensor structures: 'New' indicates new vibrissa geometry ; 'Old' indicates old vibrissa geometry ; blue cross - new vibrissa at $U^* = 3$; red square - new vibrissa at $U^* = 4$; black diamond - new vibrissa at $U^* = 5$; green inverted triangle - new vibrissa at $U^* = 6$; teal triangle - old vibrissa at $U^* = 3$; dark yellow circle - old vibrissa at $U^* = 4$; violet plus - old vibrissa at $U^* = 5$; grey asterisk - old vibrissa at $U^* = 12$ 329
- 8.15 Response frequency ratios of oscillation of trailing freely vibrating in crossflow new and old vibrissa geometry at $Re = 500$ (based on characteristic diameter of trailing sensor structure) for various reduced velocities (U^*). Legend for trailing sensor structures: 'New' indicates new vibrissa geometry ; 'Old' indicates old vibrissa geometry ; blue cross - new vibrissa at $U^* = 3$; red square - new vibrissa at $U^* = 4$; black diamond - new vibrissa at $U^* = 5$; green inverted triangle - new vibrissa at $U^* = 6$; teal triangle - old vibrissa at $U^* = 3$; dark yellow circle - old vibrissa at $U^* = 4$; violet left-oriented-triangle - old vibrissa at $U^* = 5$; grey right-oriented-triangle - old vibrissa at $U^* = 12$ 330
- 8.16 Scaled response frequency ratios of oscillation of trailing freely vibrating in cross-flow new and old vibrissa geometry at $Re = 500$ (based on characteristic diameter of trailing sensor structure) for various reduced velocities (U^*). Legend for trailing sensor structures: 'New' indicates new vibrissa geometry ; 'Old' indicates old vibrissa geometry ; blue cross - new vibrissa at $U^* = 3$; red square - new vibrissa at $U^* = 4$; black diamond - new vibrissa at $U^* = 5$; green inverted triangle - new vibrissa at $U^* = 6$; teal triangle - old vibrissa at $U^* = 3$; dark yellow circle - old vibrissa at $U^* = 4$; violet left-oriented-triangle - old vibrissa at $U^* = 5$; grey right-oriented-triangle - old vibrissa at $U^* = 12$ 331

- 8.17 Amplitudes of oscillation (left) and response frequency ratios (right) of the trailing new vibrissa geometry at $U^* = 3, 4$, and 5 , $Re = 500$, $m^* = 1.04$ for inline spacing (S) of $3D_1$ and $4D_1$ in the wake of a wake-generator circular cylinder with $CDR = 1.5$. Inline spacing measured between trailing edge of wake generator to leading edge of trailing structure. Legend : S = $3D_1$ - inline spacing of $3D_1$; S = $4D_1$ - inline spacing of $4D_1$; 1st-S = $3D_1$ - primary response frequency for inline spacing of $3D_1$; 1st-S = $4D_1$ - primary response frequency for inline spacing of $4D_1$; 2nd-S = $3D_1$ - secondary response frequency for inline spacing of $3D_1$; 3rd-S = $3D_1$ - tertiary response frequency for inline spacing of $3D_1$ 332
- 8.18 Isometric view of the vortex cores in the wake of the wake-generator circular cylinder and trailing freely vibrating in crossflow new vibrissa geometry at inline spacing of $3D_1$ (left) and $4D_1$ (right), $Re = 500$ and $U^* = 5$ 332
- 9.1 Top view of vortex cores in the wake of tandem circular cylinders with stationary upstream circular cylinder and freely vibrating in crossflow trailing circular cylinder at an inline spacing of $0.5D$, freestream flow velocity (U) = $1D/s$ and $Re = 500$. Top : before motion of trailing circular cylinder ($t^* = 180$) ; bottom : after streamwise motion of trailing circular cylinder away from upstream circular cylinder at $u = 1D/s$ (relative to the upstream stationary circular cylinder) at $t^* = 187.4$ 336
- 9.2 RMS amplitude ratios of the approaching and distancing sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) relative to a wake-generator circular cylinder at $CDR = 1.0$ for various sideways displacement and $Re = 500$ (based on wake-generator circular cylinder). Top left : approaching sensor at $1D/s$; top right : approaching sensor at $2D/s$; bottom left : distancing sensor at $1D/s$; bottom right : distancing sensor at $2D/s$ 339
- 9.3 Mean displacement ratios of the approaching and distancing sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) relative to a wake-generator circular cylinder at $CDR = 1.0$ for various sideways displacement and $Re = 500$ (based on wake-generator circular cylinder). Top left : approaching sensor at $1D/s$; top right : approaching sensor at $2D/s$; bottom left : distancing sensor at $1D/s$; bottom right : distancing sensor at $2D/s$ 340
- 9.4 Dominant response frequency (detected frequency, not frequency of encounter) of the approaching sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) relative to a wake-generator circular cylinder at $CDR = 1.0$ for various sideways displacement and $Re = 500$ (based on wake-generator circular cylinder). 342
- 9.5 Vortex cores in the wake of the wake-generator circular cylinder at $CDR = 1.0$ and Wake-Induced Vibration (WIV) on trailing circular cylinder with $G/D = 0.0$ and approaching velocity of $1D/s$. Top left : $t^* = 186.6$ - before encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; top right : $t^* = 187.4$ - encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; bottom left : $t^* = 188.8$ - encounter of approaching circular with counter-clockwise-rotating vortex tubes ; bottom right : $t^* = 189.4$ - aftermath of the encounter of approaching circular cylinder with counter-clockwise rotating vortex tubes. 343

9.6	Vortex cores in the wake of the wake-generator circular cylinder at CDR = 1.0 and Wake-Induced Vibration (WIV) on trailing circular cylinder with $G/D = 1.5$ and approaching velocity of 1D/s. Top left : $t^* = 186$ - before encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; top right : $t^* = 187$ - encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; bottom left : $t^* = 188.4$ - encounter of approaching circular with counter-clockwise-rotating vortex tubes ; bottom right : $t^* = 189.6$ - aftermath of the encounter of approaching circular cylinder with counter-clockwise rotating vortex tubes.	344
9.7	Vortex cores in the wake of the wake-generator circular cylinder at CDR = 1.0 and Wake-Induced Vibration (WIV) on trailing circular cylinder with $G/D = 1.0$ and approaching velocity of 1D/s. Top left : $t^* = 186.2$ - before encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; top right : $t^* = 187$ - encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; bottom left : $t^* = 188.4$ - close-encounter of approaching circular with counter-clockwise-rotating vortex tubes ; bottom right : $t^* = 189.2$ - aftermath of the close-encounter of approaching circular cylinder with counter-clockwise rotating vortex tubes.	345
9.8	Scaled (to its respective one-sided maximum value) lift forcing and motion response of the approaching circular cylinder at a velocity of 1D/s for various sideways displacement (G/D). Top left : $G/D = 0.0$; top right: $G/D = 1.0$; bottom : $G/D = 1.5$	347
9.9	Pressure profiles in the wake of the wake-generator circular cylinder and WIV on approaching circular cylinder at a velocity of 2D/s and initial sideways displacement of 0.5D. A plane slice is located in the midspan of the simulation domain in order to strengthen the coloration of the pressure profiles.	348
9.10	Pressure profiles in the wake of the wake-generator circular cylinder and WIV on approaching old vibrissa geometry at a velocity of 1D/s and initial sideways displacement of 0.5D. Left : pressure profile in saddle plane of major axis ; right : pressure profile in nodal plane of major axis	348
9.11	RMS amplitude ratio relative to D_2 (top), mean Displacement relative to D_2 (middle) and dominant response frequencies (bottom) of the trailing circular cylinder at various sideways displacements and relative velocities (to stationary wake-generator circular cylinder) in the <i>Kármán</i> street of a wake-generator circular cylinder at CDR = 3.0 and $Re = 1,500$ (based on characteristic diameter of wake-generator circular cylinder, D_1). Relative velocity is based on D_2/s while sideways displacement is based on D_1	352
B.1	Schematic diagram of discretisation with QUICK scheme and its corresponding variable definitions with variable grid spacing.	372
C.1	Depiction of coherent structures in the wake of geometry 4 (vibrissa-like geometry with no major axis undulations with axis ratio 0.54) at simulation time, $t^* = 202$, at $x/D = 6.2$ (left) [and 7.2 - (right), details in figure C.2)], where x is measured from the centroid of the geometry . Top left : Side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) denoted by black dotted lines ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.	376

- C.2 Depiction of coherent structures in the wake of geometry 4 (vibrissa-like geometry with no major axis undulations with axis ratio 0.54) at simulation time, $t^* = 202$, at $x/D = 7.2$ (right plane cut - refer to figure C.1), where x is measured from the centroid of the geometry . Top left : overview of the vortex cores in the right plane-cut ; top right : enlarged view of the top half of the vortex cores and tangential velocity vector in the right plane-cut ; bottom : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the right plane-cut. Legend: orange area denotes cross-sectional area of vortex cores. 377
- C.3 Depiction of coherent structures in the wake of geometry 5 (vibrissa-like geometry with no minor axis undulations) at simulation time, $t^* = 225$, at $x/D = 5.3$ (left) [6.3 (middle) and 7.05 - (right), details in figures C.4, C.5)], where x is measured from the centroid of the geometry . Top left : Side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) denoted by black dotted lines ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores. 378
- C.4 Depiction of coherent structures in the wake of geometry 5 (vibrissa-like geometry with no minor axis undulations) at simulation time, $t^* = 225$, at $x/D = 6.3$ (middle plane cut - refer to figure C.3), where x is measured from the centroid of the geometry . Top left : overview of the vortex cores in the middle plane-cut ; top right : overview of the vortex cores and tangential velocity vector in the middle plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the middle plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the middle plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores. 379
- C.5 Depiction of coherent structures in the wake of geometry 5 (vibrissa-like geometry with no minor axis undulations) at simulation time, $t^* = 225$, at $x/D = 7.05$ (right plane cut - refer to figure C.3), where x is measured from the centroid of the geometry . Top left : overview of the vortex cores in the right plane-cut ; top right : overview of the vortex cores and tangential velocity vector in the right plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the right plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the right plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores. 380
- C.6 Depiction of coherent structures in the wake of geometry 9 (old vibrissa geometry) at simulation time, $t^* = 228$, at $x/D = 6.47$ (left) [and 7.67 - (right), details in figures C.7)], where x is measured from the centroid of the geometry . Top left : Side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) denoted by black dotted lines ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: green area denotes cross-sectional area of vortex cores. 381

- C.7 Depiction of coherent structures in the wake of geometry 9 (old vibrissa geometry) at simulation time, $t^* = 228$, at $x/D = 7.67$ (right plane cut - refer to figure C.6), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores in the right plane-cut ; top right : overview of the vortex cores and tangential velocity vector in the right plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the right plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the right plane-cut. Legend: green area denotes cross-sectional area of vortex cores. 382
- D.1 Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x) where x is measured from the centroid of the geometry. Top left : overview of vortex cores and tangential flow velocity vectors at $x/D = 0.35$; top right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.35$ in the vicinity of $z/D = 6.15$ (measured from the base of the circular cylinder) ; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.4$ in the vicinity of $z/D = 6.15$; middle right: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.45$; bottom: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.5$ in the vicinity of $z/D = 6.15$. Legend: orange area denotes cross-sectional area of vortex cores. 384
- D.2 Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x) where x is measured from the centroid of the geometry. Top left : enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.57$ in the vicinity of $z/D = 6.15$; top right: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.58$; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.58$ in the vicinity of $z/D = 6.15$; middle right: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.6$; bottom left: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.65$; bottom right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.65$ in the vicinity of $z/D = 6.15$. Legend: orange area denotes cross-sectional area of vortex cores. 386
- D.3 Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x) where x is measured from the centroid of the geometry. Top left : enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.1$ in the vicinity of $z/D = 6.15$; top right: overview of vortex cores and tangential flow velocity vectors at $x/D = 1.3$; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.37$ in the vicinity of $z/D = 6.15$; middle right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.4$ in the vicinity of $z/D = 6.15$; bottom left: overview of vortex cores and tangential flow velocity vectors at $x/D = 1.4$; bottom right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.7$ in the vicinity of $z/D = 6.15$. Legend: orange area denotes cross-sectional area of vortex cores. 387

D.4	Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x), where x is measured from the centroid of the geometry, and in the vicinity of $z/D = 6.15$, where z is measured from the base of the circular cylinder. Top left : enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.1$; top right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.2$; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.3$; middle right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.5$; bottom left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.8$; bottom right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 3.0$. Legend: orange area denotes cross-sectional area of vortex cores.	388
D.5	Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at $x/D = 3.3$. Left: Spanwise overview ; right : enlarged spanwise view in the vicinity of $z = 6.15$	389
E.1	Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the plane-cut (perpendicular to streamwise) at $x/D = 6.6$, $t^* = 194$ and $U^* = 5.97$. Left : overview of bulk of vortical braids for each nodal plane of the minor axis ; right : enlarged view of the upper half of the corresponding (to the left figure) bulk of vortical braids. The bulk of vortical braids at this plane-cut is found to be symmetric about the peak of the nodal plane. Hence, the lower half of the bulk of vortical braids is found to be a mirror-image of the upper half of the vortical braids about the peak of the corresponding nodal plane. Top : vortex cores about the first nodal plane of minor axis (from the top of geometry) ; middle : vortex cores about the second nodal plane of minor axis (mid-span of geometry) ; bottom : vortex cores about the third nodal plane. Legend: orange area denotes vortex cores.	392
E.2	Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the plane-cut (perpendicular to streamwise) at $x/D = 8.9$, $t^* = 194$ and $U^* = 5.97$. Top left : overview of bulk of vortical braids for the third nodal plane of the minor axis (from the top, for the first and second nodal planes, refer to figure 6.61) ; top right : enlarged view of the upper half of the bulk of vortical braids ; bottom left : enlarged view of the lower half of the bulk of vortical braids ; bottom right : enlarged view of the bulk of vortical braids at the spanwise end (last nodal plane) of the geometry. Legend: orange area denotes vortex cores.	393
E.3	Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the first nodal plane of the plane-cut (perpendicular to streamwise) at $x/D = 11.8$, $t^* = 194$ and $U^* = 5.97$. Top left : overview of bulk of vortical braids for the first nodal plane of the minor axis (from the top) ; top right : enlarged view of the upper half of the bulk of vortical braids ; bottom : enlarged view of the lower half of the bulk of vortical braids. Legend: orange area denotes vortex cores.	394
E.4	Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the second nodal plane of the plane-cut (perpendicular to streamwise) at $x/D = 11.8$, $t^* = 194$ and $U^* = 5.97$. Left : overview of bulk of vortical braids for the first nodal plane of the minor axis (from the top) ; right : enlarged view about the centre of the bulk of vortical braids. Legend: orange area denotes vortex cores.	395

E.5	Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the plane-cut (perpendicular to streamwise) at $x/D = 11.8$, $t^* = 194$ and $U^* = 5.97$. Top left : overview of bulk of vortical braids for the third nodal plane of the minor axis (from the top) ; top right : enlarged view of the upper half of the bulk of vortical braids ; bottom left : enlarged view of the lower half of the bulk of vortical braids ; bottom right : enlarged view of the bulk of vortical braids at the spanwise end (last nodal plane) of the geometry. Legend: orange area denotes vortex cores.	396
E.6	Vortex cores and location of plane-cuts in the wake of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) for $Re = 500$, $U^* = 4.8$ and natural frequency ratio (f_{nr}) = 2.0. Top left : side view of the wake and location of plane cuts (from left to right) $x/D = 4.2$, 5.6, and 6.0 for $t^* = 234$; top right : side view of the wake and location of plane cuts (from left to right) $x/D = 4.6$, 5.2, and 6.2 for $t^* = 246$; bottom : side view of the wake and location of plane cuts (from left to right) $x/D = 4.4$ for $t^* = 284$	397
E.7	Vortex cores and tangential velocity vector of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) for $Re = 500$, $t^* = 234$ and $U^* = 4.80$. Top left : overview of in the plane-cut at $x/D = 4.2$; top right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 4.2$; middle left : overview of in the plane-cut at $x/D = 5.6$; middle right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 5.6$; bottom left : overview of in the plane-cut at $x/D = 6.0$; bottom right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 6.0$. Legend: orange area denotes vortex cores.	398
E.8	Vortex cores and tangential velocity vector of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) for $Re = 500$, $t^* = 246$ and $U^* = 4.80$. Top left : overview of in the plane-cut at $x/D = 4.6$; top right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 4.6$; middle left : overview of in the plane-cut at $x/D = 5.2$; middle right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 5.2$; bottom left : overview of in the plane-cut at $x/D = 6.2$; bottom right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 6.2$. Legend: orange area denotes vortex cores.	399
E.9	Vortex cores and tangential velocity vector of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) in the plane-cut at $x/D = 4.4$ for $Re = 500$, $t^* = 246$, and $U^* = 4.80$. Left : overview of in the plane-cut ; right : enlarged view of a period of vortical braids in the plane-cut. Legend: orange area denotes vortex cores.	400
F.1	Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing circular cylinder at inline spacing of 4D and CDR = 0.6. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture)..	402
F.2	Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing circular cylinder at inline spacing of 4D and CDR = 2.6. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture).	405

F.3	Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing elliptical cylinder at inline spacing of $4D$ and $CDR = 0.6$. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture)..	406
F.4	Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing elliptical cylinder at inline spacing of $4D$ and $CDR = 2.8$. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture).	409

List of Tables

3.1	Overall chemical composition of harbour seal vibrissae. '1st', '2nd', and '3rd' represent experimental results from first, second and third runs, respectively. Weight(%) indicates the weight fraction of the element to the overall weight of the vibrissa. Atomic (%) indicates the atomic fraction of the element to the overall number of atoms in the vibrissa.	63
3.2	Surface chemical composition of harbour seal vibrissae obtained from X-Ray Photoelectron Spectrometer (XPS). '1st', '2nd', and '3rd' represent experimental results from first, second and third runs, respectively. Weight(%) indicates the weight fraction of the element to the overall weight of the vibrissa. Atomic (%) indicates the atomic fraction of the element to the overall number of atoms in the vibrissa.	64
4.1	Normalised dominant response frequencies of the vibrissa models mounted on soft (S), medium (M) and hard (H) holding materials towed at three speeds (100 mm/s, 200 mm/s, 300 mm/s). Legend for first column (first two letters): CC - circular cylinder model, EC - elliptical curved cylinder model, SW - straight vibrissa model, CW - curved vibrissa model; (last three letters): 100 indicates towing speed of 100 mm/s, 200 indicates towing speed of 200 mm/s and 300 indicates towing speed of 300 mm/s. f_1^* indicates the first dominant response frequency, f_2^* indicates the second dominant response frequency, U^* indicates the corresponding reduced velocity. . . .	104
4.2	Nominal dominant response frequencies of the vibrissa models mounted on soft (S), medium (M) and hard (H) holding materials towed at three speeds (100 mm/s, 200 mm/s, 300 mm/s). Legend for first column (first two letters): CC - circular cylinder model, EC - elliptical curved cylinder model, SW - straight vibrissa model, CW - curved vibrissa model; (last three letters): 100 indicates towing speed of 100 mm/s, 200 indicates towing speed of 200 mm/s and 300 indicates towing speed of 300 mm/s. f_1^* indicates the first dominant response frequency, f_2^* indicates the second dominant response frequency, U^* indicates the corresponding reduced velocity. f_1 and f_2 are in Hertz.	104
5.1	Drag coefficient of harbour seal vibrissa for various grid size combinations and its corresponding computational cost with state-of-the-art single core computer. The grid size is normalised to the characteristic diameter of the vibrissa (the minor diameter of the vibrissa) with dx,dy and dz representing the grid size in streamwise, crossflow and spanwise direction respectively.	127
5.2	Comparison of mean drag coefficient, RMS lift coefficient and Strouhal frequency on a stationary circular cylinder at Re = 100 and 200 [93].	132
6.1	Detailed dimensions of simulated geometries with variable definitions depicted in figure 2.3. All characteristic dimensions will be normalised to 1.0 for numerical simulations with a minimum of 20 grid points across this characteristic diameter. . .	150

Chapter 1

Introduction

1.1 Background

In recent years, underwater explorations have received more attentions with the prevalence of the uniqueness of the deep ocean. Such explorations are coupled with the need to find more sources of energy in the offshore region. These explorations pose major challenges to the current oceanic technologies available to man. Underwater navigation is known to be absolutely essential for such explorations. However, the lack of visibility in deep oceans has left mankind with a different frontier to challenge than that of space. Through persistent observation on the method of navigation of oceanic creatures, a novel method of underwater navigation was studied with the ultimate aim (of the project, not the thesis) of artificially re-creating such sensing and therefore, navigating abilities. One of such novel inspiration comes from the observation of the sensing methodology of fishes that uses their lateral lines (a pressure-sensing organ) for vicinity detection [71]. However, many challenges still remains particularly for clean and informative sensor in turbulent and unsteady wakes. As such, a novel underwater sensing methodology is studied to complement the pressure-sensing methodology for underwater navigation. This novel method is inspired by the harbour seals, which is capable of sensing only with their vibrissae [26]. Furthermore, such sensing is performed while the seals are moving. The robustness of the seals' sensing ability is further illustrated by the seal's ability to sense ageing water disturbances ($t > 35s$) [99]. The independence of the initial orientation of the seal with respect to the water disturbances is also shown [80]. Figure 1.1 depicts the muzzle of a harbour seal with its vibrissae.

Figure 1.1: Harbour seal and its vibrissae on the muzzle in New England Aquarium, Boston, Massachusetts, USA (Copyright: Heather Beem [MIT Towing Tank])

The sensitivity of the underwater sensing principle of the harbour seals is found to originate from the Vortex-Induced Forces (VIV) reduction properties of their vibrissae. [40]. Such reduction is hypothesized to increase the Signal-To-Noise Ratio (SNR) as the vibrissa is exposed to wakes (other than its own). These studies may provide a basis for adoption of the underwater sensing ability of the harbour seals as a novel underwater sensing technology. As such, the objective of this study is to understand the underwater sensing mechanism of the harbour seal vibrissae. Once such understanding is attained, its adoption as technologies for underwater exploration will be natural.

1.2 Objectives

The objective of this study is to understand the underwater sensing principle of the harbour seal vibrissae. In order to achieve this objective, completion of sub-objectives are necessary, as follows:

- * To establish and confirm the Vortex-Induced Vibration (VIV) reduction ability of the vibrissa;
- * To identify the most significant feature of the vibrissa for VIV reduction;
- * To establish the performance of a single vibrissa in a known wake;
- * To establish the effects of spatial positioning of the vibrissa with respect to the wake;
- * To establish the response of the vibrissa to variation of known-wake generator; and
- * To define response of the vibrissa at non-zero velocity relative to the known wake.

The completion of these sub-objectives will provide a better understanding of the underwater sensing principle of the harbour seals' vibrissae. For Chapter 4 to Chapter 8 of this thesis, each chapter poses a hypothesis to be tested. It is followed by the experiments (or simulations) designed to test the hypothesis and the confirmation of the corresponding hypothesis. Each hypothesis is composed of, at least, one sub-objective. hence, the evaluation and conclusion on each hypothesis would represent evaluation and fulfillment of the corresponding sub-objective(s).

1.3 Structure of The Thesis

This report covers the experimental and numerical studies on a single harbour seal vibrissa as it is exposed to its own wakes and other forms of perturbations. The layout of the report are as follows:

- * **Chapter 1** describes the introduction and motivation of the current study. The objective and corresponding sub-objectives are also described in the chapter. The depth of the subsequent discussions in the report are briefly illustrated in the chapter.
- * **Chapter 2** recounts the previous studies related to the current topic. A brief description on various types of biomimicry sensing technology is described. Given the novelty of the current study, directly-correlated studies are yet to be found to the date of writing of this thesis. However, biological studies, which have inspired the current study, are reviewed. Studies on the geometrical effects are also touched upon in correlation to the multi-faceted features of the geometry of the vibrissa.
- * **Chapter 3** examines the properties of the harbour seal vibrissae. As previous studies on the material properties of harbour seal vibrissae are yet to be published, the author took the initiative of measuring the mechanical properties of the harbour seal vibrissae and some elemental chemical compositions of the vibrissae. This chapter shows the methodology of the experiments, the experimental results and the implications of the results on the underwater sensing of the harbour seal vibrissae.
- * **Chapter 4** reports the experimental studies performed in the completion of this thesis. This chapter is subdivided into the experimental setup, methods of data analysis, results and discussions as

well as conclusion. The dynamic similarity method is used to scaled up the experimental setup and it is further illustrated in Appendix A.

- * **Chapter 5** discusses the methodology inscribed in the numerical simulation. Details of the simulation method are presented in this section as well as geometrical construct of the harbour seal vibrissae and their variant. Grid design and validation cases for stationary and freely vibrating cases are also presented in this chapter. The final section of this chapter is devoted to description on the methodology of data analysis.
- * **Chapter 6** details the studies on the effects of geometrical variation of the harbour seal vibrissae on the mean drag coefficient and RMS lift coefficient as well as the Strouhal frequency of the corresponding geometry. The purpose of this studies is to discover the most significant feature in reducing (Vortex-Induced Forces) VIF and VIV. This section can be further subdivided into stationary structures simulations, freely vibrating in crossflow (1D) structures simulations, and freely vibrating in streamwise and crossflow (2D) structures simulations. Analyses on the dynamics of the response of the structures are also presented along with studies on the wake topology and instabilities in the wake of the structures. Discussions of results and conclusion on the geometrical effects on the VIV response of each structure are also presented.
- * **Chapter 7** illustrates the effects of angle of attack on the response of the vibrissa. Similar to Chapter 6 , this chapter can be subdivided into the similar category (stationary, 1D freely vibrating and 2D freely vibrating setup). The force and/or motion responses of the vibrissae are described in conjunction with variation of the natural frequency of the system.
- * **Chapter 8** discusses the response of a freely vibrating in crossflow vibrissa/elliptical cylinder/circular cylinder (as a sensor) as the sensory structure trails in the wake of a stationary circular cylinder. Studies on the effects of spatial positioning and relative sizes of the wake generator to the sensor are also described in this chapter. Finally, studies on the effects of the natural frequency of the system are also described.
- * **Chapter 9** describes the effect of approaching and distancing on the response of trailing circular cylinder/ elliptical cylinder/ old vibrissa geometry as it moves in the *Kármán* street of the wake-generator circular cylinder. The simulations in this chapter is used to model the response

of the vibrissae on the muzzle of the harbour seals as they track the wakes of their vibrissae (regular and unstable wakes). Study on the effects of approaching or distancing velocity on the response of the trailing sensor structure is also performed along with studies on the effects of the Dimension Ratio between the wake-generator circular cylinder and the trailing sensor structure.

- * **Chapter 10** provides the conclusion of the thesis and recommended future work for advances in understanding the underwater sensing principle of the harbour seal vibrissae.

Chapter 2

Literature Review

From dawn to dusk, it was one of the many possible analogies that the author takes on mankind's exploration from space to the ocean. The differences in these frontiers were so resounding, but yet, so similar in some of their challenges. These persistent challenges have led mankind to seek for solutions to overcome them. The solutions to such challenges were often embodied in biological entities of mother nature. Biologically inspired methods have dominated the sphere of airborne explorations. Since then, these inspirations have extended its reach to underwater exploration. One of the biologically-inspired methods for tackling zero-vision in underwater environment was through the use of pressure sensing [71]. In order to navigate within their surroundings (even during schooling), the fishes was found to utilise their lateral line (a pressure sensing organ). This method has been adopted as a technology for underwater navigation [55, 56]. Kottapalli et al had closely mimicked the superficial neuromast of the lateral line of fishes through the use of Liquid Crystal Polymer (LCP). Their results had indicated a good vicinity sensing methodology. However, challenges in oceanic sensing include sensing in turbulent environment. As such, pressure based sensing method may encounter difficulties in performing its task in such turbulent condition. A novel biologically-inspired principle of the harbour seals vibrissae was researched in order to solve such turbulent wake sensing challenge. Biological studies have shown that harbour seals were capable of performing underwater sensing and tracking with only their vibrissae [26, 27]. On an earlier study, Dehnhardt & Kaminski had identified the tactile sensing ability of the harbour seals with their vibrissae [26]. They found that the seals were capable of differentiating sizes of discs through tactile sensing with their vibrissae. Through experiments with a static seal following the motion of a vibrating sphere, Dehnhardt et al found that the sensitivity of the seal vibrissae to perturbations were similar to the hydrodynamic

sensing ability of other aquatic animals [27]. They had also found that the seals were capable of sensing water velocity of up to $245 \mu\text{m/s}$. Previous studies by Renouf and Mills & Renouf have indicated the responsiveness of the seals vibrissae to perturbations [77, 69]. Renouf observed the fishing behaviour of seals captured as yearlings and as pups. It was found that the yearling-captured seals responded to the presence of fishes by extending their necks and following the hydrodynamic trails of the fishes almost perfectly. He had also observed that yearling-captured seals were significantly affected in their tracking of fishes by removal of their vibrissae. However, visibility reduction was found to be ineffective in hampering the seals fish-tracking ability. Mills & Renouf performed a more controlled experiments on the harbour seals with a steel rod perturbing the seals' vibrissae at various frequencies from 100 Hz to 2500 Hz. The seals had two forced choices for indicating the presence or absence of the perturbations. They found that the seal vibrissae were most sensitive to (higher-frequency) perturbations (lowest amplitude threshold) at the frequency of 1000 Hz for a $3 \mu\text{m}$ amplitude of perturbation.

Further biological studies had also shown the extreme sensitivity of the sensing principle of the harbour seal vibrissae. Notably, it was found to be capable of sensing the wake of an artificial fish-like fin after 35 seconds of time delay [99]. The seal vibrissae were also found to be capable of sensing a helical vortices generated by a miniature submarine [28]. These tests were conducted on naive and trained seals with minimum differences in performance when tracking the wakes of the submarine [28]. Figure 2.1 depicts the top view of vibrissae and the top and side views of the undulations on the harbour seal vibrissae. It was important to note that the geometry of the vibrissae were elliptical in cross-section and were found to consist of many features including undulations in major and minor axes, offset angle between its leading edge and trailing edge in major axis, and spanwise variations of the offset angle. From observation of 80 undulations on seals vibrissae, it was found that the minimum diameter of the vibrissae varies (due to undulations) between 0.24 mm to 0.29 mm [40]. The maximum diameter of the vibrissae was found to vary (due to undulations) between 0.475 mm to 0.595 mm. The offset angle on the major axis was found to vary between 15.27° (at the smallest cross section) and 17.6° (at the widest cross section). Figure 2.2 shows the model vibrissa obtained from measurements of 80 undulations of the seal vibrissae [40].

As the hydrodynamic trails were generated by the miniature submarine, auditory cues were obstructed with pink noise from headphones (wore by the seals) and visual cues were impeded through

Figure 2.1: Harbour seal vibrissae from New England Aquarium, Boston, Massachusetts, USA: Top - Top view of full length and segmented vibrissae; Middle - Top view on the major axis undulations of a vibrissa; Bottom - Side view on the minor axis undulations of a vibrissa.

Figure 2.2: Ideal vibrissa model and definitions of its geometrical parameters. From left to right : top view of the minor axis of the vibrissa, top view of the major axis of the vibrissa, and isometric view of the vibrissa. [40].

blindfolding of the seals. The engine of the miniature submarine was turned off prior to the seal's entry into the water. For all the submarine tracking tests, the seals were observed to follow the hydrodynamic trails of the miniature submarine regardless of the type of trails generated (straight, curved, sudden turns). Dehnhardt et al also performed control tests where stocking masks were used to impede the motion of the seal vibrissae. In these control tests, the seals were found to be incapable of performing any tracking. A different control test was also performed with the engine of the miniature submarine turned on after generating the hydrodynamic trails. For these tests, the seals were observed to swim straight to the submarine. It was also noted that the seals seldom lost the hydrodynamic trails once it had sensed the trails. In another study, the independence of the initial orientation of the harbour seals with respect to the hydrodynamic trails were shown by Schulte-Pelkum et al [80]. In this study, another harbour seal was used as a wake generator while the tracking harbour seal was allowed to dive into the water after ≈ 14 seconds of time delay. Schulte-Pelkum et al observed two distinct tracking behaviour of the seals. Figure 2.3 shows the two trail-tracking behaviour of the seals in these experiments [80]. The first pattern was observed to be a near-perfect alignment between the generated trail and the tracking path of the trailing-seal (termed 'linear pattern'). The second pattern was observed to be an undulatory path about the generated trail (termed 'undulatory

Figure 2.3: Wake-tracking behaviour of trailing harbour seal while tracking a wake generated by a fellow harbour seal. A) Linear pattern : the wake-tracking seal adheres almost-perfectly to the wake-generated by the fellow harbour seal; B) Undulatory pattern : wake-tracking seal was observed to criss-cross the wake generated by the fellow harbour seal, however, the amplitude of undulation was found not to exceed one body length [80].

Figure 2.4: Search pattern of wake-tracking harbour seal at various initial angular orientation to the wake generated by a fellow harbour seal. For explanation on linear pattern and undulatory pattern, please refer to Fig. 2.3.) [80].

pattern'). For the second pattern, it was also noted that, the furthest point of such pattern was never greater than one body length. Although the seal's tracking ability was found to be independent of its relative initial orientation to the wake, the pattern of search was found to be strongly related to its initial orientation. Figure 2.4 depicts the hydrodynamic-trail searching behaviour of the harbour seal as a function of the relative orientation of the seal to the hydrodynamic trail.

Further biological study had also shown a better response of the harbour seals with their vibrissae to vibratory stimulant as compared to the response of the California sea lion with their vibrissae as

Figure 2.5: Wake-generator paddles for experiments on the shape discrimination through wake analysis by harbour seals. From left to right : flat paddle, undulated paddle, circular paddle and triangular paddle. The height of the paddles were 30 cm [101].

it was exposed to the wakes of the same miniature submarine [36]. The most significant difference in performance was observed after a time delay of greater than 5 second were introduced in the wake-tracking experiment. After such time delay, the sea lion was observed to fail in tracking the wakes of the miniature submarine. In comparison, Dehnhardt et al and Wieskotten et al observed that harbour seals were still capable of excellent wake-tracking up to 35 seconds of time delay. From a closer observation, Hanke et al observed that the vibrissae of the California Sea Lion was nearly circular cylindrical in shape with no visible bristles along their lengths [40]. However, the vibrissae of the harbour seal was found to be elliptical in cross-section. Undulations on their minor and major axes were also found along their lengths [40]. These features were distinguishably different from the vibrissae of many other creatures. Wieskotten et al performed further studies by designing paddles of various shapes to be tracked by the seal. Figure 2.5 shows the experimental paddle used to generate wakes for shape discrimination experiment on harbour seals. The experimental paddles consisted of a flat (standard) paddle, an undulatory paddle, a circular paddle and a triangular paddle. Each paddle was rotationally dragged across the window, where the seal will be able to enter half-way. From figure 2.5, significant geometrical distinction between the paddles were observed. However, the vortical features in the wakes of these towed paddles were considerably less distinct, with exception of the undulatory paddle. Figure 2.6 illustrates the wakes behind the standard paddle, circular paddle and triangular paddle [32, 113, 104].

Figure 2.6: Flow visualisations of flow around (Top) vertical flat plate, (Middle) circular cylinder, (Bottom) triangular cylinder [32, 104, 113].

From figure 2.6, it can be observed that there were strong similarity between the wakes of the standard, circular and triangular paddles. The wakes in these paddles were observed to form a von-*Kármán* street. Due to such qualitative similarity, it was expected that the seal will experience difficulty in discriminating these paddles. However, from Wieskotten et al's experimental result, the seal was found to be capable of discriminating these paddles. Figure 2.7 shows the results of shape-discrimination experiments for paddles of similar sizes. With reference to figure 2.7, it can be observed that the seals were only capable of differentiating the standard paddle to the other paddles. Discrimination among other shapes were found to be less convincing. Referring to figure 2.6, this was interesting as the vortical arrangements of the flat paddle, circular paddle and triangular paddle were qualitatively similar. However, a closer observation on the wakes properties of these paddles indicated an essential property of the wakes for the sensing of the vibrissa. In particular, at the given Re , the Strouhal number (St) of the flat paddles were observed to be distinct from the Strouhal Number of the circular and triangular paddles. With regards to the undulatory paddle, due to the

Figure 2.7: Shape-discrimination experimental results on harbour seal. The wake-generators were flat paddle, undulated paddle, circular paddle and triangular paddle [101] (see Figure 2.5 for shape illustrations).

significant spanwise variation of the geometry, the wake of the structure was expected to consist of a complex coherent structure. As the study in this thesis will show (refer to chapter 6), such undulatory surface will most likely trigger flow-induced vibration at low frequencies (reduced frequency ($f^* = fD/U$) < 0.8). Figure 2.8 depicts the Strouhal Number of the flat, circular and triangular paddles as a function of the Reynolds Number [8, 73, 65]. From figure 2.8, it can be observed that the Strouhal Number of the flat plate was approximately 0.12 at $Re(O^4)$ while the Strouhal Number of the circular cylinder and triangular cylinder were approximately 0.2 and 0.18, respectively. Such distinction in the vortex shedding frequency was an indication on the importance of the characteristic frequency of the perturbations to the underwater sensing ability of the harbour seal vibrissae. Due to the similarity in results between the shape discrimination of flat paddle and other paddles (refer to figure 2.7), Wieskotten et al performed further experiments on size-shape discrimination between the flat paddle and other paddles [101]. Figure 2.9 demonstrates their experimental results for size-shape discrimination tasks on harbour seals.

While the results signify the seal's ability to differentiate between flat and triangular paddles, the results for other paddles were found to be similar. Notably, the seals made 70% correct choices when discriminating between the flat and triangular paddles. However, the seals also made 60% and 55% when discriminating between the flat and circular as well as flat and undulatory paddles, respectively. However, despite the findings of the robustness of the seals' underwater sensing ability, Wieskotten et al found that the seals were less sensitive to the gliding phases of the miniature submarine [100]. Figure 2.10 shows the success rate of the seal-tracking burst and glide experiments with miniature

Figure 2.8: Strouhal number (St) of flat plate (top left), circular cylinder (top right) and triangular cylinder (bottom) as a function of Reynolds number (Re) [8, 73, 65].

submarine for time delay range of 5 to 30 seconds. It can be observed that there were significant decline in the success rate of the wake-tracking seal as time delay of 15 s and larger was introduced. This was as expected due to the less energetic wake generated by the miniature submarine in gliding phases (no energy transfer from propeller to fluid). Interestingly, Wieskotten et al found that the widths of wakes during the burst and glide phases were not very distinct. It was noticed that the glide phases resulted in narrower street as compared to the wakes of the burst phases. However, significant difference in the water velocities were observed as time delay was introduced between the start of the submarine and the start of the seal. Figure 2.11 depicts the wake-tracking experiments of the seal for burst and glide phases.

An earlier study by Dykes [29] had shown that the nerves at the base of the vibrissae were capable of sensing a limited range of static deflection and vibratory perturbations. These nerves were termed Slowly Adapting (SA) fibres and Rapidly Adapting (RA) fibres, respectively. Dykes performed a

Figure 2.9: Size-shape discrimination experiments on harbour seal between flat paddle and other paddles. The width of the paddles were varied randomly between 2,4,5 and 7 cm [101].

Figure 2.10: Success rates of wake-tracking by harbour seal for wakes generated by bursting (solid line) and gliding (dashed line) miniature submarine at various time delay (between the start of the submarine and start of the seal). Numbers on top (for burst) and on bottom (for glide) shows the number of trials for each time delay and types of wakes [100].

live-seal operation to tap the impulses at the base of the vibrissae, as variation of perturbations were introduced on the vibrissae of the seal. Dykes found that the impulses were transmitted by the nerves for static deflections up to 20 degrees. Beyond this limit, no impulses were observed on the nerves of the seals. For vibratory response, the RA fibres were found to generate very strong impulses at lower frequencies of perturbation. For perturbations at 250 Hz, a slight decline (0.5%) in number of responding fibres were observed. For a perturbation at 500 Hz, a considerable decline (80%) of the responding fibres were observed. Dykes found that perturbations at 1000 Hz and higher were beyond the sensing ability of the harbour seals vibrissae (only < 20% responsive RA fibres). His

Figure 2.11: Snapshot from wake-tracking experiment on the seal for burst (left) and glide (right) wakes of a miniature submarine after 10 and 20 seconds, respectively. Magenta dots - path of seal, blue dots - burst phase of submarine, yellow - glide phase of submarine [100].

experimental results was observed to be consistent with the results of Miersch et al [67]. Miersch et al performed experiments with the harbour seal and sea lion vibrissa trailing a circular cylinder. Through their force measurements, they found that the seal vibrissae responded with a larger Signal To Noise Ratio (SNR) as compared to the sea lion vibrissae. A significantly larger amount of noise was also observed on the response of the sea lion vibrissae. However, not all studies have resulted in similar conclusion as Dykes' and Miersch's results [29, 67]. Earlier study by Mills & Renouf have indicated that the sensitivity threshold of the harbour seal vibrissae were optimum at 1000 Hz and higher. Their experimental setup involved sinusoidal perturbation on the seal vibrissae (through steel rod) between the frequency range of 100 Hz and 2500 Hz. However, from existing studies on the diets (preys, e.g. fish, jellyfish and etc) of harbour seals, it can be shown that such frequency range were well above the natural sensing frequency of the vibrissae [88, 22, 82]. Figure 2.12 illustrated the results of Mills & Renouf and Dykes on the sensitivity of the harbour seal vibrissae at various frequency of perturbations. Despite all the findings on the underwater sensing ability of the harbour seal vibrissae, there were limited number of studies on the tracking (and preying) of harbour seals on fishes [77]. Although certain similarities can be found, the structure of the wakes of a fish was observed to be distinct from the wakes of structures that had been used as wake-generators (wakes of a harbour seal, circular cylinder, fish-like fin, and propeller) [88, 28, 80, 101]. Figure 2.13 illustrates the z-vorticity field of wakes of a Giant Danio fish [88]. With reference to figure 2.13, it can be observed that the wakes of fishes consist of counter-rotating vortices in a staggered and continuous

Figure 2.12: Sensitivity of harbour seal vibrissae at various frequency of perturbations. Left : amplitude threshold of the harbour seal vibrissa sensing at frequency range (100 to 2500 Hz), Mills & Renouf observed that below the amplitude threshold at a given frequency, the seal was incapable of signifying the absence or presence of the perturbation. Right : the percentage of rapidly adapting (RA) fibres responding to perturbations at various frequencies [69, 29].

Figure 2.13: Two-dimensional (XY-plane) flow visualization of the z-vorticity field in the wake of a swimming Giant Danio. Green - irrotational flow field, blue - counterclockwise rotation, red - clockwise rotation [88].

arrangement. Similar arrangements were not observed for circular cylinder (see figure 2.6), propeller, fish-like fin and harbour seal (the wake of the harbour seal was suspected to be distinct due to the different anatomy and swimming posture) [99, 80, 44].

Through their simulations on stationary circular cylinder, elliptical cylinder and seal vibrissa, Hanke et al found that the geometry of the vibrissa reduces forcing on the body due to shedding of vortices [40]. The dynamic forces of the vibrissae was found to be very small as compared to

the dynamic forces of the circular cylinder. They have also performed experiments with a head-mounted camera on the harbour seal for analysis of the vibratory property of the vibrissae as the seals were swimming. No vibrations on the seal vibrissae were observed (or resolved) from the camera. Hanke et al suggested that this was due to the minute vibration of the vibrissa which was below the detectable level of the head-mounted camera. From our study (refer to Chapter 6), it was found that such reduction in vibration was a result of reduction in forcing from vortex shedding. This occurs mainly due to the spatial delay in vortex shedding (in contrast to common observation of the spanwise desynchronization in vortex shedding). This result may be achieved through analysis on the studies of flow around various geometries. In particular, the vortex shedding around a tapered circular cylinder have shown significant spanwise desynchronization in vortex shedding [33, 34, 76, 83]. Gaster observed that the local vortex shedding frequency was dependent on the local diameter of the structure. Vortex shedding was observed to occur in cells with a predominant frequency governing each cell. Further explanation on the connection between the cells in cellular vortex shedding was described by Techet et al [83]. Figure 2.14 depicts the vortex shedding on the tapered cylinder and the illustration on the connection between the cells of the vortex shedding. From Figure 2.14, vortex shedding cells can be observed in the wake of the tapered cylinder. Papangelou observed that the configurations of the vortex shedding cells were a function of Reynolds number (Re)[76]. Tip Reynolds Number was found to be irrelevant to the configuration of the vortex shedding cells. However, the frequency jump between the cells were found to be a function of flow speed, taper angle and kinematic viscosity. Techet et al also observed similar correlations between the frequency jump to the amplitude ratio (A/D), reduced frequency (fD/U) and Reynolds Number. The frequency jump was observed to be consistent from one cell to another. The physical explanation of frequency jumps between the cells was found in the form of vortex splits connecting one frequency cell to another. Such variation in spanwise vortex shedding frequency resulted in the modulation of the amplitude of response of the tapered circular cylinder. The modulation was found to be conveyed in the form of low-frequency oscillations of the tapered circular cylinder. Despite the modulations, Gaster found that the vortex shedding in each cell was regular and periodic with a frequency jump observed between the cells(see Figure 2.14) [34]. Such modulation was observed to weaken the von *Kármán* street in the wake of the tapered cylinder as compared to that of the circular cylinder. Figure 2.15 shows the comparison between the von *Kármán* street of a circular cylinder and a

Figure 2.14: Vortex shedding of a tapered circular cylinder. Left : smoke visualisations of the vortex shedding on a tapered circular cylinder at maximum $Re = 140$; Right : a sketch of the occurrence of vortex split between the patches of the vortex shedding on the tapered circular cylinder [76, 83].

tapered cylinder.

From Figure 2.15, it can be observed that the coherent structure in the wake of the tapered cylinder was less persistent as compared to that of the circular cylinder. Interestingly, Techet et al found that spanwise desynchronisation of vortex shedding was not mandatory for tapered circular cylinder. At certain frequencies and beyond certain amplitude of oscillation threshold, a spanwise-uniform 2S (two single vortices shed per cycle) or 2P (two pairs of vortices shed per cycle) shedding mode were observed for the flow around the tapered circular cylinder [83, 106]. Although the occurrence of vortex split was pioneered through observation of tapered cylinder, such occurrence was also found in flow around other geometries, most notably around circular cylinders [53]. König et al observed that cellular vortex shedding occurred on flow around circular cylinders due to end effects (of the circular cylinder). Figure 2.16 illustrated the three shedding cells of the circular cylinder in laminar flow regime. Flow around the vibrissa was also observed to experience a significant spanwise desynchronization in vortex shedding with a significant deformation of the vortex tubes as well as the braids in the wakes (refer to Chapter 5). As aforementioned, although such desynchronization may reduce the (non-uniform) forcing on the structure due to vortex shedding, it was found that the

Figure 2.15: Comparison of the coherence of vortical structures in the von *Kármán* street of the circular cylinder and tapered circular cylinder [33].

Figure 2.16: Vortex shedding cells at the near-wall region of a circular cylinder with three cells. From left to right (far from wall to near wall): high frequency, medium frequency, and low frequency, respectively [53].

reduction in VIV of the vibrissa was mainly contributed due to the spatial delay in vortex shedding (refer to Chapter 6).

Further evidence on the effectiveness of spatial delay in vortex shedding for VIV reduction was illustrated in studies on the flow around circular cylinder fitted with helical strakes [109]. Figure 2.17 shows the z-vorticity average time field of flow past a helical strakes fitted circular cylinder.

The introduction of helical strakes was found to significantly reduce VIV on the circular cylinder. Analysis of the flow around helical strakes fitted circular cylinder shows a significant spatial delay of vortex shedding in the wake of the structure [54]. However, as aforementioned, the geometry of the seal vibrissae was multi-faceted. Previous studies have shown that curvature of a circular cylinder (oriented with its horizontal section as trailing edge) had minimum effect on the lift coefficient (C_L) of the structure [68, 25]. Figure 2.18 illustrates the comparison between the wakes of a stationary circular cylinder and a stationary curved circular cylinder at low Reynolds Number. Although there was a small difference in the Re , the wakes of the straight and curved circular cylinder at the two regimes were observed to be very similar to each other. The regularity and absence of braids in the wakes of the lower Reynolds Number of straight and curved circular cylinder was observed. For higher Re , the vortical braids were similarly observed for straight and curved circular cylinder. A noted distinction was observed near the end of the curvature where cellular shedding was observed for curved circular cylinder. As such, with reference to these studies, the author believes that the other features on the geometry of the vibrissae took precedence in reducing VIV rather than the curvature of the vibrissae. Nuzzi et al investigated on the effect of three-dimensionality of a circular cylinder on the flow around the structure at $Re = 145$ [74]. Figure 2.19 depicts the circular cylinder with mild cross-sectional variation in Nuzzi et al's experiments. Mild taper was introduced in the mid-section of the circular cylinder. Through flow visualization, Nuzzi et al observed that the vortex shedding in the wake of such mildly-tapered cylinder resulted in period doubling (and therefore, vortex split) in the tapered section of the cylinder. From forced vibration experiments, they found that global lock-in on the mildly-tapered cylinder was possible for certain excitation frequencies. Further observation revealed a longer vortex formation length in the region of smaller diameter of the mildly-tapered cylinder. Consequently, the vortex shedding was pioneered in the region of the smallest diameter of the cylinder. Figure 2.20 illustrates the globally locked-in vortex shedding of the mildly-tapered cylinder at $Re = 145$. With reference to figure 2.20, the lead of the vortices in the mid-section was found to increase significantly as the wakes of the cylinder travelled further downstream. The phase-locking of flow was found to occur for the entire span of the wakes up to the distance of $24 D$, where D was the characteristic diameter of the untapered section of the mildly-tapered cylinder. For cases where period-doubling was observed, formation of tightly wound helical vortices was observed in the wake of the cylinder. Figure 2.21 illustrates the time evolution of the tightly wound (diamond-

Figure 2.17: The z-vorticity field of a flow past a helical strakes fitted circular cylinder. Left : reduced velocity ($U^* = U/fD = 3$); Middle : $U^* = 6.0$; Right : $U^* = 9.0$ [54].

shaped) helical vortices. During the period-doubling, it was observed that the vortex shedding in the spanwise direction alternated between the absence and existence of vortex-split. The results of Nuzzi et al was fundamental for understanding of the flow around the seal vibrissae, which contains multiple-two-axial variations in the spanwise direction [74]. A more relevant study to the geometry of the vibrissa was performed by Ahmed & Bays-Muchmore [2]. Similar results to Nuzzi et al was observed on the longer vortex formation length of the wavy cylinder on the saddle point (section with minimum diameter of the wavy cylinder). The reverse was found to be true with the shortest vortex formation length observed on the nodal point (section with maximum diameter of the wavy cylinder). From their experimental results, it was found that a greater distortion of separation lines was observed for shorter wavelengths. Analysis on the sectional drag coefficients concluded that drag coefficients were larger for nodal section as compared to saddle section. Further study revealed a greater stability of the flow in the saddle point as compared to the nodal point [2, 1]. This was due to the narrower wake formation on the nodal points as well as the suppression of turbulence due to spanwise strain of the flow field. The suppression of vortex pairing in the nodal points was also attributed to the spanwise strain of the flow field. Such strain on the flow field was a consequence of the geometrical waviness of the trailing edge of the wavy cylinder. This was found to create spanwise vorticity originating from saddle points to nodal points.

Bearman & Tombazis evaluated the effects of the forces and flow around a structure with a wavy trailing edge and a cross-sectional area of a half-ellipse [6]. Contrary to the findings of Ahmed & Bays-Muchmore, Bearman & Tombazis observed smaller drag on the nodal points of the structure [2, 6]. Overall drag reduction of the structure was also noted by Bearman & Tombazis. Further reduction in the drag coefficient was observed with the increase in the amplitude of the waves in the

Figure 2.18: Visualisations of flow around straight circular cylinder (Top) and flow around curved circular cylinder (Bottom). Top left : flow around straight circular cylinder at $Re = 200$; Top right : flow around straight circular cylinder at $Re = 275$; Bottom left : flow around curved circular cylinder at $Re = 100$; Bottom right : flow around curved circular cylinder at $Re = 500$ [104, 68].

Figure 2.20: Globally locked-in vortex shedding on the mildly-tapered cylinder (see figure 2.19). The mid-section (smallest diameter) of the cylinder span was observed to lead the vortex shedding of the mildly-tapered cylinder [74].

Figure 2.21: Globally locked-in vortex shedding on the mildly-tapered cylinder (see figure 2.19). The mid-section (smallest diameter) of the cylinder span was observed to lead the vortex shedding of the mildly-tapered cylinder [74].

Figure 2.22: Vortex shedding modes of a wavy-trailing-edged half-ellipse. The shaded areas represented the positions of vortex splits (or dislocations). f_1 was the first vortex shedding frequency and f_2 was the second vortex shedding frequency, where $f_1 \neq f_2$ [85].

Bearman & Owen [7]. From their experimental results, a 30% in drag reduction was observed for a square cylinder with a wavy leading edge as compared to a square cylinder. Suppression of vortex shedding was observed to occur for the ratio of amplitude of waviness to wavelength of waviness between 0.03 and 0.045. Figure 2.23 depicts the plot of drag coefficients of thin plates and rectangular cylinder with wavy leading edge, C_D , as a function of the wave steepness (w/L), where w was two times of the amplitude of the waves and L was the wavelength of the waves. With reference to Figure 2.23, it can be observed that there was a threshold of wave steepness between 0.03 and 0.06 for optimum drag reduction on wavy leading-edged rectangular cylinder. Further increase in wave steepness of a rectangular cylinder was found to be ineffective in reducing the drag. Although an increase in wave steepness was found to reduce drag force on a wavy leading-edged thin plates, the reduction was found to be more gradual. A near-linear decrease in drag was observed as the wave steepness was increased from 0 to 0.09. Further increase in the wave steepness of the wavy leading-edged thin plates was found to be ineffective in reducing the drag. Interestingly, despite the different drag coefficients between rectangular cylinder and thin plates with no waves, the saturated drag coefficients of the wavy leading-edged rectangular cylinder and thin plates was found to collapse into a single value (from $w/L > 0.09$). It was important to note that such optimum value of wave steepness was dependent on Reynolds Number. Reduction in VIV was also observed for circular cylinder with hemisphere bumps located helically on the surface of the cylinder. The drag of this bumpy cylinder was observed to be reduced by approximately 25%. It was important to note that the bumps on the

leading edge should be larger than the bumps on the trailing edge for VIV reduction. Darekar & Sherwin performed low Reynolds Number simulations (from 10 to 150) on a square cylinder with wavy leading and trailing edges that were in phase with each other [24]. They simulated only one wavelength of the structure, as such, the full dynamics of the flow around the wavy square cylinder may not be fully represented. Similar to findings by Ahmed et al, spanwise vorticity was found in the wake of the wavy square cylinder [2, 1]. This spanwise vorticity was found to result in the breakdown of von Kármán street. Consequently, the drag coefficient of the structure was reduced by 16% as the near wake of the structure becomes steady and symmetric. Figure 2.24 and 2.25 illustrate the variation of the drag coefficients and RMS lift coefficients as functions of wave steepness, W/λ , and wavelength, λ/D , for small and large wavelengths, respectively, where W was twice the amplitude of the waves, D was the length of the sides of the square section and λ was the wavelength of the sinusoidal wave on the square section. From Figure 2.24, it can be observed that the reduction in drag and lift coefficients are, at least, as equally affected by a change in wavelength as a change in wave steepness for small wavelengths. As observed from figures 2.24 and 2.25, such occurrence was most obvious for $\lambda_D = 5.6$. For other wavelengths, the effects of wave steepness were observed to be more profound. The effect of wave steepness was most prominent for $W/\lambda < 0.1$. Darekar & Sherwin found that $\lambda/D = 5.6$ was a critical wavelength, where changes in wake structure were observed for wavelengths above and below this critical wavelength. Through their analysis, it was found that a square cylinder with wavy leading edge was more sensitive to wave steepness than a square cylinder with a wavy trailing edge. This was because the flow has a longer time of evolution for a square cylinder with wavy leading edge.

Lam et al have studied the effects of undulations of the diameter of a circular cylinder at $Re = 20,000 - 50,000$ [61]. They observed a 20% drag reduction on the wavy cylinder as compared to circular cylinders. The reduction in drag of the wavy cylinder was found to be a function of the degree of obliqueness of the wavy cylinder, defined as $a^2/\lambda D_{min}$, where a was the amplitude of the waves, λ was the wavelength and D_{min} was the minimum diameter of the wavy cylinder. From their experimental flow visualisation, it was observed that the geometric saddles led the separation of flow around the wavy cylinder. Such lead in flow separation was extended as the obliqueness of the wavy cylinder was increased. This was similar to the findings by Nuzzi et al on a mildly tapered circular cylinder where vortex shedding was found to occur on the saddle point (minimum cross

Figure 2.23: Variation of drag coefficients (C_D) as a function of wave steepness, w/L , w was two times the amplitude of the waves and L was the wavelength of the waves. Triangle mark - thin plates with wavy leading edge; square mark - rectangular cylinder with wavy leading edge [7].

Figure 2.24: Mean drag coefficients (C_D) and root mean square lift coefficients ($C_{L,RMS}$) as functions of wave steepness, W/λ , and wavelength, λ/D for small wavelengths ($\lambda \leq 5.6$), where λ was the wavelength of the undulations and W was twice the amplitude of the undulations on the wavy square cylinder, and D was the width of the square section perpendicular to the flow [24].

sectional area). Lam et al also found a similarity between the frequency of vortex shedding of a wavy cylinder and a circular cylinder. However, the lift coefficient of the wavy cylinder was found to be significantly reduced as compared to a circular cylinder. From flow visualisation, Lam et al observed similar vortex formation lengths to the results of Ahmed et al [2, 1, 62]. Longer and wider vortex formation region of the wavy cylinder was identified as compared to that of a circular cylinder. This was particularly prominent in the nodal sections of the wavy cylinder. Similar to previous studies, the spanwise vorticity was found to originate from the saddle point towards the neighbouring nodal

Figure 2.25: Mean drag coefficients (C_D) and root mean square lift coefficients ($C_{L,RMS}$) as functions of wave steepness, W/λ , and wavelength, λ/D for large wavelengths ($\lambda \geq 5.6$), where λ was the wavelength of the undulations and W was twice the amplitude of the undulations on the wavy square cylinder, and D was the width of the square section perpendicular to the flow [24].

planes. Figure 2.26 shows the velocity plot of the wake behind the wavy cylinders at various spanwise positions for $Re = 3,000$. From Figure 2.26, a wider spread on saddle planes of the wavy cylinder was observed as compared to the circular cylinder from the flow velocity profile. The reverse was also found to be true for the nodal planes of the wavy cylinder. Such occurrence was observed in the flow up to $Re = 9,000$. It can also be seen that the point of minimum flow velocity occur much later in the wake of a wavy cylinder as compared to the circular cylinder. The recover distance of the wavy cylinder was also observed to be greater than that of a circular cylinder. However, the three-dimensional vortical structures of the wavy cylinder was found to breakdown much more easily due to interaction amongst themselves. As mentioned earlier, flow separation was found to occur earlier in saddle planes. Figure 2.27 depicts the streamlines of flow around a wavy cylinder. From figure 2.27, it can be seen that the separation line followed the shape of the trailing edge. The streamlines were observed to be parallel to the freestream flow only along the saddle and nodal points of the wavy cylinder. In order to identify the geometrical effects of the waves, Lam et al performed flow simulations for various geometries [59]. Figure 2.28 illustrates the mean separation lines of the flow around a circular cylinder, wavy cylinder with small wave amplitude and wavy cylinder with large wave amplitude. With reference to figure 2.28, Lam et al believed that the wavy separation line on the wavy cylinder was the main cause of wider and longer vortex formation lengths. Although the separation line vary along the span of the wavy cylinder, it was observed that the mean separation line of the wavy cylinder occurred in the proximity of the separation line of the circular cylinder. Figure

Figure 2.26: Comparison of flow velocities in the wake of the wavy cylinder at $Re = 3,000$. U was defined as the local flow velocity, U_∞ was the freestream flow velocity, x was the perpendicular distance to the axis of the cylinder and d was the mean diameter of the wavy cylinder [62].

2.29 illustrates the evolution of the wakes behind the wavy cylinder at various spanwise positions.

From figure 2.29, it can be observed that mushroom-like structures were observed in the spanwise direction of the wake of the wavy cylinder. This was similar to the wake of the circular cylinder as observed from figure 2.30 [62, 104]. However, from comparison between figures 2.29 and 2.30, the wake of the wavy cylinder exhibited more disorganised streamlines with greater turbulent mixing as compared to the wake of the circular cylinder. Further evaluations were performed on the wake of the wavy cylinder at various span. Figures 2.31 and 2.32 represent the wakes of the wavy cylinder for various Reynolds Number at nodal and saddle planes, respectively. From comparison between figures 2.31 and 2.32, it can be observed that the vortex shedding patterns were very similar at the experimented Reynolds Number. However, the width of the wake was observed to be considerably different with the wider wake observed on the saddle planes in the wake of the wavy cylinder. Such occurrence was observed for flow around a wavy cylinder for $Re = 200$ to $2,000$. Lam et al also found that the vortex formation length in the wake of the wavy cylinder was longer on the saddle planes as compared to nodal planes [62]. Based on their measurements on Reynolds stresses, Lam et al found that a greater turbulence mixing was observed in the nodal planes. From their further studies, Lam et

Figure 2.27: Visualisation of streamline and separation points for flow around a wavy cylinder. U_∞ was the freestream flow velocity, a was the amplitude of the undulations, λ was the wavelength of the vibrissa and D_m was the mean diameter of the undulatory cylinder [62].

Figure 2.28: Separation lines of flow around circular cylinder (left), wavy cylinder with small wave amplitude (middle) and wavy cylinder with large wave amplitude (right). See figure 2.27 for variables definition [59].

Figure 2.29: Spanwise view on the time evolution of the wakes of a wavy cylinder at $Re = 600$. The wavy cylinder was indicated by the wavy white lines on the left of each figure. U_∞ was the free stream flow velocity, t was the time elapsed in second, and T was the shedding period of the vortices in second [62].

al found the significant effects of the wavelengths on the root-mean-square lift coefficient ($C_{L,RMS}$) and mean drag coefficient (C_D) coefficients of the wavy cylinder [59]. Figure 2.33 depicts effects of the wavelength and the amplitude of the waves of a wavy cylinder on its mean drag coefficient (C_D), root-mean-square lift coefficient ($C_{L,RMS}$), and Strouhal number (St). From figure 2.33, it can be observed that optimum reduction of drag and lift coefficients was achieved for wavelength (λ/D_m) of 1.9, where λ was the wavelength and D_m was the mean diameter of the wavy cylinder. Regardless of the wave configuration, a similar Strouhal Number between a circular cylinder and wavy cylinder was also found. For reduction of ($C_{L,RMS}$) and (C_D), a greater wave amplitude was seen to yield greater reduction on these force coefficients. However, such result was not observed for the effects of wavelength on the reduction of ($C_{L,RMS}$) and (C_D). It was found that reductions of ($C_{L,RMS}$) and (C_D) were only observed for certain optimal values of wavelength (λ/D_m). Such optimal regions on the effects of wavelength was observed to coincide with the region where the vortex shedding frequency (St) of the wavy cylinder deviated at a larger value from the vortex shedding frequency (St) of a circular cylinder, albeit still a small deviation relative to its absolute value. Figure 2.34 illustrates the vortex formation length of the wavy cylinder at various wavelength and wave amplitude configurations at $Re = 3000$. indent With reference to figure 2.34, it can be seen that the vortex formation length increased with wave amplitude. This was consistent with the corresponding reduction of ($C_{L,RMS}$) and (C_D) as steady and longer wakes were formed in the trailing edge of the body. This result in greater distance between the trailing edge of the wavy cylinder and the location of rolling-up of the vortices. Such greater distance allowed for lower forcing on the wavy cylinder as the rolled-up vortices shed from the near-field wake of the wavy cylinder. From figure 2.34, it can also be observed that vortex formation length was less conclusively correlated to the wavelengths. A trend shift in the vortex formation length as a function of wave amplitude was found as the wavelength of the wavy cylinder was modified. However, more detailed study will necessary for conclusive results.

In their latest effort, Lam et al performed studies on the effects of wavelength and wave amplitude of wavy cylinder through numerical simulation at $Re = 100$. For reduction of ($C_{L,RMS}$) and (C_D), they found a region of optimum wavelength (λ/D_m) between 2.5 and 6 for $Re = 100$ as compared to their previous experimental results where optimum wavelength (λ/D_m) of 1.9 at $Re = 3,000$ was noted. Such result indicated a strong dependency of the optimum wavelength of the wavy cylinder to the Reynolds Number for reduction of ($C_{L,RMS}$) and (C_D). Figure 2.35 documented the variations

Figure 2.30: Spanwise view of the mode B vortex shedding in the wake of a circular cylinder. Mushroom-like structures were observed along the span of the cylinder. The triple arrows indicate the direction of freestream flow, and L indicated the length of the circular cylinder [104].

of $(C_{L,RMS})$ and (C_D) for various wavelength and wave amplitude configuration. It was important to note that these findings were Reynolds Number dependent as shown through the variation of optimum wavelength for various Re [59, 60]. From figure 2.35, it can be observed that the most significant reduction of $(C_{L,RMS})$ and (C_D) occurred at $5 \leq \lambda/D_m \leq 6$. It was important to note that $(C_{L,RMS})$ and (C_D) were amplified as the wavelength of the undulations were reduced below certain values. Lam et al classified the force response of the wavy cylinder based on the wake structures of the wavy cylinder. Figures 2.36 and 2.37 illustrate the classifications of flow around a wavy cylinder and their corresponding force response as well as wakes around the wavy cylinder, respectively [60]. With reference to figures 2.36 and 2.37, it can be observed that Lam et al divided the flow regimes due to variation in characteristics of wakes and force responses into five regimes. These regimes were characterised as follows:

- * Regime I A, II A : quasi-steady two dimensional wake with no variability in spanwise direction;

Figure 2.31: Flow visualisations of the wakes of a wavy cylinder at nodal planes for various Reynolds Number. The Reynolds Number was defined based on the local Re , which was dependent on the local diameter of the wavy cylinder [62].

Figure 2.32: Flow visualisations of the wakes of a wavy cylinder at saddle planes for various Reynolds Number. The Reynolds Number was defined based on the local Re , which was dependent on the local diameter of the wavy cylinder [62].

Figure 2.33: Variability of the : a) mean drag coefficient(C_D), b) root-mean-square lift coefficient ($C_{L,RMS}$), and c) Strouhal number of the wavy cylinder at various wavelength (λ) and wave amplitude (a) combinations for $Re = 3,000$, where D_m was the mean diameter of the wavy cylinder [59].

- * Regime I B : free shear layer elongation with spatially delayed vortex shedding;
- * Regime I C and II C : steady and symmetrical wake with no vortex shedding;
- * Regime II B and III A : Fully three-dimensional wake with distortion of the wake in spanwise direction;
- * Regime III B : Complicated three-dimensional wake with spanwise expansion of the wake pattern with no visible vortex shedding in saddle plane.

It was evident that regimes with no vortex shedding experience minimum ($C_{L,RMS}$) and (C_D) such as regime I C and II C (most reduced ($C_{L,RMS}$) and (C_D)). As the saddle planes for Regime III

Figure 2.34: Vortex formation length of circular and wavy cylinder for various wavelength and wave amplitude configurations at various spanwise planes (saddle and nodal planes) for $Re = 3,000$. See figure 2.33 for variables definition [59].

Figure 2.35: Variability of the mean drag coefficient (C_D) [left] and root-mean-square lift coefficient ($C_{L,RMS}$) [right] of the wavy cylinder at various wavelength (λ) and wave amplitude (a) combinations for $Re = 100$. See figure 2.33 for variables definition [60].

C do not exhibit vortex shedding, evident reduction of ($C_{L,RMS}$) and (C_D) was also observed. Only Regime I A and II A exhibited no reduction and slight reduction of ($C_{L,RMS}$) and (C_D), respectively. Explanations for such occurrence were found in the wakes of the wavy cylinder in these regimes. Such wakes were seen to be very similar to the wake of a circular cylinder with no distortion or minimum spanwise distortion in the spanwise direction of the wakes. In turn, these regimes were governed by the topology of the undulations on the wavy cylinder. With reference to figure 2.36, it can be seen that the wave amplitude governed the variation within Regime I, II and III with smaller contribution observed from the effects of wavelength variation. However, the shift from one regime

Figure 2.36: Regimes of flow around a wavy cylinder as functions of wave amplitude (a/D_m) and wavelength (λ/D_m) due to variability of the wakes and force response of the wavy cylinder at $Re = 100$. See figure 2.33 for variables definition [60].

to another (I to II or II to III and their combinations) were strongly affected by the wavelength of the undulation with smaller effects from the wave amplitude. From all the reviewed studies on the effects of geometrical variation of a circular cylinder, the wakes of a seal vibrissa was observed to slightly resemble the wake of a wavy square cylinder reported by Darekar & Sherwin (refer to Chapter 6) [24]. Similar to findings of Lam et al, distortions in the spanwise direction of the wake was also observed in the wake of the seal vibrissa. However, regardless of the regime, the form of the wake was significantly different with the wakes of the seal vibrissae, which form a structure approximating horseshoe vortices (refer to Chapter 6). The reviewed literatures in this section will be further discussed and compared to the wakes and forcing as well as motion response of the seal vibrissa to their wakes in subsequent chapters.

From their measurements, Hanke et al also found an offset angle between the leading edge and trailing edge of the vibrissa [40]. Such offset angle was observed to vary along the length of the vibrissa. To the date of this thesis, studies on the effects of such offset angle on the forces due to vortex shedding were yet to be published. Hence, this thesis will also explore such geometrical effects on the force and motion response of the seal vibrissa to their vortex shedding.

Figure 2.37: Definitions of the flow regimes around a wavy cylinder due to various characteristics of wakes and force responses. See figure 2.33 for variables definition [60].

Chapter 3

Properties of Harbour Seal Whiskers

This chapter is devoted to the evaluation of the properties of the harbour seal whiskers. To the date of the writing of this thesis, no publications on the detail of mechanical and chemical properties of the harbour seal whiskers are yet to be found. The importance of the properties of the harbour seal whiskers could not be overstated in attempting to understand the underwater sensing principle of harbour seals. As such, this chapter is dedicated to the essential properties of the harbour seal whiskers for understanding of the underwater sensing principle of the harbour seal whiskers. This chapter evaluates the mechanical, physical and chemical properties of the harbour seal whiskers.

3.1 Mechanical Properties

This section describes the methods of vibrissae acquisition and preparations prior to experiments as well as experimental methodologies for measuring the mechanical properties of the vibrissae and their natural frequencies. It is divided into 5 subsections, namely: vibrissae preparations, measurements of mechanical properties, strain control test, immersion time control test, and measurements on the natural frequency of the vibrissae. It should be noted that the latter measurement is performed with a different experimental setup from the experimental setup for measurement of the other properties. For measurements of natural frequency of the vibrissae, the vibrissae are clamped rigidly at their bases. For measurements of other properties on Dynamic Mechanical Analysis (DMA) machine, the vibrissae are rigidly clamped on both ends. DMA machine is chosen for experiments to determine the mechanical properties of the vibrissae due to its robustness and ability to perform dynamic testing on the vibrissae [8]. This is due to the nature of the dynamic disturbances that the vibrissae on the

muzzle of the seals are exposed to as they are used for sensing. The DMA machine is also capable of analyzing the viscoelastic nature of materials. The DMA experiments are performed with frequency sweep (increasing frequency from 1 Hz to 150 Hz).

3.1.1 Vibrissae Preparations

The harbour seal vibrissae were obtained from New England Aquarium, Boston, Massachusetts, USA [3]. These vibrissae are shed periodically by the seals. Hence, they are obtained from the breeding area of the seals with no stress inflicted on the seal. As the author does not intend to harm the seals, the author focused on working with the limited number of vibrissae samples obtained from the floor in the breeding area of the seals. From onsite observation (also from figure 1.1), the vibrissae on the muzzle of the seals vary from the shortest (on the foremost of the muzzle) to the longest (on the rearmost of the muzzle) vibrissae. The majority of the vibrissae from the aquarium are within the classification of medium to long vibrissae. As such, most of the experiments in this project will utilize the long vibrissae. For experiments with DMA machine, two vibrissae of approximate length of 8 cm (long vibrissae) are used. Prior to the experiments, the vibrissae are segmented into four parts of 2 cm each. Figure 2.1 shows the two samples of seal vibrissae used in the DMA experiments. All vibrissae are immersed in water for 48 hours to remove surface bound contaminants. In the next 48 hours, the vibrissae are allowed to dry at room temperature. In order to determine the sensitivity of the properties of the seal vibrissae to immersing solution, four different media are chosen to represent the media that real vibrissae are commonly exposed to. These media include air, tap water, saline water and Hanks' Balanced Salt Solution (HBSS). For experiments on vibrissa segments immersed in water, saline water, and HBSS, the vibrissae are immersed in these fluids for 48 hours before the commencement of the experiments. Immersion of vibrissae in water and saline water allows for determination on the significance of salinity of immersing fluid on the properties of the vibrissae. The saline water is created by mixing 35 grams of salt in 1 liter of soft water and is similar to seawater in salt content. In order to examine the vibrissae properties in nourishing substances similar to the vibrissae on the seals, HBSS is chosen as one of the immersing solutions. The main reason for using HBSS for nutrition supply for the cells of the vibrissae is due to its use in supplying nutrition for cell culturing [41]. It is also a non-reactive substance with capability of maintaining a consistent pH, which will maintain the consistency of the physiology of the vibrissae. Upon removal from

Figure 3.1: Variation of normalized minor and major diameter of the short, medium, and long vibrissae along their normalized lengths.

immersing fluid, the vibrissa segments are immediately placed in DMA machine for experiments. For determination of the natural frequencies of the vibrissae, three vibrissae of lengths 1.8 cm, 5.2 cm and 7.5 cm are used. They are termed 'short', 'medium', and 'long' vibrissae, respectively. For each of these vibrissae, eight equidistant measurements on their major and minor axis are performed along their lengths with digital vernier caliper. These measurements represent the profile of the vibrissae when they are modeled as tapered elliptical beams. Figure 3.1 shows the variation of the normalized major and minor diameter for the three vibrissae along their normalized lengths. It can be observed that the major and minor diameters have decreasing and undulating dimensions from the base to the tip of the vibrissae. For all vibrissae, except the major diameter of the short vibrissa, their dimensions are within enclosed range of $0.02 < b/L < 0.2$.

3.1.2 Measurements of Mechanical Properties

As the seals sensed perturbation through the vibrations of their vibrissae, the relevant mechanical properties to be evaluated are the modulus of elasticity, viscoelasticity, and damping. Experiments and measurements were performed with DMA machine fitted with fiber tension clamp. The vibrissa segments were rigidly clamped at both ends to avoid slip during DMA tension experiments. For

experiments on vibrissa segments immersed in fluids, the segments were removed from the fluid and clamped on the DMA machine within 5 seconds. The DMA machine was calibrated to ensure the tightness of the clamp as well as the accuracy of the displacements, forces and the masses of the parts in the experiments. After calibrations, the vibrissae were subjected to periodic applications of axial tensile force in frequency sweep experiments from 1 to 150 Hz at an interval of 5.64 Hz. During these experiments, the vibrissa is subjected to constant strain of 0.001%. A small strain is chosen for the experiments to avoid plastic deformations on the vibrissae, which would permanently damage the vibrissa as well as modify their properties. Each experiment was repeated three times for four different segments of a single vibrissa (see 'Vibrissae preparations' for vibrissa segments used in DMA). In these experiments, stress and strain measurements are outputs of the DMA experiments. The ratio of stress to strain provided a good estimate of the effective modulus of elasticity of the vibrissae necessary for the understanding of the large-scale dynamics of the vibrissae. The variability of the estimated modulus of elasticity as a function of frequency was an indication of the viscoelasticity of the vibrissae. Viscoelasticity was a measure of tendency of a solid to behave more fluid-like under stress. As biological entities such as muscles and rat vibrissae have exhibited viscoelastic behavior under tension, it is imperative for us to evaluate the viscoelasticity of the vibrissae [82, 17, 35, 49]. It should be noted that the viscoelasticity of the vibrissae also represented the effects of the geometry of the vibrissae on the stress-strain property of the vibrissae. This was represented as the overall variability of the strain rate of the vibrissae under small deformations. As it would be described on 'Results' section, the vibrissae are non-viscoelastic, hence, the estimated modulus of elasticity provides an accurate assessment of the modulus of elasticity of the vibrissae. The damping of the vibrissa is represented by tangent delta with delta defined as the phase angle between the stress and strain indicating the energy dissipation by the material [66]. From here onwards, the variable 'tangent delta' will be abbreviated as 'tan delta'. Tan delta is also a measurement output from the DMA experiments. As the vibrissa were divided into four segments, the following indications and their respective vibrissa segments were used throughout the this thesis. The first vibrissa segment is the segment located closest to the muzzle of the seal, while the fourth segment is closest to the free end (tip) of the vibrissa. The second and third segments are located consecutively in between first and fourth segments along the length of the vibrissa.

Figure 3.2: Modulus of Elasticity of harbour seal vibrissa subjected to axial strain of 0.001% and 0.005%.

3.1.3 Strain Control Test

In order to ensure that the strain applied during the experiments does not cause plastic deformations, control tests are conducted with varying strains on saline water-immersed vibrissa. The vibrissa is clamped on both ends to the DMA machine and is subjected to 0.001% and 0.005% strain applied through frequency range of 1 Hz to 200 Hz with force applied as the aforementioned vibrissae experiments with DMA Machine (See 'Measurements of Mechanical Properties'). Figure 3.2 illustrates the comparison between the modulus of elasticity measured in the control tests.

It can be observed that the agreement between the two curves is excellent for frequency of force application between 1 to 93 Hz. This agreement reduces dramatically beyond 93 Hz where the frequency of force application approaches the natural frequency of the system. However, in nature, it is unlikely that the vibrissae will be subjected to signals with frequencies beyond 93 Hz underwater. Hence, this paper prioritizes the properties of the vibrissae at frequencies below or equal to 93 Hz. Measurement on properties beyond 93 Hz will be performed, however, these measurements will not be as significant as properties at their respective lower frequencies.

Figure 3.3: Variations of modulus of elasticity for vibrissae segments with duration of immersion in saline water. First segment represents the vibrissa segment closest to the muzzle of the seal, while fourth segment represents the free end (tip) of the vibrissa. Second and third vibrissa segments are sequentially located between first and fourth vibrissa segments.

3.1.4 Immersion Time Control Test

For vibrissae immersed in solutions, it is necessary to determine their variability with duration of immersion. Regions with minimal variability of properties with time will be established as the standard immersion time. The immersing solution in the control tests is the saline water due to its proximity to the natural habitat of the harbour seals. The dry vibrissae segments represent segments with 'zero' immersion hour. Other durations of immersion experimented are 48 hours and 168 hours, respectively. It is important to note that the seals commonly dive for 10 to 20 minutes, which is equivalent to the duration of vibrissae immersion in seawater [86]. However, the fluid in the cavity of the vibrissae may also significantly affect the vibrissae properties. Hence, longer duration of immersion is tested to ensure consistency and accuracy of the properties. For each vibrissa segment and duration of immersion, three measurements are recorded for frequencies from 1 to 150 Hz. For consistency tests on a shorter time span, a randomly chosen vibrissa segment is tested for one-hour immersion time. The chosen segment is the second segment and three measurements are recorded for frequency 1 Hz to 153 Hz. Figure 3.3 shows the variation of mean modulus of elasticity for vibrissae segments with duration of immersion in saline water.

From figure 3.3, it can be observed that modification on the modulus of elasticity of the second segment between 1-hour and 48-hours duration of immersion is negligible in comparison to the observed variation in the first hour. A 168-hours immersion is found to soften the vibrissae significantly. It is believed that a 168-hours duration of immersion results in changes that are unrealistic (as well as in comparison to the duration of diving of the seal). Hence, the standard duration of immersion of the vibrissae segments for our DMA experiments is established to be 48 hours.

3.1.5 Determining The Natural Frequencies of The Vibrissae

For a sensing system to detect perturbations, it is imperative for such sensors to have natural frequencies that are distant from their target frequencies. As such, the response of the vibrissae will strongly depend on their natural frequencies. An experimental setup is built to determine the natural frequencies of the vibrissae. Figure 3.4 depicts the schematic diagram of the experimental setup.

The experimental setup consists of a Brel and Kjr Type 4810 mini-shaker, Brel and Kjr Type 2718 power amplifier, Agilent 33120 Function waveform generator, Tektronix TDS 1012 oscilloscope and Photron Fastcam 1024 PCI high-speed camera. The camera is operated at 10,000 fps (frames per second) with a resolution field of 256 x 256 pixels. The uncertainty level is estimated at 2.5 %. It is important to note that the field of view of the high-speed camera is on the plane of vibration of the vibrissae. For these experiments, the vibrissa is clamped firmly at one end with a four-way clamp. The other end of the clamp is tightened into a metal holder, which is screwed to a mini-shaker. The waveform generator creates the motion input for the mini-shaker through prior amplification with a power amplifier. In order to ensure the correct waveform, an oscilloscope is connected to the output of the waveform generator to visualize the waveform transmitted to the metal holder. The experiment is recorded with a high-speed camera, for subsequent analysis in determining the occurrence of the natural frequencies of the vibrissae. For these experiments, the input wave is sinusoidal wave with frequency of oscillation varying from 1 to 200 Hz with an interval of 1 Hz. Once the initial experiment is completed, the region of frequency of oscillation with higher amplitude of oscillation is repeated with an interval of 0.1 Hz. Experiments on the effects of angular orientations of the vibrissae on their natural frequencies are also evaluated. However, our results indicate that the natural frequencies of the vibrissae are independent of their angular orientation. More discussions on the implications of the vibrissae neutrality to its angular orientation will be presented in Results

Figure 3.4: The overall experimental setup for determining the natural frequency of the harbour seal vibrissae. Setup: Light, high-speed camera, power amplifier, mini-shaker and vibrissa, waveform generator, oscilloscope. The field of view of camera is on the plane of vibration of the vibrissae.

section of this chapter. From the analytical model derived in Results section, it can be observed that the density of the vibrissae is an important variable in determining their natural frequencies. The density is calculated as the ratio of the mass of the vibrissae segments to their volume. In order to increase measurement accuracy, individual densities of the vibrissae segments are utilized instead of overall density. The masses of the vibrissae segments are measured with Sartorius TE64 microbalance coupled microgram resolution. The volume is estimated with vibrissae modeled as tapered elliptical cylinder coupled with earlier geometrical measurements.

3.2 Results

This section is divided into 3 subcategories as follows: the properties of vibrissae in various solutions, the spatial variation of the properties of vibrissae, and the natural frequency of the vibrissae.

3.2.1 Properties of Vibrissae in Various Solutions

As aforementioned, given the importance of bending for the underwater sensing of the seals, the evaluated properties are modulus of elasticity and damping of the vibrissae. However, it is also important to determine the viscoelasticity of the vibrissae. Hence, this paper attempts to qualitatively describe the viscoelasticity of the vibrissae.

Modulus of Elasticity

An important factor affecting the bending of the vibrissae during sensing is their bending stiffness. As the same vibrissa segment is tested in various solutions, the bending stiffness of the vibrissa segments is only affected by its modulus of elasticity. Figure 3.5 shows the modulus of elasticity of first vibrissa segment immersed in various fluids subjected to axial tensile force at varying frequencies (see 'Vibrissae preparations').

As expected, the largest modulus of elasticity is observed on dry vibrissa, where the molecules of the vibrissa are not in bondage with molecules of the solution. Such bondage will result in lowering of stiffness and increase in flexibility due to the entrapped water molecules in the vibrissa, as observed in vibrissa immersed in water based solutions. Consequently, saline water-immersed vibrissa and HBSS-immersed vibrissa show lower moduli of elasticity as compared to dry vibrissa with an average of 3.78 GPa. A water-immersed vibrissa is found to experience a significant decrease in modulus of elasticity, approximately 47% of the modulus of elasticity of dry vibrissa. This is most likely due to the lower surface tension of water (as compared to salt water), which allows for greater penetration of the water molecules into the cavity between the vibrissae microstructure. Significant changes on the moduli of elasticity of the vibrissae are observed for frequency of force application beyond 140 Hz. This is due to the proximity to the frequency of force applications to the natural frequency (between 130 and 180 Hz, as shown in figure 3.2 and as indicated in Section 3.1.3) of the vibrissa fixed at both ends. Minimal variations are observed for the moduli of elasticity of the vibrissa at frequencies of

Figure 3.5: Modulus of Elasticity of vibrissa segments immersed in dry, water, saline water and HBSS at varying frequency of axial force applications.

force application below 140 Hz.

Viscoelasticity of Vibrissae Segments

As viscoelasticity describes the variation of the mechanical properties under external forcing, it is imperative to establish qualitative understanding on the viscoelasticity of the vibrissae. Such understanding will provide a greater insight on the variation of the mechanical properties of the vibrissae during their period of sensing. With reference to figure 3.5, qualitative description on the viscoelasticity of the vibrissae can be obtained from observations on the variability of the modulus of elasticity at varying frequency of force application. The variability of the modulus of elasticity is calculated as the standard deviation in statistical analysis. It is noted that the standard deviation (variability) of the dry, saline water-immersed, HBSS-immersed, and water-immersed vibrissae are 3.9%, 7.3%, 6.3%, and 13.78% of their respective average modulus of elasticity (with exclusion of resonance region, $f_n > 140\text{Hz}$). As such, with exception of water-immersed vibrissae, the vibrissae exhibited small variability in their modulus of elasticity under varying forcing. Hence, it can be qualitatively deduced that the vibrissae have negligible viscoelasticity. Consequently, the variations of the modulus of elasticity of the vibrissae during underwater sensing can be neglected, with exception for the vibrissae

Figure 3.6: Damping of vibrissa segments immersed in dry, water, saline water and HBSS at varying frequency of axial force applications.

immersed in water.

Damping

As the seals' sensing depends on the bending of their vibrissae, the amount of perturbations transmitted to the base of the vibrissae is very important. Such transmission is dependent on the damping properties of the vibrissae. As aforementioned, tan delta indicates the amount of damping in the vibrissae for DMA experiments. Figure 3.6 shows the variations of tan delta for vibrissa segments immersed in various fluids at varying frequencies of force application.

If a material has tan delta of less than 0.01, the material is classified as glassy. For materials with tan delta between 0.1 and 1.2, the material is classified as leather-like. From figure 3.6, it can be observed that, for frequency range of 1 Hz to 120 Hz, the tan deltas of harbour seal vibrissae are between 0.03 and 0.06. Hence, it can be deduced that the damping behavior of the vibrissae is relatively small as it is between glass-like and low damping leather-like materials. Immersion of vibrissae in solutions is observed to increase tan delta by approximately 100% with minute differences when different solutions are used as immersing agent. However, similarity to modulus of elasticity is observed as the frequency of force application approaches the natural frequency of the system at 140

Hz, the damping of the vibrissae is found to change significantly. The damping of the vibrissae may be important in studies with strong emphasis on accuracy of forcing. However, for the purpose of understanding the seal's underwater sensing principle, the damping will be considered as negligible.

3.2.2 Spatial Variation of The Properties of Vibrissae

The previous subsection evaluates the variations of the mechanical properties of the vibrissae segments when they are immersed in different relevant solutions. This subsection provides description on the mechanical properties of the overall vibrissae. The vibrissae are divided into four segments of equal lengths and are tested individually. In this subsection, our study focuses on the properties of the vibrissae in dry condition and immersed in saline water. These media are chosen due to their strong relevance as the habitats of the harbour seals. Evaluations of the properties of the vibrissae in water or HBSS are considered as less relevant due to their limited availability in natural habitat of harbour seals and their irregular effects on the vibrissae, respectively. The results for each vibrissa segment in the following experiments are taken from average between 1 Hz to 93 Hz for reasons stated previously (See Materials & Methods). Measurement results on vibrissa segments near the base to the tip of the vibrissae will be presented.

Modulus of Elasticity

As the vibrissae are segmented, the mechanical properties of the vibrissa are presented for each segment of the vibrissae. Figure 3.7 shows the average of the modulus of elasticity of the four dry and saline water vibrissae segments along their lengths (same vibrissae segments are used for experiments on different media, see 'Materials & Methods').

Unlike in rat whisker [43], the modulus of elasticity of dry seal vibrissae are found to vary along the length. It can be observed that the decrease of the modulus of elasticity from the base (~ 5.8 GPa) to the free-end tip (~ 2 GPa) of the vibrissae is very significant and it follows a near-linear trajectory. Similar variation on modulus of elasticity along the length of the vibrissae is observed for saline water immersed vibrissae. The line of modulus of elasticity for saline water-immersed vibrissae is nearly parallel to the line of modulus of elasticity for dry vibrissae. However, the modulus of elasticity of the saline water-immersed vibrissae is always smaller than the modulus of elasticity of the dry vibrissae. This is as expected because the voids between the cross-networks of the keratin in dry

Figure 3.7: Average Modulus of Elasticity of dry (blue) and saline water-immersed (red) vibrissae segments. See figure 3.3 for information on spatial positions of vibrissa segments.

vibrissae will undergo greater stress under equivalent strain with the saline water-immersed vibrissae, whose similar voids are entrained with electrolytes. Given the linear variation of the modulus of elasticity and the tapering profile of the vibrissae, bending stiffness (EI) of the vibrissae is found to decrease very significantly along their lengths. The decrease in bending stiffness follows a fifth-order exponential decay curve, as shown in figure 3.11. The significant decrease in bending stiffness allows for greater transduction of the perturbations near the tip of the vibrissae, as smaller resistance (inertia) is encountered during such transduction. However, it is necessary to note that the seal vibrissae simply achieve an optimum balance between their inertia and transmission efficiency. It is erroneous to deduce that only with smaller inertia, a better transduction could be achieved. For dry and saline water-immersed vibrissae, the greatest variation in modulus of elasticity is observed between first segment and second segment of the vibrissae. This indicates that the largest bending stiffness is near the base of the vibrissae. It provides strong reinforcement for the structural design of the vibrissae to ensure survival in harsh conditions. Despite the greater bending stiffness at the base, it is important to note that the transmission of the signals from the free-tip ends there. As such, significant signal dissipation is expected to occur near the base of the vibrissae.

Figure 3.8: Average damping of dry (blue) and saline water-immersed (red) vibrissae segments. See figure 3.3 for information on spatial positioning of vibrissa segments.

Damping

Although the modulus of elasticity decreases along their lengths, similar trends are not expected for damping due to their different nature of response to the introduction of water molecules in the cavity of the vibrissae microstructures. Figure 3.8 illustrates the average tan delta for dry and saline water-immersed vibrissae segments along their lengths.

Slight variations in damping along the length of the vibrissae are observed with vibrissae segments exhibiting glassy properties an insignificant damping along their lengths. However, the absolute values of tan delta for saline water immersed vibrissae are nearly 1.5 times to the tan delta of dry vibrissae. This indicates that immersion in saline water increases the damping across the length of the vibrissae. Such increase in damping is expected as the saline water fills the void in the cross-networks of the keratin. The above results show that the moduli of elasticity of the vibrissae are approximately a million times larger than the modulus of elasticity of muscle (soft tissue) [17]. It is also observed that the modulus of elasticity and spring constant of the vibrissae vary along their lengths. However, the damping properties of the vibrissae are negligible and are nearly uniform along the their lengths. Immersion of vibrissae in saline water is found to soften the vibrissae while

doubling their damping properties. However, despite this doubling, the vibrissae are still classified as material with almost negligible damping properties. As such, it is necessary to thoroughly understand the mechanical properties of the vibrissae due to its dominance in the seals' muscle-vibrissae system. It is important to note that the mechanical properties of the long vibrissae tested with DMA show a good consistency among themselves. This seems to differ from the findings on the rat vibrissae where significant variation on their mechanical properties are observed across the 24 vibrissae [49]. However, due to the limited number of vibrissae analyzed in this paper, the author believes that further analysis on vibrissae of different lengths would be necessary to achieve a more conclusive end.

3.2.3 Natural Frequency of The Vibrissae

Many models have been developed to understand the sensing ability of the rats with their vibrissae [43, 108]. These models assume that the rat vibrissae are geometrically similar to a cone or a truncated cone. The results from Yan et al cover a large variety of boundary conditions with a truncated cone model. However, the resulting equations became relatively inefficient due to the need for solution with numerical model. As such, an analytical model will be preferred here to represent a simpler and more intuitive solution. The solution obtained with this simple equation will be compared to experimental results to provide qualitative conclusion on the natural frequency of the harbour seal vibrissae. It should also be noted that the natural frequency of the vibrissae is found to be independent to their angular orientation. This implies that the centers of mass of the vibrissae are located in the region where the curvature is nearly zero. Such implication is also supported by the shape of the vibrissae geometry, which tapers from the base to the free end of the vibrissae. Hence, the mass elements of the vibrissae are greater near the base of the vibrissae.

Effects of Variation of The Modulus of Elasticity on The Natural Frequency of The Vibrissae

As the moduli of elasticity of the harbour seal vibrissae vary along their lengths, it is important for us to evaluate this effect on the response of the vibrissae. An analytical model is derived to estimate the natural frequencies of the lateral vibration of the vibrissae about their major axis in a cantilever configuration. The vibrissae are assumed to be an elastic body and the equations for estimating the natural frequency of the vibrissae about their major axes are as follows [84]:

Figure 3.9: (Left) Cross-sectional view of harbour seal vibrissa with Scanning Electron Microscope (SEM) at 70x magnification, (Right) Close up view on the hole in the vibrissa at 1000x magnification.

$$f_n = \frac{2\pi}{\lambda^2} \sqrt{\frac{EI}{A\rho}} \quad (3.1)$$

λ is dependent on the mode of vibration of interest. For calculation of the first natural mode, λ is two times the length of the vibrissae (L). For determination of the second natural mode, λ is equal to the length of the vibrissae. The underlying assumption in equation 3.1 is that the lengths of the vibrissae are significantly greater than the diameter of the vibrissae. Consequently, this equation will only be applicable to the prediction of natural frequencies of the long vibrissae. The moment of inertia and average cross sectional area are calculated from measurements on the long vibrissae with the vibrissae modeled as tapered elliptical beams. The following equations describe the method of calculating the moment of inertia and the average cross sectional area [58]:

$$I = \frac{\pi\rho L}{64} \int_0^L ab^3 dz + \frac{\pi\rho}{4L} \int_0^L abz^3 dz \quad (3.2)$$

$$A = \frac{1}{L} \int_0^L \pi abd dz \quad (3.3)$$

Effects of Cavity on Natural Frequency of Vibrissae

Through observation with Scanning Electron Microscope (SEM), interesting cross-sectional features of the harbour seal whiskers are observed. Figure 3.9 depicts the cross-sectional view of the internals of the harbour seal vibrissae.

Two vibrissae were used in cross-sectional area analysis. The vibrissae have been initially segmented

into four segments. These segments are coated with a 5 nm thick gold coating with a gold spin-coater to ensure their electrical conductivity for analysis under SEM. After spin-coating, these vibrissae segments are placed in SEM chamber for detailed observations on their physical features. With reference to figure 3.9, it can be observed that the vibrissae have cavity along their lengths. Hence, it is important to determine the effects of such cavity on the response of the vibrissae. As the response of the vibrissae to disturbance are most effective when their characteristic frequencies are far away from the natural frequency of the vibrissae, this subsection will focus on the effects of such cavity on the natural frequency of the vibrissa. With reference to Equation 3.1, the influences of the cavity on the natural frequency of the vibrissae are mainly contributed through the moment of inertia and the cross sectional area. The dimensions of the cavity are measured through images obtained from SEM, as shown in figure 3.9. It is observed that the cavity is elliptical in cross section with a span of 70% and 10% across the major and minor diameter of the vibrissae, respectively. However, the lengthwise span of such cavity is very difficult to measure. This is due to the need for constant segmentation of the vibrissae while maintaining their structural integrity even in section near the end that are very small. Hence, an assumption is made that the cavity is shaped as tapered ellipse along the length of the vibrissae and it extends up to 90% of the length of the vibrissae. Such an assumption is reasonable as the cavity serves to supply nutrition to the vibrissae cells. Hence, the cavity has to extend along the length of the vibrissae to serve this function. Given that the vibrissae are modelled with (cavity as vacuum space) and without cavity (cavity as solid), these results will represent the two extreme conditions of the seal vibrissae. In reality, as the nutrition is transported through fluidic means, the natural frequency of the vibrissae will be in the range of these extrema. Figure 3.10 shows the variation of natural frequencies of the vibrissae with moduli of elasticity and cavity.

It can be observed that the variation on the modulus of elasticity significantly affects the second and third mode natural frequencies. Smaller variations are observed on the first mode natural frequency. However, such variation is still relatively significant with the first mode natural frequency ranging from 12.4 Hz to 27.8 Hz. Given such variation, it is important to determine the Effective Modulus of Elasticity (EME) that will represent the resultant modulus of elasticity of the beam. With reference to figure 3.10, it can be observed that there are three crossing points between our experimental data and analytical model. These crossing points are the EMEs, which represent the net effect of the length-varying modulus of elasticity. It is found that the EME decreases for higher natural modes.

Figure 3.10: Variation on natural frequencies of long vibrissae with modulus of elasticity and inner cavity.

This is expected as the higher natural modes occur at greater frequencies (as compared to lower natural modes), which causes the vibrissae to be more viscoelastic, reducing the overall stiffness of the vibrissae, as observed from figure 3.2. Hence, it is possible to define the line representing the EMEs for different natural modes of vibrations. Such line shall be utilised for further studies on the responses of long vibrissae to external perturbations. From figure 3.10, it can be observed that the cavity has considerable effects on the natural frequencies of the vibrissae. The cavity increases the average natural frequencies of the vibrissae by 24.5%. Such increment is observed on all the modes of natural frequencies. Hence, for further studies on the underwater sensing ability of the seals, it is important to include the effects of the cavity in the vibrissae model. The sensing and accuracy of the response of the vibrissae to external perturbations will be considerably different with inclusions of the cavity in the relevant mathematical models. With reference to Figure 3.11, it is found that the fifth-order exponential decay of the bending stiffness of the vibrissae causes a significant dissipation as the vibration is transmitted from the base to the tip of the vibrissae.

Figure 3.12 depicts the free-tip of the long and medium vibrissae at three different resonant modes. It can be observed that the free-tip of the vibrissae undergoes minimum vibration (in forced vibration experiment at their base). This is opposite to the case of rat vibrissae, where maximum

Figure 3.11: The variation of the bending stiffness (EI) of the vibrissae along their lengths. Each vibrissa segment is plotted with the variations of their bending stiffness at varying frequencies (from 1 Hz to 150 Hz).

vibration is observed at the free-tip of the vibrissae. However, it is important to note that the perturbations initiated at the end of the vibrissa will not be intensively dissipated due to the greater inertia receiving them. Attention must also be given to the shift in the center of mass due to the lengthwise variable spread on mass elements (in comparison with a tapered elliptical beam). However, as the current study focuses on the underwater sensing ability of the harbour seal vibrissae, such discussions will be limited here. Discussions on the effects of the added mass of the vibrissae in water will be presented in Chapter 6.

Figure 3.12: Comparison between the tip displacements for different modes of vibrations at resonance of the seal vibrissa and rat vibrissa. (Top Left) First mode resonance of the long harbour seal vibrissa, (Top Middle) Second mode resonance of the long harbour seal vibrissa, (Top Right) Third mode resonance of the long harbour seal vibrissa. (Middle Left) First mode resonance of the medium length harbour seal vibrissa, (Middle Right) Second mode of resonance of the medium length harbour seal vibrissa.

3.3 Chemical Properties of Harbour Seal Vibrissae

This section described the chemical properties of the harbour seal vibrissae. The objective of such evaluation is to ensure that the vibrissae do not consist of chemical components beyond the common composition found in human hairs and vibrissae of other mammals [78]. While it is well known that human hairs are made of keratin, similar assumptions are generally taken for rat whiskers. Even though the author suspect that the harbour seal vibrissae are composed of keratin, nevertheless, detailed analysis on the overall and surface composition of the vibrissae are performed. The overall chemical composition of harbour seal vibrissae are obtained with EDS (Energy Dispersive Spectroscopy) and the surface chemical composition of the harbour seal vibrissae are obtained with (X-Ray Photoelectron Spectrometer). Table 3.1 shows the tabulated overall chemical composition of the harbour seal vibrissae obtained from the spectrum plot of EDS.

From table 3.1, it can be observed that consistency of the abundance of each element is observed among the three runs. It is found that Carbon (C) is the most abundant element on the vibrissae at

Table 3.1: Overall chemical composition of harbour seal vibrissae. '1st', '2nd', and '3rd' represent experimental results from first, second and third runs, respectively. Weight(%) indicates the weight fraction of the element to the overall weight of the vibrissa. Atomic (%) indicates the atomic fraction of the element to the overall number of atoms in the vibrissa.

Element	Weight(%)			Atomic(%)		
	1st	2nd	3rd	1st	2nd	3rd
C	61.69	62.32	61.29	69.93	70.06	69.68
O	32.02	32.77	31.89	27.25	27.66	27.22
Na	1.22	1.19	1.36	0.72	0.7	0.81
Mg	0.31	0.42	0.46	0.17	0.23	0.26
Si	0.00	0.32	0.31	0.00	0.15	0.15
S	3.19	2.06	2.99	1.35	0.87	1.27
Cl	0.87	0.24	0.76	0.33	0.09	0.29
K	0.41	0.37	0.57	0.14	0.13	0.2
Ca	0.3	0.32	0.38	0.1	0.11	0.13

62% of the weight of the vibrissa and 70% of the atomic fraction of the vibrissa. This is as expected, as the vibrissa is observed to be similar to most most hairs and vibrissae of living organisms where the main element of composition is Carbon. Such value is found to be higher than the percentage of Carbon in human hair (50%) [78]. This accounted for the higher stiffness of the vibrissae when compared to human hair. The Oxygen(O) content of the vibrissae is also observed be slightly higher than that of human hair (difference of approximately 10% in weight fraction). Sodium is also consistently found in the vibrissa, however, this element is known to be uncommon to human hair. Sulfur is also found in the vibrissae, albeit at a lower amount as compared to human hair. Finally, other chemical elements are found in even smaller quantities in the vibrissa, e.g. Magnesium, Silicon, Chlorine, Potassium and Calcium. Table 3.2 shows the surface chemical composition of the vibrissa from XPS.

It should be noted that XPS penetrates approximately 10 nm into the surface. It is found that the surface chemical composition is very similar to the overall chemical composition. This indicates that there are no special surface coating on the vibrissa which could enhance the signal capturing ability of the harbour seal vibrissae. The main distinctions between the surface and overall chemical composition are found in the content levels of Carbon and Oxygen. The carbon content on the surface of the vibrissa is observed to 10% higher than the carbon content of the vibrissa. The oxygen content on the surface of the vibrissa is found to be 10% lower than the oxygen content of the vibrissa.

Table 3.2: Surface chemical composition of harbour seal vibrissae obtained from X-Ray Photoelectron Spectrometer (XPS). '1st', '2nd', and '3rd' represent experimental results from first, second and third runs, respectively. Weight(%) indicates the weight fraction of the element to the overall weight of the vibrissa. Atomic (%) indicates the atomic fraction of the element to the overall number of atoms in the vibrissa.

Element	Weight(%)			Atomic(%)		
	1st	2nd	3rd	1st	2nd	3rd
Na 1s	1.33	0.61	0.53	0.77	0.35	0.31
O 1s	16.72	18.74	15.77	13.9	15.61	13.09
N 1s	1.82	2.63	1.34	1.73	2.5	1.27
Ca 2p	2.26	2.32	3.43	0.75	0.77	1.14
C 1s	72.58	70.72	74.07	80.38	78.45	81.93
S 2p	0.63	0.86	0.61	0.26	0.36	0.25
Si 2p	4.66	4.14	4.26	2.21	1.96	2.01

However, as the vibrissae are collected from the floor in the harbour seal habitat of New England Aquarium, it was also contaminated by human hands and surrounding air. Hence, it is suspected that the carbon and oxygen content on the surface of the vibrissa is supposedly similar to the carbon and oxygen content of the vibrissa. Additionally, as XPS only penetrates 10 nm into the surface of the vibrissa, it is possible that the surface chemical composition detected by XPS includes the contaminants unremoved by water immersion. It is also important to note that contamination on the inner section of the vibrissa is less likely given the short duration of exposure time to the environment and human contact, as compared to the surface of the vibrissa. Such contamination will most likely increase the carbon content on the surface of the vibrissa. This will result in the decrease of the oxygen content on the surface of the vibrissa as compared to the oxygen content of the vibrissa.

3.4 Discussion

In this section, the author will discuss the findings in this chapter and their implications to the understanding on the underwater sensing principle of the harbour seal whiskers. Comparison and combinations with previous experimental results will also be presented.

3.4.1 Linking present results and previous studies

For the same amount of vibrissae displacement, a larger EME will result in greater force transmitted to the base of the vibrissae. A greater force will result in larger chance of sensing perturbations by the seals. However, previous studies have indicated that increment in the threshold of deflections is necessary for the seal to sense lower frequencies perturbations [77, 69]. If it is assumed that the seal senses perturbations based on only the magnitude of the forces, such findings will be illogical. This is because of the fact that lower frequency perturbations will result in greater EME. Hence, for the same amount of deflection, greater forces are sensed at the base of the vibrissae. This clearly contradicts the finding from previous studies [69]. However, it is not possible to assume that the seal sensing principle is only based only on the vibrations of the vibrissae. This is because a previous study has shown that the seal nervous system is triggered by static deflections [29]. Hence, our results are in complete agreement that the seals are capable of sensing static deflections on their vibrissae as well as vibratory motions of their vibrissae. From comparison between the amount of responding RA fibres in Dykes' experiments and Figure 3.10, it is important to note that the onset of decrease in RA fibres coincided with the onset of the 'flat' section of EME line at approximately 250 Hz [29]. The three natural modes of vibration of the vibrissae are also found to be in the region of 100% responsiveness of RA fibres. Despite the increase in understanding of the underwater sensing ability of the harbour seal, further studies are necessary to understand the seals' ability in distinguishing the wakes of their predators and preys. Our present results indicate that the first mode resonance frequencies of the vibrissae fall in the range where 100% of RA fibres will be able to respond to [29]. As the seals diet mainly consist of fishes, it is necessary to indicate that the wakes of the fishes are three-dimensional inverted von-*Kármán* street wakes [107, 88]. Correspondingly, the Strouhal number ($St = fD/U$, where f is the frequency of vortices shed by the fish, D is the width of the fish wakes, U is the swimming speed of the fish) of a swimming fish is approximately 0.3 [91]. Hence, oscillatory responses of the vibrissa to the perturbations by the fish wakes will be expected. More importantly, the frequency of self-induced oscillations for the vibrissae must not overlap with the Strouhal number (St) of swimming fish. Recent studies have shown that the geometry of harbour seal vibrissae reduces vortex-induced forces [40]. From simulation results, the author has found that the Strouhal Number of the seal vibrissae to be ≈ 0.17 (refer to Chapter 6). This shows that the seal will be capable of sensing and responding to changes in the amplitude and/or the frequency of the

oscillations of their vibrissae. Such frequency of oscillation is also found to fall within the range of strong responses from RA fibres. These findings may provide strong ground for explaining the underwater sensing ability of the harbour seal and their high success rate in tracking wakes left by their preys or fellow seals [28, 80]. Further studies on the nervous system of seals will be necessary for confirmation on the above statements and conclusive proofs on harbour seal's underwater tracking ability.

3.4.2 Importance of the boundary conditions

Similar to rat vibrissae, the boundary conditions at the bases of the vibrissae strongly affect the resonance frequencies of the vibrissae [43, 108]. Yan et al found that the rat vibrissae model exhibit sensitivity to various boundary conditions. For freely vibrating tip and spring-modelled base, there exists a range of spring constant at the base of the vibrissae, which could significantly affect the natural frequency of the vibrissae. Extensive studies on straight circular cylinder have also shown that there are significantly differing responses due to varying damping and mass ratios of the circular cylinder [50, 51, 52, 38]. Biological studies have indicated that the seal protracted their vibrissae when attempts to track the wakes are made [28, 99, 27]. This implies that the boundary condition at the base of the vibrissa stiffen when the seals are performing underwater detection. Such implication is supported by in situ testing by author's colleague at New England Aquarium, Boston, USA. It is observed that when the seals are not feeding or sensing, their vibrissae are loosely attached to the muzzle at the base of the vibrissae. This variation of boundary condition will significantly affect the rigidity and damping at the base of the vibrissae. The complexity of the problem is further increased through our current findings on the lengthwise variations of moduli of elasticity and spring constants. This implies that perturbations introduced at certain location along the length of the vibrissae will be modified as they are transmitted along the length of the vibrissae. Fluid-vibrissae interactions provide further challenge in attempts to achieve for conclusive understanding on harbour seals' underwater sensing principle. However, a big step forward could be taken through determination on the precise boundary condition with an in situ testing at the base of the vibrissae while the seals are tracking their preys.

3.4.3 Understanding the underwater tracking ability of harbour seal

As indicated earlier in this chapter, no nerves along the vibrissae are observed. Hence, perturbations along the length of the vibrissa have to be transmitted to the base of the vibrissae for seals' sensing and interpretations. It is believed that the intensity of the wakes plays a strong role in allowing the seal to differentiate between the wakes of predators and prey. However, different swimming method and sizes will generate varying wake intensity, size as well as structure [39]. Hence, further studies are requested for greater understanding on this complex underwater tracking principle. Previous studies have indicated that the undulations on the vibrissae reduced vortex induced vibrations (VIV) [40, 67]. This implies that the vibrissae are subjected to minute self-induce vibrations as the seals are swimming. This will allow for greater sensitivity to lateral perturbations on the seal vibrissae. However, the large number of vibrissae and their interactions with each other as well as surrounding fluid will require detailed investigations to uncover the details of the underwater tracking ability of the seal.

Chapter 4

Experiments on VIV Reduction Properties of Harbour Seal Vibrissae

This chapter describes the towing experiments of vibrissae with and without wake generators. The objective of the towing experiments is to validate the hypothesis on the underwater sensing principle of the harbour seal vibrissae. Such hypothesis are described in the following section. The towing experiments are performed in the towing tank facility at Singapore-MIT Alliance for Research and Technology located in University Town, Singapore. The towing experiments are performed with 3D printed vibrissae model. As such, dimensional analysis are utilised in sizing up of the vibrissae model and the experimental setup. Two types of towing experiments will be performed. The first towing experiment involved the towing of a single vibrissa model without any wake generator upstream. This experiment is devised to understand the response of the vibrissae to their own wakes, known as Flow-Induced Vibration (FIV). The second towing experiment is designed for a single vibrissa trailing a circular cylinder as wake generator. The circular cylinder is five times of the characteristic diameter of the vibrissa model. For both types of experiments, reference structures are interchanged with vibrissa for comparison. Circular cylinder and elliptical cylinder of the same aspect ratio are utilised as the reference structures due to the extensive literature available on them. For towing experiments with wake generators, two variety of experiments are performed through the absence and presence of transverse offset positioning with respect to the wake-generator circular cylinder. Variation of speed settings are also implemented on the experimental setup, as follows:

* Towing speed of vibrissa model equal to towing speed of wake generator;

- * Towing speed of vibrissa model faster than towing speed of wake generator. This resulted in approaching vibrissa model relative to the wake generator, which represented the real-life case of harbour seals approaching their preys.
- * Towing speed of vibrissa model slower than towing speed of wake generator. This resulted in distancing vibrissa model relative to the wake generator, which represented the real-life case of harbour seals losing their preys.

4.1 Hypothesis

This section is devoted to explanation on the governing hypothesis of the underwater sensing principle of the harbour seal vibrissae. Such hypothesis will be the main objective of the experiments as well as the numerical simulations in later chapters. With reference to literatures described in Chapter 2, hypotheses on underwater sensing principle of harbour seal vibrissae are proposed. The implications of the validity of such hypothesis will also be discussed.

From earlier biological studies, it is found that the seals have more than 44 ($\pm 1-2$) vibrissae on each side of their muzzle [26]. Hence, it is likely that the harbour seals utilised more than a single vibrissa to detect and differentiate between various types and sizes of the wakes. However, in order to understand the underwater sensing principle of the harbour seal vibrissae, it is necessary to understand the response of a single vibrissa to its own wakes and to other perturbations. Experiments on single vibrissa will aim to evaluate the effect of the physical characteristics of the vibrissa and the effects of boundary condition on the sensing properties of harbour seal vibrissae. The author hypothesised that the essential sensing characteristics of the vibrissae are their amplitude and frequency of oscillation. Figure 4.1 depicts the vibrissa-fish wakes interaction, which resulted in oscillation of the vibrissa at an amplitude of 'A' and frequency of 'f' [107, 114].

With reference to figure 4.1, the wakes of the fish are depicted as dipole sheet with its intensity indicated through the colouring [57]. The hypothesis on vibrissa underwater sensing ability illustrated the coupling between the frequency of vibrations of the vibrissa and the intensity of the sensed-wakes. The moment and torsion acting on the vibrissa will inflict various amplitudes of oscillation, which is correlated to the strength of the vortical structures. In our studies, torsional forcing are not evaluated in the experiments as well as simulations. From flow visualisation, it is observed that torsional

Figure 4.1: Interaction between vibrissa and dipole sheet strength for caudal swimming fish. A and f are the amplitude and frequency of oscillation of the vibrissa due to vibrissa-fish wakes interaction [107].

motion is not observed while bending motion is clearly identifiable through the same flow visualisation. Additionally, measurements of torsion is found to significantly increase the complexity of the experimental setup as well as degrading the accuracy of the signal sensed through bending. This is because the additional degree of freedom which is found to increase the noise in the bending direction due to the vibration of the carriage tank by the motor. The geometry of the vibrissa of the seal is hypothesised to reduced Vortex-Induced Vibration (VIV). Such reduction in VIV is hypothesised to allow for significantly reduced Flow-Induced Vibrations (FIV) on the vibrissae while the harbour seal is swimming forward with no perturbations. Such minute perturbation shall be considered as noise relative to the target water disturbances to be detected. Hence, if water disturbance are present, the seal is hypothesised to be able to sense and characterise these disturbances through the amplitude and frequency of vibrations of its vibrissae. Such reduction in the 'noise' will significantly increase the Signal To Noise Ratio (SNR) of the harbour seal vibrissa sensing principle. As such, even small water perturbations could result in significant SNR due to the significant reduction in VIV. From biological findings, this hypothesis is supported as the seals are observed to protract their vibrissae to the most forward position when they attempted to sense disturbances in their vicinity [28, 80, 99].

However, in order to properly study and understand the underwater sensing principle of the har-

Figure 4.2: Detailed view on the boundary condition at the base of the vibrissae [26].

bour seals vibrissae, it is necessary to observe and evaluate the effects of boundary condition at the base of the vibrissae. Figure 4.2 shows the detailed view of the muzzle of the seal, particularly, the base of the vibrissae. As aforementioned, the seal is observed to protract their vibrissae to the most forward position while tracking the wakes. This indicated that the muscles at the base of the vibrissae are contracted, which implied a stiffer boundary condition. However, an onsite surveillance at New England Aquarium (Boston, USA) revealed loosely bound vibrissae on the muzzle of the seal as the seal is not tracking wakes. Hence, this further reinforced the necessity to understand the effects of boundary condition at the base of the vibrissae on the response of the vibrissae to their own wakes and perturbations. Previous studies by Dykes have indicated abundant nerves on at the base of the vibrissae. From figure 3.2, it can be observed that there are no nerves along the length of the vibrissae. Such finding confirmed the common biological assumptions on the absence of nerves along the length of the vibrissae. From Dykes' finding and figure 3.2, it can be implied that all perturbations on the vibrissae have to be transmitted to the base of the vibrissae for sensing by the seals. As such, the strength of perturbations at the base of the vibrissae will be dependent on the mechanical properties of the vibrissae evaluated in Chapter 3.

4.2 Experimental Setup

This section is divided into four subsections of Dynamic Scaling of the Vibrissa-Sensor experimental setup, the setup of the vibrissa-sensor and the towing experimental setup with and without wake

generator. The first subsection explained the scaling of the experimental setup in order to preserve similitude to the real harbour seal vibrissae sensing condition. The second subsection described the parts and setup of the vibrissa-sensor, which mimicked the vibrissae of harbour seals. The third subsection identified the experimental setup and the controls of the vibrissa-model towing experiments. The final subsection illustrated the experimental setup of the vibrissa-model trailing a wake-generator circular cylinder towing experiments. With reference to figure 4.3, the vibrissa bend-sensors are always translated with linear stage (stepper motor). The linear stage has a length of 3.6 m, width of 1.5m and height of 1.5m with the water filled up to the height of 1.2 m. The stepper motor is Zaber Linear Stage, LST 1500D T3 with 200 motor steps per revolution, Repetability $< 2.5 \mu m$, accuracy $375 \mu m$, maximum speed 240 mm/s, microstep size $1.9844 \mu m$, travel range 1.5 m. The bigger motor (belt-driven) is used as wake generator with maximum speed = 1 m/s. The noise on the stepper motor are characterised with experiments using short vibrissa (where the vibrissa one end is in contact with the four bend sensors that perform deflection measurements and the other end of the vibrissa is above the free surface of the water). It is found that the noise of the linear stage in PSD plot is on the order of 10^{-7} (for reference, the signal strength of VIV on a circular cylinder in a PSD plot is on the order of 10^{-2}).

4.2.1 Dynamic Scaling of The Vibrissa-Sensor Setup

As the main geometrical features of the real vibrissae are in the scale of hundreds of micrometer (μm), it is necessary to scale-up the size of the vibrissa model to ensure clean signal with minimal noise from mechanical vibration. Such mechanical vibration is inborn from the towing experimental setup where the carriage played important roles in towing the vibrissa and/or other structures. Regardless of the size of the vibrissa model, this mechanical vibration is relatively constant. For real-scale vibrissa model (and therefore, small forcing due to perturbation), the relative strength of the mechanical vibration (noise) would have been sufficient in rendering the signals of vibrissa-sensor useless. As such, this justified the need to scale-up the size of the vibrissa model. In order to preserve the dynamics of the system, the scaling-up of the vibrissa model has to be accompanied by dynamically scaling the rest of the system. Such method is achieved through dimensional analysis with Buckingham π theorem [15]. Figure 4.3 depicted the simplified towing experimental setup.

Detailed steps of the dimensional analysis can be found in Appendix A. From dimensional anal-

Figure 4.3: Simplified sketch of the vibrissa towing experimental setup.

ysis, the following non-dimensional numbers are obtained:

$$\pi_1 = \frac{\mu}{\rho U R} = \frac{1}{Re} \quad (4.1)$$

$$\pi_2 = \frac{L_w}{R} \quad (4.2)$$

$$\pi_3 = \frac{\varepsilon}{R} \quad (4.3)$$

$$\pi_4 = \frac{E}{\rho U^2} \quad (4.4)$$

$$\pi_5 = \frac{D}{\rho U^2 R^2} \quad (4.5)$$

$$\pi_6 = \frac{M}{\rho U^2 R^3} \quad (4.6)$$

$$\pi_7 = \frac{fR}{U} = f^* \quad (4.7)$$

$$\pi_8 = \frac{\rho_w}{\rho} \quad (4.8)$$

$$\pi_9 = \frac{A}{R} \quad (4.9)$$

where:

- * ρ density of fluid (kg/m^3)
- * μ dynamic viscosity (kg/ms)
- * L_w length of vibrissae (m)
- * R diameter of ellipse perpendicular to the free stream flow (m)
- * U freestream flow velocity (m/s)
- * ε surface roughness of vibrissae (m)
- * E modulus of elasticity of vibrissae (Pa)
- * ρ_w density of vibrissae (kg/m^3)
- * D drag force (N)
- * M moment acting on the vibrissae about their spanwise-axis (Nm)
- * f frequency of vibration of vibrissae in transverse direction (Hz)
- * A amplitude of vibration of vibrissae in transverse direction (m)
- * f^* reduced velocity of the vibrissae
- * Re Reynolds number

These non-dimensional equations are used for scaling the vibrissa towing experiments. From overall experimental setup analysis, it is found that the sizing of the vibrissae is dependent on the resolution of the high-speed camera (lens focal length). Based on the resolution of the high-speed camera, it is necessary to scale the vibrissa model three times of its real-life size. However, in order to scale-up the vibrissa model, it is necessary to dynamically scale the relevant parameters represented in non-dimensional equations.

With reference to figure 4.1, as the major and minor diameter of the vibrissa model are scaled to five times of the real-life sized, the Reynolds Number is supposed to be increased. However, in order to maintain the dynamic similarity, the velocity of the vibrissa in the towing experiments should be reduced to 5 times of its real-life velocity. From equation 4.2 to 4.9, it can be observed that variation of the minor diameter of the vibrissae (the characteristic diameter) will affect other variables in the following ways:

- † From equation 4.2, in order to maintain the consistent value of π_2 while scaling-up R to $3R$, it is necessary to increase L_w to $3L_w$. This implied that the length of the vibrissae should be increased to three times the length of the real vibrissae.
- † From equation 4.3, in order to maintain the consistent value of π_3 while scaling up R to $3R$, it is necessary to increase ε to 3ε . This implied that the roughness features of the vibrissae needed to be increased to three times of the roughness feature on the real vibrissae. However, as observed from chapter 3, the roughness features on the sea vibrissae are negligible.
- † From equation 4.1, it is observed that an increase in diameter (R) needed to be accompanied by a corresponding decrease of free stream flow velocity (U). With reference to equation 4.4, such tripled-decrement in velocity resulted in a decrease of nine-times on the Young's modulus in preservation of the value of π_4 .
- † From equation 4.5, it can be observed that the three times decrease of velocity and three times increase in diameter neutralised each other. Hence, from equation 4.5, it can be deduced that the drag forces obtained from these experiments with vibrissa model are dynamically similar to the corresponding forcing on real vibrissae.
- † From equation 4.6, it can be observed that the higher power of diameter (as compared to velocity)

implied that the moment on the vibrissa model are three-times of the moment on the real vibrissa for a constant value of π_6 .

† From equation 4.7, it is found that the frequencies of oscillation of the vibrissa model must be reduced by nine-times to preserve the value of π_7 .

† From equation 4.8, it is observed that the density of the vibrissa is proportional to the fluid density. Hence, as water is used for the experiments on the vibrissa towing experiment, no scaling is necessary on the density of the vibrissa model.

† From equation 4.9, it is found that the amplitude of oscillation of the vibrissa model will be three-times of the corresponding parameter on real-life vibrissa. This is a consequence of the three folding of the diameter on the vibrissa model.

From the dynamic scaling illustrated above, the experimental setup is designed with all the necessary adjustments. Properties of the vibrissa model will be measured (refer to chapter 3) and scaled accordingly.

4.2.2 Vibrissa-Sensor Setup

In this section, the setup of the vibrissa-sensor will be illustrated. The setup consists of a half-muzzle of harbour seal, holding material near the base of the vibrissa model, four bend sensors surrounding the base of the vibrissa model and vibrissa model with coin-shaped edge at its base (for contact with bend sensors) as well as DAQ board from National Instrument (Labview 7.0), NI-USB 6218 DAQ board with sampling rate of 40,000 Hz on each bend sensor. Figure 4.4 shows the vibrissa-sensor setup and the details of its parts and mountings.

It should be noted that all the parts of the vibrissa-sensor setup are 3D-printed or casted with proportionated polymer based on the aforementioned dynamic scaling. The vibrissa model, vibrissa cap, cable mounting and half-muzzle of the seal are 3D-printed with a resolution of 0.25mm. The bend sensors in this experimental setup is commercially-available sensors, Ecoflex, which based their sensing methodology on the change of resistance of their Carbon-patch as the sensors are bent. As constant current is supplied to the bend sensors, such changes will be recorded through the change in measured voltage across the electrodes. The holding materials are casted with Smooth-on, Inc. polymers [75]. From figure 4.4a, it can be observed that the cables from DAQ board is directly connected

Figure 4.4: Details of parts and mountings of vibrissa-sensor setup consisting of half-muzzle, holding material, vibrissa model, bend sensors and vibrissa cap cap: a) Overview of vibrissa-sensor setup on the 3D-printed half-muzzle of the seal; b) Detailed view of the layout of the four bend sensors setup with the vibrissa cap passively connecting the four bend sensors; c) Overview of the mounting of the vibrissa model on the vibrissa cap, where the four bend sensors are attached to; d) Close-up view of the attachment at the coin-end of the vibrissa model to the vibrissa cap and the attachment of bend sensors on the vibrissa cap; e) Close-up view of the uncapped coin edge at the base of the vibrissa model with holding materials located near the base of the vibrissa model on the 3D-printed half-muzzle of the seal.

to the tail of the bend sensors. Power supply is also directly connected to the bend sensors through similar means. With reference to figure 4.4b, it can be observed that the bend sensors are slotted through the four-legged mountings. Subsequently, these sensors are slotted through the vibrissa cap. The vibrissa cap is designed to provide consistent physical connection between the four bend sensors to the coin-edge of the vibrissa model. This is because detachment of the bend sensors from the vibrissa model will result in inaccurate signal measurements, as such, it has to be avoided. Figure 4.4c shows the overview of the capped vibrissa model without the half-muzzle of the seal. Figure 4.4d illustrates the detailed view of the attachment between the bend sensors and vibrissa cap as well as the vibrissa cap and the coin-edge of the vibrissa model. Figure 4.4e shows the attachment of the holding material on the 3D-printed half muzzle of the seal and near the base of the vibrissa model. The holding material is designed to represent the muscle conditioning on the muzzle of the seal at the base of the vibrissae. However, earlier findings have indicated that the seal is capable of adjusting (tensing or loosening) the muscles at the base of the vibrissae [28, 99]. A precise property of the holding material would be obtained through onsite measurement on the muzzle of the seal while the seal is tracking their preys. However, such measurement would not be possible due to the requirement of in situ experiment at a very specific conditioning as well as the need to match the swimming speed of the harbour seals while they are chasing their preys. As such, in order to identify the material properties of the holding material for optimum performance, empirical methods are established. Three holding materials of varying modulus of elasticity (stiffness) is casted with Smooth-on polymer [75]. These holding materials are termed soft, medium and hard holding materials. Further discussions on the holding materials will be presented in the following section of this chapter. As observed from figure 4.4b, it can be observed that there are four bend sensors perpendicularly oriented with respect to their neighbours. These bend sensors are slotted along the perimeter of the vibrissa cap. Prior to slotting through the vibrissa cap, it has also been slotted through the four-legged mounting, which allows for electrical connection through the electrodes on the sensors. As the bend sensors are perpendicularly located to their neighbours, each pair of sensors will transmit signal changes for each axial deflection of the vibrissa model (in streamwise and crossflow directions). Such deflection is conveyed through the coin-edge of the vibrissa model. This is because the coin-edge section is the only physically-connected section of the vibrissa model to the bend sensors. The coupling of sensors to their neighbours is observed to be at its minimum due to their respective perpendicular orientation.

Figure 4.5: Calibration setup of bend sensors with actuation by linear-stage.

However, each pair of sensors oriented in the same direction will transmit similar signals due to their coupling through vibrissa cap. In order to tabulate and chart the sensitivity of the bend sensor to deflections, a calibration process is setup. Figure 4.5 illustrates the bend sensor calibration setup with linear-stage actuator.

The linear-stage actuator is chosen as compared to the carriage (rack-pinion) system for towing the vibrissa model setup. This is because of the lower mechanical vibration yielded by the linear-stage as compared to the carriage system. Figure 4.6 depicts the calibration results of the bend sensors with linear-stage.

The displacement error of the linear-stage is $1.9 \mu\text{m}$ and such error could be nullified by returning the linear stage to its home position. As such, it is ideal for calibrating the bend sensors where high positional accuracy is necessary. With reference to figure 4.6, the first dip of the signal indicated the application of near-instantaneous displacement of 1mm. A short-time delay was introduced after the introduction of deflection. Such delay allowed for observation of the temporal consistency of the signal from the bend sensor undergoing a deflection. After the short-delay, the bend sensor is returned to its original position. After reaching its original position, the linear-stage actuator is rested for approximately 5s to reach its steady-rest value before subsequent test. Subsequent deflection tests are performed at an increment of 1mm with the same procedures. The largest deflection in

Figure 4.6: Raw signals of bend sensors obtained during calibration setup with linear-stage.

the bend sensor calibration is 9mm. This is because of the slip occurring on the bend sensor as a 10mm deflection is introduced by the linear-stage actuator. The signals are captured at a sampling frequency of 40kHz. From figure 4.6, it can be observed that such high sampling frequency allowed noise-capturing, which included electrical noise. It can also be observed that the calibration of the displacement as a function of the voltage can be achieved. It is also important to that the signal strength of the bend sensors are different when deflected forwardly and backwardly. Therefore, it is important to ensure the uniformity of the orientation of the bend sensors as it was contacted with the coin-edge of the vibrissa model. Furthermore, the strength of the signals from the bend sensors is also dependent on the point of application (moment arm) of the perturbations. As such, cautious approach is taken as the bend sensors are assembled and locked at its end on the experimental setup. Similar care is also taken when applying forces for calibrating the bend sensors as the point of force application during the calibration has to simulate the conditions of the bend sensors as they are acted on by the vibrissa model. For mechanical and electrical noise removal, causal filtering is applied through the application of Butterworth filter [16]. The choice of Butterworth filter is reinforced by the need of the smoothest possible response, which will allow for accurate readings on the steady values of the signals. The filter is designed as lowpass filter with passband frequency of 20 Hz and stopband frequency of 150 Hz. The passband ripple of the filter is limited to 1 dB with attenuation of 80 dB

Figure 4.7: Filtered signals of bend sensors obtained during calibration setup with linear-stage. Butterworth filter is used to filter the raw signals with passband of 20 Hz and stopband of 40 Hz. The passband ripple of the filter is limited to 1 dB with stopband-attenuation of 80 dB (refer to figure 4.6 for raw signals).

at stopband. It is important to note that the stopband frequency for filtering in experiments will be performed at 150 Hz. This is to ensure full encapsulation of the fluid-structure interaction between the structures with their wakes and the wakes of the wake-generator (characteristic frequency of the wake-generator can be estimated theoretically, Strouhal number ≈ 0.2 , and dynamically scaled). Such filter is designed in MATLAB and is applied through MATLAB after loading of the corresponding data. The electrical noise is filtered out by filtering any signal at $\approx 50\text{ Hz}$. Based on the Strouhal frequency of the circular cylinder, the highest expected frequency of oscillation is calculated to be $\approx 30\text{ Hz}$. As such, filtering signals at 50 Hz would not interfere with the target measurement results. Any frequencies above 150 Hz will be attenuated as high frequency signals are not expected from the current vibrissa-model sensor trailing a circular cylinder. More details on mechanical noise filtering could be found at "Experimental Setup" Section (Chapter 4.2). Figure 4.7 illustrates the filtered signal from the calibration of the bend sensor (raw signals depicted in figure 4.6).

As expected from filtered signals in figure 4.7, it can be observed that the filtered signals are very distinctive and clear. In order to chart and model the displacement as a function of the model, it is necessary to extract the effective values from figure 4.7. Such values are determined through extraction

Figure 4.8: Calibration plot for bend sensors with cubic polynomial fitting [Cubic equation is presented in the figure]. Green cross - data points from steady filtered signal (refer to figure 4.7; magenta line - cubic polynomial fitting curve.

of the steady voltages for each deflection of the bend sensors. The steady voltages are defined as voltages of the bend sensors right before the return of the linear-stage actuator to its original ('home') position. This meant that the bend sensors are allowed the maximum amount of stabilisation time before extraction of the voltages corresponding to the displacements. Figure 4.8 shows the plot of the steady-voltages as a function of the displacement as well as the cubic polynomial befitting the data. From figure 4.8, it can be observed that the cubic polynomial fitting curve exhibited a finely-matched fit to the data points, with exception of the 1mm displacement of the bend sensor. Such cubic equation is described as follows:

$$V = 0.0017x^3 - 0.022x^2 - 0.051x - 2.6 \quad (4.10)$$

where V is the voltage difference across the electrode of the bend sensors and x is the horizontal displacement of the bend sensors at a given perpendicular distance from their electrodes. Equation

Figure 4.9: Fundamental vibrissa models for evaluating the geometrical effects on the fluid-structure interaction in the towing vibrissa model experiments. A) straight circular cylinder, b) elliptical curved cylinder, c) straight vibrissa, d) curved (true) vibrissa.

4.10 will be used on subsequent interpretation of the experimental signals. This is because all of the experimental signals will be recorded in voltages, as such, the importance of the calibration chart could not be overstated. Figure 4.9 depicts the basic vibrissa models that are utilised for evaluating the effects of the geometry of the vibrissa models on their response to the flow around their structures.

With reference to figure 4.9, four vibrissa models are used in the towing experiments. These vibrissa models are straight circular cylinder, elliptical curved cylinder, straight vibrissa and curved (true) vibrissa. The curved vibrissa is an enlarged seal vibrissa with all its geometrical features. These vibrissa models are created for evaluation of the effects of geometrical features on the flow-induced vibration of each vibrissa model. The straight circular cylinder is utilised as the reference geometry

Figure 4.10: Holding materials in towing vibrissa experimental setup as the model of the muscle at the base of the harbour seal vibrissae. From left to right: soft, medium and hard (pink) holding materials. a) Top side of holding material, b) bottom side of holding material (not visible in figure 4.4e)

as there are extensive literature on studies of flow around circular cylinder. As such, all of the results from other vibrissa model shall be referenced to the circular cylinder. The elliptical curved cylinder is created to assess the effects of the spanwise curvature on the Flow-Induced Vibration (FIV) of the elliptical geometry. The straight vibrissa is developed to assess the effects of the undulations on the FIV response of the straight vibrissa. The curved vibrissa is intended to observe the compound effects of curvature and undulations on the FIV of the true vibrissa. For evaluation of the FIV responses, the amplitude of oscillation near the tip of the vibrissa models are evaluated. The frequency of oscillation of the tip of the vibrissa models will also be investigated. It is important to note that the deflections at the tip of the vibrissa models will be conducted to the coin-edge of the vibrissa model. As such, all the signals (measurements) will be obtained through the deflections of the bend sensors by the coin-edge of the vibrissa model. All the vibrissa models are 3D printed with ABS (Acrylonitrile Butadiene Styrene) as base material. The Modulus of Elasticity of the ABS is 1.63 GPa, which is found to match the dynamically scaled modulus elasticity of the harbour seal vibrissae very well.

As aforementioned, the holding materials in the towing experimental setup represented the muscle at the base of the vibrissae of the harbour seals. Due to significant challenges in obtaining the properties of the muscles at the base of the vibrissae while the seals are tracking their preys, three different holding materials are designed. Figure 4.10 exhibited the three holding materials of soft, medium and hard (from left to right) holding materials. The signals of the vibrissa models fitted with these holding materials will indicate the optimised holding material for signal acquisition.

Each holding materials are casted with a Smooth-on polymer of varying mechanical properties [75]. The soft, medium, and hard holding materials are casted with Smooth-on polymers Ecoflex00-10, Ecoflex00-30, and MoldMax 30, respectively. The holding materials are also purposely chosen to be viscoelastic to mimick the similar nature found in the human muscles [81]. The viscoelasticity of the holding materials could be observed through the variability of the modulus of elasticity of the holding materials as the frequency of force applications is varied. As the frequency of force applications are varied, for an ideal non-viscoelastic material, its modulus of elasticity should remain constant. The through hole on the holding materials is designed as the insertion point of the vibrissa model. Although the manufacturer provided the materials properties of the polymers for casting each holding material, Dynamic Mechanical Analysis (DMA) is performed on each holding material with compression plates. The DMA setup involved placement of the holding materials between the compression plates attached to the DMA machine on each side. DMA will ensure that proper casting had been performed and the effects of air pockets are minimum on the holding materials. The existence of air pockets could significantly alter the mechanical properties of the holding materials. DMA analyses provided the variability of the modulus of elasticity of each holding material. As such, more precise properties are available for evaluation at a given experimental conditions. Figure 4.11 depicts the modulus of elasticity of the holding materials at various frequency of force applications.

As each experiment is repeated to ensure consistency, both experimental runs for each holding material are depicted in figure 4.11. With exception of the soft holding material, the uncertainty level of the Modulus of Elasticity of the holding material is below 2%. However, greater uncertainty is observed for the measurement of the soft holding material with an uncertainty level of $\approx 5\%$ at low frequency of force application. Overall, the uncertainty level of the measurement of the modulus of elasticity of the soft holding material is well below 5%. With reference to the extensive studies on the VIV of a circular cylinder, the resonance frequency of the current vibrissa model could be estimated. Such approximation indicated the occurrence of VIV in the range of $0Hz < f_{VIV} < 40Hz$. Given the prevalence of this frequency region, the mean of the modulus of elasticity of the holding materials in this frequency range are determined. The corresponding modulus of elasticity of the soft, medium and hard holding materials are 30 kPa, 60 kPa, and 80 kPa, respectively. These values will be used as reference for future calculations.

It is important to note that the current study consider the vibrissa model with finite bending stiffness

Figure 4.11: Variations of modulus of elasticity of the holding materials (soft, medium, and hard) at various frequency of force applications for DMA experiment with compression plates.

(Modulus of Elasticity 1.63 GPa). This is not the case for previous studies where very rigid model or very flexible model are often considered. Due to such finite bending stiffness, the coupling behaviour between the fluid and structure exhibited distinct behaviours.

4.2.3 Vibrissa Model Towing Experimental Setup

In order to characterise the response of the vibrissa to its own wakes, vibrissa model towing experiments are performed. Figure 4.12 illustrates the vibrissa towing experimental setup in the Marine laboratory of Center for Environmental Sensing and Modeling (CENSAM) of Singapore-MIT Alliance for Research and Technology (SMART).

With reference to figure 4.12, it can be observed that the vibrissa model-sensor setup is attached and carried by the linear stage through a set of mechanical fixtures. As aforementioned, the linear stage yielded minimum amount of mechanical vibration (noise) as compared to the carriage of the tow tank. In order to avoid secondary vibration due to over-acceleration and insufficient steady-

Figure 4.12: Vibrissa model towing experimental setup: a) side view of experimental setup, b) close-up view on the mounting of the vibrissa model-sensor setup on linear stage.

4.3. TOWING EXPERIMENTS OF A VIBRISSA MODEL-SENSOR TRAILING AN UPSTREAM WAKE-GENERATOR

speed towing due to under-acceleration, the acceleration of the linear stage is set to 350 mm/s^2 . An over-acceleration would result in significant reduction of the accuracy of the acquired signals while an under-acceleration will result in shortening of the steady-speed signals. As depicted in the figure 4.12a, the vibrissa model-sensor setup is attached to the linear stage, which is controlled with software in the computer. This is also the case for the control of the tow tank carriage. From figure 4.12b, it can be observed that the vibrissa model-sensor setup is attached to an extensor that is capable of changing the transverse alignment of the vibrissa model. The use of extensor will be clearer in vibrissa model-sensor trailing a wake-generator experiment.

4.3 Towing Experiments of A Vibrissa Model-Sensor Trailing An Upstream Wake Generator

In this section, the setup of vibrissa model-sensor trailing an upstream wake-generator towing experiments will be described. Similar setup for vibrissa-mode sensor is devised as described in the previous section. An upstream wake-generator is placed in the upstream of the vibrissa model-sensor. The upstream wake-generator in these experiments is a circular cylinder of 10 times the size of the vibrissa model. Such size is chosen due to its proper scaling ratio between the width of fish tails and the real vibrissae of harbour seals. The circular cylinder is chosen due to the extensively studied nature of its wakes. The knowledge of the wakes of the circular cylinder will provide clear indication and ideas of the resultant motion of the vibrissa model due to the wake-structure interaction. Hence, this will reduce the possibilities of errors in analysis of the experimental results (e.g. the amplitude and frequency of oscillation of the vibrissa model due to wake-structure interaction). Such circular cylinder is mounted on the tow tank carriage through the fixtures. As the circular cylinder is solely a wake generator, no signal measurements are performed on or through it. As such, the mechanical vibration from the tow tank carriage will incur almost no effect on the vibrissa model-sensor experimental setup. The initial and final streamwise distances between the vibrissa model-sensor and wake generator are maintained at 5 times the diameter of the circular cylinder (50 times of the vibrissa model). Further evaluation on the wake-structure interaction are performed through transverse offset of $0.5D$ between the position of the vibrissa model and the wake-generator circular cylinder. Such transverse offset is expected to deliver an interesting dynamics on the oscillation of the vibrissa

Figure 4.13: Sketch of the vibrissa model-sensor trailing a wake-generator circular cylinder experimental setup with indicators of streamwise and transverse positioning with respect to the structures. It should be noted that the transverse position of the vibrissa model is adjustable through the use of extensor.

model. This is because of the head-on collision between the coherent structures in the von *Kármán* wakes of the circular cylinder and the vibrissa model. Therefore, it is essential to analyse the response of the vibrissa model due to the existence and nonexistence of head-on collision with the coherent structures of the von *Kármán* street. Such analysis will provide grounds for optimisation of the response of the vibrissa model to sensing of the flow perturbations. Figure 4.13 illustrates the sketch of such experimental setup between the vibrissa model and the wake-generator circular cylinder.

It should be noted that care is taken to ensure that the extensor would fit tightly and bolted to remove any possible sliding motion, which will displace the vibrissa model with respect to the wake-generator. Due to the limitation on the length of the linear stage (1.3m), it is not possible to study the effect of inline positioning on the response of the vibrissa model to the wake generator. Such studies are performed with simulations where the numerical domain could be increased with computational cost as its limitation. Care is also taken to ensure the prevention of end effects of the flow around circular cylinder. As such, the height of the circular cylinder is significantly longer (3x) to the length of the vibrissa model. Equal or different towing speeds are applied to the experimental setup in order to

understand the effects of approaching and distancing vibrissa models relative to the wake-generator circular cylinder. Such experimental settings will provide insight into the response of the harbour seal vibrissae as the harbour seals approach or loses their preys.

4.4 Results And Discussions

4.4.1 Single Vibrissa Model-Sensor Towing

In order to characterize the response of the various vibrissa models to their own wakes, single vibrissa-model towing experiments are conducted. With reference to previous studies, figure 4.14 illustrates the plot of frequency response ratio (to natural frequency of the experimental setup) against the reduced velocity for a uniform circular cylinder. [51].

The frequency response ratio is defined as the ratio of the frequency of oscillation of the structure to the natural frequency of the system, mathematically written as:

$$f^* = \frac{f_{VIV}}{f_n} \quad (4.11)$$

where f_{VIV} is the dominant response frequency of the vortex-induced vibration of the structure (in

Figure 4.14: Comparisons of the experimental data on frequency response ratio for towing cantilevered uniform circular cylinder ($m^* = 1.2$, where m^* is mass ratio, as defined in equation 4.13) with the three different holding materials to the previous study ($m^* = 2.4$) [51]. f^* is the frequency response ratio, defined in equation 4.11 and U^* is the reduced velocity, defined by equation 4.12

this case, the vibrissa model) and f_n is the natural frequency of the structure. The reduced velocity can be defined as follows:

$$U^* = \frac{U}{f_n D} \quad (4.12)$$

With reference to figure 4.14, it can be observed that the response of the circular cylinder (vibrissa model) exhibited similar response to the previous study on a cantilevered circular cylinder [51]. It is important to note that such agreement is true, despite the difference in mass ratio, where mass ratio is defined as the ratio of the mass of the structure subjected to external flow (m_{sys}) to the added mass of the body due to the movements of the displaced fluid in the vicinity of the body (m_a). Mathematically, the mass ratio can be written as:

$$m^* = \frac{m_{sys}}{m_a} \quad (4.13)$$

As expected, the response of the vibrissa model with hard holding materials is the closest to the results of previous study on a cantilevered cylinder. This is due to the closer hardness of the holding materials to a fixed boundary condition at one of the ends of the circular cylinder. Consequently, the response of the circular cylinder with soft holding materials showed the largest deviation from previous studies. It is necessary to invoke that the mass ratio between the vibrissa model (and the real vibrissae) are different to the mass ratio of the circular cylinder of previous study. The mass ratio of the circular cylinder of previous study is 2.4 while the mass ratio of the vibrissa model is set to 1.04. As such, from previous studies, it is expected that the profile of the frequency ratio will differ slightly.

As shown in equations 4.11, 4.12, it is necessary to determine the natural frequency of the system. As the vibrissa model is configured as a cantilever, it is possible to determine the natural frequency of the vibrissa model setup using such model. Figure 4.15 depicts the cantilevered beam setup representing the vibrissa model.

With reference to figure 4.15, the author has the Euler-Lagrange's beam equation as follows:

$$EI \frac{d^4 y}{dx^4} + \mu \frac{d^2 y}{dt^2} = q \quad (4.14)$$

where y is the lateral displacement of the beam (m), q is the force per unit length acting on the beam

Figure 4.15: Schematic diagram of beam for formulation of the Euler-Lagrange beam equation

(N/m), and μ is the mass of the beam per unit length (kg/m). Assuming a steady condition of the beam, the second term on the left hand side (LHS) of equation 4.14 will be negligible due to the absence of the rate of change of the displacement (and velocity) with time. As such, the author has:

$$EI \frac{d^4 y}{dx^4} = q \quad (4.15)$$

Based on the definition, the variable q and $\frac{d^4 y}{dx^4}$ can be approximated as follows:

$$\begin{aligned} q &\sim \frac{F}{l} \\ \frac{d^4 y}{dx^4} &\sim \frac{y}{l^4} \end{aligned} \quad (4.16)$$

where l is the length of the beam (m). Substituting the approximations in equation 4.16 into equation eq:415, the author has:

$$EI \frac{y}{l^4} \sim \frac{F}{l} \quad (4.17)$$

The force (F) acting on the beam on equation 4.17 can be calculated as the product of the stiffness constant of the system (k) and the lateral displacement of the beam (y). Hence, substituting this definition into equation 4.17, the author has:

$$\begin{aligned} EI \frac{y}{l^4} &\sim \frac{ky}{l} \\ \frac{EI}{l^3} &\sim k \\ k &= \frac{n_0 EI}{l^3} \end{aligned} \quad (4.18)$$

where n_0 is the proportionality constant. The stiffness constant of the system is mainly governed by

the stiffness of the holding material. As such, it is necessary to quantify such stiffness constant of the holding material. Hence, DMA experiments are performed on the holding materials with compression clamps. Axial compressions are applied on the holding material through the compression clamps at various frequencies of force application. However, it is noted that the vibrissa models impose bending moment on the holding materials (instead of pure compression and/or tension) as they are deflected. As such, it is necessary to calibrate the proportionality constant in order to match the bending of the holding material to the deflection of the vibrissa model due to the different affected zone. In order to perform such calibration, it is assumed that the holding materials have homogeneous properties. The following is the calibration procedure of the holding materials.

Upon force applications by the vibrissa model, deflections on the holding material will ensue. Such deflection is followed by the change of the volume of the holding material. Hence, it can be observed that force applications resulted in volumetric changes. Mathematically, this can be written as:

$$\begin{aligned} F &= k\delta x \\ F &= k \frac{\delta V}{A} \end{aligned} \tag{4.19}$$

δx is the local deflections of the holding material due to force application, δV is the volumetric changes of the holding material due to force application and A is the area of the force application. For a given force, through comparison between the axial and transverse force applications on the holding materials, it can be observed that the area of force application differs. Hence, the corresponding stiffness constants is found to be different as well (due to the differing amount of resisting/affected volume of materials). Hence, for a small amount of deflection, it can be assumed that the volumetric changes will be the same for different types of force application. Hence, re-arranging equation 4.19, the author has:

$$\frac{F}{\delta V} = \frac{k}{A} \tag{4.20}$$

Applying equation 4.20 for axial and bending forces and deformations, the author has:

$$\frac{F}{\delta V} = \frac{k_a}{A_a} \tag{4.21}$$

$$\frac{F}{\delta V} = \frac{k_b}{A_b} \quad (4.22)$$

Combining equations 4.21 and 4.22, the author has:

$$\begin{aligned} \frac{k_a}{A_a} &= \frac{k_b}{A_b} \\ k_b &= k_a \frac{A_b}{A_a} \end{aligned} \quad (4.23)$$

Due to the nature of the axial and bending deformations, the change in the ratio of A_b/A_a is negligible for small deformations. As such, this ratio could be represented by a constant value of n_A . Substituting this definition into equation 4.23, the author has:

$$k_b = n_A k_a \quad (4.24)$$

Equation 4.24 illustrated a linear relationship between the stiffness constant of the holding materials due to transverse force and axial force. Substituting equation 4.24 into equation 4.18, the author has:

$$n_A k_a = \frac{n_0 EI}{l^3} k_a = \frac{n_0 EI}{n_A l^3} \quad (4.25)$$

For the sake of simplicity and with reference to equation 4.25, k_a shall be written as k and a new proportionality constant, n , is defined as $n = n_0/n_A$, as such:

$$k = \frac{n EI}{l^3} \quad (4.26)$$

The value of n is dependent on the holding material at the base of the vibrissa model. As such, it is experimentally determined with the circular cylinder as the vibrissa model with corresponding holding material near its base.

Khalak & Williamson derived an estimate of the amplitude of VIV of a circular cylinder in open water through simplified model [52]. The equation representing the amplitude of VIV of a circular cylinder is as follows:

$$A^* = \frac{1}{4\pi^3} \frac{C_Y \sin \phi}{(m^* + C_A)\zeta} \left(\frac{U^*}{f^*}\right)^2 f^* \quad (4.27)$$

where

$$f^* = \sqrt{\frac{m^* + C_A}{m^* + C_{EA}}} \quad (4.28)$$

and

$$C_{EA} = \frac{1}{2\pi^3} \frac{C_Y \cos \phi}{A^*} \left(\frac{U^*}{f^*}\right)^2 \quad (4.29)$$

With reference only to equations 4.27, 4.28, 4.29, the variables in these equations are defined as follows:

- A^* - amplitude ratio (A/D)
- C_Y - lift coefficient
- ϕ - phase angle between the motion of the body to the fluid-forcing on the body
- m^* - mass ratio (mass of structure/mass of displaced fluid)
- U^* - reduced velocity (U/f D)
- C_A - potential flow added mass coefficient (1.0 for circular cylinder)
- ζ - damping of the system
- C_{EA} - effective added mass coefficient

The introduction of holding materials to the vibrissa model-sensor setup resulted in higher damping of the experimental setup. From equations 4.27, 4.28, 4.29, it can be observed that as the damping (ζ) increases, the amplitude of VIV will decrease from the inversely proportional relationship. Figure 4.16 illustrated the comparison of the amplitudes of VIV of a circular cylinder model with holding materials to the previous study.

With reference to figure 4.16 and as mentioned earlier, the amplitude of VIV of the circular cylinder with holding materials is found to significantly decrease as compared to that of the circular cylinder

Figure 4.16: Comparison between the present experimental VIV data of the circular cylinder model with holding material at its base ($m^*=1.04$) and the previous study of the towing of a uniform circular cylinder ($m^*=1.2$) [52].

with no holding materials. Regardless of the type of holding materials at the base of the circular cylinder (vibrissa model), reduction in amplitude of VIV is observed. However, it is also important to note that the Reynolds Number of the current study ($Re = 400$ to 1200) is significantly lower than the Reynolds Number of the previous study ($Re = 5000$ to 15000) [52]. From previous studies, such difference in Reynolds Number is found to minimally contribute to the difference in amplitude of VIV [105]. Figure 4.17 depicts the normalized response frequencies of the towed various vibrissa models with the soft, medium and hard holding materials for three reduced velocities (corresponding to towing velocity of 100mm/s , 200mm/s and 300mm/s).

For figure 4.17, it is important to note that markers at normalized frequency 'zero' indicated that no observed dominant response frequency of VIV are observed for the given vibrissa model and holding material. As such, regardless of the holding material at its base, it can be observed that no dominant response frequency of VIV are observed for straight vibrissa. However, curved elliptical cylinder and curved vibrissa exhibited dominant response frequency of VIV when they are equipped with soft holding material and are towed at higher velocities (200mm/s and 300mm/s). However, from figure 4.17, it can be observed that dominant response frequency of VIV is detected for circular cylinder with various combination of holding materials and towing speeds, except for medium and

Figure 4.17: Normalised response frequency of four of the vibrissa models to VIV at three reduced velocities for each vibrissa model (corresponding to towing speed of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material and blue - circular cylinder, red - curved elliptical cylinder, green - straight vibrissa, magenta - curved vibrissa.

soft holding material at towing speed of 300 mm/s. From equation of motion of a body mounted on a circular cylinder, the natural frequency of the system, f_n , can be defined as:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m_T}} \quad (4.30)$$

where $m_T = m_{sys} + m_a$ with m_{sys} as the mass of the body(system) and m_a as the added mass of the structure. From equation 4.30, it can be observed that the natural frequency of the system is strongly dependent on the stiffness of the holding material. As such, with reference to figure 4.11, it can be observed that the moduli of elasticity among the holding materials vary significantly. Such variation corresponded to variation in stiffness constants of the holding materials. From equation 4.30, it is found that variation of stiffness constant will result in the variation of the natural frequency of the holding material. As such, the soft holding material is expected to resonate at lower frequencies among the three holding materials. Hence, with reference to equation 4.11, for a given frequency of VIV of the vibrissa model, the normalised response frequency of VIV will be higher for soft holding materials as compared to other holding materials. Such response is observed in figure 4.17,

Figure 4.18: Amplitude Ratio of VIV of the vibrissa models for the three reduced velocities of each vibrissa models (corresponding to towing velocities of 100 mm/s, 200 mm/s, 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material and blue - circular cylinder, red - curved elliptical cylinder, green - straight vibrissa, magenta - curved vibrissa.

where all the large dominant response frequencies ($f^* > 2$) are found on vibrissa models with soft holding material. From figure 4.17, it can also be observed that vibrissa models with curvature and/or elliptical cross section experienced higher frequency of vibrations if vibrations occurred. From figure 4.17, it can also be observed that the curved vibrissa has a slightly higher normalised frequency of VIV as compared to the curved elliptical cylinder. Such phenomena is instigated by the combination of geometrical undulations and the curvature of the curved vibrissa.

Figure 4.18 depicts the amplitude ratio of the VIV of the vibrissa models as a function of the reduced velocity (corresponding to towing speed of 100 mm/s, 200 mm/s and 300 mm/s).

where amplitude ratio (A^*) is defined as A_{VIV}/D with A_{VIV} as the amplitude of VIV of the vibrissa model and D as the characteristic diameter of the vibrissa model (minor diameter of vibrissa model with elliptical cross sectional area and RMS-diameter for undulatory vibrissa models). With reference to chapter 6, it will be conclusive that the largest amplitude of VIV is observed on circular cylinder with undulatory vibrissa exhibiting the smallest amplitude of VIV. With reference to figure 4.18, it can be observed that the largest amplitude of VIV is observed for circular cylinder towed

at 100 mm/s with soft holding material. An interesting exception is observed for circular cylinder towed at 300 mm/s. Regardless of the holding materials, it is found that the amplitude of VIV of the circular cylinder is lowest at 300 mm/s as compared to the circular cylinder towed at 100 mm/s and 200 mm/s. Such occurrence is most likely due to the transition of the flow from one regime to another (vortex shedding mode A and B for towing speeds of 100 mm/s and 200 mm/s, respectively, to shear-layer instability regime for towing speed of 300 mm/s) [104, 105]. For most experimental settings, the medium holding material is found to result in the largest reduction of amplitudes of VIV. As the holding material is designed to model the muscle at the base of the vibrissae, it shall provide the optimum performance for noise (VIV) reduction in order to optimise sensing (Signal-to-Noise-Ratio). As such, it is empirically deduced that the medium holding material exhibited the near-optimum condition for VIV reduction and therefore, underwater sensing of the harbour seal vibrissae. The amplitudes of VIV on straight and curved vibrissa models are significantly smaller than the amplitudes of VIV of the circular cylinder and curved elliptical cylinders. As such, it can be deduced that the undulations on the vibrissa models played a significant role in reducing the amplitudes of VIV. This should be further validated and explored in chapter 6. From figure 4.18, it can also be observed that the towing speed has various effects on the variety of experimental setup. Similar pattern are observed for straight and curved vibrissa models with hard and soft holding materials.

From figures 4.17, 4.18, it can be concluded that the normalised frequency of VIV is inversely proportional to the modulus of elasticity of the holding material (boundary condition). Similar to previous studies, the frequency of VIV is found to be directly proportional to the towing speed of the vibrissa models. It is also observed that no observable frequency of VIV is found on vibrissa models with undulations, which suggested that such geometrical features significantly reduce VIV. The elliptical cross-section is found to reduce VIV to a smaller extent as compared to the VIV reduction due to the undulations. The curvature of the geometry of the vibrissa models is found to minimally affect the VIV of the vibrissa models. The largest amplitudes of VIV are observed on towed circular cylinder and the smallest amplitudes of VIV are observed for straight and curved vibrissa models. It is also empirically concluded that the medium holding material represented the best model of the muscle condition at the base of the real vibrissae due to its significant reduction of the amplitudes of VIV, regardless of the type of vibrissa models.

4.4.2 Single Vibrissa Model-Sensor Towing with Upstream Wake Generators

This subsection describes the experiments on single vibrissa model-sensor towed in the wake of a circular cylinder. The diameter of the circular cylinder is 13 mm (10 times of the characteristic diameter of the trailing vibrissa models). The upstream circular cylinder acted as a wake-generator whose wakes are to be tracked by the trailing vibrissa models. Similar to earlier subsection, the vibrissa models involved in this study are circular cylinder, elliptical curved cylinder, straight vibrissa and curved vibrissa models. The objective of such is experiment is to identify the response of each vibrissa model to the wake of the upstream circular cylinder. As indicated earlier, the circular cylinder has a regularly spaced unsteady wake (von *Kármán* street) [89, 104]. As such, it is important for the vibrissa models to capture such pattern through their dominant response frequencies and amplitude of oscillations (Wake-Induced Vibrations, WIV). The amplitudes and frequency of oscillations obtained in these experiments will be compared to the corresponding results from the above subsection. Such comparison will provide the insights on the sensitivity and robustness of each vibrissa model as a sensing device. The spacing between the vibrissa model and the wake generator is 10 times of the characteristic diameter of the circular cylinder. Figure 4.19 illustrate the ratio of amplitudes of VIV for the vibrissa models immersed in the wake of a circular cylinder towed at similar speeds against reduced velocities (corresponding to towing velocity of 100 mm/s, 200 mm/s and 300 mm/s).

The ratio of amplitude of VIV for the vibrissa models is defined as the ratio of the amplitude of the vibrissa model immersed in the wakes of the upstream circular cylinder to the amplitude of the towed corresponding vibrissa model without wake generator. This amplitude ratio indicates the sensitivity of each vibrissa model to the wakes of the upstream circular cylinder with respect to its own VIV. As such, the VIV of the vibrissa models is considered as 'noise' relative to the wake-induced vibrations, which is considered as the 'signal'. Hence, by definition, the ratio of amplitudes of VIV corresponds to the Signal-to-Noise Ratio (SNR) performance of each vibrissa model for a given experimental setup. With reference to figure 4.19, it can be observed that the highest SNR is observed for curved vibrissa model with medium holding material. It is important to note that all the largest SNRs are observed for vibrissa models with medium holding materials. The lowest SNR is observed for circular cylinder models. For circular cylinder model mounted on hard holding material, the SNR is 1.04, which indicated an increase of 4% in the amplitude of oscillation of the circular cylinder model as it is subjected to the wakes of the upstream circular cylinder. Such minute

Figure 4.19: Ratio of Amplitudes of VIV of the vibrissa models trailing a ten times larger circular cylinder traveling at equivalent nominal speeds for the three reduced velocities of each vibrissa models (corresponding to towing velocities of 100 mm/s, 200 mm/s, 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material and blue - circular cylinder, red - curved elliptical cylinder, green - straight vibrissa, magenta - curved vibrissa.

increment is expected as the VIV of the circular cylinder model is considerably large (refer to figure 4.18). As such, the resulting SNR is expected to be small due to the weak coupling between the wakes of the upstream circular cylinder and the response of the trailing circular cylinder model. Vibrissa models with elliptical cross section exhibited very high SNR when equipped with medium holding materials. However, small SNR are observed on the same vibrissa models with hard holding materials. This indicated that the hard holding material provided excessive stiffness and damping, which resulted in the significant dissipation of the energy transmitted by the perturbations. Minimal SNR variation is also observed as the vibrissa models with hard holding materials are towed higher velocities. This indicated a relative insensitiveness of these vibrissa model settings to speed variation which will prove to be a major setback for underwater sensors. Vibrissa models with soft holding materials exhibited very low SNRs as compared to medium holding material due to the large 'noise' of such settings. From figure 4.19, it can be deduced that the most suitable settings (highest SNR) are vibrissa models with undulations and configured with medium hardness holding material. Such conclusion is found to agree with the proposed hypothesis on passive reduction of VIV through

geometrical means (early in this chapter).

Table 4.1 shows the dominant response frequencies of the vibrissa models trailing an upstream circular cylinder with various holding materials and towing speeds. From table 4.1, it can be observed that the dominant response frequency peaks of the vibrissa models on medium holding materials are very consistent at 0.416 (first peak) and 1.247 (second peak). The use of medium holding material is also found to result in single dominant response frequency for most vibrissa models (except circular cylinder). As such, it is evident that medium holding material is further validated as the more suitable boundary condition at the base of the vibrissa model. This is because of the consistent and concise dominant response frequency acquired through its use. For vibrissa models mounted on hard and medium holding materials, the second dominant response frequency is always within the vicinity of lock-in region. Lock-in region is defined as the region of response of the structure where its dominant response frequency coincided with the natural frequency of the system [105]. However, it is also observed that the first dominant frequency peak is far below of the lock-in region. For vibrissa models with soft holding material, the first dominant response frequency is observed to be slightly higher than the response frequency of the vibrissa models with medium and hard holding material. Similar proximity of the second dominant response frequency to lock-in region is not observed. With reference to equation 4.28, it can be seen that theoretical model indicated the independence of the response frequency to the holding material. However, it is necessary to point out that the natural frequency of the system (vibrissa model-holding material) is strongly influenced by the holding material. As such, it is evident that the normalised dominant response frequency would be a function of the stiffness of the holding material. Table 4.2 shows the nominal dominant response frequencies of the towed vibrissa models with the soft, medium and hard holding materials at the three towing speeds (100 mm/s, 200 mm/s and 300 mm/s).

From table 4.2, it can be observed that the dominant response frequencies of the vibrissa models are very consistent for a given towing speed. Despite significant changes of the experimental settings, the dominant response frequencies are found to be always between 2.441 and 7.324 Hz. Such consistency merits further investigation. The vibrissa models with soft holding material is found to exhibit greater variation of their dominant response frequencies. It is also observed that vibrissa models equipped with soft or hard holding materials often exhibited dual dominant response frequencies as compared to vibrissa models equipped with medium holding material. Vibrissa model with medium holding

Trail CC	H			M			S		
	f_1^*	f_2^*	U^*	f_1^*	f_2^*	U^*	f_1^*	f_2^*	U^*
CC-100	0.368	1.105	5.030	0.416	1.247	5.677	0.000	0.000	7.695
EC-100	0.391	1.162	8.872	0.416	0.000	10.018	0.599	1.761	13.580
SW-100	0.368	1.116	8.872	0.416	0.000	10.018	0.617	1.849	13.580
CW-100	0.368	1.105	8.872	0.416	0.000	10.018	0.564	1.726	13.580
CC-200	0.480	1.105	10.060	0.000	1.247	11.354	0.000	0.000	15.390
EC-200	0.000	1.128	17.745	0.000	1.247	20.036	0.000	1.775	27.159
SW-200	0.391	1.128	17.745	0.416	1.247	20.036	0.599	1.832	27.159
CW-200	0.000	1.231	17.745	0.416	1.247	20.036	0.000	1.691	27.159
CC-300	0.480	1.105	15.080	0.000	1.345	17.031	0.612	1.985	23.085
EC-300	0.368	1.116	26.617	0.416	0.000	30.054	0.533	1.691	40.739
SW-300	0.357	1.128	26.617	0.416	0.000	30.054	0.564	1.761	40.739
CW-300	0.391	1.197	26.617	0.416	0.000	30.054	0.564	1.691	40.739

Table 4.1: Normalised dominant response frequencies of the vibrissa models mounted on soft (S), medium (M) and hard (H) holding materials towed at three speeds (100 mm/s, 200 mm/s, 300 mm/s). Legend for first column (first two letters): CC - circular cylinder model, EC - elliptical curved cylinder model, SW - straight vibrissa model, CW - curved vibrissa model; (last three letters): 100 indicates towing speed of 100 mm/s, 200 indicates towing speed of 200 mm/s and 300 indicates towing speed of 300 mm/s. f_1^* indicates the first dominant response frequency, f_2^* indicates the second dominant response frequency, U^* indicates the corresponding reduced velocity.

Trail CC	H			M			S		
	f_1	f_2	$U(m/s)$	f_1	f_2	$U(m/s)$	f_1	f_2	$U(m/s)$
CC-100	2.441	7.324	0.1	2.441	7.324	0.1	0.000	0.000	0.1
EC-100	2.594	7.706	0.1	2.441	0.000	0.1	2.594	7.629	0.1
SW-100	2.441	7.401	0.1	2.441	0.000	0.1	2.671	8.011	0.1
CW-100	2.441	7.324	0.1	2.441	0.000	0.1	2.441	7.477	0.1
CC-200	3.185	7.324	0.2	0.000	7.324	11.354	0.000	0.000	0.2
EC-200	0.000	7.477	0.2	0.000	7.324	20.036	0.000	7.691	0.2
SW-200	2.594	7.477	0.2	2.441	7.324	20.036	2.594	7.935	0.2
CW-200	0.000	8.162	0.2	2.441	7.324	20.036	0.000	7.324	0.2
CC-300	3.185	7.324	0.3	0.000	7.900	0.3	2.65	8.6	0.3
EC-300	2.441	7.400	0.3	2.441	0.000	0.3	2.309	7.324	0.3
SW-300	2.365	7.477	0.3	2.441	0.000	0.3	2.441	7.629	0.3
CW-300	2.594	7.935	0.3	2.441	0.000	0.3	2.441	7.324	0.3

Table 4.2: Nominal dominant response frequencies of the vibrissa models mounted on soft (S), medium (M) and hard (H) holding materials towed at three speeds (100 mm/s, 200 mm/s, 300 mm/s). Legend for first column (first two letters): CC - circular cylinder model, EC - elliptical curved cylinder model, SW - straight vibrissa model, CW - curved vibrissa model; (last three letters): 100 indicates towing speed of 100 mm/s, 200 indicates towing speed of 200 mm/s and 300 indicates towing speed of 300 mm/s. f_1^* indicates the first dominant response frequency, f_2^* indicates the second dominant response frequency, U^* indicates the corresponding reduced velocity. f_1 and f_2 are in Hertz.

Figure 4.20: Normalised dominant response frequency of the towed circular cylinder trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) -medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.

material responded almost always with a single dominant response frequency. It is important to note that the uncertainty level of measurements in tables ?? are well below 5%. Figure 4.20 depicts the plot of normalised frequency peaks against reduced velocities for circular cylindrical model towed with and without wake-generator circular cylinder.

From figure 4.20, it is found that the first dominant response frequencies are well below the lock-in region for the tested experimental settings. For circular cylinder model equipped with hard holding materials at the three reduced velocities, the first dominant response frequencies are found to be locked-in. The second response frequencies are found to deviate significantly from the VIV frequency of the circular cylinder model. These results indicated that strong signals (vibration) are obtained from the VIV of the circular cylinder model, instead of the WIV from the wake-generator circular cylinder. For towing speeds of 100 mm/s and 200 mm/s, the circular cylinder mode mounted on soft holding material is found to be unable in capturing any dominant response frequencies related to the characteristic frequencies of the wake-generator circular cylinder. Figure 4.21 shows the plot of normalised frequency peaks against reduced velocities for the curved elliptical cylinder towed with

Figure 4.21: Normalised dominant response frequency of the towed curved elliptical cylinder trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.

and without wake-generator circular cylinder.

With reference to figure 4.21, single dominant response frequency is observed for all elliptical curved cylinder mounted on medium holding material trailing a wake-generator circular cylinder. The use of hard and soft holding material is found to almost always result in dual dominant response frequencies. However, these dominant response frequencies are found to be very consistent for the three towing speeds. The implementation of soft holding material is observed to increase the normalised frequency captured by the vibrissa model due to the lower natural frequency associated with the soft holding material. Figure 4.22 shows the plot of normalised frequency peaks against reduced velocities for the straight vibrissa model towed with and without wake-generator circular cylinder.

From figure 4.22, similar dual dominant response frequencies to the elliptical curved cylinder is observed for straight vibrissa model trailing a wake-generator circular cylinder. Implementation of soft holding material is found to result in an interesting trend with decreasing first and second dominant response frequencies as the towing speeds are increased. It is also observed that the dominant response frequency reduces as the modulus of elasticity of the holding material on the straight vib-

Figure 4.22: Normalised dominant response frequency of the towed straight vibrissa model trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) -medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.

rissa model decreases. Figure 4.23 shows the plot of normalised frequency peaks against reduced velocities for the curved vibrissa model towed with and without wake-generator circular cylinder.

From figure 4.23, it can be observed that the first dominant response frequency of the curved vibrissa model is very consistent for all the holding materials. Such consistency is only partially observed for straight vibrissa model and curved elliptical cylinder. Singular dominant response frequencies are prevalent with the use of the medium holding material, with exception for towing speed of 200 mm/s. In contrast, for soft and hard holding materials, singular dominant response frequency is observed only for towing speed of 200 mm/s. For curved vibrissa model with hard or medium holding material, the second dominant response frequencies are found to be within the vicinity of the lock-in regime. However, significant deviation from lock-in are observed on the second dominant response frequency of the curved vibrissa model equipped with soft holding material.

In essence, the above results have strongly show that the undulatory geometry of the vibrissa model are capable of performing underwater detecting of the wake-generator through trailing in its wakes. The performance of such detection is found to robust and consistent as the undulatory vibrissa

Figure 4.23: Normalised dominant response frequency of the towed curved vibrissa model trailing a ten times larger wake-generator circular cylinder against reduced velocity (corresponding to towing speeds of 100 mm/s, 200 mm/s and 300 mm/s). Legend: (x) - hard holding material, (o) - medium holding material, (+) - soft holding material, blue - dominant response frequency without wake generator, green - first dominant response frequency with wake generator, dark green - second dominant response frequency with wake generator.

models are mounted on medium holding material. It is also found that the use of holding material did not necessarily result in larger amplitudes of oscillation due to the consequence on the flow regime around the vibrissa model. Inconsistent dominant response frequencies are also observed when the vibrissa models are equipped with soft holding material. Further advantage of the use of medium holding material is found through the singularity of the dominant response frequency. The SNRs of the undulatory vibrissa models mounted on medium holding material are observed to be significantly higher than the corresponding SNRs of circular cylinder model and elliptical curved cylinder. For the given Re , such result conclusively show that the undulatory vibrissa models are optimum for underwater detection and tracking with mounting on medium holding material.

Chapter 5

Methodology of Numerical Simulation

This chapter elaborates the numerical simulation methods and methodology of analysis for the simulation results. This chapter is divided into few sections as follows:

- * The first section describes the numerical methodology for discretisation of the Navier-Stokes equation and the implementation of immersed boundary method as well as calculation of forces on the body,
- * The second section explains the simulation settings and domain for all the simulation performed in presented in subsequent chapters. This section is further divided into three subsections mainly:
 - [+] The domain and environmental settings for single body simulations,
 - [+] The domain and environmental settings for two bodies aligned in streamwise direction simulations with zero relative velocity,
 - [+] The domain and environmental settings for two bodies aligned in streamwise direction simulations with non-zero relative velocity,
- * The third section illustrates the grid design for flow simulations and code validation for BDIM,
- * The four section focuses on the methodology of analysis of the simulation results such as flow visualisation, force calculations and analysis.

The simulation codes are written by a PhD graduate of Massachusetts Institute of Technology), which forms the body of his thesis [94]. The code has also been published on multiple occasions [few

selected publications are cited due to the depth of description of the code] [97, 93, 96, 95]. All simulations are conducted on a workstation (a 12-cores Macintosh Workstation at CENSAM [Center for Environmental Sensing and Modeling], SMART [Singapore-MIT Alliance for Research and Technology]), Oscar (HPC facility of Brown University) and NTU HPC facility (special thanks to Melvin Soh). Further details of each chapter will be presented in the above chronological manner of numerical simulation.

5.1 Cartesian-grid based solver for problems in fluid-structure interaction

The studies in this paper are conducted based on numerical simulations with the ‘Boundary Data Immersion Method (BDIM)’ simulation code [94, 97, 93, 96]. The code was developed with immersed boundary method, which allows for structural motion with respect to the fluid without the necessity of grid re-meshing. Instead, a meta-equation and its subset equations, which define the domains of solid, fluid and transitional region is developed [93]. These equations are shown as follows:

$$M_\epsilon(\phi_\epsilon) = B_\epsilon(\phi_\epsilon) + F_\epsilon(\phi_\epsilon) + S_\epsilon(\phi_\epsilon) \quad (5.1)$$

where ϕ_ϵ is a function of $(\mathbf{x}, t)$ and is the solution to the met-equation $M_\epsilon(\phi_\epsilon)$, $\mathbf{x}$ is the position vector in the spatial domain, t is the simulation time, $M_\epsilon(\phi_\epsilon)$ is the analytic meta-equation, $B_\epsilon(\phi_\epsilon)$ is the governing field equations in the solid domain, $F_\epsilon(\phi_\epsilon)$ is the governing field equation in the fluid domain and $S_\epsilon(\phi_\epsilon)$ is the interfacial conditions between solid and fluid domains. Figure 2.1 illustrates the sold, fluid, and interfacial domain definitions as expressed in equation 5.1.

From equation 5.1, it can be observed that the meta-equation has the following properties:

$$\text{for } \mathbf{x} \in \Omega_b^{-\epsilon} : M_\epsilon(\phi_\epsilon) = B(\phi) + O(\epsilon) \quad (5.2)$$

$$\text{for } \mathbf{x} \in \Omega_f^{-\epsilon} : M_\epsilon(\phi_\epsilon) = F(\phi) + O(\epsilon) \quad (5.3)$$

$$\text{for } \mathbf{x} \in \Omega_s^{+\epsilon} : \epsilon M_\epsilon(\phi_\epsilon) = S(\phi) + O(\epsilon) \quad (5.4)$$

Figure 5.1: Two-domain problem definition in Boundary Data Immersion Method (BDIM), Ω_b : solid domain, Ω_f : fluid domain, Ω_s : solid to fluid interfacial domain, 2ϵ : the width of the solid-fluid interfacial region.

where Ω_b is the solid domain, Ω_f is the fluid domain, Ω_s is the solid to fluid interfacial domain, 2ϵ is the width of the solid-fluid interfacial region. $\Omega_s^{+\epsilon}$ is defined as the smooth boundary region in the simulation domain. The $-\epsilon$ superscript indicates the exclusion of the smooth boundary interface, e.g. $\Omega_b^{-\epsilon} = \Omega_b - \Omega_s^{+\epsilon}$, similarly for $\Omega_f^{-\epsilon}$.

By definition, the equations representing the body $[B(\phi)]$, fluid $[F(\phi)]$, and interfacial region $[S(\phi)]$ can be defined as follows:

$$B_\epsilon(\phi, \mathbf{x}, t) = \int_{\Omega_b} B(\phi, x_b, t) K(\mathbf{x}, x_b) dx_b \quad (5.5)$$

$$F_\epsilon(\phi, \mathbf{x}, t) = \int_{\Omega_f} F(\phi, x_f, t) K(\mathbf{x}, x_f) dx_f \quad (5.6)$$

$$S_\epsilon(\phi, \mathbf{x}, t) = \int_{\Omega_s} S(\phi, x_s, t) K(\mathbf{x}, x_s) dx_s \quad (5.7)$$

where K_ϵ is the nascent delta integration kernel, which has unit zeroth moment and compact support for ϵ , defined as:

$$\text{for } |x - y| > \epsilon : \quad K_\epsilon(\mathbf{x}, \mathbf{y}) = 0, \quad (5.8)$$

$$\int_{\Omega} K_\epsilon(\mathbf{x}, \mathbf{y}) dy = 1 \quad (5.9)$$

Through simplification and approximation of the integral kernel in meta-equations 5.5 5.6, 5.7, the

author has:

$$B_\epsilon(\phi, \mathbf{x}, t) \approx B(\phi, x_b, t)\delta_\epsilon^B(\mathbf{x}, \mathbf{t}) \quad (5.10)$$

$$F_\epsilon(\phi, \mathbf{x}, t) \approx F(\phi, x_f, t)\delta_\epsilon^F(\mathbf{x}, \mathbf{t}) \quad (5.11)$$

$$S_\epsilon(\phi, \mathbf{x}, t) \approx S(\phi, x_s, t)\delta_\epsilon^S(\mathbf{x}, \mathbf{t}) \quad (5.12)$$

By substituting the approximations in equations 5.10, 5.11, 5.12 into equation eq:501, the author has:

$$M_\epsilon(\phi_\epsilon) = \delta_\epsilon^B B_\epsilon(\phi_\epsilon) + \delta_\epsilon^F F_\epsilon(\phi_\epsilon) + \delta_\epsilon^S S_\epsilon(\phi_\epsilon) \quad (5.13)$$

where

$$\delta_\epsilon^B(d) = \begin{cases} 0.5[1 - \sin(\frac{\pi d}{2\epsilon})] & \text{for } |d| < \epsilon \\ 1 & \text{for } d < -\epsilon \\ 0 & \text{else} \end{cases} \quad (5.14)$$

$$\delta_\epsilon^F(d) = [1 - \delta_\epsilon^B(d)] \quad (5.15)$$

$$\delta_\epsilon^S(d) = \begin{cases} \frac{1}{2\epsilon}[1 + \cos(\frac{\pi d}{\epsilon})] & \text{for } |d| < \epsilon \\ 0 & \text{else} \end{cases} \quad (5.16)$$

where d is the distance from the interfacial boundary to the grid points. Figure 5.2 illustrates the behaviour of the approximation of the kernel integral δ_ϵ^B , δ_ϵ^F , and δ_ϵ^S .

In order to calculate the forces acting on the structure from the fluid, a one-sided formulation of Derivative Informed Kernel (DIK) integration for pressure force, F_p , is derived with the final equation as follows [94]:

$$F_p = \int_{\Omega} [p - d\hat{\mathbf{n}}(\mathbf{r} - \frac{\mathbf{U} - \mathbf{u}_0}{\Delta t})] \hat{\mathbf{n}} \delta_\epsilon^+ d\mathbf{x} \quad (5.17)$$

Figure 5.2: Behaviour of the kernel integrals, δ_ϵ for a 1D case. For points $x \leq -\epsilon$ within Ω_b , the author has $\delta_\epsilon^B = 1$ and $\delta_\epsilon^F, \delta_\epsilon^S = 0$. Similarly, for $x > \epsilon$, the author has $\delta_\epsilon^F = 1$ and $\delta_\epsilon^B, \delta_\epsilon^S = 0$. Within the smoothing region, points less than ϵ from σ_s , $\delta_\epsilon^F, \delta_\epsilon^B, \delta_\epsilon^S = O(1)$.

where $\mathbf{U}$ is the prescribed displacement rate and u_0 is the velocity field at $t-\Delta t$. Equation 5.17 calculates the pressure force on the structure by integrating over the Cartesian-grid through the distance function, d , and the pressure boundary condition. This integration will inherently determine the pressure, p , on the boundary of the solid-fluid interface (Ω_s), specifically of the fluid near the surface of the solid (δ_ϵ^+). $\hat{\mathbf{n}}$ is defined as the normal vector of the solid surface and $\mathbf{r}$ is the aggregate of all non-pressure terms in the governing equation of the flow field.

For turbulence model, BDIM utilizes the Implicit Large Eddy Simulation (ILES) model [10, 12]. This implies that BDIM resolves the large and medium-scale eddies and it relies on its convective model to represent the small-scale eddies and their interactions. The numerical dissipation in the convective models is found to be an effective implicit subgrid model to represent the interaction of the small-scale eddies [12]. For its convective model, BDIM utilizes Quadratic Upstream Interpolation for Convective Kinematics (QUICK) scheme [63]. Details of the ILES and QUICK scheme can be found in Appendix B. As such, BDIM has a third-order accuracy for spatial integration and a second-order accuracy for temporal integration. BDIM has been rigorously validated on numerous occasions for internal and external flow over a structure, involving fluid-structure interaction [42, 94, 97, 93, 96].

5.2 Simulation domain and setup

As there are numerous simulations performed in our studies, various simulation setups are created in order to optimise the performance and computational cost of these simulations. Consequently, this section is further divided into three subsections as follows:

- * Single body simulations
- * Two bodies aligned in streamwise direction simulations with zero relative velocity
- * Two bodies aligned in streamwise direction simulations with non-zero relative velocity

Each subsection is dedicated to the description and justifications of the simulation setting and variables. For single body simulations, an additional subsubsection is introduced to explain the implementation of freely vibrating body simulations. Such method of implementation is applicable to other simulation settings involving a one-dimension or two-dimension freely vibrating body.

5.2.1 Single body simulations

Chapters 6,7 involve studies on many single body simulations under various boundary conditions and settings. This subsection describes the various simulation settings for the single body simulations. The simulation domain is designed with criterion of computational efficiency and accuracy. As such, extra care is taken to ensure the completion of these objectives. Figure 5.3 shows the configuration of the simulation domain for single body simulations.

It is important to note that the simulation domain in figure 5.3 is designated for a stationary single body simulation. For simulations on freely vibrating in transverse and/or streamwise directions body, wider simulation domain is enforced at $5.3D$ on each side of the simulation domain ($4.5D$ on stationary simulation, refer to figure 5.3) where D is the characteristic diameter of the structure. As the seal vibrissae and seal vibrissae-like geometries are elliptical in cross section, the characteristic diameter of the structure is defined as the diameter which is oriented perpendicularly with respect to the freestream flow. It is necessary to note that the seal vibrissa is still classified as bluff body due to its significantly higher pressure drag coefficient as compared to friction drag coefficient. This is despite its more streamlined geometry as compared to circular cylinder. With reference to figure 5.3, it can be observed that the simulation domain is symmetrical about the crossflow direction with a

Figure 5.3: Simulation domain for stationary single-body simulations. The freestream flow is parallel to the major axis of the structure. The crossflow width on each side of the domain is $4.5D$, where D is the characteristic diameter of the structure (defined as the diameter of the structure that is perpendicular to the freestream flow). The lengths of the domain upstream and downstream of the body are $4.0D$ and $10.0D$, respectively. The height of the simulation domain is $12.2D$.

length of $4.5D$ on each side. Symmetry of the simulation domain in crossflow direction is essential in maintaining a no-pressure difference condition on both sides in the crossflow direction of the body. The length of the upstream and downstream of the domain with respect to the body is, at least, $4D$ and $10D$, respectively. Slightly larger upstream and downstream lengths are commonly utilised for simulations on elliptical-based structures $4.5D$ and $10.5D$, respectively. Such increase is introduced to counteract the longer body length of the elliptical structure in streamwise direction. On the spanwise length of the simulation domain, care is taken to always simulate 4 wavelengths of undulations for vibrissa and vibrissa-like structures. This led to the spanwise length of approximately $25D$ of the simulation domain. In order to maintain uniformity, similar spanwise length are applied to simulations of straight (non-undulatory) structures. Such spanwise length is also essential in ensuring the consistency of the wake structure of the circular and elliptical cylinders, regardless of changes in their spanwise length [104]. The size of the simulation domain has been rigorously tested and refined with a circular cylinder as reference simulation body. Simulations of flow around circular cylinder are performed with various simulation domain sizes. The drag and lift coefficient of the simulations

are compared to the previous experimental results to ensure equivalence of the simulation to flow over the structures in open water [73, 104]. Inlet boundary condition is introduced in the entry plane of the freestream flow and outlet condition is applied to the exit plane of the freestream flow. The perpendicular planes to crossflow are designed to preserve the mass flow rate into or out of the planes [94]. Such boundary condition is essential in the prevention of reflecting flow on these boundaries of the simulation domain. For planes in the spanwise boundaries, periodic boundary condition is applied. Such boundary indicates that the simulation is performed at an infinitely long structure, which avoided the effects of solid boundaries on the wakes and response of the structure.

In order to preserve the dynamic similarity between the simulations and the real vibrissae on the harbour seal, the Reynolds Number and other non-dimensional quantities are designed to be equal to their corresponding values on real vibrissae. The Reynolds Number of the simulations is fixed at 500, which is equivalent to the Reynolds Number on the real vibrissae. However, the control variable for such consistent Reynolds Number is the kinematic viscosity of the fluid. As such, it is possible to maintain the freestream flow velocity and the characteristic diameter of the structure for easier scaling of other non-dimensional quantities. It is important to note that the simulations are always conducted in dimensionless manner. The following dimensionless variables are the control variable of the simulation:

* Freestream flow : U in $\frac{D}{t^*}$

* Dimensionless (Simulation) Time : $t^* = \frac{tU}{D}$ - simulation end time is defined in dimensionless time

* Reynolds number : $Re = \frac{UD}{\nu}$

* Mass ratio : $m^* = \frac{m_{body}}{m_{fluid}}$

* Reduced velocity $U^* = \frac{U}{f_{ny}D}$

* Reduced frequency $f_n^* = \frac{f_{ny}D}{U}$

where ν is the kinematic viscosity of the fluid, m_{body} is the mass of the simulated body in the simulation domain, m_{fluid} mass of the displaced fluid by the body, f_{ny} is the natural frequency of the mass-spring system in transverse (crossflow) direction. It should be noted that mass ratio, reduced velocity, and reduced frequency are control variables for freely vibrating body simulations.

For two-dimensional freely vibrating simulations, it is necessary to define the mass ratio of the system in streamwise and crossflow directions (if they have directional dependency) as well as the natural frequency of the system in streamwise and crossflow directions. The dimensionless yield variables of the simulations are as follows:

* Strouhal number: $St = \frac{f_{vs}D}{U}$

* Lift coefficient $C_L = \frac{Lift}{\frac{1}{2}\rho U^2 D}$

* Drag coefficient $C_D = \frac{Drag}{\frac{1}{2}\rho U^2 D}$

* Frequency ratio $f_r = \frac{f_r}{f_{ny}}$

* True reduced velocity $U_r^* = \frac{U}{f_r D}$

* Amplitude ratio $A^* = \frac{A}{D}$

where f_{vs} is the vortex shedding frequency of the body, Lift is the net transverse force acting on the body, Drag is the net streamwise force acting on the body, f_r is the response frequency of the structure, A is the amplitude of response of the structure. Strouhal Number is only applicable to simulation of stationary structures.

For freely vibrating simulations, it is necessary to understand the response of the structure at various natural frequency of the system. As such, the natural frequency of the structure in the simulation is varied with reduced velocity and reduced frequency as the non-dimensional variables. The mass ratio of the structure in the simulations is maintained at 1.04, equivalent to the mass ratio of the harbour seal vibrissae. The Reynolds Number of all the simulations is 500 with a freestream flow of 1D per 1 solution (simulation) time. The solution time of the simulation is dissimilar to the real (wall) time. The initial solution time is 0.000 with an end time of at least, 300. A longer simulation time is performed for validation and analysis on the steady amplitude time for a simulation structure and settings (refer to subsection on '5.3 Grid Design and Validations'). It is necessary to indicate that the simulation end time of 300 is often significantly longer than the necessary time of steady response of the structure (> 10 steady oscillations). The characteristic dimension (D) of the object is maintained at 1.0 which represents the reference length dimension of all the simulations (similar to 1 meter of the SI unit for length). From extensive studies on flow around stationary circular cylinder, it is found

that the Strouhal Number of a circular cylinder in open water is approximately 0.2 for a wide range of Reynolds Number ($150 < Re < 20,000$) [73]. For freely vibrating structures, the frequency of vortex shedding is commonly equivalent to the dominant frequency of lift coefficient. It is important to note that frequency ratio, true reduced velocity and amplitude ratio are yield variables of freely vibrating body simulations. For two-dimension freely vibrating structure, each direction of vibration would yield a frequency ratio, amplitude ratio and true reduced velocity. The true reduced velocity of the simulation is important for comparison between simulations of varying mass ratio [105, 45, 46]. The significance of frequency ratio is related to the occurrence of lock-in, which indicated the equivalence of the dominant response frequency to the natural frequency of experimental setup [106]. In essence, lock-in represented resonance of the vortex-induced vibration of the structure. For circular cylinder, such resonance is found to be very persistent and is strongly linked to the persistency of the von *Kármán* street in the wake of the circular cylinder [105]. The amplitude ratio represented the amplitude of response of the structure to its VIV. However, similar to the frequency ratio, the significance of amplitude ratio is aroused through the occurrence of resonance. It is also important to note that no fixed time step is applied to the simulation. The time step in the simulation is determined through Courant-Friedrichs-Lewy condition, defined as follows [21]:

$$C = \frac{U\Delta t}{\Delta x} \leq C_{max} \quad (5.18)$$

With reference to equation 5.18, it is necessary to indicate the control variable of this equation is Δx , C_{max} , and U . As such, variable time step will be applied with dependence on the value of the 'C' relative to C_{max} . The C_{max} of all the simulations is limited to 0.8. This resulted on average time step of approximate 0.03 (non-dimensionless time, for $U = 1D/s$ and $\Delta_x = 0.05D$). Detailed study on the setting of CFL criterion of the simulation code could be found in Weymouth [94].

Freely Vibrating Simulations

This subsection describes the implementation method of the free motion simulation on a structure. The free motion of the simulation on a structure is implemented through a mass-spring-dashpot system. Figure 5.4 illustrates the top view of such mass-spring-dashpot system.

With reference to figure 5.4, it is important to note that no damping(B) is implemented in a free motion system, as such, the damping of the system is zero. Although, the simulation code has the

Figure 5.4: Top view of mass-spring-dashpot system for implementation of free motion on a structure (m) where k is the spring constant, and B is the damping coefficient. However, as the simulated body is freely vibrating, the damping coefficient (B) is zero. The spring and dashpot system is applied at the centre of mass of the structure, which prevents the introduction of moment on the body.

functionality of providing damping on the system. The spring constant of the system(k) and mass of the structure(m) is the determining factor of the natural frequency of the system, as follows:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (5.19)$$

where $m = m^*V$ with V as the volume of the structure. The natural frequency from equation 5.19 is substituted into equations for calculating reduced velocity (or reduced frequency) for non-dimensionalisation shown in previous section. For a two-dimension freely vibrating structure, a streamwise natural frequency to crossflow natural frequency ratio (termed 'natural frequency ratio') is a crucial parameter and is defined as '2.0' for the purpose of our study. Such value is chosen with reference to previous studies, which indicated the condition of resonance occurring at such natural frequency ratio [112, 23]. Hence, the streamwise spring constant is four times of the crossflow spring constant. It is also important to note that the volume of the structure is calculated by the simulation code as the structure is loaded into the code. After determining the volume, multiplication of this value to mass ratio and density of the fluid is performed in order to determine the 'real' mass of the structure. In subsequent evaluation, the author has found that the published wavelength of the seal vibrissa is inaccurate [40]. As such, measurements on the wavelength of the vibrissa indicated a significantly longer wavelength (λ) of 4mm instead of 1.8mm. Additional simulations on the new vibrissa geometry are performed with a similar simulation domain in figure 5.4 with greater depth of 27.1D. It should be shown in chapter 6 that the new vibrissa geometry is capable of greater VIV reduction as compared to the old vibrissa geometry. As such, the results of the old vibrissa geometry

Figure 5.5: Simulation domain for tandem structures with no relative velocity between the structures. The freestream flow is parallel to the major axis of the structure. The crossflow width on each side of the domain is $5.5D_1$, where D_1 and D_2 are the characteristic diameters of the upstream and downstream structure, respectively (defined as the diameter of the structure that is perpendicular to the freestream flow). The lengths of the domain are divided into three sections, with the most upstream section at $4.0D_1$ (from the center of the upstream structure) and most downstream section at $4.5D_1$ (from the trailing edge of the trailing structure). A constant gap of $4D_1$ is maintained between the trailing edge of the upstream structure and the leading edge of the trailing structure. The height of the simulation domain is $12.2D_2$.

represented the worst case scenario for the response of the new vibrissa geometry (a case of worst sensitivity). Such case is similarly applicable to simulations on tandem structures with and without relative velocities between the structures.

5.2.2 Two tandem bodies simulation with no relative velocity

For simulation on tandem bodies with no relative velocity between the two bodies, a different domain is designed. Such domain is longer and wider as the trailing structure is allowed to freely vibrate in crossflow direction. Figure 5.5 depicts the simulation domain for the two tandem bodies with no relative velocity.

With reference to figure 5.4, the simulation domain for tandem bodies are longer and wider with the same depth (height). The greater length is necessary for a greater pressure convergence and accuracy. It is also necessary to allow for sufficient spacing in the wake of the trailing structure, which will avoid pressure build up near the exit plane of the flow. Such pressure build up will affect the accuracy of the simulation. Hence, the length of the domain downstream of the trailing cylinder is designed to be $4.5D_1$ from the trailing edge of the trailing structure, where D_1 is the characteristic diameter of the wake-generator circular cylinder. A gap of $4D_1$ is designated between the trailing edge of the wake-generator circular cylinder and the leading edge of the trailing structure. Such gap will ensure the full-development of von *Kármán* street from the wake-generator circular cylinder [110, 111]. In turn, the von *Kármán* street shall provide unsteady but stable characteristic wakes for the detection by the wake structure. The greater width ($5.5D$ on each side) of the simulation domain is to ensure the accuracy of the response of the trailing structure to Wake-Induced Vibration (on simulations where free vibration in transverse direction are allowed for the trailing structure). However, the wake-generator circular cylinder (upstream) is always stationary. Simulations are also performed with various relative transverse positioning between the wake-generator circular cylinder and the trailing structure. Such positioning ranges from $0.0D$ (inline) to $1.5D$ with an increment of $0.25D$, where D is the characteristic diameter of the wake-generator circular cylinder. Such positions are designated on the trailing structure to evaluate the effects of the wakes of the circular cylinder on the structure. Previous studies have indicated that the main wake affected region of a circular cylinder is $\sim 1.0D$ [110]. As the objective of the tandem structure simulations is to assess the sensitivity of the vibrissa geometry, reference comparison is necessary. Three trailing structures are designated i.e. circular cylinder, elliptical cylinder and vibrissa. Each simulation will consist of one of the aforementioned trailing structure in the wake of the upstream circular cylinder. Two types of settings are configured for the trailing structure. In order to study the transverse positional sensitivity of the trailing structure to the wake of the upstream circular cylinder, simulations with stationary trailing structures are performed at various Reynolds Number. Such simulation should result in the rate of change of the steady and fluctuating lift coefficient as its transverse position is varied. Simulations are also performed with a freely vibrating in transverse direction trailing structure. Such simulations are necessary in order to assess the response of the trailing structure to WIV, which possess strong similarity to a harbour seal tracking wakes with its vibrissae. Further studies are also performed

through variation of the relative sizes of the structure. The ratio of the characteristic diameter of the wake-generator circular cylinder (D_1) to the characteristic diameter of the trailing structure (D_2) is defined as Characteristic Dimension Ratio (CDR), mathematically written as:

$$CDR = \frac{D_1}{D_2} \quad (5.20)$$

CDR is varied between 0.6 to 3.0 with proper scaling of the simulation domain (refer to figure 5.4). For easier visualization, an analogy of shrinking trailing cylinder with respect to the simulation domain and upstream circular cylinder can be used. For freely vibrating in transverse direction simulation, similar implementation method described in previous subsection is applied. The natural frequency of the trailing structure in crossflow direction is determined to be equivalent to the resonant frequency of the structure in open water, as follows:

- * For circular cylinder, $f^* = 0.1847$
- * For elliptical cylinder, $f^* = 0.1847$
- * For old vibrissa, $f^* = 0.083$
- * For new vibrissa, $f^* = 0.083$

Similar to single body simulations, error on the wavelength of the vibrissa geometry is identified after the completion of the studies on the tandem structures. As such, additional simulations are performed to assess and compared the performance of the new vibrissa geometry and old vibrissa geometry to WIV.

5.2.3 Two tandem bodies simulation with non-zero relative velocity

This subsection describes the simulation settings for tandem bodies with non-zero relative velocities. Such simulation is intended for study on the characteristic response of the harbour seal vibrissae as the seal approaches its prey or loses the hydrodynamic trail of its prey. As such, in comparison to simulations with two tandem structures with zero relative velocity, the simulation domain for two tandem structure with non-zero relative velocity is significantly longer. However, the width and depth of the simulation domain is equivalent due to the same WIV and settings of the simulation. The length of the simulation domain is increased to $\approx 30D_1$ and divided into three section. As aforementioned, two

different types of simulations are performed to simulate the approaching and distancing harbour seal from its prey. The first type of simulation involves a trailing structure (i.e. circular cylinder, elliptical cylinder or vibrissa) approaching the wake-generator circular cylinder from a large distance. The second type of simulation involves a trailing structure starting closely to the wake-generator circular cylinder and moving away from it. For approaching type simulation, the initial lengths of the simulation domain are divided as $4.0D_1$ on the most upstream section (from the center of the wake-generator circular cylinder), $2.0D_1$ most downstream section (from the the trailing edge of the downstream structure) and $23.0D_1$ between the trailing edge of the wake-generator circular cylinder to the leading edge of the trailing edge structure. Figure 5.6 illustrates the simulation domain for the approaching type simulation. The simulation is allowed to run to a simulation time of t^* (based on D_1) = 180 for fully developed von *Kármán* street of the wake-generator circular cylinder. At $t^* = 180$, the trailing structure is moved at a designated velocity ($U_{trail} = (1D \text{ or } 2D)/\text{Solution time}$) towards the wake-generator circular cylinder. As the travelling distance of the trailing structure is limited by the computational cost, the travelling velocity of the trailing structure is consequently limited by its travelling distance. The average acceleration of the trailing structure is set at a constant value of $0.4D/(\text{Solutiontime})^2$. In order to avoid the occurrence of secondary instability, the velocity profile during acceleration is set to follow a sine curve. At a distance of $0.5D_1$ between the trailing edge of the wake-generator circular cylinder and the leading edge of the trailing structure, the simulation is terminated. For distancing type (second type) simulation, a reversal condition of spacing is true. The initial lengths of the simulation domain is similar to approaching type simulations with exception of $0.5D_1$ gap between the trailing edge of the wake-generator circular cylinder and the leading edge of the trailing structure. Similar simulation time is allowed before the initiation of the motion of the trailing structure ($t^*(\text{based on } D_1) = 180$). Sine and cosine velocity profile is implemented for the acceleration of the trailing structure away from the wake-generator circular cylinder with similar acceleration and velocities of trailing structure simulated. The simulation will be terminated when the gap between the trailing edge of the wake-generator circular cylinder and the leading edge of the trailing structure is $23.0D_1$. It should be noted that $Re = 500$ is maintained for all the simulations with respect to the characteristic dimensions of the trailing structure. This is because of the main purpose of the study on understanding the response of the seal vibrissa to a stable perturbation. As such, the dynamic similarity on the seal vibrissae is of utmost importance. However, care is taken to

Figure 5.6: Simulation domain for tandem structures with no relative velocity between the structures. The freestream flow is parallel to the major axis of the structure. The crossflow width on each side of the domain is $5.5D_1$, where D_1 and D_2 are the characteristic diameters of the upstream and downstream structure, respectively (defined as the diameter of the structure that is perpendicular to the freestream flow). The lengths of the domain are divided into three sections, with the most upstream section at $4.0D_1$ (from the center of the upstream structure) and most downstream section at $4.5D_1$ (from the trailing edge of the trailing structure). A constant gap of $4D_1$ is maintained between the trailing edge of the upstream structure and the leading edge of the trailing structure. The height of the simulation domain is $12.2D_2$.

ensure that the vortex shedding of the upstream (larger or smaller) circular cylinder would occur in a stable von *Kármán* street [104]. As described in the above subsections, additional simulations on the new vibrissa geometry trailing a wake-generator circular cylinder are performed and compared to the response of the old vibrissa geometry trailing a wake-generator circular cylinder. Simulation on the correct geometry will be performed on an equivalent domain with greater depth ($27.1D_2$ instead of $12.2D_2$).

5.3 Grid design and validations

As indicated in the first section of this chapter, ILES is implemented as the turbulence model in the simulation code. As such, care should be taken in the grid design of the simulation domain. This is due to the strong dependence of the accuracy of the ILES on the grid design, primarily the dissipation of turbulence afforded in the simulations [94]. Accordingly, a minimum of 10 grid points across the main features of the simulated structures is the sufficient condition for accurate simulation results. However, in order to ensure better accuracy at an optimised computational cost, the grid is designed with a minimum of 20 grid points across the main features of the simulated structures. As such, a minimum grid size of 0.05 is applied on the streamwise and crossflow direction of the simulation domain. However, as the spanwise length of the structures are significantly greater than the width and the length of the simulation domain, a minimum grid size of 0.1 is applied. This coincided with the significantly larger wavelength of the vibrissae (wavelength of old vibrissa $[\lambda/D_{min}] \approx 3.6$, wavelength of new vibrissa $[\lambda/D_{min}] \approx 8.0$, where D_{min} is the minimum minor diameter of the undulatory vibrissae) [40]. Hence, the fineness of the spanwise grid is not the limiting factor in the grid design. The grid in the vicinity of the vibrissa is designed to be the finest primarily in the area 1.5D upstream, 3.5D downstream, 1.2D on each crossflow side and the entire length of the simulated structure (all dimensions measured from the centre of the structure to be analysed). A constant expansion rate of 1% is applied on the grid sizes in the streamwise and crossflow direction outside the designated fine-grid area. Figure 5.7 illustrates mesh for the single body simulation domain.

For two tandem structures with zero relative velocity, a larger simulation domain is required (refer to figure 5.4). As such, it is also necessary for the von *Kármán* street of the wake-generator circular cylinder and the response of the trailing structure to be as accurate as possible. Such requirement is imposed on the grid design through extension of the fine-grid region to the entire streamwise length of the simulation domain. Widening of the fine-grid region along the crossflow direction is also performed to a length of 1.7D on each side of the structure in crossflow direction. As the spanwise grids are well-refined on its entire length, no further refinement in spanwise direction are performed. For two tandem structures with non-zero relative velocity, the necessary requirement of accurate von *Kármán* street of the wake-generator circular cylinder and response of the trailing structure should be applicable. As such, similar to the case of two tandem structures with zero relative velocity, an extension on the fine grid area is applied. Hence, the entire streamwise length of the simulation

Figure 5.7: Mesh of the simulation domain for single body simulation. Grid sizes in streamwise, crossflow and spanwise directions are 0.05, 0.05, and 0.1D, respectively (refer to table 5.1 for grid refinement). Fine-grid area can be identified through the region of darker coloration.

Schematic No.	dx * dy * dz	Drag Coefficient (C_D)	Computational Time (hours)
1	0.1 * 0.1 * 0.1	0.5	192
2	0.05 * 0.05 * 0.1	0.49	1024
3	0.05 * 0.05 * 0.05	0.51	2688
4	0.025 * 0.025 * 0.1	Unstable	123008
5	0.025 * 0.025 * 0.05	0.5	776320

Table 5.1: Drag coefficient of harbour seal vibrissa for various grid size combinations and its corresponding computational cost with state-of-the-art single core computer. The grid size is normalised to the characteristic diameter of the vibrissa (the minor diameter of the vibrissa) with dx,dy and dz representing the grid size in streamwise, crossflow and spanwise direction respectively.

is defined as the fine-grid region with similar width of fine-grid in crossflow direction to the two tandem structures with zero relative velocity. Figure 5.8 depicts the mesh of the simulation domain for approaching and distancing trailing structure to a wake-generator circular cylinder.

For figures 5.7,5.8, the regions of finest-grid is represented by the regions of darker coloration. As aforementioned, a 1% expansion of the grid size in every direction is implemented from the finest-grid region to the edges of the simulation domain. Such methodology would ensure high accuracy of flow around the structures and their responses (priority of near-field flow) while balancing the computational cost of the simulations (through reduction of accuracy in the farthest section of the flow in the simulation domain). For optimisation of computational efficiency and accuracy, grid refinement tests are performed. The grid refinement tests are performed on flow around the new vibrissa geometry at $Re = 500$. Various grid sizes are simulated with comparison of the mean drag coefficient of the vibrissa geometry. The simulation is performed for $t^* = 500$, where the drag coefficient has reached a steady value for a significant number of oscillations (> 50 oscillations). Table 5.1 depicts the drag coefficient and computational cost of the harbour seal vibrissa for various grid fineness.

With reference to table tab:501, it is observed that schematic 1 and 2 possessed the necessary accuracy for the simulation with reasonable computational cost with schematic 1 requiring significantly less computational time as compared to schematic 2. However, from flow visualisation of similarly-sized grids on flow around circular cylinder (similar simulation settings), certain features of the wake of the circular cylinder is inaccurate. Notably, the vortical braids in the wake of the circular cylinder are found to be irregularly spaced. Figure 5.9 shows the comparison between the stable von *Kármán* street of circular cylinder with grid sizes in schematic 1 and 2.

With reference to figure 5.9, it should be noted that the grid sizes of the left circular cylinder is based on three-quarter (from top to bottom in spanwise direction) of schematic 1 and one-quarter

Figure 5.8: Mesh of the simulation domain for approaching and distancing vibrissa to the wake-generator circular cylinder simulation. Grid sizes in streamwise, crossflow and spanwise directions are 0.05, 0.05, and $0.1 D_2$, where D_2 is the characteristic diameter of the trailing structure ($D_2 = D_1 = D$ for two tandem structures of equivalent characteristic diameter). Fine grid area can be identified through the regions of darker coloration.

Figure 5.9: Isometric view (Top) and Top View (Bottom) of coherent structure (vortex cores) in the steady wake of a freely vibrating in transverse direction circular cylinder at $Re = 500$ for grid sizes based on (Left) Schematic 1 for the top three-quarter of the structure and Schematic 2 for the rest and (Right) Schematic 2 for the entire spanwise length of the structure (refer to table 5.1 for details on the schematic).

(lower one-quarter in spanwise direction) of schematic 2. Such configuration is created in order to depict a direct consequence of the two schematics of grid sizes. The top and bottom figures in figure 5.9 depict the isometric and top view of the steady flow around freely vibrating in transverse direction circular cylinder, respectively. The settings of the simulation is similar to the settings of the

simulation for harbour seal vibrissa (refer to above subsection on 'Single body simulations'). From figure 5.9, vortex tubes can be found in the figures on the left and right. However, it can be observed that the vortical braids are much more consistent on the lower quarter of the left figures. On the top three-quarter of the left figure, the vortical braids are nearly non-existent. Such is not the case for the right figures in figure 5.9. Regular vortical braids are observable along the spanwise length of the circular cylinder of the right figures, similar to the observed experimental results shown in figure 5.9 [104]. Such difference in the topology of the wake concluded the necessary grid sizing for capturing the details of response and wake of all the simulations with the BDIM code. Hence, despite the higher computational cost, schematic 2 is chosen for all the grid sizing of the aforementioned simulations due to its accuracy in determination of responses of the structure and wake topology of the structure.

As mentioned earlier, a cost-effective yet accurate simulation end time is required to perform the planned simulations. Additionally, accuracy of the simulation would prove to be the key component in balancing the cost-effectiveness of the planned simulations. The author of the code has performed validations on the code for flow around a stationary circular cylinder at $Re = 100$ and 200 with comparison to previous experiments. Table 5.2 shows comparison between the RMS lift coefficient, mean drag coefficient and the Strouhal frequency of the stationary circular cylinder at $Re = 100$ and 200 . From table 5.2, it can be observed that the code is well validated within the low Reynolds number regime, particularly for flow around a circular cylinder. Figure 5.10 depicts the time series of the lift and drag coefficient of a circular cylinder and an old vibrissa exposed to a freestream flow (separate simulation for each) at $Re = 500$. From the results of figure 5.10, the simulations are performed for a simulation time (t^*) of 3000. This is approximately 600 steady cycles of oscillation (one steady oscillation period for circular cylinder, $T^* = TU/D$, 5 where T is the period of one steady-oscillation in seconds) [72]. Hence, such simulations are significantly longer than the necessary duration of simulation for achievement of steady oscillations. Equivalently, an experiment on a 3 km long towing tank will be necessary in order to achieve a similar time series of results (for towing speed of 1m/s). From figure 5.10, It can be observed that the drag and lift coefficients of the circular cylinder and old vibrissa model attained steady amplitudes during the early stages of the simulation. The drag coefficients of the circular cylinder and old vibrissa geometry show minor fluctuations as compared to the fluctuations of their corresponding lift coefficients. For further validations, comparison between mean drag coefficient, RMS lift coefficients, and Strouhal frequency of the current simulation of flow

around a stationary circular cylinder at $Re = 500$ and previous experimental/simulation studies are conducted. The mean drag coefficient of the circular cylinder is found to be 1.095 and the RMS lift coefficient of the circular cylinder is observed to be 0.4254. Comparison between the drag coefficient of the circular cylinder from this simulation and previous works is found to conform very well [98, 70]. Comparison on the RMS lift coefficient of the circular cylinder in this simulations is also conducted with strong agreement observed relative to previous studies [70, 72]. The author would like to note that while literature review on circular cylinder is ample, limited number of papers are found on studies of flow around circular cylinder at low Reynolds Number (such as $Re = 500$) and with similar aspect ratio as the current simulations, in particular, experimental work. The dominant response frequency of the lift force (corresponds to Strouhal Number) is found to be 0.1962 and the dominant response frequency of the drag coefficient is found to be 0.4056. The dominant response frequency of the lift coefficient of the circular cylinder is found to strongly agree to findings by Roshko [79]. From previous studies, it was also observed that the dominant frequency of the drag coefficient is approximately twice to the dominant frequency of the lift coefficient, which is the consequence of the nature of the *Kármán* street [104]. Such result is expected from previous studies on the influence of vortex shedding on a circular cylinder to its drag and lift coefficients [103, 104, 73]. A brief fluctuation is observed on the lift coefficient of the circular cylinder in the vicinity of $t^* = 2000$. However, a long period of steady amplitude is observed on the plot of lift coefficient of the circular cylinder before and after this fluctuation. Analysis of the phase plot and Poincare Map of the drag and lift forcing have also revealed a consistent limit-cycle response. As such, doubt is cast aside on the conclusiveness of the steady-amplitude observed in the plot of the lift coefficient of the circular cylinder. With reference to the plot of lift coefficient of the old vibrissa geometry, it can be observed that amplitudes of oscillation do not remain completely steady for each oscillation cycle. However, a careful observation shows that this is due to the significantly smaller lift coefficient of the old vibrissa geometry as compared to the circular cylinder (an order of magnitude smaller). The RMS lift coefficient and mean drag coefficient of the old vibrissa geometry is found to be 0.0264 and 0.6487, respectively. The dominant frequency of the lift coefficient of the old vibrissa is found to be 0.1807. However, no dominant frequency of drag coefficient is observed for the old vibrissa. Small nominal fluctuations on the time series plot of the lift coefficient of the old vibrissa may seem large due to the very small nominal value of the lift coefficients. Consequently, the old vibrissa geometry is also

Authors	Re = 100			Re = 200		
	C_D	C_L	St	C_D	C_L	St
Liu et al. [64]	1.35 ± 0.012	± 0.339	0.164	1.31 ± 0.049	± 0.69	0.192
Choi et al. [19]	1.34 ± 0.011	± 0.315	0.164	1.36 ± 0.048	± 0.64	0.191
Liu et al. [18]	1.35 ± 0.012	± 0.303	0.167	1.37 ± 0.051	± 0.71	0.198
Current code (BDIM) [93]	1.35 ± 0.013	± 0.330	0.167	1.34 ± 0.050	± 0.69	0.195

Table 5.2: Comparison of mean drag coefficient, RMS lift coefficient and Strouhal frequency on a stationary circular cylinder at Re = 100 and 200 [93].

found to achieve a quasi-steady amplitude state during the early stages of the simulation. Such result is confirmed through observations of the drag coefficient of the old vibrissa geometry where steady-displacement of the drag coefficient is found to occur during the first ten percentile of the simulation time. This shows that steady-amplitude results can be achieved through simulations of significantly shorter duration. In order to determine a more concise reading of the plots, zoomed-in view of the images in figure 5.10 is depicted in figure 5.11.

From figure 5.11, it can be observed that the drag and lift coefficients of the circular cylinder reached steady-amplitude at $t^* = 250$. The drag and lift coefficients of the old vibrissa geometry is observed to achieve steady-amplitude at $t^* = 90$. Hence, it is seen that the elliptical cross-section of the old vibrissa geometry significantly reduced the transient period of response of the geometry. Such results indicated that a simulation time of 300 will be sufficient for most structures except for circular cylinder. However, for a freely vibrating in transverse direction circular cylinder, a significantly shorter transient period is observed. This is due to the greater efficiency of power transfer between the fluid and the circular cylinder. Hence, a shorter transient period is possible due to the nature of the fluid-structure interaction. Figure 5.12 shows the time series of the motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow at Re = 500 and reduced velocity, $U^* = 5$. From figure 5.12, it can be confirmed that the transient period of a freely vibrating in transverse direction circular cylinder is significantly shorter than that of a stationary circular cylinder. This is due to the smoother power transfer between the circular cylinder and its surrounding fluid. The steady-amplitude is found to occur at $t^* > 50$. This further justified the sufficiency of a simulation end time of $t^* = 300$. In concurrence with the stationary results, the transient period of a freely vibrating in transverse direction old vibrissa geometry is found to be shorter than that of the freely vibrating in transverse direction circular cylinder.

Weymouth et al have also performed various code validations to Poiseuille flow [94]. Their

Figure 5.10: Time series of the force responses of a circular cylinder (Left) and an old vibrissa (Right) in flow around the structures at $Re = 500$ (separate simulation for each structure) up to simulation time, $t^* = 3000$. Each structure is stationary relative to the freestream flow. Top: Drag Coefficients (C_D); Bottom: Lift Coefficients (C_L).

validations have shown the great accuracy of the simulation code for external around structures and internal flow in structure. For our studies, further validation on freely vibrating in transverse direction circular cylinder is performed. However, due to the low Reynolds number ($Re = 500$) of the current study, limited comparison could be found for flow around a freely vibrating circular cylinder (most studies are found to occur at Reynolds number 10^4 to 10^6) [105]. For a freely vibrating in crossflow circular cylinder, validations are performed through comparison with Direct Numerical Simulation (DNS) results (at $Re = 500$) and experimental results ($Re = 3,000$ to $12,000$) [13, 50, 51, 52]. Figure 5.13 shows comparison between the steady-amplitude of response and dominant response frequency ratio of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $m^* = 1.04$ for various reduced velocity based on simulation results with the current code and Direct Numerical Simulation (DNS) [90, 30, 31, 20, 13].

Figure 5.11: Time series of the force responses of a circular cylinder (Left) and an old vibrissa (Right) in flow around the structures at $Re = 500$ (separate simulation for each structure) up to simulation time, $t^* = 500$. Each structure is stationary relative to the freestream flow. Top: Drag Coefficients (C_D); Bottom: Lift Coefficients (C_L).

Figure 5.12: Motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow at $Re = 500$ and $U^* = 5$. Left : up to $t^* = 1900$; Right : up to $t^* = 500$.

Figure 5.13: Motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow at $Re = 500$ and $m^* = 1.04$ for various reduced velocity. Top: Steady-amplitude ratio (A/D); Bottom: dominant response frequency ratio [13].

With reference to figure 5.13, it can be observed that the amplitude of response and dominant response frequency of the circular cylinder in the freestream flow are well matched between our simulation results and the simulation results of DNS for the various overlapping reduced velocity. Small deviations at larger reduced velocity ($U^* = 8$ and 9) are observed (deviations may seem large due to the scale of the y-axis). The wake topology of the circular cylinder in our simulation results is also found to match the expected wake topology from experimental wake visualisation (refer to top

right figure in figure 2.18. It is important to note that simulations results of DNS with spectral element method has been shown to be very accurate in correlation with the experimental results of similar settings [90, 20]. Hence, as strong agreements are observed on the amplitude of response and dominant response frequency, it provides further validation of the simulation code (on top of validations by Weymouth [94, 97, 93]). For further validation, figure 5.14 shows comparison of the response of a freely vibrating in crossflow circular cylinder for $Re = 3,000 - 19,000$ from our simulations, experiments (performed by Heather Rachel Beem, MIT Tow Tank) and previous experimental studies.

From figure 5.14, good agreements can be observed between the amplitude ratio from the simulations results and experimental results of similar settings (compare blue-cross, red-plus and magenta-box signs). Similar amplitude of oscillation is also observed for experiments and simulations of various mass ratio and Reynolds Number. From previous studies, it is expected that a circular cylinder with lower mass ratio will allow for greater region of lock-in (lock-in occur at more reduced velocities). As such, it can be seen that our simulation results matched the experimental results. However, a small difference in the width of lock-in region can be observed through comparison between the simulation results of the circular cylinder at $Re = 5000$ & $m^* = 10.1$ (blue-cross) and experimental results at $Re = 3000 - 12000$ & $m^* = 10.1$ (magenta-box). Such difference implied that the simulation code underestimated the occurrence of lock-in at slightly higher reduced velocities, hence, care should be taken for results at higher reduced velocities above resonance. This difference could be due to the non-zero damping in the real experimental setup and the variability of the material properties of the spring at various frequency of oscillation. These two factors are non-existent in the case of simulations where the damping is always zero with a consistent spring constant. However, as our objectives is to understand the underwater sensing principle of the harbour seal vibrissae, priority is given to understanding of the response of the harbour seal vibrissae in worst case scenario, where resonance of the structure occurs. As such, the accuracy of the simulation results at resonance is of utmost importance and it has been shown in figure 5.13, 5.14 that our simulation code possessed a very good accuracy at resonance. Chapter 8 will provide further validations for two tandem stationary circular cylinder in a freestream flow with comparison to experimental studies [4].

Figure 5.14: Motion response of a freely vibrating in transverse direction circular cylinder to VIV in a freestream flow for various reduced velocity, Reynolds Number and mass ratio. Top: Steady-amplitude ratio (A/D); Bottom: dominant response frequency ratio. The first, second and third signs in the legend represents our simulation (first and second signs) results and experimental results (third sign, conducted by Heather Rachel Beem, MIT Tow Tank), respectively.

5.4 Analysis method of forces and responses of the structures

In order to understand the various nature of the problem, it is necessary to employ a wide variety of methods to evaluate aspects of the results. In this section, descriptions of methodology for flow visualisation, force analysis and motion analysis are elaborated.

5.4.1 Flow visualisation

As flow visualisations is very important for understanding of the physics of the flow, careful observation and methodology for identification of the vortex cores and coherent structures are developed. From previous studies, it is found that intuitive ways of identifying vortex cores has failed to in clearly identifying these cores [48]. Minimum pressure method has been found to fail due to the possible existence of pressure minimum in an unsteady irrotational motion. Such occurrence would not have fulfilled the first criteria of vortex core, both of which are described as follows:

1. A net circulation (net vorticity) should be finite for vortex cores. Therefore, regions of potential flow are excluded from vortex cores, and a potential flow vortex is a vortex with no cross-section
2. The geometry of the identified vortex core must be Galilean invariant (fulfil Galilean relativity)

A pressure minimum method of vortex core identification may also fail in representing the correct vortex cores such as the occurrence of sink and source, where pressure minimum existed but vortex cores did not [48]. Additionally, for mixing layer, iso-pressure surface could not clearly distinguish between the ribs and rolls of the wake. Figure 5.15 depicts the ribs and rolls in a mixing layer through iso-pressure surface identification and λ_2 identification.

From figure 5.15, it can be observed that the topology of the vortex cores in a mixing layer is different with minimum pressure identification method and with λ_2 criterion. A significantly clearer indication of the ribs and rolls are visible with λ_2 criterion as compared to the minimum pressure method. It is also important to note that the minimum pressure varied with time for unsteady flows, as such, the nominal value of the iso-pressure surface has to be varied every time step. Hence, the pressure minimum method also posed problems of practicality. Another intuitive method is developed through the use of streamlines and pathlines. However, streamlines and pathlines are Galilean variant, as such,

Figure 5.15: Comparison of the topology of vortical structures for a mixing layer with: a) iso-pressure surface based on minimum pressure identification, b) iso-surface of $\lambda_2 = 0$ criterion [48].

they did not fulfil the necessary criteria of vortex cores. Furthermore, such method would fail in a turbulent flow with numerous vortices travelling at different speeds [48]. Plotting of iso-vorticity surface is found to fail in a shear flow where the shear flow in the background is comparable to the vorticity magnitude of the vortices. Hence, a different method is proposed, termed λ_2 criterion [48]. Such method bore similarity to the method of Q-factor [47]. However, the λ_2 criterion method is believed to perform better in identification of a vortex core, notably on conically symmetric vortex, Bodewadt vortex and axisymmetrical axial vortex within a vortex ring [48]. The λ_2 criterion can be defined through the following method. Taking the gradient of Navier-Stokes equation, the author has:

$$a_{i,j} = -\frac{1}{\rho} p_{i,j} + \nu u_{i,jkk'} \quad (5.21)$$

where $a_{i,j}$ is the acceleration gradient, $p_{i,j}$ is symmetric, ν is the diffusion (viscous) coefficient and $u_{i,jkk'}$ is the gradient of the Laplacian operator for the convective velocity field. As such, $a_{i,j}$ could be decomposed in to symmetric and asymmetric parts shown as:

$$a_{i,j} = \left[\frac{DS_{i,j}}{Dt} + \Omega_{ik}\Omega_{kj} + S_{ik}S_{kj} \right] + \left[\frac{DQ_{ij}}{Dt} + \Omega_{ik}S_{kj} + S_{ik}Q_{kj} \right] \quad (5.22)$$

where

$$S_{ij} = \frac{1}{2}(u_{i,j} + u_{j,i}) \quad (5.23)$$

$$\Omega_{ij} = \frac{1}{2}(u_{i,j} - u_{j,i}) \quad (5.24)$$

In equation 5.22, the first square bracket represents all the symmetric terms of the acceleration gradient and the second square bracket represents the asymmetric terms of the acceleration gradient (vorticity transport equation). Hence, substituting the symmetric terms of the acceleration gradient into equation 5.21, the author has:

$$\frac{DS_{i,j}}{Dt} + \Omega_{ik}\Omega_{kj} + S_{ik}S_{kj} = -\frac{1}{\rho}p_{i,j} \quad (5.25)$$

In order to define the minimum pressure region due to vortical motion, only $S^2 + \Omega^2$ shall be considered as unsteady strain and diffusion may produce minimum pressure with no vortical motion. As such, a vortex core is defined as a connected region with two negative eigenvalues of $S^2 + \Omega^2$. Such definition excluded the possibility of vortex core centred at a wall (common error of iso-vorticity surface identification method) as $S^2 + \Omega^2$ vanished at a wall. Hence, such λ_2 criterion will be implemented for all the vortex-core visualisation around the structures. Figure 5.16 shows a sample view of vortex-core identified with λ_2 criterion in the flow around a stationary circular cylinder trailed by another circular cylinder at $Re = 500$.

For figure 5.16, the visualisation of the vortex cores are performed with Tecplot 360. The vortex core can be easily identified in the software by plotting the $\lambda_2 = 0$ iso-surface. In figure 5.16, the vortex cores (i.e. vortex braids and tubes) can be clearly identified in the flow. Such identification would have been less prominent and troublesome to identify with minimum pressure or iso-vorticity surface methods.

5.4.2 Force analysis

As indicated earlier in this chapter, the outcome of the simulations include streamwise, crossflow, and spanwise forces acting on the body, as calculated by equation 5.17. However, the forces acting on the body due to the flows are dependent on the geometry and mass of the body. As such, it is necessary to normalise these forces into force coefficient for easier comparison across geometries

Figure 5.16: Stable von *Kármán* street in the wake of a stationary circular cylinder trailed by another circular cylinder (far right) identified at $Re = 500$ with λ_2 criterion.

and masses. Such equations are presented as follows (for drag and lift coefficient, similar definitions have been shown earlier):

$$C_D = \frac{Drag}{\frac{1}{2}\rho U^2 D} \quad (5.26)$$

$$C_L = \frac{Lift}{\frac{1}{2}\rho U^2 D} \quad (5.27)$$

$$C_Z = \frac{Z - Force}{\frac{1}{2}\rho U^2 D} \quad (5.28)$$

where Drag, Lift, Z-force are the forces acting on the structure in streamwise, crossflow and spanwise directions, respectively. The force coefficients are plotted as a function of the simulation time. It

is necessary to point out that the simulations resulted in pressure forces and viscous forces. The summation of pressure and viscous force resulted in the Total Force on the body, which is represented as Drag, Lift, and Z-Force in equation 5.28.

From the plot of variation of these forces with time, it is possible to determine the transient and steady period of the force response. Statistical parameter such as mean, root-mean-square (RMS) and variance of each forces are calculable, although the first two methods are the most useful methods for understanding the underwater sensing ability of the harbour seal vibrissae. In order to determine the variability of the force or motion responses of the structure, normalised standard deviation are calculated. However, in order to properly normalised and define such variability, a unique method is devised. Equation 5.29 described the definition of the normalised standard deviation of response (ϕ_{std})

$$\phi_{std} = \frac{\sigma_{rms}}{\phi_{rms}} \quad (5.29)$$

where σ_{rms} and ϕ_{rms} are defined as follows:

$$\sigma_{rms} = \sqrt{\frac{1}{N} \sum_i^N (\phi_i - \mu)^2} \quad (5.30)$$

$$\phi_{rms} = \sqrt{\frac{1}{N} \sum_i^N \phi_i^2} \quad (5.31)$$

where the variables in equations 5.29, 5.31 are defined as follows:

- * N - overall number of data
- * ϕ_i - i^{th} datum of the N number of data
- * ϕ_{rms} - Root-Mean-Square (RMS) value of the data ϕ_i
- * μ - statistical mean of the data ϕ_i
- * σ_{rms} - standard deviation of the data ϕ_i with respect to their RMS value (instead of statistical mean)
- * ϕ_{std} - normalised standard deviation of σ_{rms} with respect to the RMS for dataset ϕ_i

It should be noted that the RMS values in this report are calculated with the conventional method, as shown in equation 5.31. Such definition of normalised standard deviation (ϕ_{std}) is created in order to

define the variability of the data about the RMS value of the responses. This is because the statistical standard deviation (and therefore) variance is performed about the mean value of the dataset (responses). As such, for lift coefficient and motion response of structures undergoing vortex shedding and VIV, respectively, the mean of such value would be very close to 'zero'. For direct comparison between different geometries, normalisation of such standard deviation to its mean value of ≈ 0 would incur significant error due to the minuscule denominator. Therefore, no significance and conclusive evidence could be drawn from such definition. Variability of the responses (data) about their RMS value represent a quantity of measure on the steadiness of the amplitude of the responses (force and/or motion) with time. Hence, for direct comparison between different geometries, normalisation of such redefined standard deviation to its RMS value is found to be desirable due to the use of such RMS value as reference, in earlier calculation. The physical implication maybe clearer as contemplation on the implementation of this parameter on a sinusoidal wave, which should yield $\phi_{std} = \sqrt{2}$. A value greater than that of the sinusoidal would imply a greater overall variation of the responses in comparison to the sinusoidal wave, and vice versa. For further investigation, a Fast Fourier Transform is performed on the force coefficients in temporal domain to yield the corresponding coefficients in frequency domain. Welch method is applied for such conversion from temporal to frequency domain. The MATLAB(2013) software is used for the application of Welch method. The Welch method is found to allow for greater accuracy in estimation the power spectral density (PSD) of a signal [92]. The Welch method is applied with 50% overlap between each windows and a division of 8 windows for the entire signals. For better accuracy, it is essential to inform the Welch method routine in MATLAB of the sampling frequency of the data. However, as previously indicated, the stability of the simulations and its time step are dependent on the Courant-Friedrich-Levy number [21]. Hence, the time step will vary with simulation time. Figure 5.17 shows the variation of time step with simulation time for flow around a stationary circular cylinder at $Re = 500$.

From figure 5.17, it can be observed that the time step converged almost immediately after the simulations are initiated. The time step is found to stabilize around 0.04 dimensionless time. Despite some variation, it can be observed that the variation of the time step is minimum almost immediately after the initiation of the simulation. As such, the time-step average of the simulation is calculated from the data with only minor error due to the small variability of the time step. The inverse of the time-step average is taken as the sampling frequency of the data.

Figure 5.17: Variation of simulation time step with simulation time as a function of Courant-Friedrich-Levy number [21]. This simulation is performed on the flow around stationary circular cylinder at $Re = 500$.

From the power spectral density (PSD) plot of the drag, lift, z-force and the motion response of the bodies, their corresponding dominant response frequencies could be identified. Although the dominant response frequencies could be identified, the type of response of the system may not be obvious, e.g. periodic, unstable but predictable or chaotic. As such, it is important to identify the phase plot of the system in order to determine the behavioural response of the system due to VIV. Figure 5.18 shows phase plot of the lift coefficient on the stationary circular cylinder due to freestream flow at $Re = 500$.

Such method could also be applied on other force coefficient and the motion response of the structures. On figures 5.18,5.19, the black colour indicated the initial response of the circular cylinder at time ≈ 0 . As the simulation progresses, the lift coefficient of the circular cylinder brightened towards bright red. Hence, the darker colour indicated the lift coefficient response of the circular cylinder at earlier simulation time. The brighter red colour indicated the lift coefficient of the circular cylinder at a later simulation time. From figure 5.18, it can be observed that the lift coefficient of the circular cylinder responded in a predictable manner although not periodic. It should be noted that the y-axis in figure 5.18 is the rate of change of lift coefficient of the circular cylinder with time. A more concise method of indicating the type of response is found in Poincare Map. Poincare map created a mapping of the phase plot into a lower-dimensional space through identifying the points in the phase plots

Figure 5.18: Phase plot of the lift coefficient of a stationary circular cylinder in a freestream flow at $Re = 500$. From black to bright red marker $\sim$ from initial simulation time to the end of simulation time.

Figure 5.19: Poincare map of the lift coefficient of a stationary circular cylinder in a freestream flow at $Re = 500$. From black to bright red $\sim$ from initial simulation time to the end of simulation time.

which passes through the points of origin or the points in the vicinity of origin. Such method can be obtained through performing a plane cut on the x-positive, x-negative, y-positive, or y-negative planar axis with dependence on the location of the first point. Figure 5.19 shows the Poincare map of the lift coefficient of the circular cylinder in a freestream flow at $Re = 500$.

From figure 5.19, it can be observed that the x-axis is the current lift coefficient of the circular cylinder and the y-axis is the fast-forward one-cycle value of the lift coefficient of the circular cylinder. From

these labels, it could be observed that the Poincare Map represents a map of characteristic of the system and a map for predicting the future state of the system (the state of the system in this example is the lift coefficient of the circular cylinder). As such, it can be observed that the lift coefficient response of the circular cylinder is unsteady but predictable as it followed a certain trajectory. Such trajectory acted as indicator of the trajectory of the future response of the lift coefficient of the circular cylinder. It shall be visible in chapter 6 that the lift coefficient of the circular cylinder is not a chaotic system.

Chapter 6

VIV Reducing Properties of Harbour Seal Vibrissae Geometry

As described in chapter 2, one of the most unique features of the underwater sensing ability of the harbour seals is the geometry of their vibrissae (refer to figure 2.1). From experiments in chapter 4, it is found that the geometry of the harbour seal vibrissae reduces VIV. From previous studies, it is also found that the geometry of the harbour seal vibrissae contains many features, as follows [40]:

- * Undulations in minor axis
- * Undulations in major axis
- * Offset angle between the waves in the leading edge and trailing edge
- * Spanwise variation of the offset angle

Hence, the geometrical effects of each feature on VIV reduction is unclear. It is the objective of this chapter to explore the effects of each feature on the force and motion response of the corresponding structures. In order to achieve this objective, simulations on various vibrissa-like geometries and vibrissa geometry will be performed. With reference to the objective of this chapter, the following vibrissa-like geometries are created:

- * Vibrissa-like geometry with no major axis undulations [Axis Ratio = $D/L = 0.44$]
- * Vibrissa-like geometry with no major axis undulations [Axis Ratio = $D/L = 0.54$]
- * Vibrissa-like geometry with no minor axis undulations

- * Vibrissa-like geometry with no offset angle
- * Vibrissa-like geometry with constant angle of 17.6° offset angle
- * Vibrissa-like geometry with constant angle of 15.27° offset angle

It should be noted that the features on the vibrissa-like geometry above are based on the old vibrissa model. This is due to the initial error in publication and assessment [40]. The error is only found two months before the completion of this thesis due to the numerous experience of Professor Triantafyllou in VIV. From Floquet stability analysis, it is found that the stable ratio of average diameter to the wavelength of the vibrissa is ≈ 5 to 6 while the reported ratio of average diameter to wavelength of vibrissa is ≈ 2.5 [5]. Figure 2.3 illustrates the measurement legend definitions of the seal vibrissae from previous publication [40]. The parameters of the seal vibrissae are as follows: M (half-period of the undulations) = 0.91mm. a and b (radii of the major and minor axis of the vibrissa model in maximum cross sectional area, respectively) = 0.595 mm and 0.24 mm. k and l (the radii of the major and minor axis of the vibrissa model in the throat section, respectively) = 0.475 mm and 0.29 mm. Offset angle between the leading edge and trailing edge in the major axis of the vibrissa model at maximum cross sectional area (α) and minimum cross sectional area (β) are 15.27° and 17.60° , respectively. Therefore, from proper measurement, it is found that the correct ratio of average diameter to wavelength is approximately 5.5, which lies in the range of stable ratio of average diameter to wavelength based on Floquet Stability analysis. This also implied that the offset angle between the leading edge and trailing edge in the major axis of the vibrissa is approximately half of the old vibrissa geometry, namely, 8.73° and 8° at the minimum and maximum cross sectional area, respectively. Subsequent analysis shows that the force and motion response of the new vibrissa geometry is better in reducing VIV and therefore, as mentioned earlier, do not discredit the findings with the old vibrissa model for the purpose of understanding the underwater sensing principle of harbour seal vibrissae. Notably, comparison between the response of the old vibrissa model and the new vibrissa model will be described in each subsection as validation. Simulations on the old vibrissa model and the new vibrissa model will also be performed. For reference comparison, simulations on circular cylinder and elliptical cylinders will be conducted. Figure 6.1 shows the reference geometries, vibrissa models, and vibrissa-like geometries to be simulated and studied for evaluation of the geometrical effect on the VIV reduction properties of each structure. Table 6.1 depicts the detail

Figure 6.1: Simulated geometries for studies on the effects of geometrical features on the VIV reduction properties of vibrissa-like geometries. Black-line: the profile of the vibrissa-like geometry in major axis (straight or undulatory), Red-line: the profile of the vibrissa like geometry in minor axis (straight or undulatory). For reference comparison, (I) circular cylinder and (II) elliptical cylinder are simulated. Vibrissa-like geometries with all-but-one feature of the harbour seal vibrissa: (III) vibrissa-like geometry with no major axis undulations (Axis Ratio = $D_{min}/D_{maj} = 0.44$, where D_{min} and D_{maj} are average minor and major diameters of the vibrissa, respectively), (IV) vibrissa-like geometry with no major axis undulations (Axis Ratio = 0.54), (V) vibrissa-like geometry with no minor axis undulations, (VI) vibrissa-like geometry with no offset angle, (VII) vibrissa-like geometry with constant offset angle of 15.27° , (VIII) vibrissa-like geometry with constant offset angle of 17.6° . True vibrissa geometry: (IX) Old vibrissa geometry, (X) new vibrissa geometry.

dimensions of all the simulated geometries. It is important to note that all the dimensions in table 6.1 will be scaled to a characteristic diameter of 1.0 (in order to maintain uniformity of the dimensionless parameter, e.g. Strouhal number, Reynolds Number and etc).

This chapter will be divided into three main sections, notably, simulations of flow around fixed structures, simulations of flow around freely vibrating in crossflow structures, and simulations of flow around freely vibrating in streamwise and crossflow structures. As described in chapter 5, all the simulations are performed at $Re = 500$ (unless otherwise mentioned). For freely vibrating simulations, the the mass ratio of the structure is 1.04 (in streamwise and crossflow directions) and the normalised natural frequency of the system is set at the vortex shedding frequency of the corresponding structure in stationary simulations. Periodic boundary conditions are applied on the spanwise ends of the geometries with inlet and outlet condition on the upstream and downstream limit-planes of the structures. Conservation of mass flux is applied on both limit-planes in the crossflow direction

Geometry No.	a	b	k	l	M	α	β
I	0.5	0.5	0.5	0.5	0	0°	0°
II	1.0	0.5	1.0	0.5	0	0°	0°
III	0.595	0.24	0.595	0.29	0.91	N.A	N.A
IV	0.475	0.24	0.475	0.29	0.91	N.A	N.A
V	0.595	0.27	0.475	0.27	0.91	0°	0°
VI	0.595	0.24	0.475	0.29	0.91	0°	0°
VII	0.595	0.24	0.475	0.29	0.91	15.27°	15.27°
VIII	0.595	0.24	0.475	0.29	0.91	17.6°	17.6°
IX	0.595	0.24	0.475	0.29	0.91	15.27°	17.6°
X	0.595	0.24	0.475	0.29	1.82	15.27°	17.6°

Table 6.1: Detailed dimensions of simulated geometries with variable definitions depicted in figure 2.3. All characteristic dimensions will be normalised to 1.0 for numerical simulations with a minimum of 20 grid points across this characteristic diameter.

of the structure. Subsequent use of the terms '1D vibration' and '2D vibration' shall represent 'freely vibrating in transverse direction' and 'freely vibrating in streamwise and crossflow directions' setup of the simulated structures, respectively.

6.1 Simulations of Stationary Geometry

In this section, evaluations on the effects of geometrical features on the force response of the structure will be conducted. The simulated geometries can be found above (refer to figure 6.1). All the simulations are performed with the experimental setup described in the introductory remarks of the chapter. From the simulations results, the essential force coefficients for each structure are tabulated and plotted, such as the mean drag coefficient, the RMS lift coefficient of the structure, and the normalised standard deviation (to RMS value) of the lift coefficient with respect to the RMS lift coefficient. Figure 6.2 shows the mean drag coefficient, RMS lift coefficient and normalised standard deviation (ϕ_{std}) for each of the geometry illustrated in figure 2.3.

For figure 6.2, the numbering of the geometry is the same as the Roman numeral-equivalent for the geometries in figure ???. As such, from figure 6.2, it can be observed that the $\bar{C}_D$ are comparable for geometries with elliptical cross-section (geometries 2 to 10). The $\bar{C}_D$ of the circular cylinder is found to be significantly higher than the $\bar{C}_D$ of the other elliptical geometries. This is mainly due to the greater pressure gradient between the leading edge and trailing edge of the circular cylinder as compared to the corresponding pressure gradient for elliptical based cross-sectional geometries. Figure 6.3 depicts the top view of the pressure coefficient profiles for circular cylinder and elliptical

Figure 6.2: Quantitative measures of the force coefficients to their corresponding geometries for simulations of flow around stationary geometries at $Re = 500$ (for legend on geometries, refer to figure ?? where the number for each geometry correspond to its Roman numerals-equivalent in figure ??). (Top Left) Mean Drag Coefficient ($\bar{C}_D$) of the simulated geometries, (Top Right) Normalised standard deviation (relative to RMS value, not mean, of the lift coefficient) to RMS value (ϕ_{rms}) for the simulated geometries, (Bottom Left) RMS Lift Coefficient ($C_{L,RMS}$) for simulated geometries, (Bottom Right) Zoomed-in plot on the $C_{L,RMS}$ for geometry 6 to 10.

cylinder during their steady period of response.

The pressure coefficient is mathematically defined as:

$$C_p = \frac{P}{\frac{1}{2}\rho U^2} \quad (6.1)$$

From figure 6.3, it can be observed that the adverse pressure gradient between the leading edge and trailing edge of the circular cylinder is ≈ 1.4 , which is significantly higher than the corresponding value on elliptical cylinder ≈ 0.425 . As such, the lower $\bar{C}_D$ of elliptical-based cross section geometry is expected. From figure 6.3, it can also be observed that the streamwise velocity profile exhibited a similar conclusion with greater difference in streamwise velocity between the leading and trailing

Figure 6.3: Top view of the pressure coefficient (refer to equation 6.1) and streamwise velocity profiles of a stationary circular cylinder (Top) and elliptical cylinder (Bottom) in a freestream flow at $Re = 500$ during their steady period of response. Left (top and bottom): Pressure profile ; Right (top and bottom) : Streamwise velocity profile. Note: 'p' in the legend indicates pressure coefficient and 'u' indicates the normalised streamwise flow velocity (freestream flow velocity, $U = 1$).

edges of the circular cylinder as compared to the corresponding value in the elliptical cylinder. It should be noted that the near-field wake of the circular cylinder is more concentrated with visibly closer vortex shedding occurring near the body (vortex shedding occurred at $0.5D$ from the trailing edge of the circular cylinder). However, the vortex shedding on an elliptical cylinder is found to occur at a later position of $2D_{min}$ (or $1D_{maj}$) from the trailing edge of the elliptical cylinder, where D_{min} and D_{maj} represent the minor and major diameters of the elliptical cylinder. Such difference in the vortex formation length further accounted for the lower pressure gradient as well as reduction in the lift coefficient of the elliptical cylinder as compared to the circular cylinder, as shown in Figure 6.3. From figure 6.3, it can also be observed that the axis ratio of the vibrissa-like geometries with no major axis undulations is found to result in minute difference in drag coefficient. Such difference in

axis ratio of 0.1 (0.44 for Geometry 3 and 0.54 for Geometry 4) would have rendered Geometry 3 to be the more streamlined body. However, significant difference in $C_{L,RMS}$ and ϕ_{std} is observed between the lift coefficient of geometry 3 and geometry 4. In contrast, from figure 6.2, it can be observed that the drag coefficient of the vibrissa-like geometry with no minor axis undulations is relatively higher than the other elliptical-based geometries. Figures 6.4 and 6.5 show the pressure profile of the flow around geometry 5 (vibrissa-like geometry with no minor axis undulations) and geometry 4 (vibrissa-like geometry with no major axis undulations with axis ratio 0.44), respectively.

As it is noted earlier, offset angle between the leading edge and trailing edge in the major axis existed on the seal vibrissae. Similarly, geometry 5 (vibrissa-like geometry with no minor axis undulations) consisted of such offset angle. Hence, the planes of analysis in figure 6.4 is divided into four planes. These planes consist of two planes in the nodal section, widest cross section, (one plane at the peak of the leading edge and one plane at the peak of the trailing edge) and two planes in the saddle section, narrowest cross section (one plane at the trough of the leading edge and one plane at the trough of the trailing edge). Unlike geometry 5, geometry 4 (vibrissa-like geometry with no major axis undulations) did not have any offset angle in its minor and major axis. As such, only two planes of pressure profile are necessary for representation of the pressure profiles as shown in Figure 6.5. From figure 6.4, it can be observed that the low pressure region is longer in the nodal planes as compared to the saddle planes. The width of the low pressure region is found to be similar near the centroid of the geometry. However, significant tapering of the low pressure region is observed in the nodal planes with a nearly uniform width observed in the saddle planes of the geometry. The shape of the low pressure region is found to be very similar on the nodal plane at the peak of the leading edge and the nodal plane at the peak of the trailing edge. Such similarity is also observed in the saddle plane. It is important to note that the thin lines in the streamwise direction shown in the pressure plots (figures 6.4, 6.5) is due to the imperfection in the visualisation and not in the simulation results. Comparing figure 6.4 and figure 6.5, it can be observed that the low pressure region of geometry 4 is significantly shorter as compared to the low pressure region of geometry 5. The width of such low pressure region is found to be similar to the corresponding region in geometry 5. However, no observable difference is perceived in between the near-field low pressure region of the nodal and saddle planes of geometry 4. Interestingly, the pressure in the low-pressure region of the nodal plane is found to be slightly higher than the corresponding pressure in the saddle plane. This indicated a

Figure 6.4: Top view of the pressure coefficient (refer to equation 6.1) profiles of a vibrissa-like geometry with no minor axis undulations in a freestream flow at $Re = 500$ during their steady period of response. As there are offset angle between the leading edge and trailing edge of the major axis, four planes are presented. Top left : plane of widest cross section (node) at the peak of the leading edge ; Top right : plane of widest cross section (node) at the peak of the trailing edge ;Bottom left : plane of narrowest cross section (saddle) at the trough of the leading edge ;Bottom right : plane of cross section (node) at the trough of the leading edge. Note: 'p' in the legend indicates pressure coefficient.

higher flow velocity in the saddle plane as compared to the nodal plane. Hence, such occurrence is found to concur to the minimum energy principle where the fluid particles tended towards the path of lesser obstacle (smaller cross section). From figure 6.5, it is also observed that a single coherent structure (approximately by regions of very low pressure - dark blue) is shed in the nodal plane while at the same instance, two coherent structures are shed in the saddle plane. This implied the occurrence of vortex splitting between the nodal planes and the saddle planes due to the necessary connection between the coherent structure in the nodal plane and saddle planes. Similar observation is attained

Figure 6.5: Top view of the pressure coefficient (refer to equation 6.1) profiles of a vibrissa-like geometry with no major axis undulations and axis ratio of 0.54 in a freestream flow at $Re = 500$ during their steady period of response. Left : Nodal plane (widest cross section) ; Right : Saddle plane (narrowest cross section). Note: 'p' in the legend indicates pressure coefficient.

for geometry 5 (refer to figure 6.4), with a significantly lower pressure in coherent structure of the nodal plane as compared to the saddle planes. Due to the larger low pressure region in the wake of geometry 5 as compared to geometry 4, the aggregate effect of such adverse pressure gradient is shown to result in a larger drag coefficient for geometry 5 as compared to geometry 4. Figure 6.6 illustrates the spanwise view of the pressure coefficient profile of stationary geometry 5 and geometry 4 in a freestream flow at $Re = 500$.

From figure 6.6, it can be concluded that the low pressure region of geometry 5 ($\approx 2D_{maj}$) is significantly longer than the low pressure region of geometry 4 ($\approx 1.5D_{maj}$). A greater spanwise pressure periodicity is also observed for geometry 4 as compared to geometry 5 with a lower pressure in the near-field of geometry 4 as compared to geometry 5. A further cue on the larger drag coefficient of geometry 5 as compared to geometry 4 could be found in the subsequent propagation of the pressure flow beyond the near-field wake. From figure 6.6, it can be observed that the low pressure front of geometry 4 is regularly propagated into the far field wake as compared to the corresponding propagation of geometry 5. Although the propagation is visible for geometry 5, it is observed that the pressure in the far field wake decayed significantly as compared to the near-field wake. This indicated that a significant amount of the kinetic energy of the flow has been consumed in the creation of the low pressure region in the near-field wake of geometry 4 as compared to geometry 5. From figure

Figure 6.6: Spanwise view of the pressure coefficient (refer to equation 6.1) profiles of a geometry 5 (left) and geometry 4 (right), refer to figure 6.1, in the plane of symmetry (through the centroid of the geometries) for a freestream flow at $Re = 500$ during their steady period of response. Note: 'p' in the legend indicates pressure coefficient.

6.2, it can also be observed that the drag coefficient of the new vibrissa geometry (geometry 10) is considerably lower than the elliptical-based geometries. Hence, further analysis is necessary in order to reveal the key to a further reduction in drag coefficient (smaller than elliptical cylinder). Figure 6.7 depicts the pressure coefficient profiles of the flow around the new vibrissa geometry during its steady period of response at $Re = 500$.

From figure 6.7, it can be observed that the low pressure region in the wake of the vibrissa is periodic in the spanwise direction. Such regularity is found to persist to at least $17D_{min}$ downstream from the trailing edge of the new vibrissa geometry. It can also be observed that the minimum pressure wake topology is significantly different between the nodal and saddle plane of the vibrissa. The low pressure region in the nodal plane is found to be very widely spread into a near circular manner. No shedding of coherent structure is apparent in the nodal plane of the vibrissa. However, two shedding of coherent structures are observed in the saddle plane of the vibrissa. It is also observed that the shedding of these coherent structure occurs almost simultaneously at $\approx 2D_{maj}$ from the trailing edge of the vibrissa. The adverse pressure gradient between the leading edge and the trailing edge of the vibrissa is found to be significantly smaller than the pressure gradient of geometry 4 and geometry 5. The weakening of the adverse pressure gradient is found to be the consequence of significant spanwise flow velocity as the flow is redirected as it passed through the vibrissa. Figure

Figure 6.7: in the plane of symmetry (through the centroid of the geometries) for a freestream flow at $Re = 500$ during their steady period of response. Note: 'p' in the legend indicates pressure coefficient.

6.8 depicts the comparison between the spanwise flow velocities of the vibrissa and elliptical cylinder.

The normalised spanwise flow velocity (w) is mathematically defined as follows:

$$w = \frac{w'}{U} \quad (6.2)$$

where w' is the nominal spanwise flow velocity. From comparison of the spanwise flow velocity of the flow around the elliptical cylinder and the new vibrissa geometry in figure 6.8, it can be observed that the spanwise flow velocity of the vibrissa geometry is $\approx 30x$ of the spanwise flow velocity of the elliptical cylinder. This depicts the significance of the spanwise flow velocity in the wake of the new vibrissa geometry which significantly reduced the adverse pressure gradient and therefore, the drag coefficient. In order to better visualise the evolution of the spanwise flow velocity in the wake of the vibrissa, figure 6.9 shows the spatial evolution of the spanwise flow velocity in the wake of the new vibrissa geometry.

The first (top left-most) plot in figure 6.9 is significantly de-colourized for better visualisation of the colour coding in the plot due to the subtlety of the spanwise flow velocity about the centroid of the new vibrissa geometry. The colour of the rest of the plots corresponded to the spanwise flow velocity quantified by the legend in the bottom right-most plot. For figure 6.9, as the plane of pressure representation is moved away from the body, a light translucency is maintained for referencing to the geometry of the vibrissa. A significant spanwise flow velocity is observed from $4.15D_{min}$, which is sufficiently far from the trailing edge of the structure. This occurrence is of great significance towards reduction in the drag coefficient of the new vibrissa geometry. This is because as the spanwise flow velocity dissipates due to geometrical effect, pressure build-up will occur. Occurrence of such pressure build-up near the trailing edge of the body would have resulted in significantly higher drag coefficient. However, through spatial delay in the strengthening of the spanwise flow velocity, the location of pressure build-up is spatially delayed downstream. Thus, no significant drag coefficient is observed. In essence, the flow is found to move through the new vibrissa geometry as if the new vibrissa geometry is more streamlined (similar to an airfoil) than its actual geometry. Such occurrence should not be of great significance in the wake of the old vibrissa model as the drag coefficient is found to be considerably higher than the new vibrissa geometry. Figure 6.10 illustrates the spanwise flow velocity profile of the old vibrissa model at $Re = 500$.

From figure 6.10, it can be observed that the spanwise flow velocity of the old vibrissa model is

Figure 6.8: Isometric view of the spanwise flow velocity (w) of the flow around a stationary elliptical cylinder (Top) and new vibrissa geometry (Bottom) at $Re = 500$. The plane perpendicular to y -axis is located about the centroid of the geometries and the plane perpendicular to the x -axis is chosen to reflect the location of the largest spanwise flow velocity. Note: ' w ' in the legend indicates normalised spanwise velocity, defined in equation 6.2.

Figure 6.9: Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.0 (at trailing edge of vibrissa), 2.4, 3.45, 4.15, 4.85, 5.9 ; bottom-ow from left to bottom right : $x/D = 6.25, 6.6, 7.3, 8.35, 9.05, 10.1$, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).

relatively lower and more irregular as compared to the spanwise flow velocity of the new vibrissa geometry. Significant variation of the spanwise flow velocity can be observed along the spanwise and streamwise directions of the flow field. The variation is observed to persist up to, at least, $10.6D_{min}$. Figure 6.11 illustrates the spatial evolution of the spanwise flow velocity in the wake of the old vibrissa geometry during its steady period at $Re = 500$.

Figure 6.10: Isometric view of the spanwise flow velocity (w) of the flow around a stationary old vibrissa geometry at $Re = 500$. The plane perpendicular to y -axis is located about the centroid of the geometries and the plane perpendicular to the x -axis is chosen to reflect the location of the largest spanwise flow velocity. Note: ' w ' in the legend indicates normalised spanwise velocity, defined in equation 6.2.

By comparing figure 6.9 and figure 6.11, it can be observed that the spanwise flow velocity of the new vibrissa model is significantly more regular as compared to that of the old vibrissa geometry. The spanwise flow velocity profile near the trailing edge of the new vibrissa geometry is found to be similar to the corresponding profile for old vibrissa geometry. However, the subsequent progression of the spanwise flow is found to diverge with a regular occurrence of flow towards the saddle from the neighbouring nodal points on the new vibrissa geometry and irregular occurrence of spanwise flow on the old vibrissa geometry. Such regularity in the spanwise flow of the new vibrissa geometry is found to be result in efficient dissipation of spanwise flow velocities (as indicated by the weakening spanwise flow velocity as the flow travelled away from the body in figure 6.9). Consequently, a large pressure build-up occurred in the region of spanwise velocity dissipation, which is located in the wake of the new vibrissa geometry. Hence, a smaller adverse pressure gradient is resulted for the new vibrissa geometry as compared to the old vibrissa geometry. Figure 6.12 shows the lowest-pressure planes (perpendicular to the freestream flow) in the wake of the correct and old vibrissa geometries for flow around the respective geometries at $Re = 500$.

Figure 6.11: Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.14 (near trailing edge of old vibrissa model), 1.52, 1.9, 3.04, 3.8 ; middle-row from left to right : $x/D = 4.56, 5.7, 6.46, 6.84, 7.22, 7.98$; bottom-row from left to right : 8.74, 9.12, 9.88, 10.64, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).

Figure 6.12: Lowest-pressure planes (perpendicular to the freestream flow) in the wake of new vibrissa geometry (left) and old vibrissa geometry (right) for flow around the respective geometries at $Re = 500$.

The lowest-pressure plane during the steady period of force response ($t^* = 216$) of the correct and old vibrissa geometries are found at $3.46D_{min}$ and $2.72D_{min}$, respectively. From figure 6.12, it can be observed that the lowest-pressure plane of the old vibrissa geometry consist of a large number of low pressure zones. In contrast, only small zones of low pressures are observed in the lowest-pressure plane of the new vibrissa geometry. Interestingly, in figure 6.12 the low pressure zones of the new vibrissa geometry is found to be coupled with a higher pressure zones (positive pressure coefficient). This conclusively showed that the drag coefficient of the new vibrissa geometry is smaller than the old vibrissa geometry (and therefore, other geometries in the current studies) due to the lower adverse pressure gradient on the body as it effectively dissipated the spanwise flow velocity.

Through observation of figure 6.2, it can be observed that the $C_{L,RMS}$ of the elliptical cylinder is significantly lower than the $C_{L,RMS}$ of the circular cylinder. In turn, geometry 3 (vibrissa-like geometry with no major axis undulations with axis ratio of 0.44) and geometry 5 (vibrissa-like geometry with no minor axis undulations) are observed to respond with considerably lower lift coefficient. Although strong similarity are found between geometry 3 and 4, the $C_{L,RMS}$ of geometry 4 is found to be reasonably higher than the $C_{L,RMS}$ of geometry 3. Interestingly, $C_{L,RMS}$ of geometry 6 to geometry 10 (vibrissa-like models with undulations in minor and major axes) are found to be significantly smaller than the low $C_{L,RMS}$ of geometry 3 and 5. An even further reduction of $C_{L,RMS}$ is observed for the case of the new vibrissa geometry. Hence, an extremely significant reduction of $C_{L,RMS}$ is observed through comparison of the lift force response of a circular cylinder and the new vibrissa geometry at $Re = 500$. For a greater understanding on the reductions of $C_{L,RMS}$ for these geometries, a step-by-step approach is implemented from the reduction of $C_{L,RMS}$ of elliptical cylinder as compared to the circular cylinder to the observed $C_{L,RMS}$ reduction of geometry 3 and 5 as compared to an elliptical cylinder and geometry 4. A next-step analysis on the $C_{L,RMS}$ reduction of geometry 6 (to 10) as compared to geometry 3 and 5 are performed with the final-step analysis performed on the reduction of $C_{L,RMS}$ of new vibrissa geometry as compared to other simulated geometries. Figure 6.13 depicts the vortex cores (based on λ_2 criterion) for freestream flow around a circular cylinder and an elliptical cylinder.

With reference to figure 6.13, it can be observed that the vortex shedding of a circular cylinder occurred at a closer proximity to the body as compared to the vortex shedding of an elliptical cylinder. At $Re = 500$, it is observed that the vortex shedding of a circular cylinder occurred at $0.5D$ (equivalent vortex formation length, $L_F/D = 2.0$) from the trailing edge of the circular cylinder (measured to the point of detachment of the first fully-formed vortex). In comparison, the vortex shedding of an elliptical cylinder is found to occur at $\approx 1D_{min}$ from the trailing edge of the elliptical cylinder at $Re = 500$. The proximity of the vortex shedding to the body is of great significance to the lift force on the body. This is because a spatially delayed shedding would imply a larger amount of fluid required between the forcing due to the vortex shedding on the body. Such larger body of fluid would result in a dampening effect on the forcing due to vortex shedding, which would result in lower reaction force on the body due to the forcing from vortex shedding. Such dampening of forcing due to vortex shedding can be observed in the pressure profiles in figure 6.12. The influence of the

Figure 6.13: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a circular cylinder and elliptical cylinder at $Re = 500$ during steady period of force response. Top and bottom left : Isometric and top views of vortex cores for flow around a circular cylinder, respectively ; Top and bottom right : Isometric and top views of vortex cores for flow around an elliptical cylinder.

pressure minimum region (represented by the first pressure minimum region in the wake of the body - coherent structure) on the elliptical cylinder is found to be significantly lowered through comparison between the pressure field on the body and the pressure field of the coherent structure. In contrast, the coherent structure (represented by first pressure minimum region) in the wake of a circular cylinder is found to have a profound influence on the body with some dampening effect. Additionally, the minimum pressures of the vortices in the circular cylinder are observed to be significantly lower than the corresponding pressures for an elliptical cylinder. Hence, a more significant forcing on the body due to vortex shedding would be prominent for a given distance between the body and the location of vortex shedding. Figures 6.3, 6.13 show the pressure field and the vortex cores plots of a circular

cylinder and elliptical cylinder right before the moment when the nearly-fully-formed-vortex are to be shed. It is important to note that the coherent structures in the wake of the circular cylinder in figure 6.13 consisted of vortex tubes and braids, which are observed in previous studies [104]. However, the wake of an elliptical cylinder is observed to consist of only vortex tubes with no braids. This is mainly due to the more streamlined profile of the elliptical cylinder as compared to circular cylinder, which deviated perturbation from occurring and propagating as braids.

A next-step comparison is performed between geometry 3 and geometry 4 as well as elliptical cylinder. From figure 6.2, it can be observed that the $C_{L,RMS}$ of geometry 4 is nearly-equivalent to the $C_{L,RMS}$ of an elliptical cylinder and $C_{L,RMS}$ of geometry 3 (and 5) is (are) significantly lower than the $C_{L,RMS}$ of geometry 4 and elliptical cylinder. Figure 6.14 shows the comparison between the topology of vortex cores in the wake of geometry 3 and geometry 4.

From figure 6.14, it is observed that the topology of the wakes of geometry 3 and 4 shared strong similarity. However, a closer observation revealed the regular occurrence of vortical braids in the wake of geometry 4 with a less regular occurrence of vortical braids in the wake of geometry 3. From the top views of the coherent structures in figure 6.14, similar spacing between vortex tubes and lateral width of the wakes are observed for geometry 3 and geometry 4. Hence, this indicated that the qualitative difference of the wake topology between geometry 3 and geometry 4 represented the key evidence for the explanation of the difference in the lift force response ($C_{L,RMS}$). Figure 6.15 illustrates the side view of the vortex cores in the wakes of geometry 3 and 4 during their steady period of response at $Re = 500$.

From figure 6.15, it is observed that the wake of geometry 3 consisted of less number of vortical braids as compared to the wake of geometry 4. This is found to be consistent with the aforementioned definition on the more streamlined body of geometry 3 as compared to geometry 4. From recollection of figure 6.13 on the comparison of the coherent structures in the wake of a circular cylinder and an elliptical cylinder, it can be observed that the wake of an elliptical cylinder is devoided from any vortical braids, which are present in the wake of a circular cylinder. Earlier in this section, it has been established that such occurrence of vortical braids is a relevant consequence of the perturbations resulting from a bluff body. Hence, as geometry 3 is comparatively more streamlined as compared to geometry 4, the lower number of vortical braids occurrence in the wake of geometry 3 is found to be consistent with the earlier justification (on comparison between flow around a circular cylinder

Figure 6.14: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a vibrissa-like geometry with no major axis undulations of axis ratio 0.44 (geometry 3) and a vibrissa-like geometry with no major axis undulations of axis ratio 0.54 (geometry 4) at $Re = 500$ during steady period of force response. Top and bottom left : Isometric and top views of vortex cores for flow around a geometry 3, respectively ; Top and bottom right : Isometric and top views of vortex cores for flow around a geometry 4.

and an elliptical cylinder). For a more conclusive approach, figure 6.16 depicts the pressure profiles of flow around geometry 3 and geometry 4 in the nodal and saddle planes at $Re = 500$ during their steady period of response.

From a quick observation of figure 6.16, it is found that the pressure profiles in the nodal and saddle planes of geometry 3 shared some similarity to the corresponding profile of geometry 4. However, a closer observation revealed a stronger minimum pressure region in the nodal and saddle planes of the wake of geometry 4 (as compared to the nodal and saddle planes in geometry 3). The spacing between the trailing edge of the geometries and the first minimum pressure region is found

Figure 6.15: Side view of the coherent structures (identification based on λ_2 criterion) of flow around geometry 3 (left) and 4 (right) at $Re = 500$ during their steady period of force response.

Figure 6.16: Pressure coefficient profiles of flow around a vibrissa-like geometry with no major axis undulation for axis ratio of 0.44 (geometry 3) at the nodal plane (left) and saddle plane (right) of the geometry for freestream flow at $Re = 500$. Top figures : geometry 3 ; bottom figures : geometry 4 - reprint from figure 6.5 for easier comparison.

to be equivalent ($\approx 1D_{min}$ in nodal planes and $\approx 2D_{min}$ in saddle planes). A slight disparity is also observed in the low pressure region. From figure 6.16, it can be observed that the low pressure region of geometry 4 maintained a uniform width from the Y-line across the centroid of the geometry to the Y-line across the first minimum pressure region. In comparison, the width of the low pressure region of geometry 3 is found to decrease from the Y-line across the centroid of the geometry to the Y-line across the first minimum pressure region. It is also observed that the pressure profiles in the nodal and saddle planes of geometry 3 shared stronger similarity (especially in the near-field wake of the body) as compared to the corresponding profiles for geometry 4. Hence, it has been shown that the lower $C_{L,RMS}$ of geometry 3 (as compared to that of geometry 4) is mainly contributed by the higher pressure in the first minimum-pressure region in the wake of the geometry. Other contributing factors are found in the streamwise profile of the low pressure region in the near-field wake of the respective geometries as well as the disparity between the pressure profiles in the nodal and saddle planes of the respective geometries.

From further observations of figure 6.2, it is observed that the lift coefficient ($C_{L,RMS}$) of the vibrissa-like geometry with dual-axial undulations are significantly lower than the $C_{L,RMS}$ of a circular cylinder, elliptical cylinder or vibrissa-like geometries with single-axial undulations. Further analysis is performed through comparison between the topology of the vortex cores in the wake of geometry 6 (vibrissa-like geometry with no offset angle) and geometry 8 (vibrissa-like geometry with constant offset angle of 17.6°), as shown in figure 6.17 (descriptions of flow around geometry 7 will be provided later in this section).

From figure 6.17, it can be observed that the topology of the wakes of geometry 6 and 8 shared strong similarity in their formations of vortical braids and tubes. Only minor differences are observed in the proximity of the top end of geometries from the side view of the vortex cores of geometry 6 and 8. Hence, with reference to the $C_{L,RMS}$ of these geometries in figure 6.2, it can be deduced that analysis on the force response of one of these geometries will be sufficient in the explanation of the significantly lower $C_{L,RMS}$ of the vibrissa-like geometry with dual-axial undulations (geometry 6 to 8). Figure 6.18 shows the pressure profiles of the flow around geometry 6 (vibrissa-like geometry with no offset angle) in the nodal and saddle planes.

By comparing figure 6.16 and figure 6.18, it can be observed that the pressure of the minimum pressure region of geometry 6 is significantly higher than the corresponding pressure for geometry

Figure 6.17: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a vibrissa-like geometry with no offset angle (geometry 6) and a vibrissa-like geometry with constant offset angle of 17.6° (geometry 8) at $Re = 500$ during steady period of force response. Top, middle and bottom left : Isometric, side and top views of vortex cores for flow around a geometry 6, respectively ; Top, middle and bottom right : Isometric, side and top views of vortex cores for flow around a geometry 8.

Figure 6.18: Pressure profiles of flow around geometry 6 (vibrissa-like geometry with no offset angle) at the nodal (left) and saddle planes (right) for $Re = 500$.

3 ($C_{p,min}$ (geometry 6) = -0.25, $C_{p,min}$ (geometry 3) = -0.35). The uniformity of pressure profile between the nodal and saddle planes of geometry 3 is not observed in geometry 6. For geometry 6 (refer to figure 6.18), it is observed that the region of lowest pressure is found in the saddle plane of the near-field wake and in nodal plane of the region downstream from the near-field wake. Correspondence of low pressure region between the nodal and saddle planes are observed for geometry 6, where the lowest pressure region in nodal and saddle plane is observed to be spatially correlated. However, the low pressure region in the near field wake is observed to be different for the nodal and saddle planes of geometry 6. The saddle plane is observed to consist of a wider but shorter low pressure region as compared the longer but narrower low pressure region in the nodal plane for the near-field wake of the body. Hence, it is found that the smaller $C_{L,RMS}$ is found to be the result of a combination of the higher minimum pressure and the irregular pressure distribution between the nodal and saddle planes of geometry 6. Although the first minimum pressure region occurred at equivalent distance in the saddle planes of geometry 6 (as compared to geometry 3), it is observed that such distance is doubled in the nodal planes of geometry 6. The higher pressure in the minimum pressure region is observed on the saddle plane of geometry 6. The higher pressure in the near-field low pressure region of geometry 6 as compared to the corresponding pressure for geometry 3 is believed to be the consequence of the greater perturbation of the spanwise flow velocity of geometry 6.

Through comparison of figure 6.19 and figure 6.20, it can be observed that the perturbation in the spanwise flow velocity of geometry 6 (figure 6.19) is more significant (in magnitude and number of occurrences) as compared to the corresponding velocity of geometry 3 (refer to figure 6.20). As the

Figure 6.19: Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.09 (near trailing edge of old vibrissa model), 1.36, 2.17, 2.99, 3.53, 4.07 ; middle-row from left to right : $x/D = 4.62, 4.89, 5.43, 5.97, 6.24, 6.52, 7.33$; bottom-row from left to right : 7.87, 8.42, 8.69, 9.23, 9.50, 10.32, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).

Figure 6.20: Rear view of the spanwise flow velocity of the flow around a stationary new vibrissa geometry at $Re = 500$ for various spatial positions at $t^* = 216$ (stable flow). Significant discolouration is implemented on the top left-most figure to better visualise the colour on the plot. All other plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.09 (near trailing edge of old vibrissa model), 1.36, 1.63, 1.90, 2.99, 3.80, 4.07 ; middle-row from left to right : $x/D = 4.34$, 4.62, 5.70, 5.97, 6.24, 6.79, 7.33 ; bottom-row from left to right : 7.87, 8.15, 8.42, 8.96, 9.50, 10.05, where D is the characteristic diameter in the simulation (minor diameter of vibrissa geometry).

spanwise flow velocity annihilates each other (through the meeting of positive and negative spanwise flow velocity), pressure build-up occurred in the wake of the respective bodies. Hence, as the wake of geometry 6 consisted of a larger number of spanwise perturbation at a greater magnitude, the resulting pressure build-up increased more of the overall pressure in the low pressure region in the wake of geometry 6.

From figure 6.2, despite the similarity of lift and drag coefficient of geometry 6, 7, 8, 9, it is very significant to assess (and compare) the wake topology and response of geometry 9 (old vibrissa geometry) and geometry 10 (new vibrissa geometry), in lieu of comparison between geometry 6 and geometry 10. This is due to the stronger correlation between geometry 9 and geometry 10, where the main differencing parameter is the wavelength of the geometry (3.6 and 6.8, respectively). As aforementioned, since the lift and drag coefficients of geometry 6, 7, 8 and 9 are relatively similar, comparison study between the force responses of geometry 9 and geometry 10 should be representative of the comparison between the force responses of geometry 6, 7, or 8 and geometry 10. Figure 6.21 shows the coherent structures in the wake of the old and new vibrissa geometries.

Through comparison between figure 6.21 and figure 6.22, it is found that spanwise periodicity existed in the wake of the new vibrissa geometry. However, a significantly less regular wake topology is observed for the old vibrissa. From the top views of the respective figures, despite the significant spanwise deformations, discernible vortex tubes could be observed. Interestingly, despite the similarity of their geometrical parameter, the difference in the topology of the wakes of the old and new vibrissa geometries are observed to be very significant. The spanwise regularity in the wake of the new vibrissa could be further elaborated through observations on the side view of its wakes. From figure 6.22, it can be observed that the bulk of vortical braids in the middle of the geometry is found to shed ahead of its neighbouring bulk of vortical braids. Additionally, this is found to be true for the bulk of vortical braids near both ends of the new vibrissa geometry, which are observed to lead in shedding along with the bulk of vortical braids in the middle of the geometry. In comparison, from figure 6.21, it can be observed from the side view of the wake of the old vibrissa that a significant number of criss-cross occurred in the first spanwise-oriented vortical braids from the near-field wake of the body. As the pressure profiles of the new vibrissa geometry are depicted in figure 6.7, further comparison of such profiles to the old vibrissa geometry is necessary for better description on the difference in lift force response of the respective geometries to the flow. Figure 6.23 shows the pressure

Figure 6.21: Depiction of vortex cores (identification based on λ_2 criterion) of flow around the old vibrissa geometry (geometry 9) at $Re = 500$ during steady period of force response. Top left : Isometric view, top right : side view, bottom : top view.

profiles in the nodal and saddle planes of the leading and trailing edges of the old vibrissa model.

By comparing figure 6.7 and figure 6.23, it can be observed that the pressure profiles of the new and old vibrissa geometries in the nodal and saddle planes are significantly different. While no minimum pressure region are observed in the nodal plane of the new vibrissa geometry, strong minimum pressure region is observed on the nodal plane at the peak of the leading edge of the old vibrissa geometry. Higher minimum pressure are observed at the downstream edge of the near-field low pressure region on the nodal plane at the peak of the trailing edge. In contrast, the lower pressure at minimum pressure region is observed in the saddle plane at the trough of the trailing edge instead of the trough of the leading edge. Longer near-field low pressure region is observed in the nodal plane as compared to the saddle planes with strong correlation observed between the location of the minimum pressure region in the nodal and saddle planes. Further comparison between the side view

Figure 6.22: Depiction of vortex cores (identification based on λ_2 criterion) of flow around the new vibrissa geometry (geometry 10) at $Re = 500$ during steady period of force response. Top left : Isometric view, top right : side view, bottom : top view.

of the pressure profiles about the plane of symmetry of each geometry (figure 6.7 and figure 6.23), shows a much stronger regularity in the wake of the new vibrissa geometry as compared to the old vibrissa. However, certain staggered arrangement is observed in the spanwise pressure profile of the wake of the old vibrissa. The lower $C_{L,RMS}$ of the new vibrissa geometry is observed to be due to the greater spacing between the trailing edge of the geometry to the first minimum pressure region in the wake of the geometry ($4D_{min}$) as compared to the corresponding spacing of the old vibrissa

Figure 6.23: Pressure profiles of flow around the old vibrissa geometry at $Re = 500$ during its steady period of oscillation. Top left : top view of nodal plane at the peak of leading edge; top right : top view of nodal plane at the peak of trailing edge ; middle left : top view of saddle plane at the trough of leading edge ; middle right : top view of saddle plane at the trough of trailing edge ; bottom : side view of the plane of symmetry about the centroid of the geometry.

geometry ($3D_{min}$). The higher pressure of the low minimum pressure region of the new vibrissa geometry (and hence, smaller pressure difference between the pressure around the body and the minimum pressure region) is found to contribute to the lowering of the $C_{L,RMS}$ of the new vibrissa geometry. The phase difference between the vortex shedding at one period of undulations to another of the new vibrissa geometry is observed to reduce the $C_{L,RMS}$ of the new vibrissa geometry. Such phase difference can be observed from figures 6.7, 6.22. From figure 6.7, such phase difference is observed from the occurrence of minimum pressure region in the saddle plane of the geometry and the absence of minimum pressure region in the nodal plane. This implied the occurrence of vortex shedding in the saddle plane with no occurrence of vortex shedding in the nodal plane at the given time. Such phase difference can also be observed from figure 6.22. In the side view of the vortex cores of figure 6.22, the bulk of vortices in the middle span of the new vibrissa geometry is observed to lead the shedding as compared to its neighbouring vortices. Similarly, the bulk of vortices at the top end and bottom end of the simulation domain are observed to lead the shedding as compared to its lower and upper neighbours, respectively. Hence, it can be deduced that the vortex shedding of the new vibrissa geometry occurred in a desynchronized pattern along the spanwise direction. Figure 6.24 shows the crossflow velocity component of flow around the old and new vibrissa geometries during their steady period of oscillation.

From figure 6.24, the regularity aspects of the crossflow velocity of the old and new vibrissa geometries can be observed. Despite the significant spanwise variation, the crossflow velocity profile of the old vibrissa geometry is observed to occur almost simultaneously in the spanwise direction. A close observation on the corresponding plot for the new vibrissa geometry revealed a slight phase difference along the spanwise of the geometry. Strong correlation is found to persist for the crossflow velocities along the spanwise direction of the old vibrissa geometry. For the new vibrissa geometry, only very weak-correlation is observed to persist between the crossflow velocities along the spanwise direction of the new vibrissa geometry. A strong correlation between the crossflow velocity along the spanwise direction represented a strong indication of the synchrony of the forcing on the body due to vortex shedding. The weakly-correlated crossflow velocity of the new vibrissa geometry has served to further reinforce the earlier findings of the desynchronisation of the vortex shedding between each undulatory period of the new vibrissa geometry.

Figure 6.24: Isometric view of the iso-surface of the crossflow velocity of the old (left) and new (right) vibrissa geometries during their steady period of oscillation for flow at $Re = 500$.

6.1.1 Instability of the coherent structures

Secondary instabilities in the wake of a stationary circular cylinder has undergone profound investigations [104]. The instability of vortical tubes in the wakes of the circular cylinder is found to result in undulatory geometry in the wake of the circular cylinder. Subsequently, these undulations are observed to develop into vortical braids with certain spanwise arrangements along the length of the circular cylinder. The undulations of the vortical tubes of the circular cylinder and the vortical braids connecting these vortical tubes could be observed in figure 6.13. Figure 6.25 revealed the cross-sectional cut of a plane perpendicular to the streamwise direction on the vortical braids and their corresponding tangential velocity vectors.

From figure 6.25, the plane-cut on the midst of the vortical braids is performed in order to identify the rotational direction of each vortical braids. Hence, from figure 6.25, it can be observed that the vortical braids always occurred in pairs (clockwise and counter-clockwise rotational pairs). The center-to-center spacing of the vortical braids is found to vary between $0.625D$ to $1.875D$, where D is the characteristic diameter of the circular cylinder. This is found to be similar to previous observations on the wavelength of the secondary instability (vortical braids) [5, 104]. A sparser spacing is

Figure 6.25: Direction of rotation of vortical braids based on tangential velocity vector on the plane-cut perpendicular to the streamwise (x -axis) in the wake of a circular cylinder. Grey areas in top right and bottom figures indicated the vortex cores based on identification with λ_2 criterion and arrows indicated the tangential velocity vectors. The planes are lightly transparent, as such, the light grey region in the background of top right and bottom figures represented the circular cylinder (rear-view). Top left : Isometric view of coherent structures in the wake of the circular cylinder, the vertical line indicates the location of plane-cut (perpendicular to x -axis) and the horizontal lines acted as the separator of the the spanwise zone (top, middle and left) ; top right : vortical braids and tangential velocity vector in the 'top' zone of the plane-cut; bottom left : vortical braids and tangential velocity vector in the 'middle' zone of the plane-cut ; bottom right : vortical braids and tangential velocity vector in the 'bottom' zone of the plane-cut.

observed near the top ends of the circular cylinder with significantly smaller spacing observed in the middle span of the circular cylinder. The physical significance of the vortical braids is correlated to their weakening of the vorticity in the vortex tubes. Such weakening of vortex tubes resulted in lowering of the lift coefficients on the circular cylinder. As the vortical braids occurred at large numbers, their importance in reducing the lift coefficients (through weakening of the vorticity in the vortex tubes) is evident. In contrast, the wake of the elliptical cylinder is observed not to consist of vortical braids due to the stable nature of the vortical tubes (as the elliptical cylinder is more streamlined which resulted in higher minimum pressure regions). However, as regularity of the wake patterns are observable in the wake of the new vibrissa geometry, further evaluations are necessary in order to assess the profiles of the vortical braids in the wakes of the new vibrissa geometry. Unlike in the case of circular cylinder, the instability (undulations in the wake of the circular cylinder) of the new vibrissa geometry are imposed through its geometrical construct. As such, the evolution and consequence of such physical imposition are of great interest. Figure 6.26 illustrates the plane-cut (perpendicular to x-axis) in the middle of the vortical braids and their corresponding tangential velocity vectors. From figure 6.26, regularity and phase difference between the vortex shedding along the spanwise directions of the new vibrissa geometry can be further observed and confirmed. The spanwise spectra of the vorticity in the plane in Figure 6.27 revealed the occurrence of multiple wavelengths. Notably, normalised wavelengths (λ_{span}/D) observed in the plane are 1.0, 1.25, 1.85, 2.06, and 3.29 (which corresponds to $\lambda_{span}/\lambda = 0.14, 0.17, 0.26, 0.29, 0.46$). The last two wavelengths are observed to represent the dominant wavelengths of the vortical braids. It is interesting to note that no wavelength is found to correspond exactly with the wavelengths of the undulations on the new vibrissa geometry. From rear-view observations of the vortical structures in the wake of the new vibrissa geometry, clearer views of the form of the vortex cores can be observed, notably from the first complete bulk of the vortical braids near the top end of the new vibrissa geometry. Three major vortical structures could be clearly identified. It can also be observed that the bulks of vortical braids are found to always be centred about the saddle plane of the major axis and the nodal plane of the minor axis. The largest vortex core is found to occur in the midst of the bulk of the vortical braids. By performing a cross-sectional cut through the vortical braids (plane perpendicular to streamwise direction), details of the number of vortical structures and their corresponding sizes could be determined. From observation on the right-most of figure 6.26, the periodicity, symmetry and consistency

Figure 6.26: Illustration of the vortex cores (based on λ_2 criterion) of the new vibrissa geometry in a freestream flow at $Re = 500$ during its steady period of response ($t^* = 219.1$). Left: rear-view vortex cores ; Middle : Isometric view of the vortex cores and the location of plane-cut (perpendicular to streamwise) ; Right : view of the contours in the plane cut (orange region - vortex cores).

of the vortex cores can be observed along the spanwise direction (from the top to bottom beginning with first complete bulk of vortical braids : left-oriented bulk of vortical braids, right-oriented bulk of vortical braids and right-oriented bulk of vortical braids). Figure 6.27 illustrates the vortex cores and tangential velocity vector in a single bulk of vortical braids in the plane-cut shown in figure 6.26.

In order to identify the polarity of the vortical braids (positive polarity - counter clockwise rotation and negative polarity - clockwise rotation), the tangential velocity vectors are plotted in the field of analysis. Contour plots in figure 6.27 show the vortex cores and tangential velocity vector in the plane-cut for a single bulk of vortical braids. From these enlarged and focused views, polarity of each vortical braids could be determined from their respective tangential velocity vectors. As reference comparison, the vortical braids in the wake of circular cylinder occurred in pairs with each pair consisting of vortical braids with opposite polarity (refer to figure 6.25). Figure 6.28 illustrated the deduced polarity of each of the vortical braids depicted in figure 6.27. It should be noted that further

Figure 6.27: Enlarged view of the first complete bulk of vortical braids in figure 6.26 from the plane-cut (perpendicular to streamwise) through the midst of the vortical braids. The planes are lightly transparent with the new vibrissa geometry lightly visible in the background (located in the middle of each figure), rear-view. Legend: orange area - vortex cores, black arrows - tangential velocity vectors. Top left : plot of a single bulk of vortical braids in the plane-cut ; top right : plot of vortex cores and tangential velocity vector in the central region of bulk of the vortical braids ; bottom left : top half plot of the vortex cores and tangential velocity vector in the top half of the bulk of the vortical braids ; bottom right : bottom half plot of the vortex cores and tangential velocity vector in the top half of the bulk of the vortical braids.

analysis on the other bulk of vortical braids revealed similar arrangement of vortical braids and their corresponding polarities. With reference to figure 6.28, it can be observed that the small orange region near the top end of the plane is not assigned with any polarity. This is due to the minimum circulation observed through the tangential velocity vector (indicated in figure 6.27). Such occurrence implied the possibility of the existence of a weak vortical braids. Furthermore, the two vortical braids at the bottom of figure 6.28 are observed to consist of a single polarity despite their spatial distance.

Figure 6.28: Sketch of the polarities of the vortical braids observed in a single period of undulation of the new vibrissa geometry. Legend : orange area - vortex cores.

Such deduction is achieved through careful analysis of the tangential velocity vector in figure 6.27, which indicated a strong correlation between the two vortical braids. Beyond these two anomalies, the arrangement and polarity of the vortical braids fulfil the criteria of opposite polarity pairing of the vortical braids. Such opposite polarity pairing is observed from the top-most clockwise designated small vortical braids followed by counterclockwise large vortical braids and the subsequent opposite polarised vortical braids. It should also be noted that the arrangement and polarity of the vortical braids are found to be symmetrical about the horizontal line spanning through the middle of figure 6.28.

Through comparison between the wake topology of vibrissa-like geometries with dual-axial undulations (geometry 6 to 9, refer to figures 6.13, 6.14, 6.15, 6.17, 6.21, 6.22), the wake topology of the geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) is observed to preserve the vortical tubes and braids topology unlike other vibrissa-like geometries with dual-axial undulations. Such preservation of vortical tubes and braids topology is also observed for flow around a circular cylinder, elliptical cylinder and vibrissa-like geometry with single-axial undulations (refer to figures 6.15, 6.17). Figures 6.29, 6.30 show the wake topology and tangential velocity vector of

flow around geometry 7 at $t^* = 221$ and 219, respectively. From figure 6.29, it can be observed that certain periodicity and regularities are preserved in the wake of the geometry 7. The prevalent normalised wavelengths of the vortical braids in the plane of the wake at $x/D = 6.67$ are 1.74, 2.17, and 4.35 (corresponding to $\lambda_{span}/\lambda = 0.48, 0.60$, and 1.20). Interestingly, the smallest normalised wavelength is found to be most dominant in this plane of the wake. However, such regularity of the vortical braids and tubes are only visible in the wake of geometry 7 (not in the wake of geometry 6, 8 or 9). Such occurrence may indicate a perturbation threshold imposed through the physical presence of the geometry of the body on the flow around the body. For geometry 7, the constant offset angle of 15.26° is found to result in strong perturbation, which allowed the flow around the body to return to the periodic regularity with certain similarity to the wake of the circular cylinder. However, for geometry 8 and 9, the larger offset angle (17.6°) and variation of offset angle, respectively, are observed to result in significant perturbation resulting in non-recoverable irregularity in the spanwise arrangement of the wakes of these structures. In the case of geometry 6, the absence of offset angle is found to result in the perfect misalignment between the nodal plane of the major axis and the nodal plane of the minor axis. Such significant misalignment is found to exacerbate the perturbation resulting in irregularity in the wake of the body. Given the regularity of the vortex cores in the wake of geometry 7, further analysis on the polarities of each of the vortical braids (cores) are performed. Figure 6.30 shows the coherent structures and the cross-sectional cut (plane perpendicular to streamwise direction) through the vortical braids in the wakes at $x/D = 4.9$ (measured from the centroid of the geometry) at the steady period of force response, $t^* = 221$.

Interestingly, from figure 6.29, a spanwise repeatable vortical patterns are observed (as indicated by the existence of dominant wavelengths). Two-pairs of vortical braids are observable in a period of oscillation of the geometry (corresponding to a period of oscillation in the wake on the spanwise direction). Such pairs indicated the existence of two vortical braids of each polarity which can be clearly observed in the middle of the bottom left figure in figure 6.29. In order for such arrangement to be stable, it is observed that the vortical braids of similar polarity are arranged diagonally in a square arrangement of the four vortical braids. In spanwise direction, such two pair vortical braids are followed by a large vortical braid or a pair of vortical braids (of opposite polarity to each other). In order to establish the evolutionary aspect of these vortical arrangement, further analysis are performed in the wake of the geometry 7. Figure 6.31 shows the continuation of the spatial evolution of

Figure 6.29: Depiction of coherent structures in the wake of geometry 7 (vibrissa-like geometry with constant offset angle 15.6°) at simulation time, $t^* = 221$, at $x/D = 6.67$. top right : side view of the vortex cores and the location of plane-cut (perpendicular to streamwise) ; top right : overview of the vortex cores and tangential velocity vector in the plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.

the vortex cores in the wake of geometry 7 at $t^* = 219$ and $x/D = 5.7$.

Observations of figure 6.30 shows very repeatable patterns in the spanwise direction of the wake of geometry 7. Similar to the topology observed in figure 6.29, two-pairs of vortical braids are observed with clearer large pair of neighbouring vortical braids (of opposite polarity to each other). However, more wavelengths are identified in the spectra of the vorticity in this plane. These normalised wavelengths are 1.0, 1.5, 2.85, and 4.8 (corresponding $\lambda_{span}/\lambda = 0.28, 0.42, 0.79 \text{ and } 1.33$).

Figure 6.30: Depiction of coherent structures in the wake of geometry 7 (vibrissa-like geometry with constant offset angle 15.6°) at simulation time, $t^* = 219$, at $x/D = 4.9$ (left) [and 5.7 - (right), details in figure 6.31)], where x is measured from the centroid of the geometry . top right : side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.

Diagonal arrangement are also observed in the arrangement of the two-pairs of vortical braids. This pattern is found to be repeated over four periods in the spanwise direction (equivalent to the number of undulations in the geometry). Further observations revealed the consistent occurrence of the large pair of vortical braids centred about the saddle plane of the minor axis and occurrence of the two-pairs of vortical braids in the nodal plane of the minor axis. Interestingly, the vortex tubes is observed to form a 'mushroom-shaped' wake in a period of undulation, which connected the large

Figure 6.31: Depiction of coherent structures in the wake of geometry 7 (vibrissa-like geometry with constant offset angle 15.6°) at simulation time, $t^* = 219$, at $x/D = 5.7$ (refer to top left figure in figure 6.30), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores and tangential velocity vector in the right plane-cut ; top right : enlarged view of the top third of the vortex cores and tangential velocity vector in the right plane-cut ; bottom left : enlarged view of the middle third of the vortex cores and tangential velocity vector in the right plane-cut ; bottom right : enlarged view of the lower third of the vortex cores and tangential velocity vector in the right plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.

pair of vortical braids. The subsequent spatial evolution of the wake topology is observed to result in two major vortical structures (of opposite polarity relative to each other). This pair of very large vortical braids are observed to result from the merging of the large pair of vortical braids and two-pairs of vortical braids. Subsequent evolution of this vortical arrangements will result in the vortical arrangements depicted in figure 6.29. With reference to figure 6.31, it is interesting to note that the normalised wavelengths of the vortical braids in the wake of geometry 7 are found to uniformly low-

ered to 0.83, 1.15, 2.5 and 3.0 (corresponding to $\lambda_{span}/\lambda = 0.23, 0.32, 0.69$, and 0.83), respectively. However, this is not the case when comparing the normalised wavelengths of the vortical braids at $x/D = 5.7$ to the corresponding wavelengths at the plane $x/D = 6.67$ (refer to figure 6.29). Hence, a more in-depth discussion on the occurrence of vortical splits/merging shall be inline when studies on the wake stability of these geometries are conducted.

Further illustrations of the polarity of the vortical braids of geometry 4, 5, and 9 could be found in appendix C of this thesis. As geometry 3 and 4 shared strong physical similarity, only geometry 4 will be evaluated due to the stronger and more abundant occurrence of vortical braids. As geometry 6, 8, and 9 (geometry 7 evaluated in above subsection) shared similar wake topology and force responses, only geometry 9 will be evaluated due to its relevance to the new vibrissa geometry. For subsequent sections on 1D and 2D freely vibrating bodies, discussion on Instability of the wake topology would be integrated as part of the discussion of each geometry (unlike in this section, where a separate subsection on instability is created for better definition and clarity).

6.1.2 Dominant lift force response frequency

For each simulated geometry (refer to figure 6.2), a dominant lift force response frequency is observed. Such response frequency represents the vortex shedding frequency of the structure, which would be normalised with D and U , where D is the characteristic diameter of the simulation and U is the freestream flow velocity. Such normalisation will result in a nondimensional number, Strouhal Number. As the lift force response frequency is a significant parameter in structural design as well as understanding the underwater sensing principle of the harbour seal vibrissa, analysis on these frequencies are necessary. Figure 6.32 shows the Power Spectral Density (PSD) plot of the lift force of a circular cylinder based on Welch Method Fourier Transform (refer to chapter 5).

With reference to figure 6.32, the PSD of the viscous lift forces, pressure lift coefficient and total lift coefficient can be observed. While no dominant response frequency is found for the viscous lift forces, a single dominant response frequency is observed for the pressure lift forces (and therefore, the total lift forces). The dominant response frequency peak of the lift force on the circular cylinder is found to be a sharp peak with a wide range of frequency encapsulated in the area of the peak. Such breadth on the peak of the dominant response frequency indicated variability in the lift force response of flow around the circular cylinder. Such variability is attributed to the strength of the vortex tubes

Figure 6.32: Power Spectral Density (PSD) plot of the lift force (blue – viscous, red – pressure and green – total) on a stationary circular cylinder at $Re = 500$.

Figure 6.33: Sketch of the polarities of the vortical braids observed in a single period of undulation of the new vibrissa geometry. Legend : orange area - vortex cores.

shed from the flow around the circular cylinder. In turn, the strength of the vortex tubes is strongly correlated with the number and strength of the vortical braids connecting the neighbouring vortical tubes. As the flow around the circular cylinder is known to be unsteady, the number and strength of the vortical braids is known to vary for each cycle of vortex shedding. For each of the simulated geometry, the dominant lift force frequencies are identified and plotted in figure 6.33.

From figure 6.33, minimal variations are observed on the dominant lift force frequencies of most of the simulated geometries. Considerable deviations are observed only for geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) and geometry 10 (new vibrissa geometry). As

the dominant lift force frequency is equivalent to the vortex shedding frequency of the body, any variation would imply variation in the frequency of vortex shedding of the body. Consequently, the spacing of vortex tubes in the wake of the body would vary due to such difference in frequency of vortex shedding. However, due the relatively small variation in the dominant lift force frequency, the corresponding variation in the spacing between the vortex tubes will be minute. Additionally, for geometry 6, 8 and 9, significant deformation of the vortex tubes are observed. As such, proper definition of the existence of vortical tubes may be elusive. However, it is conclusive that the variation in frequency of vortex shedding (dominant lift force frequency) would result in changes on the spacing between the dominant feature of vortex shedding (mainly vortex tubes). As such, the spacing of the vortex tubes for geometry 7 and 10 would be slightly greater than the spacing between the vortex tubes in other simulated geometries.

6.2 Simulations of 1D Vibration Geometry

Despite the strong findings of lift coefficient reduction properties of various geometries, no conclusion on freely vibrating body could be achieved. Although the lift coefficients of the bodies acted as precursor to the VIV response, the corresponding reduction in $C_{L,RMS}$ may not be translated to reduction in the VIV (Vortex-Induced Vibration) response of the body. As such, in order to assess the VIV response and their corresponding reduction of VIV, simulations of flow around the freely vibrating geometries are performed. The geometries simulated are equivalent to the stationary geometry simulations (refer to figure 6.1). As priorities are given to geometries related to the underwater sensing of the harbour seal vibrissae, detailed evaluations are conducted for old and new vibrissa geometries (and circular cylinder as reference). Assessment on other geometries are conducted near their resonant frequencies (which are approximated to be equivalent to the vortex shedding frequencies in the corresponding stationary geometry simulations). By setting the natural frequency of the system to the dynamically scaled (refer to chapter 5) vortex shedding frequency, the mass-spring-dashpot system is set to the approximate condition for occurrence of resonance.

As extensive studies have been performed on circular cylinder, various comparison between experimental results and simulation results are performed for circular cylinder. With reference to figure 5.13, it is shown that there are various response amplitude and frequencies of a freely vibrating cir-

cular cylinder with various simulation settings. Various simulation settings are tested for flow around circular cylinder. From dynamic similarity (refer to 5), the Reynolds Number is known to be a function of the freestream flow velocity (U), characteristic diameter of the simulated geometry (D), and kinematic viscosity of the fluid (ν). Similarly, the reduced velocity (U^*) is known to be a function of freestream flow velocity (U), characteristic diameter of the simulated geometry (D), and the natural frequency of the system (f_n). Despite the adherence of scaling to the dynamic similarity, real case tests are necessary in order to ensure the preservation of such dynamic similarity beyond the theoretical standpoint. Hence, simulations of flow around circular cylinder are performed with various parameter settings which ensure dynamic similarity. For example, variation of Reynolds Number could be achieved through modification of freestream flow velocity (U), characteristic diameter of the simulation (D) or kinematic viscosity of the fluid (ν). With reference to figure 5.13, it can be observed that simulations are performed for flow around circular cylinder are performed at $Re = 1000, 800, 667, 571$ and 500 through variation of the kinematic viscosity of the fluid (and therefore, maintained the corresponding reduced velocity (U^*) due to the independence of U^* from the kinematic viscosity of the fluid). From figure 5.13 regardless of the modified parameter, it can be observed that the amplitude of response exhibited strong similarity for $500 \leq Re \leq 1000$. Agreement between the two methodology of varying Reynolds Number is also observed on the dominant response frequency of the circular cylinder. The effects of Reynolds Number of the flow around the circular cylinder on the amplitude of response is evaluated by comparing simulation results at various U^* at $Re = 500$ and 5000 . From figure 5.13, it is found that the amplitude and dominant response frequency of the circular cylinder are very similar for freestream flow at $Re = 500$ and 5000 . A significant difference is observed in the upper branch of the amplitude of response of the circular cylinder. Larger amplitude ($A/D \approx 0.68$) is observed for $Re = 5000$ at $U^* = 5.5$ and 5.75 as compared to the corresponding amplitude at $Re = 500$. In order to assess the effect of variation of mass ratio and natural frequency with Re , further simulations are performed for analysis on the amplitude of response and dominant frequency of response of a freely vibrating in crossflow circular cylinder. With reference to figure 5.14, simulations and experiments of various Reynolds Number (of the freestream flow) and mass ratios (of the circular cylinder) are performed. The first set of data in the legend pertained to simulations performed with constant $Re = 5000$ and $m^* = 10.1$ for all U^* . Such mass ratio of the circular cylinder is equivalent to the experimental settings of a freely vibrating in crossflow circular

cylinder by Khalak & Williamson [51]. This similarity in setting allowed for comparison and validation between the simulation data and experimental data. The second set of data in the legend is related to the simulations of a freely vibrating in crossflow circular cylinder with a linearly varying Reynolds Number to the $U^*(Re = U^* * 1000)$ and a constant $m^* = 10.1$. This simulation setting is found to be more relevant to the experimental settings of Khalak & Williamson. This is due to the increase in their towing velocity (equivalent to increment in Re) with increase in their corresponding U^* . The third set of data in the legend pertained to the experimental data obtained by Heather Rachel Beem of MIT Tow Tank group (our group) with a freely vibrating in crossflow circular cylinder at $5200 \leq Re \leq 19000$ (variation correspond to changes in reduced velocity due to varying towing speed) and $m^* = 1.4$. The fourth, fifth and sixth set of data are experimental data of freely vibrating in crossflow circular cylinder by Khalak & Williamson and Govardhan & Williamson [51, 52, 38]. From figure 5.14, it can be observed than strong correlation are observed between the first set of data and the fourth set of data. Similar amplitude of response are observed for most of the data as the simulation settings are relatively equivalent (same mass ratio with reasonable difference in Re). Discrepancies are observed for $5.25 \leq U^* \leq 6.5$ where the simulation data implied a larger amplitude of response of the circular cylinder as compared to the corresponding response in experimental data. For $U^* > 6.5$, the simulation results indicated smaller amplitudes of response as compared to the corresponding response in experimental data. It should be noted that the amplitude response of a circular cylinder against reduced velocity is commonly classified into three regions: initial branch, upper branch and lower branch [105]. For easier identification, the first data set will be utilised as reference example. The initial branch, upper branch and lower branch on the first data set could be defined within $U^* \leq 4.5, 4.5 < U^* \leq 6.5, U^* > 6.5$. Similar pattern can be observed for all the data set in figure 5.14. With reference to figure 5.14, from previous studies, it is well understood that a larger mass ratio would result in a wider lock-in region (region of response (U^*) where the dominant frequency of response is equivalent to the natural frequency of the system) [105]. A wider lock-in region would also result in a wider region of large amplitude of response, as shown in figure 5.14. Similarity is also observed between the first set of data and the second set of data. Only minor differences in the amplitude of response are observed in the region of $4.5 \leq U^* < 5$. It should also be noted that the third set of data (experiment by Heather Beem) is obtained from forced vibration experiment on a circular cylinder with 'zero' $C_{L,v}$, where $C_{L,v}$ is the lift force in phase with velocity.

The lift force in phase with velocity represented the occurrence of power transfer between the fluid and the body with exciting effect on the motion response of the body relevant to positive $C_{L,v}$ and damping effect on the motion response of the body relevant to negative value of $C_{L,v}$ [37]. A 'zero' $C_{L,v}$ would imply an average of no power transfer occurring between the body and the fluid, which represents a very close definition to freely vibrating bodies. However, due to the nature of freely vibrating bodies, a non-zero value of $C_{L,v}$ would generally be observed due to a slight imbalance in the power transfer from the body to the fluid and the fluid to the body. With relevance to the dominant response frequency plot in figure 5.14, only simulation data (first and second set of data) are represented due to the availability of the corresponding data (natural frequency and frequency response ratios of experimental data by Khalak & Williamson and Govardhan & Williamson are often unavailable). For experimental data (by Heather Beem), response frequency ratio of the current definition would be irrelevant due to the nature of the forced vibration experiments and their resulting data. From the dominant response frequency plot in figure 5.14, it is observed that variation in Reynolds Number do not incur significant variation in the dominant response frequency of the freely vibrating in crossflow circular cylinder and therefore, width of the lock-in region for various reduced velocities (U^*). It is important to note that the amplitude of response of the circular cylinder plotted in figures 5.13, 5.14 is based on the peak amplitude of response of the circular cylinder regardless of its temporal variability. However, from figure 5.12, it is found that the first amplitude of response is observed to be larger than the stable amplitude of response. Such occurrence is observed to be true for most bluff bodies due to the large kinetic energy and large vortex tubes shedded in the first instance of motion. Consequently, a larger motion is resulted as a reaction to the shedding of such large vortical tubes (consequence of Newton's Third Law). As such, for greater consistency in the comparison of various cases of freely vibrating bodies, RMS displacement instead of peak displacement would be plotted as the amplitude of response of the body, as suggested by Hover, Techet & Triantafyllou [45].

Figure 6.34 illustrates the amplitude ratio, dominant response frequency, the corresponding frequency ratio of the various simulated geometry (geometry 2 to 10) near their resonant condition. With reference to figure 6.34, it is found that the introduction of undulations in the major axis only (geometry 5) failed to reduce the amplitude of VIV (as compared to elliptical cylinder). However, increase in the dominant response frequency ratio of a freely vibrating geometry 5 is observed as compared to elliptical cylinder. In comparison, the amplitudes of VIV of geometry 3 (vibrissa-like geometry

with no major axis undulations and axis ratio of 0.44) and geometry 4 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.54). The larger amplitude of oscillation of geometry 4 as compared to geometry 3 is expected due to the bluffer nature of geometry 4. Although the amplitude of a freely vibrating geometry 3 is already an order of magnitude smaller than the corresponding amplitude of elliptical cylinder, the amplitude of oscillation of geometry 6 to 9 are observed to be even smaller than it. However, comparison between their dominant response frequencies illustrated significantly different responses. The response frequency of a freely vibrating geometry 3 ($f_r D/U$) is ≈ 0.22 , which is significantly higher than the response frequencies of freely vibrating geometry 6, 8, 9, and 10 ($\approx 0.16 - 0.18$). Interestingly, the response frequencies of geometry 2 and geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) are notably lower as compared to the corresponding frequencies of other geometries (≈ 0.14 and 0.1 , respectively). These anomalies indicated a larger spacing between the dominant vortices (commonly vortex tubes) in the wake of these geometries. Correspondingly, with exception to freely vibrating geometry 2 and geometry 7, all other freely vibrating geometries are observed to respond near the lock-in region of their structures ($f_r/f_n \approx 1.0$). As the lift coefficient of the geometries are directly correlated to their amplitudes of oscillation, comparisons between $C_{L,RMS}$ of the stationary geometries and their corresponding amplitudes of oscillation in a freely vibrating in crossflow setup would provide strong indications of the modification of the fluid dynamics around the structure. Through comparison between figure 6.2 and figure 6.34, strong similarities are observed. Most notably, the amplitudes of oscillation of freely vibrating geometries 3, 4, 6, 7, 8, 9, 10 are found to be significantly lower than the amplitude of oscillation of geometry 2, which is directly correlated to their corresponding comparison of $C_{L,RMS}$ for stationary setup. However, certain variations are also observed, most notably, the similarity between the amplitudes of geometry 2 and geometry 5, which are not observed in the comparison between the $C_{L,RMS}$ of their stationary simulations. Significantly lower amplitudes of oscillation of geometries 3 and 4 as compared to the amplitude of oscillation of geometry 2 (and, therefore geometry 5) are also found to be incongruent to the corresponding $C_{L,RMS}$ of their stationary simulations. Further investigation on the details of variation of amplitudes and dominant response frequencies of the geometries will be presented later in this section.

Analysis on the wake topology of the freely vibrating in crossflow circular cylinder and comparison to its corresponding wake topology for stationary circular cylinder should provide a reference

Figure 6.34: VIV response of geometry 2 to 10 near their resonance condition (such condition is achieved through equivalence of the natural frequency of the system to the frequency of vortex shedding of the corresponding geometries in stationary simulations at similar $Re = 500$). Top : amplitude ratio of VIV ; middle : dominant response frequency ratio (f_r/f_n) ; bottom : ratio of the dominant frequency response to the natural frequency of system.

point on the possible variation in the wake topology of a geometry. Figure 6.35 illustrates the coherent structures in the wake of a freely vibrating in crossflow circular cylinder in a freestream flow at $Re = 500$ and $U^* = 5$ during its amplitude of motion in positive Y-direction of its stable period of response. Note: upward movement in crossflow direction - positive Y, downward movement in crossflow direction - negative Y.

Through observations of figure 6.35, significant spanwise periodicity and regularity can be observed in the wake of a freely vibrating in crossflow circular cylinder. The spacing between the vortical braids (and vortical tubes) are observed to be highly regular with a consistent spacing of $\approx 0.5D$ (center to center). As compared to flow around a stationary circular cylinder at the same Re , the spacing of the vortical braids in the wake of a stationary circular cylinder is observed to vary more significantly ($\approx 0.5D$ to $2D$). Such regularity in shedding of vortex tubes and the arrangement of vortical braids are a consequence of the feedback of power between the fluid and the body. Allowance of 1D free motion on the circular cylinder is found to give rise to the possibility of power transfer from the body to the fluid and vice versa. Unlike in freely vibrating circular cylinder, stationary circular cylinder only allowed for one way power transfer (from fluid to the body). Consequently, greater perturbations are resulted with more observed irregularity in the vortex tubes and braids. With reference to figure 6.35, it can also be observed that the instability of vortex tubes in the wake of a freely vibrating in crossflow circular cylinder is less prominent as observed through the minute undulations on the vortex tubes in its wake (as compared to that of the stationary circular cylinder). Similarly, this is found to be a consequence of the lower perturbation on the freely vibrating circular cylinder. As the instability of the vortex tubes in freely vibrating circular cylinder is less prominent, a further consequence can be observed through the regularity of the thickness of the vortical braids and their arrangements. Figure 6.36 shows the polarity of the vortical braids along the spanwise direction in the wake of the freely vibrating in crossflow circular cylinder corresponding to figure 6.35.

From figure 6.36, it can be observed that the vortical braids is found to occur in pairs of opposite polarity (for every positively polarised (counter-clockwise) vortical braids, the neighbouring vortical braids on each side would be negatively polarised (clockwise) vortical braids and vice versa). Such occurrence is found to be consistent with the previous finding on the polarity of vortical braids in the wake of stationary circular cylinder. However, the pertinent question is aroused onn the onset and the cause of vortical braids (and therefore, instability in the wake of the freely vibrating circular

Figure 6.35: Arrangement of coherent structures ('P+S' arrangement) in the wake of a freely vibrating in crossflow circular cylinder (Left) during its amplitude of motion in positive Y direction at its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 5.2$ and simulation time, $t^* = 224$ and stationary circular cylinder (Right) [reprint from figure 6.13 for ease of comparison]. Top: Isometric view ; middle : side view ; bottom : top view.

Figure 6.36: Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder, $t^* = 224$, at $x/D = 6.2$ (refer to top left figure in figure 6.35), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores and tangential velocity vector in the plane-cut ; top right : enlarged view of the top third of the vortex cores and tangential velocity vector in the plane-cut ; bottom left : enlarged view of the middle third of the vortex cores and tangential velocity vector in the plane-cut ; bottom right : enlarged view of the lower third of the vortex cores and tangential velocity vector in the plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores.

cylinder). The occurrence of highly-regular vortical braids in the wake of the freely vibrating circular cylinder depicted a significant deviation from the irregularly-spaced vortical braids in the wake of the stationary circular cylinder. A consistent normalised wavelength of 0.75 is observed throughout the span in the wake of the freely vibrating circular cylinder. As such, analysis of the onset of the instability in the wake of the freely vibrating circular cylinder is necessary to determine the nature of the regularity of the vortical braids in its wake (refer to appendix D). However, despite the known polarity of the vortical braids, the polarity of the vortex tubes connected by the braids are yet to be

Figure 6.37: Top view depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder, $t^* = 224$, in a plane-cut (perpendicular to spanwise) at $z/D = 8.7$ (refer to top left figure in figure 6.35), where z is measured from the base of the geometry. Legend: cyan area denotes cross-sectional area of vortex cores.

determined. As the vortical braids are almost always perpendicularly oriented to the vortex tubes, different point of reference is necessary to determine the polarity of the vortical tubes in the wake of the freely vibrating in crossflow circular cylinder. Figure 6.37 shows the top view of a z -plane in the wake of the freely vibrating circular cylinder at $z/D = 8.7$ (z measured from the base of the circular cylinder) at $t^* = 224$. Uniformity of the polarity of vortex tubes in the wake of the freely vibrating circular cylinder is observed along the spanwise length of the circular cylinder.

It should be noted that the vortical arrangements observed in the wake of the stationary and freely vibrating in crossflow circular cylinder are '2S' and 'P+S' arrangement, respectively [106]. In a '2S' wake arrangement of a circular cylinder, a single pair of vortices are created in one cycle of

vortex shedding. For 'P+S' arrangement, asymmetric vortex tubes profile are observed about the plane through the centroid of the geometry with parallel orientation to the streamwise direction. This resulted in two vortex tubes shed in one of the half-cycle oscillatory motion and a single large vortex tubes generated in the other half-cycle of the oscillatory motion of the body. Through comparison in figure 6.35, similar lateral widths of the wake of the stationary and freely vibrating in crossflow circular cylinder are observed. However, the arrangement of the vortical tubes are notably different. In the wake of stationary circular cylinder, the centroids of the vortex tubes are observed to be aligned with the centroid of the circular cylinder. However, the centroids of the vortex tubes of the freely vibrating circular cylinder are observed to be aligned about the axis through the outer diameter (furthest point on the circular cylinder at its amplitude of motion about its neutral axis) of the circular cylinder at its amplitudes of motion of the circular cylinder on each respective sides (in crossflow direction).

In comparison to the circular cylinder, the wake of a freely vibrating in crossflow elliptical cylinder is observed to consist of no vortical braids, similar to the wake of a stationary elliptical cylinder at equivalent $Re(=500)$. Figure 6.38 depicts the vortex cores (identified with λ_2 criterion) in the wake of a freely vibrating in crossflow elliptical cylinder during its amplitude of motion in negative Y direction of its stable period of response for $U^* = 4.81$ and $Re = 500$ [48].

In contrast to the case of circular cylinder, no significant difference is observed between the wake topology of a freely vibrating in crossflow elliptical cylinder and stationary elliptical cylinder. From figure 6.34, it can also be observed that the dominant response frequency of a freely vibrating in crossflow elliptical cylinder is below the corresponding frequency of other geometries (except geometry 7). This indicated a slightly wider spacing between one vortex core to another in the wake of a circular cylinder. Such occurrence is a characteristic occurrence of a streamlined body where the rolling up of the vortices require longer duration period before its shedding due to the weaker pressure gradient between the leading edge and trailing edge of the geometry. However, it is important to note that a weak secondary frequency peak is observed at 0.2074, which corresponds to the dominant frequency of lift coefficient of a stationary elliptical cylinder. No vortical braids and similar spacing between vortex tubes are observed through comparison between the wake topology of the freely vibrating in crossflow elliptical cylinder and stationary elliptical cylinder. A '2S' shedding pattern is observed in the wake of the freely vibrating elliptical cylinder and stationary elliptical cylinder with similar alignment of the corresponding vortex tubes to the centroid of the geometry. Slightly

Figure 6.38: Arrangement of coherent structures in the wake of a freely vibrating in crossflow elliptical cylinder (Left) during its amplitude of motion in negative Y direction of its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 4.81$ and simulation time, $t^* = 274$ and stationary elliptical cylinder (Right) [reprint from figure 6.13 for ease of comparison]. Top: Isometric view ; bottom : top view.

wider vortex tubes are observed in the near-field wake of the freely vibrating in crossflow elliptical cylinder as compared to the corresponding width of the stationary elliptical cylinder. This suggested a slightly higher circulation in the vortex tubes of the freely vibrating elliptical cylinder. However, due to the minute difference, its significance should not be exacerbated. Figure 6.39 shows the pressure profile in a freely vibrating circular cylinder and elliptical cylinder for freestream flow of $Re = 500$ at $U^* = 5.2$ and $U^* = 4.81$, respectively.

By comparing figure 6.39 and figure 6.3, despite the similarity of their vortical arrangements in their wake, difference in the pressure profile of the freely vibrating elliptical cylinder and stationary elliptical cylinder can be observed. Elongation of the low pressure region (and therefore, vortex formation length) in the near-field wake of the body is observed along with higher minimum pres-

Figure 6.39: Pressure profiles in the wake of a freely vibrating in crossflow circular cylinder (left) and elliptical cylinder (right) at $U^* = 5.2$ and $U^* = 4.81$. Top : freely vibrating body ; bottom : stationary body (reprint from figure 6.3).

sure near the core of the first vortex of the freely vibrating elliptical cylinder (as compared to the stationary elliptical cylinder). A more significant difference in the pressure profiles of the wakes is observed between the wake of a freely vibrating and stationary circular cylinder. Despite maintaining of the same distance between the minimum pressure regions (vortex cores) in the wake of a freely vibrating circular cylinder (as compared to the corresponding distance for stationary circular cylinder), a different alignment between the vortices are observed in figure 6.39. As aforementioned, the vortex cores of a stationary circular cylinder is aligned about the centroid of the circular cylinder. For freely vibrating circular cylinder, the pair of large vortex cores (larger vortex tube of 'P' vortex-pair and farther opposite-polarity vortex tube) are found to be aligned to a 45° line with respect to the streamwise direction. A larger low pressure region is observed in the near-field wake of the freely vibrating circular cylinder with closer proximity between the centre of the low pressure region to

the trailing edge of the body. The modifications of the wakes in the vicinity of the freely vibrating circular cylinder and elliptical cylinder are found to be the resultant of the greater relative movement between the body and the low pressure region (as compared to stationary body simulations) with a more significant modification observed for circular cylinder due to its larger motion response (A/D).

Through comparison between figure 6.34 and figure 6.2, significantly smaller amplitude of oscillation of geometry 3 is observed as compared to the amplitude of oscillation of elliptical cylinder. Such finding is disproportionate to the stationary simulation results, due the significant difference in the amplitude of response. However, such result is expected due to the overall more streamlined nature of geometry 3 as compared to geometry 2, which resulted in the higher minimum pressure region in the wake of a freely vibrating geometry 3. Additionally, the existence of the minor axis undulations resulted in further increment in the pressure at the minimum pressure region (shed vortices). As such, the reaction forcing on the body due to the shedding of these vortices resulted in a significantly lower amplitude of oscillation. Similarity between the dominant response frequency of a freely vibrating geometry 3 and dominant lift force frequency of a stationary elliptical cylinder. From figure 6.34, significantly smaller amplitude of oscillation of a freely vibrating geometry 4 as compared to a freely vibrating elliptical cylinder is also observed although this difference is slightly smaller as compared to the amplitude difference between freely vibrating geometry 3 and elliptical cylinder. Hence, analysis of the pressure and velocity fields of geometry 4 should conclusively show the higher minimum pressure region (vortices) shed from the low pressure region in the near-field wake of the body. Such analysis will be described later in this section. Similar to elliptical cylinder, geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) is observed to share strong similarity between the wake topology of a freely vibrating in crossflow geometry 3 and stationary geometry 3. From comparison between figure 6.34 and figure 6.2, relative difference (as compared to the responses of other geometries) is observed between the $C_{L,RMS}$ of a stationary geometry 4 and the amplitude ratio of a freely vibrating in crossflow geometry 4. Figure 6.40 shows the vortex cores (identified with λ_2 criterion) in the wake of a freely vibrating in crossflow geometry 4 during its amplitude of motion in positive Y direction of its stable period of response for $U^* = 4.76$ and $Re = 500$ [48].

From figure 6.40, it can be observed that the certain topological changes occurred in the wake of a freely vibrating in crossflow geometry 4 as compared to the corresponding topology in stationary

Figure 6.40: Arrangement of coherent structures in the wake of a freely vibrating in crossflow geometry 4 (Left) during its amplitude of motion in negative Y direction at its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 4.76$ and simulation time, $t^* = 185$ and stationary circular cylinder (Right) [reprint from figures 6.14, 6.15 for ease of comparison]. Top: Isometric view ; middle : side view ; bottom : top view.

Figure 6.41: Side view of the alternate arrangement of coherent structures in the wake of a freely vibrating in crossflow geometry 4 during its amplitude of motion in negative Y direction at its stable period of oscillation for freestream flow at $Re = 500$, $U^* = 4.76$ and simulation time, $t^* = 251$.

geometry 4. Although the vortical braids in the wakes of stationary geometry 4 is observed to form a 'W'-shaped vortex cores in the spanwise direction, the wake of the freely vibrating in crossflow geometry 4 shows a regular occurrence of 'U'-shaped vortex cores in the spanwise direction. Slight difference on the deformation of the vortex tubes are also observed with slight bending observed along the spanwise direction of the wake of the stationary geometry 4. However, the wake of the freely vibrating geometry 4 is observed to switch between two states of U-shaped vortical braids and U-shaped vortical braids with two-ribs (vortical braids) in the middle. Figure 6.41 shows the alternate U-shaped vortical braids with two-ribs in its midst as the wake of a freely vibrating in crossflow geometry 4.

Despite the intriguing nature of the instability in the wake of geometry 4, a necessary explanation of the alternation between U-shape and ribbed-U-shape vortical braids is more urgent. As such, through depiction of the pressure profile about the plane of symmetry (perpendicular to Y-axis) of the freely vibrating geometry 4, the cause of the alternation of the wake of geometry 4 is defined. Figure 6.42 shows the pressure profiles in the wake of freely vibrating in crossflow geometry 4 at its nodal and saddle planes and the plane of symmetry (perpendicular to crossflow) for $t^* = 185$ and $t^* = 251$.

Figure 6.42 compared the pressure profile about the plane of symmetry through the centroid

Figure 6.42: Pressure profiles in the wake of freely vibrating in crossflow geometry 4 at $t^* = 185$ (left), 216 (right) for $Re = 500$. top right : side view about the plane of symmetry through the centroid of geometry 4 at neutral position (perpendicular to crossflow) for $t^* = 185$; top right : side view about the plane of symmetry through the centroid of geometry 4 at neutral position for $t^* = 251$; middle left : top view of the pressure profiles in the nodal plane of geometry at $t^* = 185$; middle right : top view of the pressure profiles in the nodal plane of geometry at $t^* = 251$; bottom left : top view of the pressure profiles in the saddle plane of geometry at $t^* = 185$; bottom right : top view of the pressure profiles in the saddle plane of geometry at $t^* = 251$.

of geometry 4 at neutral position (perpendicular to the crossflow direction), which represented the cross sectional cut through all the vortex tubes in the wake of the freely vibrating geometry 4. Such cross sectional plane-cut allowed depiction of the pressure profiles about the origin of the vortical braids. It should be noted that the plane of symmetry through vortex tubes is the plane of greatest instability due to the mixing of vorticity (and velocity vectors), which entered the plane from both sides (positive and negative Y-directions). Hence, with reference to figure 6.42, it can be observed that the U-shaped vortical braids are the resultant of regular spanwise pressure profile in the near-wake region of geometry 4. As the body is freely vibrating, the wake of freely vibrating geometry 4 would be unstable (variation with time) but it is capable of maintaining periodicity in its wake. However, a certain perturbation (in the form of spanwise flow velocity for geometry 4) would result in the amplification and propagation of such perturbation, which subsequently led to the tipping of the periodicity in the wake of geometry 4. Such perturbation is observed to result in a less uniform-but-periodic spanwise pressure distribution. The redistribution of pressure is observed to result in the tilting of the lower pressure region to the lower end of the spanwise minimum pressure region. Hence, this resulted in the additional two-vortical braids in the midst of the U-shaped vortical braids due to the imbalance in the resulting spanwise flow velocity upon the creation of the U-shaped vortical braids. From the pressure profiles at nodal and saddle planes in figure 6.42, significant differences in the pressure profiles at nodal plane could be observed. For U-shaped vortical braids, a longer and wider low pressure region in the near-field wake (as compared to ribbed-U-shape vortical braids) is observed with perceivable distinction to the pressure profile in the saddle plane. From figure 6.42, a shorter but similar width of low pressure region is observed in the nodal and saddle plane of the pressure profiles at $t^* = 251$. Through comparison between the nodal and saddle planes in the wake of geometry 4 at $t^* = 185$ and $t^* = 251$, a farther location of first minimum pressure could be observed for U-shaped vortical braids, as figure 6.42 suggested. This illustrated the kinetic energy of the fluid in the vortex tube and implied a higher stability of the U-shaped vortical braids as compared to the ribbed-U-shaped vortical braids. Through figure 6.42, a better understanding on the occurrence of U-shaped and ribbed-U-shaped vortical braids are achieved. However, analysis of the polarity of the vortical braids is necessary in determining the effect of the ribs on the polarity of the ribbed-U-shape vortical braids. Figure 6.43 shows the vortex core and tangential velocity vector of the vortical braids in U-shaped ($x/D = 6.3$) and ribbed-U-shaped wake structure ($x/D = 7.1$).

Figure 6.43: Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 4 in the plane-cut (perpendicular to streamwise) for U-shaped vortical braids arrangement (top), refer to figure 6.40, and ribbed-U-shaped vortical braids arrangement (bottom), refer to figure 6.41. Top left: cross-section of vortex cores at a plane-cut $x/D = 6.3$ at $t^* = 185$; top right: enlarged view of one period of U-shaped vortical braids in the plane-cut $x/D = 6.3$ at $t^* = 185$; bottom left: cross-section of vortex cores at a plane-cut $x/D = 7.1$ at $t^* = 251$; bottom right: enlarged view of two periods of U-shaped vortical braids in the plane-cut $x/D = 7.1$ at $t^* = 251$. Legend: orange areas denote vortex cores.

As figure 6.43 illustrate the cross-sectional cut through the vortical braids for both types of wake topology, polarity of the vortical braids could be determined through analysis of the tangential velocity vector. A spectra analysis of these tangential velocity vector revealed the wavelengths of the vortical braids in this wake. Notably, the normalised wavelengths are 0.75, 1.85, and 3.7 (corresponding to $\lambda_{span}/\lambda = 0.21, 0.51, \text{ and } 1.03$). From figure 6.43, it can be observed that both forms of wake of freely vibrating geometry 4 occurred periodically along the spanwise of the body. For U-shaped vortical braids, analysis of one bulk of the vortical braids revealed a unique three-vortices

pairing. Theoretically, a delicate balance of vortical strengths is necessary to achieve a stable occurrence of such configuration. From figure 6.43, it can be observed that two large vortices of opposite polarity occurred in the vicinity of a smaller vortex, which is located at the nodal plane of geometry 4. No vortex (vortical braids) is observed to occur in the saddle plane of geometry 4 for U-shaped wake configuration. It should be noted that the smaller vortex is found to always appear at a closer proximity to the negatively-polarised vortex (clockwise rotation). As expected, the smaller vortex is oppositely-polarised to the large vortex nearer to it. In comparison, for ribbed-U-shaped wake configuration, similar vortices are observed near the nodal region of the geometry. The vortical ribs (braids) in the midst of the ribbed-U-shaped configuration is found to always occur in the saddle plane of the geometry, where no vortices are observed to occur in U-shaped wake configuration. This implied that the greater strength of the large vortices in the nodal plane region for ribbed-U-shape configuration which suffice the creation of a pair of smaller oppositely polarised vortices. Hence, through comparison between the two wake configurations, it can be deduced that the ribbed-U-shape wake configuration is the result of addition of a pair of weakly-polarised vortices and the U-shape wake configuration. As the pair of weakly-polarised vortices is a consequence of the two large neighbouring vortices, it can be concluded that the occurrence of the ribs (alternation between the wake configuration) is simply a result of the fluctuation in the strength of the vortices at the neighbouring nodal points.

From figure 6.34, it can be observed that the dominant response frequency of geometry 5 (vibrissa-like geometry with no minor axis undulations) is relatively smaller than the corresponding frequency for other geometries (except geometries 2 and 7). This is despite of the similarity in the initial configuration of the simulation. Hence, analysis on the wake topology of the freely vibrating geometry 5 is performed in order to obtain insights into the lower dominant response frequency of geometry 5. Figure 6.44 shows the vortex cores in the wake of a freely vibrating geometry 5 at $Re = 500$, $t^* = 200$, and $U^* = 5.22$.

Through comparison between figure 6.44 and figure 6.40, some similarity maybe perceived between the wake topology of the freely vibrating geometry 4 and geometry 5. However, detailed observations revealed different nature and connection of vortical braids and vortex tubes. While the wake of freely vibrating geometry 4 is observed to form U-shaped or ribbed-U-shaped vortical structures for every period of undulations, a different wake configuration is observed in the wake of freely

Figure 6.44: Depiction of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 5 (vibrissa-like geometry with no minor axis undulations) at $Re = 500$, $t^* = 200$, and $U^* = 5.22$. Top left : isometric view ; top right : side view ; bottom : top view.

vibrating geometry 5 with observed regularity for each period of undulations along the spanwise direction of the wake. Figure 6.45 shows the pressure profiles of the wakes of the freely vibrating in crossflow geometry 5 in the plane of symmetry (perpendicular to crossflow) and in the nodal and saddle planes of the geometry.

With reference to figure 6.50, regularity of the spanwise pressure profile can be observed in the low pressure region in the near field wake of the freely vibrating geometry 5. From the spanwise pressure profile, it is seen that the saddle planes consist of shorter but lower minimum pressure region in the near field wake as compared to the nodal planes. The undulations in the major axis is observed to create divergence of the minimum pressure region from saddle planes towards neighbouring nodal planes. However, due to such divergence, spanwise flow velocity is increased towards the nodal planes. As the nodal planes are neighboured by saddle planes, the spanwise flows from their neighbouring saddle planes would be opposite in velocity vector (with approximately equal magnitude). Such occurrence would increase the pressure at the nodal planes of geometry 5, in particular

Figure 6.45: Pressure profiles in the wake of freely vibrating in crossflow geometry 5 at $t^* = 199$ (top left figure only) , 200 for $Re = 500$. top right : side view about the plane of symmetry through the centroid of geometry 5 at neutral position (perpendicular to crossflow) for $t^* = 199$; top right : side view about the plane of symmetry through the centroid of geometry 5 at neutral position ; middle left : top view of the pressure profiles in the nodal plane at the peak of leading edge ; middle right : top view of the pressure profiles in the nodal plane at the peak of trailing edge ; bottom left : top view of the pressure profiles in the saddle plane at the valley of leading edge ; bottom right : top view of the pressure profiles in the saddle plane at the valley of trailing edge.

at the meeting point of the spanwise flow from their neighbouring saddle planes. Such observation can be extended into the pressure profiles at the nodal and saddle planes of geometry 5 for its peaks and troughs in the leading and trailing edges as shown in figure 6.45. From figure 6.45, it can be observed that the spacing between the minimum pressure region (vortices) are considerably sparser as compared to the corresponding spacing of other geometries (refer to figure 6.39, 6.42). The larger spacing between the vortices is found to explain the significantly lower dominant response frequency, as shown in figure 6.34. Similar pressure profiles is observed between the peak of the leading and trailing edges of the nodal planes as well as between the valleys of the leading and trailing edges of the saddle planes. Larger minimum pressure region is observed in the saddle planes as compared to the nodal planes. From comparison between figure 6.42 and figure 6.45, lower pressure minimum pressure region is observed in the wake of a freely vibrating geometry 5 as compared to the corresponding pressure for geometry 4. This explained the significantly higher amplitude of VIV of geometry 5 as compared to geometry 4. Such difference in pressure region is also observed between the pressure fields of freely vibrating geometry 4 and elliptical cylinder, as observed through comparison between figure 6.45 and figure 6.39. In spite of the understanding of the amplitude and dominant response frequency of the freely vibrating geometry 5, knowledge of the instability and regularity of the wake configuration are yet to be determined. Figure 6.46 shows the vortex cores and tangential velocity vector in the plane-cut (perpendicular to streamwise direction) of the wake of the freely vibrating geometry 5 at $x/D = 6.8$ (x measured from centroid of the body), $t^* = 200$ and $U^* = 5.22$.

While figure 6.46 illustrates the details of the vortex cores and their corresponding velocity vectors (polarity), knowledge of their relative alignment to the undulations on the major axis of geometry 5 is necessary. Such knowledge would tailor the relevance of the emerging vortical arrangements to the undulations of the major axis, whenever possible. Consequently, any emerging periodicity and regularity would be more observable through their relevance to the undulations of the geometry. Hence, figure 6.47 shows the perspective view [45° rotation about the Z-axis from the streamwise direction (0°)] of the freely vibrating geometry 5 and the plane-cut at $x/D = 6.8$ corresponding to figure 6.46.

Despite the evident regularity from the side view of the pressure profiles of freely vibrating geometry 5 in figure 6.45, such regularity was not observed from the analysis of the vortical braids

Figure 6.46: Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 5 in the plane-cut (perpendicular to streamwise) at $x/D = 6.8$, $t^* = 200$ and $U^* = 5.22$. Top left: spanwise overview ; top right: enlarged view of the first one and a half period of undulations ; bottom left: enlarged view of the subsequent 1.25 period of undulations ; bottom right: enlarged view of the final 1.25 period of undulations. Legend: orange area denotes vortex cores.

in the subsequent propagation of the coherent structures, as depicted in figure 6.46. With reference to figures 6.46, 6.47, regularity is only observed for every two periods of undulations through the occurrence of a five-vortices pairing identified at the first and third saddle planes (from the top) of the wake of geometry 5. Such vortices pairing is unique in comparison to the observed polarity pairing of the vortical braids in circular cylinder (two-vortices pair) and geometry 4 (three-vortices pair). For the first five-vortices pairing at the first nodal plane of geometry 5 (from the top), composition of the vortices could be broken down into a four-vortices pairing and a single vortex based on their proximity relative to each other. The configuration of the four-vortices pairing is observed

Figure 6.47: Perspective view [45° rotation from streamwise direction about Z-axis (spanwise)] of geometry 5 and the corresponding plane-cut at $x/D = 6.8$ (refer to figure 6.46) depicting the cross-section of the vortex cores and tangential velocity vector of the plane.

to form a rhombus (as compared to a square-like configuration in figure 6.30 for stationary geometry 7). However, the arrangements of vortices is found to abide the diagonally similar polarity, which is observed in the four-vortices pairing of stationary geometry 7 (refer to figure 6.30). In completing the discussion on the regular occurrence of the vortical braids in the wake of geometry 7 and with reference to figure 6.46, the normalised wavelengths of the vortical braids in the wake of geometry 5 are 0.74, 1.11 and 5.86 (corresponding to $\lambda_{span}/\lambda = 0.20, 0.30$, and 1.63). In the wake of the freely vibrating geometry 5, the four-vortices pair is found to act with a net vorticity, which is found to result in further pairing with a single vortex of positive polarity. This implied that the four-vortices pairing created a net negative polarity, which is observable through the tangent velocity vectors where larger (in magnitude) velocity vectors are observed for clockwise oriented vortices. Similar analysis on the five-vortices pairing at the third saddle plane (from the top) revealed a square configuration of the four-vortices pair with a further pairing to a large negatively-polarised vortex. This implied a net counter-clockwise vorticity in the four-vortices pair at the third saddle plane, which is found to be of opposite polarity as compared to the single vortex at the first saddle plane of geometry 5. Similar diagonally-equal polarity is observed for the four-vortices pairing in the third saddle plane of geometry 5. With reference to figures 6.46, 6.47, five vertically-aligned vortices are observed in the vicinity of the second saddle plane (from the top) of geometry 5. These vortices are observed to occur

from the neighbouring nodal points of geometry 5 with a top to bottom polarity of positive-negative-positive-negative-positive polarities. In comparison, at the neighbouring nodal points of the first and third saddle planes (where five-vortices pairing are observed), a single vortex (part of five-vortices pairing) is found to occur. It is important to note that, as periodic boundary condition is applied at the spanwise ends of the simulation domains, the top and bottom ends of the geometry could represent another saddle plane for geometry 5. Through similar analysis, observation of five vertically aligned vortices are noted, which is found to be similar to the arrangement of vortices in the second saddle plane (from the top) of geometry 5. As such, it can be concluded that the spanwise periodicity in the wake of geometry 5 is found to occur for every two periods of undulations. However, despite the peculiarity of the spanwise configuration of the polarity of the vortices, such wake configuration is found to be stable and persistent (no form variation with time) during the stable period of VIV response of the structure.

From figure 6.34, similarity of amplitude and dominant response frequency (except geometry 7) can be observed among geometry 6 (vibrissa-like geometry with no offset angle), 7 (vibrissa-like geometry with constant offset angle of 15.26°) and 8 (vibrissa-like geometry with constant offset angle of 17.6°). Similarity in the wake topologies of freely vibrating geometry 6, 7, and 8 are also observed, as shown in figure 6.48. As such, a representative geometry is chosen in order to explain the effects of geometrical imposition of dual-axial undulations on the response and wake topology of the structure. With reference to previous subsection on simulations of stationary geometries, detailed elaborations of geometry 7 are performed for direct comparison to stationary simulation results for geometry 7. This selection is supported by the need to analyse the significant difference between the dominant response frequency of geometry 7 as compared to the corresponding response frequency of other dual-axial undulations geometries (geometries 6, 8, 9 and 10).

From figure 6.48, similarity among the topology of the wakes of geometry 6, 7 and 8 can be observed. A general U-shaped vortical braids for each period of undulations is observed in the topology of the shedded vortices. However, a more significant deformation in the vortex tube of geometry 7 is seen as compared to the vortex tubes in the wakes of geometry 6 and 8. The general topology of the wakes are observed to consist of similar configuration of vortices and additional vortical braids along the spanwise direction of the geometry. As such, the similarity of amplitude of VIV and dominant response frequency to VIV for between geometry 6 and 8 is found to coincide

Figure 6.48: Depiction of the side view of vortex cores (based on λ_2 criterion) in the wake of various geometries. Top left : geometry 6 (vibrissa-like geometry with no offset angle) at $t^* = 187$ and $U^* = 5.23$; top right : geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) at $t^* = 213$ and $U^* = 5.78$; bottom : geometry 8 (vibrissa-like geometry with constant offset angle of 17.6°) at $t^* = 211$ and $U^* = 5.14$.

with the reasoning of similarity in the wake configuration of the corresponding geometries. Different response frequency of geometry 7 is found to be strongly correlated to the deformation of the vortex tubes which is found to result in a cellular-shedding-like effect (two or more frequency responses prevail). This effect is found to manifest in the observed dual response frequency peak in the motion response of geometry 7. The second frequency peak ($f_r D/U$) is found to occur at 0.18, which is found to coincide with the natural frequency of the setup (as well as the vortex shedding frequency of a stationary geometry 7). With reference to figure 6.34, it can be seen that the amplitude of VIV of

geometry 6, 7, 8 (vibrissa-like geometries with dual-axial undulations) are significantly smaller than the amplitude of VIV of geometry 3, 4, 5 (vibrissa-like geometries with single-axial undulations). As such, it is necessary to analyse the pressure fields in the vicinity of the structure, which resulted in forcing and motion response of the corresponding geometry. Figure 6.49 shows the pressure profiles in the nodal and saddle planes of freely vibrating in crossflow geometry 7 as well as the plane symmetry about the neutral axis of the geometry 7 for $Re = 500$, $t^* = 213$, and $U^* = 5.23$.

From figure 6.49, slight differences are observed in the pressure profiles in the nodal and saddle planes of geometry 7 between their respective peaks and valleys at the leading edge and trailing edge. The perceived slight differences are the lower minimum pressure regions are observed at the peak of the trailing edge in the nodal plane (as compared to the peak at the leading edge) and at the valley of the leading edge of the saddle plane (as compared to the valley at the saddle plane). By comparing the low pressure region in the near-field wake of geometry 7 in figure 6.49, slightly wider and shorter low pressure region is observed in the saddle planes as compared to the nodal planes. However, difference in pressure profiles between the nodal and saddle planes are perceivable, as the minimum pressure region in the nodal plane occurred in the location of the first vortex shed from the body. At the corresponding location in the saddle planes, a higher pressure ($\delta C_p = 0.15$) vortex is observed. Such occurrence is also observed in the subsequent vortex in the wake of geometry 7. However, comparison between the minimum pressure region in the near-field wake of the body (not shed vortices) between the nodal and saddle planes, revealed a lower near-field minimum pressure region in the saddle plane (as compared to the nodal planes). This indicated a significant spanwise flow velocity in the wake of freely vibrating geometry 7 as the minimum pressure region is found to shift profusely from the saddle plane (right before shedding from the near-field low pressure region) to the nodal planes (location of first shed vortex). Further observation on the spanwise flow velocity will be depicted in figure 6.50. It is noted that such switched of low pressure region is also notified in the pressure profiles in the nodal and saddle planes of geometry 5, as observed in figure 6.45. Further note on figure 6.50 is on the background colouring, which is relatively 'patchy' in colouring. This is due to the 'banded' type colouring instead of continuous colouring. The use of banded colouring allowed for easier quantification of various colour indicators. However, disadvantage is brooded on the different colouring due to slight deviation from the colouring band limits. As such, colouring for the spanwise flow velocity (w/U) of 10^{-6} and 10^6 are rendered differently due to their belongings

Figure 6.49: Pressure profiles in the wake of freely vibrating in crossflow geometry 7 at $t^* = 213$ for $Re = 500$ and $U^* = 5.23$. Top left : top view of the pressure profiles in the nodal plane at the peak of leading edge ; top right : top view of the pressure profiles in the nodal plane at the peak of trailing edge ; middle left : top view of the pressure profiles in the saddle plane at the valley of leading edge ; middle right : top view of the pressure profiles in the saddle plane at the valley of trailing edge ; bottom : side view of the pressure profiles about the plane of symmetry of the body at neutral position (perpendicular to crossflow direction).

to either side of the 'zero' band limit, respectively. From the side view of the pressure profiles in figure 6.49, spanwise regularity of pressure profiles for some period of undulations are observed in the near-field wake about the plane of symmetry (in the crossflow direction). After shedding from near-field low pressure region, regularity is hastily dispersed. In comparison, the spanwise regularity of the pressure profiles of geometry 4 and 5 in figures 6.42, 6.45 and their subsequent regularity (in propagation) indicated a different nature of the wake configuration. However, from figure 6.45, despite their spanwise regularity, subsequent propagation and its corresponding pressure profiles of geometry 5 indicated a less regular pressure along the spanwise direction (as compared to the highly regular corresponding profile in the wake of geometry 4). As such, it can be deduced that the major axis undulations perturbed the flow in such a way that a less regular and higher spanwise flow velocity is observed (as compared to perturbation due to the minor axis undulations). Consequently, less regularity is observed in the pressure profiles along the length of the geometries. In congruence, as geometry 7 consisted of both major and minor axis undulations, lesser regularity is expected due to the higher prevailing spanwise flow velocity. Figure 6.50 show the spanwise flow velocity in the wake of freely vibrating geometry 4, 5 and 7 at their peak spanwise flow velocity occurrences for $t^* = 185, 200, 213$ corresponding to each geometry, respectively. From figure 6.50, confirmation on the irregularity and higher spanwise flow velocity of freely vibrating in crossflow geometry 5 and 7 are obtained. The spanwise flow velocity of freely vibrating geometry 4 is observed to highly uniform with diagonally-similar velocity vectors occurring at every nodal plane of the geometry. However, such regularity is not observed for freely vibrating geometry 5 and 7, as every nodal and saddle planes of the geometry are observed to result in different spanwise velocity profile. Higher maximum (absolute) spanwise flow velocity is also observed in geometry 5 and 7 as compared to geometry 4. Hence, the smaller amplitude of VIV of geometry 7 (and therefore, geometries 6 and 8) as compared to geometries 3, 4 and 5 are mainly contributed by the lower minimum pressure (vortices) shed from the low pressure region in the near-field wake of the geometry. However, question on the origin of the higher pressure in the pressure profiles (near-field and minimum pressure region) of geometry 7 (and therefore, geometries 6 and 8, as compared to geometries 3, 4, and 5) is found to remain despite the current understanding on the lower amplitude of VIV. In order to understand the higher pressure in the pressure profile of geometry 7 as compared to geometry 5, analyses on the spanwise flow velocity profiles are necessary. Figure 6.51 and figure 6.52 illustrates the spatial evolution of the

spanwise flow velocity profile in the wake of the freely vibrating in crossflow geometry 5 ($t^* = 200$, and $U^* = 5.22$) and geometry 7, respectively. Through comparison between figure 6.51 and figure 6.52, it can be found that certain regularity is still observed in the near-field of the wake of a freely vibrating geometry 5 at resonance and almost inexistent for geometry 7. From figure 6.51, regularity of the wake for each period of undulations is observed for geometry 5 up to $x/D = 4.6$ downstream from the centroid of the geometry. In contrast, with reference to figure 6.52, regularity of spanwise flow velocity is only momentarily observed in the wake of a freely vibrating geometry 7 at $x/D = 4.0$. Subsequent emergence of periodicity of spanwise flow velocity for geometry 7 is observed to occur in combination with significant irregular spanwise flow velocities. From further comparison between figure 6.51 and figure 6.52, it can also be observed that the spanwise flow velocity in the wake of a freely vibrating geometry 7 is found to occur earlier at $x/D = 1.8$ (as compared to geometry 5, at $x/D = 2.6$). Interestingly, strong spanwise flow velocity is observed to persist in the wake of a freely vibrating geometry 7 up to, at least, $x/D = 10.4$. In the wake of geometry 5, the spanwise flow velocity is observed to weakened significantly at $x/D = 9.6$. This indicated the stronger dissipation of spanwise flow velocity (kinetic energy of the fluid) which lead to higher pressure in the area of dissipation. As such, with reference to figures 6.45, 6.49, it can be observed that the pressure field beyond the near-field low pressure region is significantly higher in the wake of geometry 5 as compared to geometry 7. This area of higher pressure is found to coincide with the vicinity of highest spanwise flow velocity. This indicated that efficient spanwise flow velocity dissipation led to strong pressure build-up in the wake of geometry 5. In comparison, from figure 6.52, it can be found that spanwise flow velocity is introduced earlier in the wake of geometry 7 (at $x/D = 1.8$). As such, pressure build-up is initiated in the vicinity of the body due to annihilation of the spanwise flow velocity (regardless of the efficiency). Despite the earlier (spatially) initiation of pressure build-up, the dissipation of spanwise flow velocity (and therefore, corresponding pressure build-up) in the wake of geometry 7 is relatively inefficient due to the irregular occurrence of the spanwise flow velocity. As such, pressure build-up in the wake of geometry 7 is found to occur gradually (as compare to geometry 5). Hence, despite the greater physical perturbation, higher near-field pressure is observed in the wake of geometry 7 (as compared to geometry 5) with similar magnitude of pressure in the far-field (downstream end of simulation domain) wake (refer to figure 6.45, 6.49). From comparison between figure 6.51 and figure 6.52, it is also important to note that a larger number of spanwise

Figure 6.50: Spanwise flow velocity profile for various freely vibrating geometries at freestream flow of $Re = 500$. Left : geometry 4 at $t^* = 185$, $U^* = 4.76$, and $x/D = 3.8$; middle : geometry 5 $t^* = 200$, $U^* = 5.22$, and $x/D = 5.43$; right : geometry 7 at $t^* = 213$, $U^* = 5.78$, and $x/D = 5.97$.

perturbation is observed in the wake of geometry 7 with greater persistency as the flow progresses downstream (as compared to the wake of geometry 5).

As geometry 7 is distinctively different from a circular cylinder, elliptical cylinder and geometries 3, 4, and 5, analysis on the instability of its wake would provide insight into the effect of dual-axial undulations on the flow around such body. Figures 6.53 illustrates the vortex cores and tangential flow velocity in the wake of a freely vibrating geometry 7 at $U^* = 5.78$, $x/D = 7.1$ and $t^* = 213$. From a quick observation of figure 6.53, a lack of spanwise regularity is apparent from the arrangement of the vortex cores. However, a detailed observation revealed certain regularity along the nodal planes of geometry 7. In particular, the dominant normalised wavelengths are 0.88, 1.18, and 1.88 (corresponding to the $\lambda_{span}/\lambda = 0.25$, 0.33 and 0.52, respectively). The wake in the nodal planes of geometry 7 is marked with a three-vortices pair. The three-vortices pair is observed to consist of either two-vortices of negative polarity (clockwise rotation) and a vortex of positive polarity (counter-clockwise rotation) or vice versa. Regularity in saddle planes is not observed with possibility of prevalence of single vortex or two-vortices pair. A slight variability of wake topology is observed in the wake of a freely vibrating geometry 7, as shown in figure 6.54.

Such wake topology is observed to consist of more regular U-shaped vortical braids with addi-

Figure 6.51: Rear view of the spanwise flow velocity of the flow around a freely vibrating in crossflow geometry 5 at $Re = 500$, $t^* = 200$, and $U^* = 5.22$ for various spatial positions. All plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry (grey). Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 3.2, 3.8, 4.6, 5.2, 5.6, 6.0, 6.4; bottom-row from left to right: 7.2, 7.8, 8.2, 8.6, 9.0, 9.6, where D is the characteristic diameter in the simulation (minor diameter of vibrissa-like geometry).

tional vortical braids. As such, additional analysis to determine possible variability is performed. Figure 6.55 illustrates the vortex cores and tangential flow velocity in the wake of a freely vibrating geometry 7 at $U^* = 5.78$, $x/D = 7.3$ and $t^* = 265$. Through comparison and analysis of the polarity of the vortical braids, despite the perceived irregularity, certain features are observed to prevail in the wake of geometry 7. The configuration of polarity in the nodal-to-saddle planes along the span (from top to bottom with periodic boundary condition at the ends) of geometry 7 in figure 6.53 is observed to be a three-vortices pair (nodal) and single vortex (saddle), a two-vortices pair (nodal) and single vortex (saddle), a three-vortices pair (nodal) and a two-vortices pair (saddle), and a three-vortices pair (nodal) and a single vortex (saddle). Correspondingly, the configuration of polarity in the nodal-

Figure 6.52: Rear view of the spanwise flow velocity of the flow around a freely vibrating in crossflow geometry 7 at $Re = 500$, $t^* = 213$, and $U^* = 5.78$ for various spatial positions. All plots share similar colouring legend as the bottom right-most figure. Plots are made slightly transparent to allow for visibility of the vibrissa geometry (grey). Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 1.8, 2.4, 3.2, 4.0, 4.4, 4.8 ; middle-row from left to right : $x/D = 5.4, 6.0, 6.4, 7.0, 7.4, 7.6, 7.8$; bottom-row from left to right : 8.2, 8.4, 8.8, 9.6, 10.0, 10.4, where D is the characteristic diameter in the simulation (minor diameter of vibrissa-like geometry).

Figure 6.53: Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 7 in the plane-cut (perpendicular to streamwise) at $x/D = 7.1$, $t^* = 213$ and $U^* = 5.78$. Top left: spanwise overview ; top right: enlarged view of the first one and a quarter period of undulations ; bottom left: enlarged view of the subsequent 1.5 period of undulations ; bottom right: enlarged view of the final 1.25 period of undulations. Legend: orange area denotes vortex cores.

to-saddle planes along the span (from top to bottom) of geometry 7 in figure 6.55 is observed to be a two-vortices pair (nodal) and single vortex (saddle), a three-vortices pair (nodal) and single vortex (saddle), a two-vortices pair (nodal) and a single vortex (saddle), and a two-vortices pair (nodal) and a two-vortices pair (saddle). Direct comparison on dominant normalised wavelengths in the plane $x/D = 7.1$ (refer to figure 6.53) and 7.3 revealed uniformly higher normalised wavelengths at the latter plane in the wake of the geometry 7. These wavelengths are 1, 1.48, and 2.39 (corresponding to $\lambda_{span}/\lambda = 0.28, 0.41$ and 0.66). For further analysis, it is necessary to recall that the creation of a two-vortices pair is similar to the nature of a single vortex in open-flow. This is because the a single

Figure 6.54: Side view of the vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 7 at $Re = 500$, $t^* = 265$ and $U^* = 5.78$.

vortex in an open-flow would result in another vortex (or vortices) of net opposite polarity. However, the strength and quantity of such vortex (or vortices) are indeterminable. As such, despite the vortex core and tangential velocity representation, a single vortex is actually an existence with many smaller vortices of opposite polarity (as the flow at $x/D = 7.1$ and $x/D = 7.3$ is sufficiently far from the body to simulate an open-flow condition). As such, it is reasonable to note that the saddle planes in the wake of a freely vibrating geometry 7 would consist of a single vortex or a two-vortices pair with strong dependence on the net vorticity of the vortices in the neighbouring nodal planes. In turn, the vortices in nodal plane would either form a two-vortices pair or three-vortices pair with dependence on the net vorticity of the vortices in the neighbouring saddle planes. Such dependency and alternation of form are observed to explain the consistency of the wake topology in the vibrissa-like geometry with dual-axial undulations.

The old vibrissa geometry (geometry 9) is also observed to have similar wake topology as geometry 6, 7, and 8, at their respective resonance. As such, similar explanations on the consistency of its wake topology and its small amplitude of VIV are applicable. Figure 6.56 shows the vortex cores in the wake of a freely vibrating old vibrissa geometry at $Re = 500$, $U^* = 4.73$, and $t^* = 171$. The wake of old vibrissa geometry in figure 6.56 exhibited strong similarity to the wake of a stationary old vibrissa geometry (depicted in figure 6.21). However, it is important to note that an in-depth study on

Figure 6.55: Vortex cores and tangential velocity vector of a freely vibrating in crossflow geometry 7 in the plane-cut (perpendicular to streamwise) at $x/D = 7.3$, $t^* = 265$ and $U^* = 5.78$. Top left: spanwise overview ; top right: enlarged view of the first one and a half period of undulations ; bottom left: enlarged view of the subsequent 1.25 period of undulations ; bottom right: enlarged view of the final 1.25 period of undulations. Legend: orange area denotes vortex cores.

the response of the old vibrissa geometry at various reduced velocities would be necessary for proper understanding on the response of the harbour seal vibrissae for various boundary conditions. Such detailed analysis for old and new vibrissa geometries for 1D and 2D freely vibrating setup will be described in chapter 7.

In contrast to old vibrissa geometry, despite their small amplitude of VIV (refer to figure 6.34), relatively significant variation between the wake topology of a freely vibrating new vibrissa geometry and stationary vibrissa geometry are observed. Figure 6.57 depicts the vortex cores in the wake of a freely vibrating new vibrissa geometry for $Re = 500$, $U^* = 5.97$ and $t^* = 194$. Despite small

Figure 6.56: Depiction of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 9 (old vibrissa geometry) at $Re = 500$, $t^* = 171$, and $U^* = 4.73$. Top left : isometric view ; top right : side view ; bottom : top view.

congruence observed when comparing the wake topology of a freely vibrating and stationary new vibrissa geometry, an overall distinction could be observed in their wake topology. Upon careful observation of figure 6.34, it can be observed that the amplitude of VIV of the new vibrissa geometry is significantly smaller than the amplitude of VIV of geometry 6, 7, 8 and 9. Similar to earlier explanations, the pressure profiles in the wake of the new vibrissa geometry is believed to be the origin of such minuscule amplitude of VIV. However, from observation of the pressure profiles on geometry 7 and 9, significant reduction on the pressure difference between the leading edge and trailing edge of geometry 7 and 9 are observed. Further reduction in the corresponding pressure difference of the new vibrissa geometry (as compared to geometry 7 and 9) would explain the significantly lower amplitude of VIV but it would simultaneously require radical modifications of the wake topology and pressure profiles in the wake of the freely vibrating new vibrissa geometry. Figure 6.58 depicts the iso-surfaces of the pressure profiles in the wake of a freely vibrating new vibrissa geometry at $Re =$

500, $t^* = 194$ and $U^* = 5.97$.

From figure 6.58, spanwise regularity of the pressure profiles can be observed. In the midst of the span of the new vibrissa geometry, difference in pressure profiles is observed. Along the saddle planes of the new vibrissa geometry, spatially interconnected V-shaped minimum pressure braids are observed to propagated downstream. Only in the middle section of the new vibrissa geometry (which is a saddle plane) that such interconnected braids are not observed. It should be noted that the minimum pressure braids are diagonally oriented in the crossflow and spanwise directions, as seen in the top view in figure 6.58. Despite the continuous stream of interconnected minimum pressure region braids in its neighbouring saddle planes, no braids are observed in the saddle plane at the mid-span of the new vibrissa geometry, regardless of the simulation time. Such occurrence maybe related to the lower minimum pressure region occurring in the mid-span of the geometry as compared to the minimum pressure region in its neighbouring saddle planes. The saddle plane in the mid-span of the geometry is also observed to lead in vortex shedding with a higher high-pressure region countering the minimum pressure region in its spatial evolution. Despite the observed desynchronisation of vortex shedding along the spanwise length, lower minimum pressure region is not observed. The length of the near-field low pressure region of the new vibrissa is also observed to be relatively similar to the corresponding region of the old vibrissa geometry and geometry 7 (refer to figure 6.49). As such, a more careful analysis of the pressure profiles is necessary in order to reveal the origin of the significantly lower amplitude of VIV of the new vibrissa geometry. Figure 6.59 shows the spatial evolution of the pressure profiles in the wake of a freely vibrating new vibrissa geometry at $Re = 500$, $t^* = 194$, and $U^* = 5.97$.

Although simulations are performed for a downstream length of up to $x/D = 17.7$, only pressure profiles in the plane up to $x/D = 10.2$ in figure 6.59 are depicted as the origin of the significantly lower amplitude of VIV are well identified through them. From figure 6.59, it can be observed that the significant reduction in amplitude of VIV is due to the alternating position in the crossflow direction of the minimum pressure regions (and therefore, their corresponding shedding) in the wake of a freely vibrating new vibrissa geometry. As periodic boundary conditions are applied on the spanwise ends of the geometry, from figure 6.59, it can be observed that the minimum pressure region at the first nodal plane (of the minor axis, on other occasions, the nodal and saddle planes definitions are based on the major axis) occurred on the positive Y-axis (right side) at $x/D = 2.6$ and 3.8 with an alternating

Figure 6.57: Depiction of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating geometry 10 (new vibrissa geometry, left) at $Re = 500$, $t^* = 194$, and $U^* = 5.97$ and stationary new vibrissa geometry (right, reprint from figure 6.22). Top left : isometric view ; top right : side view ; bottom : top view.

Figure 6.58: Translucent iso-surface of the pressure profiles in the wake of a freely vibrating geometry 10 at $Re = 500$, $t^* = 194$, and $U^* = 5.97$. Top left : isometric view ; top right : side view ; bottom : top view.

left and right occurrence for the minimum pressure region in the subsequent nodal planes (of the minor axis). It is important to note that the nodal plane of the minor axis coincided with saddle plane of the major axis and vice versa. As such, any form of referencing of nodal plane to a respective axis is translatable to another axis. Interestingly, due to the larger amplitude of undulations in the major axis, the true nodal planes of the geometry is found to coincide with the nodal plane of the major axis. In its subsequent evolution, the minimum pressure regions of the new vibrissa geometry is observed to maintain an alternating positions (in the crossflow direction) relative to its neighbours (in the nodal planes of the minor axis). During its spatial evolution, each minimum pressure region of a nodal plane (of minor axis) is observed to alternate between positive and negative y-axis location relative to the centroid of the geometry. Such spanwise alternating position is found to be the main origin of the reduction in the amplitude of VIV of the new vibrissa geometry (unlike for stationary new vibrissa geometry where alternation occurred but less regularly). However, in addition to the

spanwise alternating position, a phase difference between the vortex shedding (stages of development of the minimum pressure region) is observed along the spanwise length of the new vibrissa geometry (refer to figure 6.59). Such phase difference of vortex shedding is also observable in figure 6.57 which represented desynchronisation of vortex shedding along the length of the geometry. As such, non-uniform forcing of the geometry along its length resulted in further reduction of amplitude of VIV on the new vibrissa geometry.

From observation of the top view of the wake of a freely vibrating new vibrissa geometry, significant spanwise mixing (with regularity) is observed with a linear increase in the lateral width of the wake. In the wake of the new vibrissa geometry, von *Kármán* street-like features are no longer observed as the vortical tubes are deconstructed and are replaced with lumps of vortical braids on staggered arrangement in the spanwise direction. Although, some features of von *Kármán* street may still be observable for flow around geometry 6, 7, 8 and 9.

While observation of the secondary instability and vortical braids of stationary new vibrissa geometry has been performed, similar analysis is necessary for the wake of a freely vibrating new vibrissa geometry. This is due to the relatively significant difference between the wake topology of a freely vibrating and a stationary new vibrissa geometry. Figure 6.60 shows the cross-section of the vortex cores at various locations $x/D = 6.6, 8.9$ and 11.8 in the wake of the freely vibrating new vibrissa geometry for $Re = 500$, $t^* = 194$, and $U^* = 5.97$. From figure 6.60, despite the observed variation of the vortex cores at various plane-cuts (spatial positions), similarity of the vortical arrangements at each nodal plane (of minor axis) for a given lump of vortical braids are observable. Distinction of wake topology is observed for the nodal plane (of minor axis) located in the mid-span of the new vibrissa geometry. Regardless of their spanwise location, it is observed that all the bulk of vortical braids are found to be symmetrical about the peak of the nodal plane (of the minor axis). Such symmetry included the polarities of the vortical braids. Figure 6.61 shows the vortex core and tangential velocity vector of bulks of vortices in the nodal plane above the midspan of the new vibrissa geometry and at the midspan of the new vibrissa geometry for the plane-cut at $x/D = 8.9$. From figure 6.61, despite the difference in the identified vortex core in the wake of a freely vibrating new vibrissa geometry at the plane-cut of $x/D = 8.9$, analysis of the arrangement of the vortical braids resulted in strong similarity between the vortices in the first and second nodal planes. For the first nodal plane of minor axis of the geometry, from top to bottom, the polarities of the vortical braids are

Figure 6.59: Rear view of the spanwise flow velocity of the flow around a freely vibrating in crossflow geometry 10 at $Re = 500$, $t^* = 194$, and $U^* = 5.97$ for various spatial positions. All plots share similar colouring legend as the bottom right-most figure. Top-row from left to right: $x/D = 0$ (midpoint of vibrissa), 2.6, 3.8, 4.6, 5.0, 5.6, 6.0 ; bottom-row from left to right : $x/D = 6.6, 7.2, 8.2, 8.6, 9.4, 10.2$, where D is the characteristic diameter in the simulation (minor diameter of vibrissa-like geometry).

Figure 6.60: Vortex cores (based on λ_2 criterion) in the wake of a freely vibrating new vibrissa geometry for $Re = 500$, $t^* = 194$, and $U^* = 5.97$. From left to right : side view of the vortex cores and location of plane-cuts (perpendicular to streamwise direction), $x/D = 6.6$, 8.9 , and 11.8 ; cross-sectional view of vortex cores in plane-cut at $x/D = 6.6$; cross-sectional view of vortex cores in plane-cut at $x/D = 8.9$; cross-sectional view of vortex cores in plane-cut at $x/D = 11.8$.

identified as follow: positive - negative - positive - negative - positive - negative. For the second nodal plane of the minor axis of the geometry (also the mid-span of the geometry), from top to bottom, the polarities of the vortical braids are found to be as follows: positive - positive - negative - negative - positive - negative - positive. More vortical braids are observed in the mid-span of the geometry as expected due to the lower minimum pressure region observed in the corresponding location. Detailed figures of the vortex cores and tangential velocity vectors for each of the bulk of vortical braids of the new vibrissa geometry at plane-cuts $x/D = 6.6$, 8.9 and 11.8 could be found in appendix E. With reference to figure 6.61, it should be noted that the dominant normalised wavelengths are 1.67, 1.83, 2.0, 6.0 and 15.75 (corresponding to $\lambda_{span}/\lambda = 0.23, 0.25, 0.28, 0.83$ and 2.20 , respectively).

Although the current study do not perform in-depth instability analysis in the wake of each geometry, it is observed that the prevalence of more normalised wavelengths in the near-field wake of the structure resulted in greater reduction in VIV (or lower lift coefficient on stationary structures). This

Figure 6.61: Vortex cores and tangential velocity vectors of the new vibrissa geometry in the plane-cut at $x/d = 8.9$ for bulk of vortical braids in the first nodal plane of minor axis (left) and second nodal plane of minor axis (right), also the mid-span of the geometry. Top : overview of bulk of vortical braids ; middle : enlarged view of the upper half of the bulk of vortical braids ; bottom enlarged view of the lower half of the bulk of the vortical braids.

is particularly true when comparing the number of identified normalised wavelengths in the wake of geometry 10 (new vibrissa geometry) to other simulated geometries, where more normalised wavelengths are observed in the wake of stationary and freely vibrating geometry 10 as compared to the corresponding wavelengths in the wake of other simulated geometries.

6.3 Simulations of 2D Vibration Geometry

From previous studies on VIV, significant difference in the responses are observed between a 1D and 2D freely vibrating circular cylinder [23]. Notably, a larger amplitude of crossflow motion (up to 1.2D) is observed as well as the third-harmonic frequency response of the circular cylinder in the crossflow direction. As the boundary conditions of the seal vibrissa could allow 2D free vibration, assessment on the responses of the geometries to 2D free vibration are necessary. The simulations are performed on all geometries (refer to figure 6.1) at $Re = 500$ with equivalent mass ratios in streamwise and crossflow direction ($m^* = 1.04$). The streamwise natural frequency is preset to twice of the crossflow natural frequency due to the nature of the occurrence of the vortex shedding. As the vortex shedding occurred in one crossflow oscillation, two vortices are shed from the body simultaneously. Due to the shedding of these vortices, two oscillations in streamwise direction are impacted on the body within one crossflow oscillation of the body. Hence, these explained the doubling of the streamwise natural frequency as compared to the crossflow natural frequency of the body. Figure 6.62 shows the streamwise and crossflow responses of the flow around the geometries at $Re = 500$ for freely vibrating in streamwise and crossflow body. From figure 6.62, it can be observed that the crossflow amplitude of oscillation of the 2D vibrating circular cylinder is the largest at its resonance (as compared to other geometries). This is expected due to the significantly larger crossflow amplitude of oscillation in a 1D freely vibrating circular cylinder (as compared to other corresponding responses of the geometries). This larger crossflow amplitude of circular cylinder (as compared to other geometries) is due to the lower minimum pressure region observed near the vortices as well as the further release of the vortices in the wake of the 2D vibrating circular cylinder. Through the setup of free vibration in streamwise and crossflow oscillations, a larger crossflow amplitude of oscillation is observed for a circular cylinder (as compared to 1D vibration). From previous studies, such result is expected due to the greater power transfer from fluid to the body (C_{Lv}). Figure 6.63 shows the

Figure 6.62: Responses of various geometries (refer to figure 6.1) to flow around these freely vibrating in streamwise geometries at their respective near-resonance conditions $Re = 500$, $m^* = 1.04$ and $f_{nx}/f_{ny} = 2.0$. Top left : streamwise and crossflow amplitudes ; top right : streamwise and crossflow frequency responses ; bottom left : normalised streamwise and crossflow response frequencies ; bottom right : ratio of streamwise to crossflow response frequencies.

wake topology of a 2D freely vibrating circular cylinder at $Re = 500$, $U^* = 5.2$, $t^* = 246$.

From figure 6.63, it can be observed that no vortical braids are observed in the wake of a 2D vibrating circular cylinder. However, through λ_2 identification, a layer of flow connecting the vortex tubes are observed. Such layer is suspected to consist of vortical braids. Figure 6.64 shows the cross-sectional view of the wake and tangential velocity vectors of a 2D vibrating circular cylinder at a plane-cut of $x/D = 2.7$ (across the vortical braids) and about the plane of symmetry about the neutral plane (across the vortex tube). Analysis of the wake of a 2D vibrating circular cylinder indicated the absence of a vortical braids in its wake. Such finding reinforced the indication of the larger crossflow amplitude of oscillation. This is due to the strengthening of the pressure force due to the shedding of the vortices on the body. Such strengthening of the pressure force is found to be a consequence of the strengthening of the vorticity in the vortex tubes due to the absence of the

vortical braids. Simultaneously, the vortical arrangement is observed to form a '2S' shedding pattern instead of the 'P+S' shedding pattern observed in the wake of a freely vibrating in crossflow circular cylinder. Figure 6.65 shows the pressure profiles in the wake of a 2D vibrating circular cylinder at $Re = 500$, $m^* = 1.04$, $t^* = 246$ and $f_{nr} = 2.0$, where f_{nr} is the ratio of the streamwise natural frequency to crossflow natural frequency of the system. From figure 6.65, it can be observed that the lower minimum pressure field is observed in the wake of 2D vibrating circular cylinder (as compared to stationary and 1D vibrating circular cylinder). Such stronger minimum pressure field is due to the absence of the vortical braids connecting the vortex tubes in the wake of a 2D vibrating circular cylinder. This further concluded the absence of the vortical braids as well as the increase in the amplitude of oscillation in the wake of a 2D vibrating circular cylinder (as observed in figures 6.63, 6.64). The absence of vortical braids also indicated the stability of the vortex tubes in the wake of the 2D vibrating circular cylinder. Such stability can be observed from the absence of undulations on the vortex tubes in the wake of a 2D vibrating circular cylinder. Interestingly, this stability is inborn from the inclusiveness of free streamwise oscillation, in addition to free crossflow oscillation, which introduced greater overall oscillation as compared to 1D vibrating circular cylinder.

As aforementioned, the natural frequency ratio (streamwise to crossflow) for the simulation setup is set to correspond to the frequency ratio between the drag and lift force on the bodies due to the nature of vortex shedding. As such, it is important to note the trajectory of motion of a 2D vibrating circular cylinder. Such trajectory would strongly indicate the type of response of the circular cylinder to the natural frequency of the setup. Figure 6.66 shows the plot of streamwise and crossflow motions (motion trajectory) of a 2D vibrating circular cylinder at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. From figure 6.66, an '8-figure' can be observed for each oscillation cycle of the crossflow motion of a 2D vibrating circular cylinder. This finding is consistent with the natural frequency ratio setup of the 2D vibrating circular cylinder. This is because an '8-figure' indicated that two streamwise oscillations occurred in a cycle of crossflow oscillation. Hence, correspondingly, two oscillation of drag coefficient occurred in one oscillation of the lift coefficient, which is relevant to the nature of vortex shedding (two vortices shed in one cycle of crossflow oscillation, resulting in a streamwise oscillation for each vortices shed). The '8-figure' motion trajectory is observed to occur consistently during the steady period of response of the 2D vibrating circular cylinder. Slight variations in the amplitude of streamwise and crossflow motion can be observed with similar period for each of the

Figure 6.63: Vortex cores (based on λ_2 criterion) in the wake of a freely vibrating in streamwise and crossflow circular cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$. Top left : isometric ; top right : side view ; bottom : top view.

streamwise oscillation (equal upper and lower half of the '8-figure').

From observations of figure 6.62, it is interesting to find that the cross flow amplitude of oscillations of a 2D vibrating elliptical cylinder (≈ 0.04) is significantly lower than the streamwise amplitude of oscillation (≈ 0.36). Such response is found to be a reversal of the response of a 2D vibrating circular cylinder. However, from figure 6.62, the streamwise and crossflow response frequency ratios maintained a value of 2.0, which is equivalent to its natural frequency ratio (streamwise to crossflow). As such, careful observation is necessary in order to reveal the nature of the amplification of the streamwise oscillation. Correspondingly, through comparison between the streamwise and crossflow amplitudes of oscillation of geometry 3 (vibrissa-like geometry with no major axis undulations with axis ratio of 0.44) and geometry 4 (vibrissa-like geometry with no major axis undulations with axis ratio of 0.54), it can also be observed that the more 'elliptical' geometry (geometry

Figure 6.64: Vortex cores and tangential velocity vectors in the wake of a 2D vibrating circular cylinder at $Re = 500$, $U^* = 5.2$, and $t^* = 246$. Top left : cross-sectional view (perpendicular to streamwise) at $x/D = 2.7$ (across vortical braids) ; top right : enlarged view of the cross-sectional view corresponding to figure on the left ; bottom left : cross-sectional view (perpendicular to crossflow) about the plane of symmetry of the geometry at the neutral axis (across vortex tube) ; bottom right : enlarged view of the cross-sectional view corresponding to figure on the left.

3, lower axis ratio) exhibited a larger streamwise amplitude of oscillation to crossflow amplitude of oscillation. However, such response is not observed for other geometries with elliptical based-cross section (geometry 4 to 10). Figure 6.67 shows the wake topology of a 2D vibrating elliptical cylinder at $Re = 500$, $m^* = 1.04$ and $t^* = 264$ on its resonance. From figure 6.67, it can be observed that the wake topology of a 2D vibrating elliptical cylinder exhibited a '2S' vortex shedding properties. No braids are observed from the vortex cores identified with λ_2 criterion and from depiction of their tangential velocity vectors. However, significant lateral spacing between the vortex tubes are observed

Figure 6.65: Pressure field in the wake of a freely vibrating in streamwise and crossflow circular cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$. Top left : isometric ; top right : side view ; bottom : top view.

Figure 6.66: Motion trajectory of a 2D vibrating circular cylinder during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Legend: Darker green - earlier time, brighter green - later time.

($\approx 3D$, center-to-center spacing) as compared to stationary and 1D freely vibrating elliptical cylinder. Such difference in spacing is due to the significant crossflow and streamwise motion in the wake of a 2D vibrating circular cylinder. Significantly lower minimum pressure is observed in the wake of a 2D vibrating elliptical cylinder as compared to the corresponding geometry for stationary and freely vibrating in crossflow setup. Figure 6.68 shows the corresponding isometric view of the pressure field and streamwise velocity of the elliptical cylinder at similar setup to figure 6.67. From figure 6.68, significant lower minimum pressure field (compare to figure 6.39) is observed for 2D vibrating elliptical cylinder. This is due to the significant streamwise and crossflow motion of the elliptical cylinder. Both the streamwise and crossflow motion would add kinetic energy (vorticity) to the shed vortices resulting in a significantly higher velocity of the fluid particles of the vortex tubes, as observed in figure 6.68. Consequently, lower minimum pressure region is observed in the vortex tubes. Uniquely, due to the difference in added mass in streamwise and crossflow direction, certain directional preference for the motion of response occurred. As the added mass in the streamwise direction is smaller as compared to the added mass in the crossflow direction (due to smaller projected diameter), preference of motion response due to the vortex shedding is found in the streamwise motion in order to obey the minimum energy principle. Hence, while larger crossflow motion is observed in the motion response of a 2D vibrating elliptical cylinder, a significantly larger streamwise amplitude is seen. Referring to figure 6.62, it can be deduced that not all elliptical based geometries sustain a larger streamwise motion response (as compared to their corresponding crossflow response). Only elliptical cylinder and geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) is observed to response with greater streamwise motion as compared to crossflow. Hence, the author acknowledged the existence of a threshold limit of axis ratio (which coincided with the definition of a body as bluff or streamlined), which is found to be within the upper vicinity of 0.5 (> 0.5 and ≤ 0.54). Above such threshold value, the motion response of the structure would be larger in crossflow and below such threshold value, a dominant streamwise motion would dominate the 2D vibratory motion of the structure. With reference to figure 6.62, consistent response frequency are observed despite the significantly higher streamwise amplitude of oscillation as compared to crossflow amplitude of oscillation. Figure 6.69 shows the motion trajectory of a 2D vibrating elliptical cylinder at $Re = 500$, $U_y^* = 5.2$ and $m^* = 1.04$ during its steady period of response.

From figure 6.69, '8-figure' motion trajectories can be observed in the motion response of a 2D

Figure 6.67: Vortex cores (based on λ_2 criterion) in the wake of a freely vibrating in streamwise and crossflow elliptical cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$. Top left : isometric ; top right : side view ; bottom : top view.

Figure 6.68: Isometric view of the pressure field (left) and streamwise velocity (right) contours in the wake of a freely vibrating in streamwise and crossflow elliptical cylinder at $Re = 500$, $U_y^* = 5.2$, and $t^* = 246$.

Figure 6.69: Motion trajectory of a 2D vibrating elliptical cylinder during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Legend: Darker green - earlier time, brighter green - later time.

vibrating elliptical cylinder. However, due to the more significant streamwise motion (as compared to crossflow motion), the 8-figure is found to be significantly distorted with perceived sharp corners on its turning points. Such motion trajectory is expected as the elliptical cylinder is observed to maintain a streamwise to crossflow response frequency ratio of 2.0 (equivalent to the natural frequency ratio of the simulation setup). For further confirmation, analysis on the wake topology of a 2D vibrating geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) will be performed. Similar to the response of a 2D vibrating elliptical cylinder, a 2D vibrating geometry 3 is observed to sustain a larger streamwise oscillation to its crossflow oscillation. The streamwise to crossflow response frequency ratio is observed to be similar to the natural frequency ratio of the simulation setup for the geometry. Figure 6.70 depicted the vortex cores in the wake of a 2D vibrating geometry 3 at $Re = 500$, $U^* = 4.8$, and $t^* = 234, 246$ and 284 .

Despite the stability of the motion responses of a 2D vibrating geometry 3, slight variation in the configurations of the vortical braids are observed. Switching between three vortical patterns are observed in the wake of a 2D vibrating geometry 3. The three vortical patterns are presented in figure 6.70. These patterns are differentiated through the spanwise arrangement of the vortical braids with similar arrangements of vortical tubes. Hence, based on the spanwise arrangement of the vortical braids, the wake patterns can be categorised as (from top to bottom of figure 6.70) three-ribbed, $\cap$ -shaped, and two-ribbed vortical arrangements. The variation of vortical arrangement is observed to follow such order: three-ribbed to $\cap$ -shaped to three-ribbed and two-ribbed to three-ribbed to

cap-shaped and continuing after the first two order of vortical arrangement. Details of instability and analysis of the vortical formations in the wake of a 2D vibrating geometry 3 can be found in appendix E. From analysis of the wake topology in the wake of a 2D vibrating geometry 3, it is found that the three-ribbed vortical braids is actually similar to the two-ribbed vortical braids with similar polarity. The mid-rib vortical braid is found to be connected to one of the side vortical braids. Hence, despite its depiction of three-ribbed vortical braids based on λ_2 identification, two vortical braids are observed through analysis on the tangential velocity vector. However, the subsequent evolution of each wake (two-ribbed and three-ribbed) showed that the λ_2 criterion correctly identified the minute difference between the two vortical configuration, which will be shown to evolve differently. Each of these vortical braids are found to be centred about the nodal plane of the minor axis of geometry 3. However, subsequent evolution of these vortical arrangements resulted in a four-vortices pair. In this state, different pairing configuration and positioning is observed. While the three-ribbed vortical braids is found to evolve into a four-vortices pair with symmetry (their centres in square configuration) located at the nodal plane of the minor axis, the two-ribbed vortical braids is found to evolve into a four-vortices pair with slight lack of symmetry (their centres in rhombic configuration) located at the saddle plane of the minor axis of the geometry. Hence, despite their resemblance in the near-field region of the geometry, their subsequent evolution illustrated significant variation. It should also be noted that the two-ribbed and three-ribbed vortical braids arrangement are strongly correlated in temporal evolution (as mentioned above). The $\cap$ -shaped vortical braids is found to consist of a persistent pair of vortical braids with similar polarity and location to the pair of vortical braids observed in the three-ribbed and two-ribbed vortical pairs.

Similar to the response of a 2D vibrating elliptical cylinder, a greater streamwise oscillation is observed for a 2D vibrating geometry 3 as compared to their corresponding crossflow oscillation. This is due to the streamlined shape of these geometries as compared to other geometries. Such streamlined shape defined the manner of power transfer between the fluid and the body. Notably, owing to the significant difference in the streamwise and crossflow added mass (estimated through potential flow, 1.0 for crossflow and 0.25 for streamwise based on major diameter) and minimum energy principle, the pressure forcing is observed to be dominant in the streamwise direction as compared to the crossflow. Figure 6.71 shows the streamwise and crossflow added mass coefficients (Cm_x and Cm_y , respectively), lift coefficient in phase with velocity of crossflow oscillation (CLv_y) and drag

Figure 6.70: Arrangement of vortex cores (based on λ_2 criterion) in the wake of a freely vibrating in stream-wise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) at $Re = 500$, $U_y^* = 4.8$, and $t^* = 234$ (top), 246 (middle), and 284 (bottom). Left: isometric view ; right : side view.

coefficient in phase with velocity of streamwise oscillation (CLv_x) for 1D and 2D freely vibrating geometries (refer to figure 6.1). From figure 6.71, it can be observed that 2D vibrating geometry 2 (elliptical cylinder) and geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) exhibited opposite signs for its added mass coefficients. Notably, their Cm_y is positive (perceived heavier mass) and Cm_x is negative with significantly negative CLv_x . While a negative CLv_x indicated a significant damping effect in the streamwise motion of the structure (power transfer from body to fluid), oppositely signed added mass coefficients indicated preferential motion of the fluid particles in the vicinity of the geometries. Primarily, despite the equivalence of mass ratio in streamwise and crossflow directions, significantly lighter perceived mass is observed in the streamwise direction. This indicated a preferential movement of the fluid flow around the geometries in the streamwise direction as compared to the crossflow direction. Significant difference between streamwise and crossflow added mass coefficients of a 2D vibrating elliptical cylinder is observed, which provided further confirmation on the preference of the fluid flow around the body (based on minimum energy principle) and therefore, a significantly larger streamwise amplitude of oscillation as compared to its crossflow amplitude of oscillation. Interestingly, from figure 6.71, increase in Cm_y is observed for 2D vibrating geometry 2 and 3 as compared to their corresponding 1D vibrating geometries. In comparison, with reference to figure 6.62, geometry 4 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.54) is observed to respond with a larger crossflow amplitude of oscillation as compared to its streamwise amplitude of oscillation. However, both amplitudes are observed to be significantly lower than the amplitudes of oscillation of geometry 3 (at least an order of magnitude smaller). A compliance of response frequency ratio of 2.0 is also observed in the motion response of geometry 4. However, similar to the frequency responses of geometries 2 and 3, frequency ratios of 0.8 to 0.9 are observed indicating a close to lock-in response for their streamwise and crossflow motions. From figure 6.71, it can also be observed that the CLv_x of geometry 4 is near-zero indicating a balance of power transfer between the body to fluid and vice versa (for geometries 2 and 3, significant power transfer observed from body to fluid). However, the Cm_x is observed to be slightly larger than that of geometry 3. This is as expected due to the bluffer configuration of geometry 4 as compared to geometry 3 (larger axis ratio on geometry 4). However, a sign reversal is observed on the Cm_y of geometry 4 as compared to the corresponding coefficient of geometries 2 and 3. Consequently, Cm_x and Cm_y of geometry 4 show similar signs, which indicated weaker

Figure 6.71: Added mass coefficients and force coefficients in phase with velocities for 1D freely vibrating (crossflow) and 2D freely vibrating (streamwise and crossflow) geometries. Legend: blue cross - added mass coefficient of crossflow motion for 2D freely vibrating geometries ; red square - added mass coefficient of streamwise motion for 2D freely vibrating geometries ; black diamond - lift coefficient in phase with velocity of crossflow motion for 2D freely vibrating geometries ; green inverted triangle - lift coefficient in phase with velocity of streamwise motion for 2D freely vibrating geometries ; teal triangle - added mass coefficient of crossflow motion for 1D freely vibrating geometries ; dark yellow circle - lift coefficient in phase with velocity of crossflow motion for 1D freely vibrating geometries.

preferential movement of the fluid particles in the vicinity of the body as the body undergoes motion. As such, a more balanced power transfer between the fluid and the body in the streamwise and cross-flow directions is observed. Figure 6.72 illustrates the motion trajectory of geometries 3 and 4 for $Re = 500$ and $U^* = 4.8$ and 4.76 , respectively.

Despite the equivalence of the response frequency ratio between geometry 3 and geometry 4, significant motion trajectory is observed between them in figure 6.72. Owing to the regularity of its streamwise motion and relative regularity of its crossflow motion, a right-skewed '8-figure' is observed in the motion trajectory of geometry 3. However, as the streamwise motion of geometry 4 is found to lack temporal regularity, significant cycle-to-cycle variation of its motion trajectory is observed. However, due to its locked dominant response frequency ratio of 2.0, '8-figure' motion trajectory is expected with slight observability (due to significant temporal variation in skewness of geometry 4) from figure 6.72.

Analyses on the wake of other 2D vibrating geometries (geometries 4 to 10) revealed similar

Figure 6.72: Motion trajectory of a 2D vibrating geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44 -left) and geometry 4 (vibrissa-like geometry with no minor axis undulations and axis ratio of 0.54 - right) during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Legend: Darker green - earlier time, brighter green -later time.

wake topology to the corresponding wake topology of 1D freely vibrating geometries. In order to determine the necessity of in-depth analysis of a 2D vibrating geometry, reference to the motion response of each geometries are further analysed from figure 6.62. With reference to figure 6.62, other 2D vibrating geometries (geometries 4 to 10) exhibited a larger crossflow (or equal) amplitude of oscillation to its corresponding streamwise amplitude of oscillation. For occasions of equal amplitude of streamwise and crossflow oscillations, the amplitude of oscillations (A/D) are significantly smaller (order of 10^{-3}) than the characteristic diameter of the geometries (minor diameter for elliptical-based geometries). While slight variations are observed in the response of the geometries, no significant variations are observed from their corresponding response under 1D freely vibrating in crossflow setup. However, with reference to figure 6.62, some exceptions are observed for geometry 9 and 10, where no streamwise dominant response frequency are observed. This is mainly due to the absence of the vortex tubes in the wake of the respective geometries. It should be recalled that the shedding of the vortices (vortex tubes) is the main cause of the vibratory motion of the bodies undergoing VIV. As such, the deconstruction of these vortex tubes (due to the significant perturbation and/or generated vortical braids) resulted in absence of consistent streamwise motion response of the structure. Further exception is also observed for geometry 7, where the dominant response frequency ratio (f_{r_x}/f_{r_y}) is observed to be relatively lower than 2.0 (≈ 1.65). From figure 6.71, larger Cm_x (absolute value) and significant change of crossflow Cm_y of a 2D vibrating geometry 7 (as compared to its 1D freely vibrating in crossflow setup) are observed. This indicated significant phase changes (phase differ-

ence between motion of the body and the lift force due to vortex shedding on the body) with a faster response (lower phase angle) on the 2D vibrating setup. This is a reversal of the trend of the Cm_y of geometry 9, where larger Cm_y (absolute value) is observed for 1D vibrating setup. Generally, introduction of an additional degree of freedom is found to reduced the Cm_y (absolute value) of the respective geometries with negative Cm_x observed. Figure 6.73 shows the motion trajectories of 2D vibrating geometry 5, 6, 7, 9, 10 during their steady period of oscillation for $Re = 500$. Observations of figure 6.73 show consistency of the occurrence of '8-figure' motion trajectory for geometries with dominant response frequency ratio of 2.0 at various level of skewness and asymmetry (dependent on the ratio of streamwise to crossflow amplitude of oscillation). Motion trajectory of a 2D vibrating geometry 7 is shown to elude an '8-figure' (relatively random and weakly patterned) due to the weak lock-in in streamwise and crossflow oscillation. However, motion trajectories of geometry 9 (old vibrissa geometry) and 10(new vibrissa geometry) are found to vary between an elliptical trajectory and '8-figure', which confirmed the absence of the dominant response frequency in their streamwise oscillations.

Figure 6.73: Motion trajectory of a 2D vibrating geometry 5, 6, 7, 9 and 10 during their steady period of oscillation at $Re = 500$, $f_{nr} = 2.0$ and $m^* = 1.04$. Top left: isometric view of motion trajectory of geometry 5 (vibrissa-like geometry with no minor axis undulations) ; top right : top view of corresponding figure on the left ; middle left : isometric view of the motion trajectory of geometry 6 (vibrissa-like geometry with no offset angle) ; middle right : top view of the motion trajectory of geometry 7 (vibrissa-like geometry with constant offset angle of 15.26°) ; bottom left : top view of the motion trajectory of geometry 9 (old vibrissa geometry) ; bottom right : top view of the motion trajectory of geometry 10 (new vibrissa geometry). Legend: Darker green - earlier time, brighter green -later time. Note : features of some motion trajectory is better visualised with isometric view, instead of top view (standard).

6.4 Response Types of 1D and 2D Freely Vibrating Geometries

Despite their frequented oscillations, certain inherent features are yet to be determined from the identification of the amplitude and dominant response frequency of their oscillations as well as analysis on the wake topology of the system. Among such inherent features, the type of responses of each geometry for a setup are left undetermined. For geometries with steady amplitude of response, periodic response would almost be instantly identifiable. However, certain type of responses (such as chaotic and non-periodic yet patterned) require further evaluation for conclusive determination. From chapter 5, methods of determining the types of responses are described through the utilisation of the phase plot and Poincare map. Figure 6.74 shows the phase plot and Poincare map of the crossflow motion response of 1D freely vibrating in crossflow geometries at $Re = 500$ and their resonance conditions. It is necessary to analyse the Poincare map in conjunction with the corresponding phase plot. This is due to their strong correlation in determining the patterns of a response of a geometry. From figure 6.74 and observations on the phase plot, it can be observed that periodic crossflow motion response are found on freely vibrating in crossflow circular cylinder, geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) and geometry 5 (vibrissa-like geometry with no minor axis undulations). From their corresponding Poincare map of figure 6.74, small pattern variability is observed on their responses. It should be noted that the Poincare map presented in this section represents only the enlarged view of the pattern. The patterns are always centred about their corresponding peak response velocity of oscillations (similarly for phase plots). Hence, it is essential to analyse the Poincare maps in conjunction with the phase plots in order to reach accurate conclusions. Despite the slight variability, it can be well-concluded that the motion responses of the freely vibrating in crossflow circular cylinder, geometry 3 and geometry 5 are, at least, quasi-periodic due to the strong repeatability of their response state. In comparison, from figure 6.74, the motion response of a freely vibrating elliptical cylinder and geometry 4 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.54) show a non-periodic but consistent pattern of star-shaped and near-elliptical stable pattern of the state of the system. However, the motion responses of geometry 6 to 9 (vibrissa-like geometry with dual-axial undulations are observed to be chaotic with no observable pattern even during their steady period of oscillation. Interestingly, from figure 6.74, the motion response of the new vibrissa geometry is observed to be initially chaotic. However, a quasi-periodic response was achieved near its more stable period of response.

Despite their identified response types on a freely vibrating in crossflow setting, the motion responses of these geometries are strongly believed to be different with an addition of another degree of freedom. Such analysis could be performed on the streamwise and crossflow oscillatory motion of the corresponding geometries on freely vibrating in streamwise and crossflow setup. Figures 6.75, 6.76 show the phase plots and Poincare maps of the crossflow and streamwise responses of 2D vibrating geometries at $Re = 500$. From phase plots in figure 6.75, it can be observed that the crossflow responses of the geometries are slightly modified with the addition of streamwise free vibration. For example, despite the consistent periodic response of the crossflow motion of a 2D vibrating circular cylinder, slight twinge in the trajectory of the state of the system are observed with deviations from circular trajectory occurring at the extremes of the states of the system. With reference to figure 6.75, significant changes in the crossflow response of 2,3 and 5 are observed. The crossflow response of geometry 3 and 5 are no longer periodic, however, quasi-periodic response are observed for geometry 3 and non-periodic but patterned response are observed for geometry 5. The patterned crossflow response of geometry 5 is observed to follow a near-elliptical trajectory. However, from observations

Figure 6.74: Phase plot (top) and Poincaré map (bottom) of the crossflow motion of freely vibrating in cross-flow geometries (refer to figure 6.1 for details of each geometry).

of the phase plots in figure 6.75, an extra turning point in each side of the state of the system are observed for geometry 2 and 3. Such coincidence is found to be strongly related to the motion response of a more streamlined geometry in a 2D vibrating setup. The crossflow motion response of geometry 4 is observed to evolve into a chaotic response with no observable pattern. Similarly chaotic response (to their response in 1D vibrating setup) is observed on the crossflow responses of geometry 6 to 9. A quasi-periodic response is similarly observed in the crossflow response of geometry 10 (new vibrissa geometry) after a longer period of stabilisation, as compared to its corresponding response in a 1D vibrating setup.

Through comparison between figure 6.75 and figure 6.76, almost no similarity is observed between the crossflow responses of the geometries to their corresponding streamwise responses. Quasi-periodic response is observed on the streamwise response of the circular cylinder. Periodic responses are observed in the streamwise motion of geometry 2 and 3. Non-periodic yet predictable responses are observed on the streamwise responses of geometry 5 and 6. Such observation indicated the im-

portance of features on minor axis on the streamwise response of the geometries. Coincidentally, geometries with non-zero offset angle in major axis and/or undulatory in minor axis are observed to respond chaotically to flow around such structures.

While geometrically imposed undulations are important in determining the response of the structures, significant emphasis should also be given to the boundary conditions (freely vibrating in cross-flow and/or streamwise directions). It has been shown in this chapter that such the addition of an extra degree of freedom in the streamwise direction could result in significant changes to the type of response of the system. The existence of single or dual-axial undulations is found to be an important parameter in reducing the amplitude of VIV. While the major axis undulations is found to produce almost no reduction in amplitudes of VIV, significant reduction is obtained through the introduction of minor axis undulations. However, combination of major and minor axis undulations is found to result in even smaller amplitude of VIV. The wavelength of the undulations is also found to strongly affect the wake topology in the wake of the structures, which, in turn, affected the response of the structure. A wavelength-to-diameter (λ/D) ratio of 6.8 for the undulations is found to more signif-

Figure 6.75: Phase plot (top) and Poincare map (bottom) of the crossflow motion of freely vibrating in stream-wise and crossflow geometries (refer to figure 6.1 for details of each geometry).

icantly reduce the forcing and response of the geometry at $Re = 500$ during its resonance. Offset angles in the major axis undulations could also be configured to affect the dominant response frequency of the geometries. Through carefully prescribing such offset angle, the singular dominant response frequency peak could be broken up into two response frequencies with the dominant frequency at approximately half of the secondary response frequency, which commonly coincided with the supposedly singular dominant response frequency.

Figure 6.76: Phase plot (top) and Poincare map (bottom) of the streamwise motion of freely vibrating in streamwise and crossflow geometries (refer to figure 6.1 for details of each geometry).

Chapter 7

Force And Motion Responses of Whisker At Angle of Attack

This chapter is dedicated to understanding the force and motion response of the new and old vibrissa geometries at various angle of attacks. Due to limitation of time and computational resources, only selective cases with strong relevance to the underwater sensing ability of the harbour seal vibrissae are simulated. Such cases include simulations of vibrissa at angle of attacks of 0° , 45° , and 90° . For each angle of attack, three simulation settings are devised, mainly, simulation on stationary structure, freely vibrating in crossflow structure (also called 1D vibration simulation), freely vibrating in streamwise and crossflow directions structure (also called 2D vibration simulation). Chapter 5 has provided elaborate information on the implementation and setup of the simulations for stationary, 1D vibrating and 2D vibrating simulations. Hence, only selective details of the simulations will be provided for each section or subsection in this chapter. In conjunction to the settings of the simulation, this chapter will be divided into four sections consisting of:

- * Stationary structure simulation,
- * 1D vibration structure simulation,
- * 2D vibration structure simulation, and
- * The implications of the responses of the seal vibrissa at various angle of attack on the underwater sensing ability of harbour seals.

For each section, elaborations on the responses (force and/or motion) of the new and old vibrissa geometries at angle of attacks of 0° , 45° , and 90° will be conducted. Unfortunately, as mentioned in chapter 5, a parametric error was found on the published old vibrissa geometry two months prior to the completion of this thesis [40]. As such, the author has a very limited time in performing additional simulations for the new vibrissa geometry at angle of attacks of 45° and 90° . Hence, selective simulations are conducted through perusing the simulation results for 1D vibration and 2D vibration old vibrissa geometry at angle of attacks of 45° and 90° . Emphasis of selection points are given to points of greatest amplitude of response and its vicinity for each geometrical setup.

Further elaborations and justification on the selection of points will be provided accordingly in the following sections. The implications of the response of the vibrissa at various simulation settings and angle of attack will be discussed in the last section of this chapter. It should be noted that the simulations are performed at relevant $Re = 500$ and, for 1D vibration and 2D vibration simulations, mass ratio (m^*) = 1.04.

7.1 Stationary Simulations

This section provided descriptions of the simulations on stationary new and old vibrissa geometry at angle of attacks (γ) of 0° , 45° , and 90° . All the simulations are performed with the geometries fixed relative to the grid at $Re = 500$. Figure 7.1 depicts the placement of the geometry relative to the simulation domain. From figure 7.1, it can be observed that all the simulation domains for the vibrissa geometries at various angle of attacks are scaled relative to the characteristic diameter of the structure in each simulation (D). By defining O as the point of origin coinciding with the centroid of the geometry, the upstream and downstream lengths of the simulation domain measured from O are $4D$ and $10D$ ($4.5D$ and $17D$ for new vibrissa geometry), respectively. The longer downstream length for new vibrissa geometry is for longer region of wake visualisation due to the longer spanwise length of the new vibrissa geometry (as compared to the old vibrissa geometry). A longer region of wake visualisation is observed to provide a more proportionate view of the wake of the new vibrissa geometry. The width of the simulation domain about O is symmetrical at $4.5D$. The height of the simulation domain is $12.2D$ ($27.1D$ for new vibrissa geometry due to approximately-doubling of the wavelength of the vibrissa). Internal lines in figure 7.1 should be neglected as they did not possess

any physical significance. These lines represented the segmentation of the computational domain during parallel computing process. Figure 7.2 shows the time history of the drag and lift coefficient of the new and old vibrissa geometries.

From figure 7.2, it can be found that the drag coefficients of the new and old vibrissa geometries

Figure 7.1: Simulation domain for vibrissa geometry at 0° (Top), 45° (Middle) and 90° (Bottom) angle of attack. By defining the centroid of the geometry in XY plane as point of origin (O), the upstream and downstream lengths (x-axis) of the domain measured from O are $4D$ and $10D$ ($4.5D$ and $17D$ for new vibrissa geometry for longer region of wake visualisation, providing better aspect ratio), respectively, where D is the characteristic diameter of the structure in the simulation. The width of the simulation domain (y-axis) is symmetrical about the x-axis passing through O at $4.5D$. The height of the simulation domain is $12.2D$ ($27.1D$ for new vibrissa geometry).

achieved steady state of ≈ 0.5 and ≈ 0.7 at $t^* \approx 100$ and ≈ 70 , respectively. As explained in chapter 6, such reduction in drag coefficient of the new vibrissa geometry is related to the efficient dissipation of the spanwise flow velocity. Such efficient dissipation led to a build-up of pressure which resulted in higher near-field low pressure region. Hence, a lower pressure difference between the leading edge and trailing edge of the new vibrissa geometry (as compared to the corresponding difference for old vibrissa geometry) is observed. From figure 7.2, it can also be observed that the lift coefficient of the new and old vibrissa geometry can only attain quasi-steady state instead of a steady state (even after 10 times longer period of simulation). The $C_{L,rms}$ of the new and old vibrissa geometries are 0.0045 and 0.0264, respectively. This indicates that the new vibrissa geometry significantly reduced the $C_{L,rms}$. The quasi-steady state of the new and old vibrissa geometry are observed to emanate at approximately the same t^* as the simulation time required to achieve steady-state of the C_D (the mean drag of the new and old vibrissa geometries are 0.4953 and 0.6487, respectively). With reference to chapter 6, it is found that the reduction of lift coefficient ($C_{L,rms}$) is due to the

Figure 7.2: Time history of Drag and Lift Coefficient of new and old vibrissa geometry. Top left – C_D of new vibrissa geometry ; top right – C_L of new vibrissa geometry ; bottom left – C_D of old vibrissa geometry; bottom right – C_L of old vibrissa geometry.

alternating occurrence in crossflow direction of the minimum pressure region along the length of the new vibrissa geometry. This will result in a lower forcing on the new vibrissa geometry as the vortices are shed from its body. No dominant frequency is observed for the drag forcing on the new and old vibrissa geometries. However, dominant response frequencies are observed on the lift forcing on the new and old vibrissa geometries ($\frac{fD}{U} = 0.1673$ and 0.1807 , respectively). Figure 7.3 illustrate the Power Spectral Density (PSD) plot of the lift forcing on the stationary new and old vibrissa geometries. From figure 7.3, the dominant frequency of lift forcing (non-dimensionalised) could be identified for new and old vibrissa geometries at 0.1673 and 0.2112 , respectively. As these frequencies do not exhibit significant deviation from the corresponding frequencies for a circular cylinder and an elliptical cylinder (≈ 0.2). The relative deviation of the dominant frequency of lift forcing of the new vibrissa geometry is due to the disappearance of the vortex tubes in the wake of the stationary new vibrissa geometry. As such, the dominant lift forcing on the body is only dependent on the vortical braids shed from the geometry. It is important to recall that the energy of the vortical braids are supplied from the vortex tubes. As such, due to the disappearance of the vortex tubes, the

Figure 7.3: Power Spectral Density (PSD) plot of the lift coefficient on the new vibrissa geometry (left) and old vibrissa geometry (right). The peak frequencies of the lift coefficients of the new and old vibrissa geometries are 0.1673 and 0.2112, respectively.

small power corresponding to the dominant lift forcing of the new vibrissa geometry is expected. For the old vibrissa geometry, despite the significant geometrical perturbation, strong deformation of the vortex tubes occur. However, it has been shown in chapter 6 (refer to figure 6.24) that the shedding of vortices along the length of the old vibrissa geometry is found. Hence, the significant deformation of the vortex tubes indicated the existent of the vortex tubes, which resulted in stronger lift forcing on the old vibrissa geometry (as compared to new vibrissa geometry). However, owing to the significant perturbations, phase difference between the nodal and saddle sections of the old vibrissa geometry existed on their corresponding shedding period. Such phase difference upon shedding is found to led to the observed significant deformation of vortex tubes in the subsequent evolution of the wake of the old vibrissa geometry.

As chapter 6 has delved with great details into the wake topology and instability of the old and new vibrissa geometries, repetitions of such contents would not be presented here.

For simulations of the stationary old and new vibrissa geometry at 45° and 90° angle of attack, analysis of the force responses as well as wake topology and instabilities will also be performed. Figure 7.4 shows the time history of the force response coefficients due to the flow around the new and old vibrissa geometries at angle of attack of 45° at $Re = 500$. From figure 7.4, it can be observed from the $C_{L,rms}$ of the new and old vibrissa geometries at 45° have non-zero mean values. This is expected due to the pressure difference between the upper (positive y-axis/crossflow direction) and lower (negative y-axis/crossflow direction) of the vibrissa geometries. Such pressure difference

(similar to an airfoil at an angle of attack) resulted in constant lift force, which is counter-acted by the spring in the mass-spring system (refer to chapter 5) of the simulation setup. From figure 7.3, it can be observed that the mean drag coefficient (C_D) and the RMS lift coefficient ($C_{L,rms}$) of the new vibrissa geometry (0.752 and 0.3358, respectively) are relatively similar to the corresponding coefficients of the old vibrissa geometry (0.6348 and 0.3244, respectively) with a slightly higher drag coefficient. The mean C_L of the old vibrissa geometry (0.3198) is found to be similar to the corresponding coefficient of the new vibrissa geometry (0.3614). Through determining the difference between the mean and RMS lift coefficient of each geometry, the RMS amplitude of lift coefficient of the corresponding geometry could be determined. Therefore, the RMS amplitude of the lift coefficients for the new and old vibrissa geometries are 0.0256 and 0.046, respectively. Hence, small amplitude of oscillations are observed on the stationary new and old vibrissa geometries. Figure 7.5 shows the wake topology of the flow around a stationary new vibrissa geometry at an angle of attack of 45° and $Re = 500$.

From observation of the wake topology of the flow around a stationary new vibrissa geometry at an angle of attack of 45° , certain spanwise regularity can be observed. Additionally, the vortex tubes are observed to be existent in the wake of the new vibrissa geometry at an angle of attack of 45° . Such existent explained the relatively large amplitude of lift coefficient of the new vibrissa geometry at an angle of attack of 45° (as compared to the corresponding amplitude of the new vibrissa geometry at an angle of attack of 0°). However, significant spanwise mixing of the wake topology is also observed with many W-shaped vortical braids observed. Despite the variant of the shapes of the vortical braids, due to the regular occurrence of the vortex tubes in the wakes of the flow around the new and old vibrissa geometries at an angle of attack of 45° , dominant frequencies of drag and lift forcing on the body are expected. Similarly, from figure 7.6, it can be observed that the wake of the old vibrissa geometry at an angle of attack of 45° exhibited a strong vortex tubes. Slightly deformed vortex tubes are observed in the wake of the old vibrissa geometries at an angle of attack of 45° . Hence, despite the difference in the wavelength of their undulations, relatively similar wake topology are found between the new and old vibrissa geometries at an angle of attack of 45° . Figure 7.7 illustrates the PSD plot of the lift and drag coefficients on the new and old vibrissa geometries at an angle of attack of 45° . From figure 7.7, it can be observed that the dominant frequency of the drag coefficients of the new and old vibrissa geometries at 45° coincided with the dominant frequency of their lift coefficients.

Figure 7.4: Time history of Drag and Lift Coefficient of new and old vibrissa geometry at 45° angle of attack. Top left – C_D of new vibrissa geometry ; top right – C_L of new vibrissa geometry ; bottom left – C_D of old vibrissa geometry; bottom right – C_L of old vibrissa geometry.

Similarity between the dominant lift coefficients of the new and old vibrissa geometries are also observed. Such similarity indicated the equivalence of the spacing of the vortices in their wakes. However, analysis of one of their wakes is necessary in order to determine the origin of equivalence of the dominant frequencies of lift and drag forcing on these geometries, which contradicts the nature of vortex shedding (two vortex tubes shed in one oscillation of the lift force, which shall result in two oscillations in the drag force).

Figure 7.8 illustrates the pressure field in the wake of a stationary new vibrissa geometry at an angle of attack of 45° during its vortex shedding from the leading edge and trailing edge. From figure 7.8, it is apparent that the timing of the shedding of vortices on the leading edge and trailing edge of the geometry would provide the necessary explanation on the equivalence of dominant frequency of drag and lift forcing on the new vibrissa geometry at an angle of attack of 45° . For vortices (corresponds to minimum pressure region in figure 7.8) shed from the trailing edge, no significant

Figure 7.5: Depiction of vortex cores (identification based on λ_2 criterion) of flow around stationary new vibrissa geometry at an angle of attack of 45° at $Re = 500$ and $t^* = 286$. Top left : isometric view ; top right : side view ; bottom : top view.

interaction is found to occur with the rolling-up shear layer from the leading edge of the geometry. Such case is different for the vortices forming (and shed) from the leading edge due to their relative positioning to the forming vortices (rolling-up shear layer) near the trailing edge of the structure. With reference to figure 7.8, it can be observed that upon its formation, the vortex from the leading edge of the structure is in constant interaction with the newly shed vortices and newly formed shear layer near the trailing edge of the structures. Such interaction is found to strongly affect the drag forcing on the structure due to vortex shedding. Figure 7.9 show the enlarged view of the drag and lift coefficient of the stationary new vibrissa geometry for one-period of shedding corresponding to figure 7.8. With reference to figure 7.9, it can be observed that a near-sinusoidal peak is observed in the trough of the C_D . Such occurrence corresponds to shedding of the vortex near the trailing edge of the geometry. This is further confirmed through observation of the C_L , where a trough is also observed in its time history. However, despite the occurrence of vortex shedding near the leading

Figure 7.6: Depiction of vortex cores (identification based on λ_2 criterion) of flow around stationary new vibrissa geometry at an angle of attack of 45° at $Re = 500$ and $t^* = 286$. Top left : isometric view ; top right : side view ; bottom : top view.

edge of the structure in the subsequent time steps (as denoted by a peak at $t^* = 147$ in the C_L plot), multiple occurrence of peaks in the C_D plot are observed. Such occurrence defied the supposed emergence of trough on the C_D of the geometry. This indicated the existence of vortex shedding near the leading edge of the structure. However, due to the combination of the interaction of the vortex (to be shed near the leading edge) with the rolling-up shear layer (near the trailing edge) and its distance from the body, significantly weaker effects are observed in the drag forcing on the body as observed from figures 7.8, 7.9. With reference to figure 7.7, such occurrence is found to explain the weak signal of the dominant frequency of drag forcing and occurrence of other weaker frequencies in the drag forcing. It should be noted that the dominant frequency of drag forcing prevailed from the vortex shedding near the trailing edge of the structure, which is observed to result in distinctively sharp streamwise forcing (trough in figure 7.9) on the structure. Hence, due to the one sided forcing (from vortex shed near the trailing edge only) on the geometry, significantly weaker signal of drag

Figure 7.7: Power Spectral Density (PSD) plot of the drag (left) and lift (right) coefficients on the new vibrissa geometry (top) and old vibrissa geometry (bottom). The peak frequencies of the drag coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively. The peak frequencies of the lift coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively.

forcing are observed (as compared to the two-sided lift forcing).

Further investigation is performed on flow over stationary new and old vibrissa geometry at an angle of attack of 90° at $Re = 500$. Analyses on the force responses of these geometries would provide the necessary information on understanding the directional sensitivity of these geometries to forcing at various angle of attacks. Figure 7.10 depicts the drag and lift coefficients on the new and old vibrissa geometries at an angle of attack of 90° for $Re = 500$. Through comparison between the force coefficients of the new and old vibrissa geometries at angle of attacks of 0° , 45° , and 90° (refer to figures 7.2, 7.4, 7.10), similar amplitude of forcing are observed for new and old vibrissa geometries at angle of attacks of 0° and 45° . Weaker directionality are observed on the new vibrissa geometry through comparison between its lift coefficient forcing at angle of attacks of 0° , 45° to 90° . Similarity in the order of magnitude of their $C_{L,rms}$ are observed between the new vibrissa

Figure 7.8: Pressure field in the wake of a stationary new vibrissa geometry at an angle of attack of 45° for one-period of force oscillation. Top left : $t^* = 145$; top right : $t^* = 147$; bottom left : $t^* = 148$; bottom right : $t^* = 149$.

Figure 7.9: Enlarged view of the corresponding drag (left) and lift coefficient (right) on the stationary new vibrissa geometry for one-period of force oscillation (corresponding to figure 7.8).

geometry at angle of attacks 0° and 90° . This is dissimilar to findings on the $C_{L,rms}$ of the old vibrissa geometry at angle of attacks of 0° , 45° and 90° . Steady increment in its $C_{L,rms}$ are observed with the largest $C_{L,rms}$ observed for old vibrissa geometry at an angle of attack of 45° (similar to the response of new vibrissa geometry). Contrary to findings on $C_{L,rms}$, the directionality of the new vibrissa geometry are easily determinable through its mean drag coefficients (C_D) for various angle of attacks tested. Increase in the mean drag coefficient (C_D) of the new vibrissa geometry is observed with increase in the angle of attack of the structure. Such increment trend is also observed for the case of the old vibrissa geometry. However, it should be noted that the VIV response of the new vibrissa geometries are found to attain similar amplitude of oscillation as the VIV response of the old vibrissa geometry. This shall be illustrated in the next sections of this chapter on freely vibrating

Figure 7.10: Time history of Drag and Lift Coefficient of new and old vibrissa geometry at 90° angle of attack. Top left – C_D of new vibrissa geometry ; top right – C_L of new vibrissa geometry ; bottom left – C_D of old vibrissa geometry; bottom right – C_L of old vibrissa geometry.

vibrissae. An important note of indication is the significant changes on the dynamics of the flow around a stationary and a freely vibrating new vibrissa geometry (which is not observed in the case of the old vibrissa geometry).

Figure 7.11 illustrates the wake topology of the new vibrissa geometry at an angle of attack of 90° at $Re = 500$. From the side view of the wake depicted in figure 7.11, it can be observed that vortex tubes are existent in the wake of a stationary new vibrissa geometry at an angle of attack of 90° (unlike new vibrissa geometry at an angle of attack of 0°). Relatively regular occurrence of vortical braids are also observed in its wake. However, spanwise deformation of the vortex tubes are observable with greater deformation observed in regions with larger regular occurrence of vortical braids.

Figure 7.11: Depiction of vortex cores (identification based on λ_2 criterion) of flow around stationary new vibrissa geometry at an angle of attack of 90° at $Re = 500$ and $t^* = 286$. Top left : isometric view ; top right : side view ; bottom : top view.

Figure 7.12: Power Spectral Density (PSD) plot of the drag (left) and lift (right) coefficients on the new vibrissa geometry (top) and old vibrissa geometry (bottom). The peak frequencies of the drag coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively. The peak frequencies of the lift coefficients of the new and old vibrissa geometries are 0.2342 and 0.2657, respectively.

7.2 Freely Vibrating Simulations (1D & 2D Vibrating)

In this section, description on the freely vibrating in crossflow (1D vibrating) and freely vibrating in streamwise and crossflow (2D vibrating) responses of the new and old vibrissa geometries at angle of attacks of 0° , 45° and 90° is elaborated. Similar to other aforementioned simulations (refer to chapter 5, 6), the Reynolds Number of the freestream flow is 500 and the mass ratio of the simulated structure is 1.04 with periodic boundary conditions applied in the spanwise ends of the geometries. While both the new and old vibrissa geometries are described, priorities are given to explanations of the motion responses of the new vibrissa geometries due to its stronger resemblance to the real seal vibrissae. The simulations performed in this section are completed for various natural frequency setup of the system (represented by reduced velocity, U^* or reduced natural frequency, f_n^*). A consistent mass ratio of 1.04 is applicable to all the freely vibrating simulations. For 2D vibrating simulations, the streamwise to crossflow natural frequency ratio of the system is maintained at 2.0 due to the inherent nature of vortex shedding, as explained in chapter 6. Despite the current understanding on the equivalence of the drag and lift force response frequency of the new and old vibrissa geometry at an angle of attack of 45° , no variation is performed on the streamwise to crossflow natural frequency ratio. This is due to the priority of similitude between the setup of the simulations for various angle of attacks on new and old vibrissa geometries. Variation on the streamwise and crossflow natural frequency ratio may result in a different responses, which would lead to difficulties in comparing their responses to responses of the corresponding geometries at angle of attacks of 0° and 90° . Figures 7.13, 7.14 illustrates the amplitude ratios and dominant response frequencies of the new and old vibrissa geometries at an angle of attack of 0° , respectively.

From figure 7.13, it can be observed that similar crossflow response patterns are observed between 1D vibrating and 2D vibrating new vibrissa geometry for $U^* < 14$. The largest crossflow amplitude (RMS) response of the 1D and 2D vibrating new vibrissa geometries are found in neighbouring points at $U^* = 3$ and 4, respectively. These similarities indicated that only slight changes in the response (and wake topology) of the vibrissa as additional DOF is introduced in the streamwise direction. However, for $U^* \geq 14$, it can be found that significant amplitudes of streamwise oscillations are observed coupled with significant deviations of the crossflow amplitude of response from its corresponding amplitude for 1D vibrating new vibrissa geometry. These streamwise oscillations are observed to greatly exceed even the largest crossflow amplitudes of oscillation. But a note should be

Figure 7.13: Characteristic response of the new vibrissa geometries at an angle of attack of 0° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.

taken on the absolute values of these amplitudes due to their minuteness as compared to the diameter of the new vibrissa geometry. Observation on the dominant response frequency of the 1D and 2D vibrating new vibrissa geometry revealed very similar response with few exceptions ($U^* = 2, 9, 10$). This confirmed the strong similarity of the response and fluid dynamics in the flow around 1D and 2D vibrating new vibrissa geometry. Figure 7.15 illustrates the wake topology of the freely vibrating in crossflow new vibrissa geometry at $Re = 500$, $m^* = 1.04$, $U^* = 5$ and $t^* = 194$. Through compar-

ison between figure 7.15 and figure 6.57, no distinction are observed between the wake topology of a freely vibrating in crossflow new vibrissa geometry at $U^* = 4$ and 5. Further investigations on the wake topology of the 1D vibrating new vibrissa geometry at other reduced velocities revealed a similar wake topology to the ones observed in figures 7.15, 6.57. This indicated that the main variability of the responses is due to the phase difference between the motion response and the lift coefficient on the geometry (similar to the well-known variation of response of a freely vibrating circular cylinder with reduced velocity) [37]. Such phenomenon of phase angle is one of the most important feature of VIV. However, these phase angles are only quantifiable for locked-in response of a geometry. However, with reference to figure 6.71, it can be observed that 1D and 2D vibrating new vibrissa geometry at resonance is found to respond with a near-zero added mass coefficients and lift coefficient in phase with velocity. As such, less significant hydrodynamic coefficients would be resulted at non-resonant conditions due to the significance and commonly-eventful resonant conditions. Despite the introduction of free vibration of the new vibrissa geometry in the streamwise direction, no observable changes in wake topology is observed despite the larger streamwise oscillation of 2D vibrating new vibrissa geometry at higher reduced velocities. Figure 7.16 illustrate the vortex cores in the wake of a 2D vibrating new vibrissa geometry for $Re = 500$ and $U^* = 11, 17$ on their stable period of response.

Observation on figure 7.14 shows similar crossflow amplitude of response between the 1D and 2D vibrating old vibrissa geometry at an angle of attack of 0° . Maximum crossflow amplitude of response is observed at $U^* = 4$ with a narrower peak observed for 1D vibrating old vibrissa geometry. Significantly smaller streamwise amplitude of oscillation are observed at various reduced velocities simulated with exception of $U^* = 16$, where a slightly greater streamwise amplitude of oscillation are observed. From figure 7.14, it can also be observed that the 2D vibrating old vibrissa geometry responded at a similar frequency to one of the two dominant response frequencies of the 1D vibrating old vibrissa geometry. It should be noted that for $U^* \leq 9$, the 2D vibrating old vibrissa geometry is observed to respond strongly to the first dominant response frequency of the 1D vibrating old vibrissa geometry. From figure 7.14, it is also important to note that the dominant response frequencies of the old vibrissa geometry are observed to follow one of the two main response frequencies which are found to always lie on lines with the slopes of 0.18 and 0.08. For 1D vibrating old vibrissa geometry, its dominant response frequency is found to consistently lie on the line with the slope of 0.18 for $U^* \leq 6$. Almost no presence of secondary frequency peak are observed in these range of reduced ve-

Figure 7.14: Characteristic response of the old vibrissa geometries at an angle of attack of 0° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.

locity. However, despite its proximity to lock-in region, no occurrence of lock-in (beyonds the range of $5 \leq U^* \leq 6$) is found on the entire range of simulated reduced velocity. For $U^* > 6$, consistent occurrences of secondary response frequency are observed, which is found to line on the line with the slope of 0.18 with the primary response frequency found on the line with the slope of 0.08. Hence, alternation of dominant response frequency is found to be dependent on the reduced velocities of the setup of the system (as observed from figure 7.14). Unlike the new vibrissa geometry, the wake

Figure 7.15: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in crossflow new vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 5$ and $t^* = 194$. Top left : isometric view ; top right : side view ; bottom : top view.

topology of the old vibrissa geometry is observed to change significantly during its resonance and non-resonance condition. Figure 7.17 depicts the wake topology of a freely vibrating in streamwise and crossflow old vibrissa geometry at $U^* = 4$ (resonance) and 10 (non-resonant condition with crossflow dominant response frequency on the line with the slope of 0.08 on figure 7.14). Observations on figure 7.17 revealed pertaining difference in wake topology of the old vibrissa geometry at resonance and non-resonance conditions. Such difference is found to be purely due to the variation of the natural frequency of the system (constant Reynolds Number and other simulation setup). The modification of the natural frequency to an equivalent value of the vortex shedding frequency of a stationary old vibrissa geometry is found to result in a larger amplitude which is found to result in more horizontal (streamwise) vortical braids. This indicated the weakening of the spanwise flow velocity due to the perturbations from the undulations of the old vibrissa geometry. Simultaneously, due to the increase in the amplitude of motion, a larger amplitude of the speed of oscillation is also observed.

Figure 7.16: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in streamwise and crossflow new vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 11$ (left) and 17 (right). Top : isometric view ; middle : side view ; bottom : top view.

Hence, greater conversion of Kinetic Energy (KE) of the fluid to the crossflow motion of the fluid is found at resonance (resulting in greater crossflow oscillation and speed). From figure 7.17, it can also be observed that the wake of the old vibrissa geometry at resonance consisted of vortex tubes and braids (with some similarity to the wake of freely vibrating in crossflow geometry 4 and 5, refer to chapter 6). However, with reference to figure 7.17, it can be observed that the wake of a 2D vibrating old vibrissa geometry at non-resonant condition is found to consist of some similarity to the wake of the 1D vibrating new vibrissa geometry. Bulk of vortical braids are also observed to be centred about the nodal plane of the minor axis (saddle plane of the major axis) for each period of undulations, similar to the bulk of vortical braids observed in the wake of a 1D vibrating new vibrissa geometry. No vortex tubes is observed in the wake of the 2D vibrating old vibrissa geometry at non-resonance condition. Interestingly, at resonance condition, a synchronised shedding of vortices is observed in the spanwise direction of the 2D vibrating old vibrissa geometry. However, at non-resonance condition, phased shedding of vortices in spanwise direction is observed, which resulted in the absence of vortex tubes in the wake of the 2D vibrating old vibrissa geometry. In comparison, observations on the wake topology of freely vibrating in crossflow old vibrissa geometry at $U^* = 8$ and 17 revealed similar yet distinctive wake topology. Figure 7.18 shows the wake topology of a 1D vibrating old vibrissa geometry at $Re = 500$ for $U^* = 8$ and 17. With reference to figure 7.14, it can be observed that, at $U^* = 8$, the dominant response frequency of the 1D vibrating old vibrissa geometry lied on the line with the slope of 0.08 and the secondary response frequency is found to lie on the line with the slope of 0.18. In comparison, at $U^* = 17$, similar amplitudes of oscillation is observed to the corresponding amplitude at $U^* = 8$. For $U^* = 17$, a single dominant response frequency is observed to lie along the line with the slope of 0.08. From figure 7.18, observations on their corresponding wake topology indicated different vortex cores in the wake of the 1D vibrating old vibrissa geometry. With reference to figure 7.18, it can be observed that the wake of the 1D vibrating old vibrissa geometry at $U^* = 17$ consisted of greater number of vortical braids at various orientation relative to streamwise direction (as compared to the corresponding geometry at $U^* = 8$). Greater disorganisation is also observed in the wake of a 1D vibrating old vibrissa geometry at $U^* = 17$ with no observable vortex tubes observed in its wake. In contrast, instances of vortex tubes are observable in the wake of a 1D vibrating old vibrissa geometry at $U^* = 8$.

As the vibrissae would act as a transducer of perturbation, it is necessary to determine the direc-

Figure 7.17: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in streamwise and crossflow old vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 4$ (left) and 10 (right). Top : isometric view ; middle : side view ; bottom : top view.

tionality of the response of the vibrissae at various angle of attacks. Identification of such response would allow determination of the directional sensitivity of the vibrissae to perturbations. As the vibrissae are attached on the muzzle of the seals, they would be exposed to various angle of inclinations. Hence, further evaluations are performed on new and old vibrissa geometries at an angle of attack

Figure 7.18: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a freely vibrating in crossflow old vibrissa geometry at an angle of attack of 0° at $Re = 500$, $U^* = 8$ (left) and 17 (right). Top : isometric view ; middle : side view ; bottom : top view.

of 45° at $Re = 500$ with similar settings to the simulations for new and old vibrissa geometries at an angle of attack of 0° . Figure 7.19 shows the amplitudes and dominant response frequencies of the new and old vibrissa geometries at an angle of attack of 45° for various reduced velocities. Due to limited resources, only selected reduced velocities in the region of resonance ($U^* = 3, 4, 5$) are

simulated for the 1D and 2D vibrating new vibrissa geometry. With reference to figure 7.19, two peaks of the crossflow amplitude of response are observed for a 2D vibrating old vibrissa geometry at an angle of attack of 45° as compared to a single peak of the crossflow amplitude of response for a 1D vibrating old vibrissa geometry at an angle of attack of 45° . It is also observed that the two peaks are observed in the streamwise amplitude of response of a 2D vibrating old vibrissa geometry. The peaks of the streamwise amplitude of response are observed to occur at the corresponding reduced velocities for the peaks of the crossflow amplitude of response. Significant reduction of crossflow and streamwise amplitudes of oscillation of the 2D vibrating old vibrissa geometry are observed for $U^* \geq 10$. However, significantly narrower range of reduced velocities representing the occurrence of resonance are observed for 1D vibrating old vibrissa geometry ($2.5 < U^* < 4$). For 1D vibrating old vibrissa geometry at an angle of attack of 45° , no dominant response frequency are observed for $U^* > 7$. However, consistent emergency of crossflow dominant response frequency is observed for the 2D vibrating old vibrissa geometry. Detectable streamwise dominant response frequency is found in the region of $3 \leq U^* \leq 4.5$. In comparison, due to the smaller number of simulated settings on the new vibrissa geometry, relative comparison to the response of the old vibrissa geometry is found to be more appropriate. For 1D vibrating new vibrissa geometry at an angle of attack of 45° , a consistent decline is observed on its amplitude of response from $U^* = 3$ to 5. The amplitudes of oscillation of the 1D vibrating new vibrissa at an angle of attack of 45° geometry are found to be consistently and significantly above the corresponding amplitudes of the old vibrissa geometry at an angle of attack of 45° . While a 2D vibrating old vibrissa geometry is found to respond with greater crossflow amplitudes of oscillation as compared to the corresponding amplitude for 1D vibrating old vibrissa geometry, similar observation is not found on old vibrissa geometry. Larger crossflow amplitudes of response are observed for 2D vibrating new vibrissa geometry at an angle of attack of 45° (as compared to 1D vibrating) at $U^* = 3$ and 5. Coincidentally, at $U^* = 4$, multiple response frequencies are observed for the streamwise and crossflow 2D vibrating vibrissa geometry at an angle of attack of 45° .

In comparison, no secondary frequencies are observed on the response of a 1D vibrating and 2D vibrating old vibrissa geometry at an angle of attack of 45° . Figure 7.20 shows the dominant streamwise and crossflow response frequencies of a 1D and 2D vibrating new vibrissa geometry at an angle of attack of 45° . With reference to figure 7.20, three and two frequency peaks are observed

Figure 7.19: Characteristic response of the old vibrissa geometries at an angle of attack of 45° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.

in the streamwise and crossflow response of a 2D vibrating new vibrissa geometry at an angle of attack of 45° . The two strongest streamwise and crossflow frequency peaks are found to occur at similar reduced frequencies (f^*), notably at $f^* \approx 0.2$ to 0.21 and ≈ 0.24 to 0.25 . However, notable difference is observed between the power corresponding to the similar dominant response frequencies for streamwise and crossflow (power signal for crossflow motion is found to be approximately one order of magnitude larger than the corresponding signal for streamwise motion). In comparison,

Figure 7.20: Power Spectral Density (PSD) plots of the 1D vibrating and 2D vibrating new vibrissa geometry at an angle of attack of 45° for $Re = 500$ and $U^* = 4$. Top left : streamwise (2D vibrating) [0.2166, 2454, 0.4187] ; top right : crossflow (2D vibrating) [0.2025, 0.2459] ; bottom : streamwise (1D vibrating) [0.2355] . All corresponding frequencies of the peaks are indicated from left to right of the plot on the square bracket of each.

only a single dominant response frequency is observed on the crossflow motion of a 1D vibrating new vibrissa geometry at 45° . In relation to figure 7.19, the occurrence of multiple streamwise and crossflow dominant response frequencies of a 2D vibrating new vibrissa geometry at an angle of attack of 45° would significantly reduce the amplitudes of oscillation of the geometry as compared to the response of the corresponding geometry with a single dominant response frequency. This is due to the break-up of the power of the motion of the geometry into three separately power bands (of response corresponding to the three dominant response frequency peaks). As such, as observed from figure 7.19, smaller streamwise and crossflow amplitudes of oscillation are observed at $U^* = 4$, where multiple streamwise and crossflow response frequencies are found. Figure 7.21 illustrates the wake topology of a 1D and 2D vibrating new vibrissa geometry at an angle of attack of 45° and

$U^* = 4$. With reference to figure 7.21, while spanwise regularity (in occurrence and orientation) is observed in the wake of a 1D vibrating new vibrissa geometry, less regularity is observed in the wake of a 2D vibrating new vibrissa geometry. From the side view of their wake topology, a significantly deformed vortex tube (in particular at the second undulations from the bottom in the near field wake) is observed in the wake of the 2D vibrating new vibrissa geometry at an angle of attack of 45° . Correspondingly, on undulatory (deformed) vortex tubes, which are trademark of vortex tubes with braids, are observed in the wake of 1D vibrating new vibrissa geometry at an angle of attack of 45° . Such significant deformation of vortex tube in the wake of the 2D vibrating new vibrissa geometry indicated cellular shedding (multiple frequency shedding), which resulted in the multiple frequencies observed in the crossflow response of the corresponding geometry. However, subsequent propagation downstream revealed a 'catching-up' of the section with most deformed vortex tubes to the rest of the vortex tubes through flow entrainment from the surrounding fluid. In contrast, as the vortex tubes are intact and coherent in the wake of the 1D vibrating new vibrissa geometry, a single dominant response frequency is observed with minor vortex splits and reorientation of vortical braids (away from streamwise direction), unlike the corresponding topology of the wake of the 2D vibrating new vibrissa geometry. In comparison, from figure 7.19, a consistently large amplitude of oscillation is observed on the 1D and 2D vibrating new vibrissa geometry at an angle of attack of 45° and $U^* = 3$. Figure 7.22 illustrates the wake topology of the 2D vibrating new vibrissa geometry at an angle of attack of 45° for $U^* = 3$. Strong occurrence of vortex tubes are observed in the wake of the 2D vibrating new vibrissa geometry at an angle of attack of 45° . Regular occurrence of vortical braids with strong streamwise orientation is also observed in the wake of the 2D new vibrissa geometry. However, with reference to previous study by Braza et al, occurrence of vortex dislocation is visible in the wake of the 2D vibrating new vibrissa geometry at an angle of attack of 45° [14]. This is due to the growth of spanwise flow velocity as depicted by the non-linear shear layer in the near-field flow of the wake. Observation on a 1D vibrating new vibrissa geometry at an angle of attack of 45° does not reveal equivalent occurrence of vortex dislocation. From the top view of the wakes of the 2D vibrating new vibrissa geometry in figure 7.22, a near symmetrical wake about the x-axis (streamwise) could be observed on the configuration of the vortex tubes and vortical braids. Interestingly, from figure 7.19, this is found to be the only occasion where the streamwise to crossflow response frequency ratio is ≈ 2.0 . As such, a similar motion response trajectory is expected due to the prevailing response

frequencies. Figure 7.23 illustrates the motion trajectory of the 2D vibrating new vibrissa geometry at an angle of attack of 45° for $U^* = 3, 4$ and 5 .

From figure 7.23, distinct motion trajectories are observed for the 2D vibrating new vibrissa geometry at an angle of attack of 45° and $U^* = 3, 4, 5$. At $U^* = 3$, a distinct '8-figure' motion trajectory is observed is a smaller upper-half as compared to its lower-half. As aforementioned, this is expected from the streamwise to crossflow dominant response frequency ratio (≈ 2.0). Additionally, at this reduced velocity, maximum crossflow amplitude of oscillation is observed (among the simulated reduced velocities). Similarly, an '8-figure' motion trajectory is observed at $U^* = 3.5$, which corresponded to the reduced velocity of maximum crossflow amplitude of response (refer to figure 7.19). For $U^* = 4$, a disfigured '8-figure' motion trajectory is observed with almost inexistent upper-half (resulting in an 'onion-like' shape). Interestingly, the motion trajectories of the new vibrissa geometry at an angle of attack of 45° is found to vary from an elliptical shape to the 'onion-like' shape. However, with the increase of reduced velocity to 5 , a pure elliptical motion trajectory is observed.

In order to assess the directionality of the new and old vibrissa geometry, it is also necessary to analyse the response of the 1D vibrating and 2D vibrating new and old vibrissa geometries at an angle of attack of 90° . However, as the new and old vibrissa geometries are oriented at an angle of attack of 90° , their major axes are perpendicularly aligned to the freestream flow. As such, bluff bodies are presented. This created a larger pressure difference between the leading edge and trailing edge of the structure. Due to the larger pressure difference, a greater amplitude of response would be expected. Figure 7.24 depicts the amplitude and dominant response frequencies of the 1D vibrating and 2D vibrating old and new vibrissa geometries at an angle of attack of 90° . Similar to the case of the 1D and 2D vibrating new vibrissa geometry at an angle of attack of 45° , simulations are performed on selected reduced velocities ($U^* = 3, 4, 6, 7$) of the 1D and 2D vibrating new vibrissa geometry with priority given to region of resonance. From figure 7.24, similar maximum crossflow amplitudes are observed between the 1D vibrating and 2D vibrating new and old vibrissa geometries at an angle of attack of 90° with a slightly lower maximum amplitude observed for 1D vibrating new vibrissa geometry. A steeper variation of crossflow amplitudes is observed for the new vibrissa geometry as compared to the old vibrissa geometry. For 2D vibrating new and old vibrissa geometry at an angle of attack of 90° , minuscule streamwise amplitudes of motion are observed (as compared to the crossflow amplitudes of motion) at various reduced velocities. With reference to figure 7.24, maxi-

Figure 7.21: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a 1D vibrating (right) and 2D vibrating (left) new vibrissa geometry at an angle of attack of 45° at $Re = 500$, $U^* = 4$ during their stable period oscillation. Top : isometric view ; middle : side view ; bottom : top view.

Figure 7.22: Depiction of vortex cores (identification based on λ_2 criterion) of flow around a 2D vibrating (left) new vibrissa geometry at an angle of attack of 45° at $Re = 500$, $U^* = 3$ during their stable period oscillation. Top : isometric view ; middle : side view ; bottom : top view.

imum crossflow amplitudes are observed at $U^* = 4$ for 1D vibrating old vibrissa geometry and 1D vibrating and 2D vibrating new vibrissa geometry at an angle of attack of 90° . For 2D vibrating old vibrissa geometry, the maximum crossflow amplitude is observed to occur at $U^* = 6$. Observations on the dominant response frequency of the 2D vibrating new vibrissa geometry at an angle of attack of 90° in figure 7.24 indicated the constant occurrence of dual frequency peak in its streamwise motion response with exception of $U^* = 7$. These frequencies are found to maintain a consistent ratio of ≈ 2.0 . In general, the dominant streamwise response frequency of the 2D vibrating new vibrissa geometry (except at $U^* = 6$) is found to correspond to the dominant response frequency of the crossflow motion. This is found to be consistent with the occurrence of an '8-figure' motion trajectory of the new vibrissa geometry at an angle of attack of 90° (at $U^* = 6$, an 'onion-like' shape trajectory is observed). No significant variation of dominant crossflow response frequency is observed between the response of a 1D vibrating and 2D vibrating new vibrissa geometry at an angle of attack of 90° .

Figure 7.23: Motion trajectory of the 2D vibrating new vibrissa geometry at an angle of attack 45° and $U^* = 3, 4$ and 5 .

In comparison, the dominant crossflow response frequency of the 1D vibrating and 2D vibrating old vibrissa geometry at an angle of attack of 90° are found to be similar to the corresponding frequency of the new vibrissa geometry. However, a significant distinction is observed in comparison between the streamwise response frequencies of the new and old vibrissa geometry. While both vibrissa geometries is found to respond with one of its two major streamwise frequencies overlapping with their dominant crossflow response frequency, a halved second response frequency (instead of doubling like the ones observed in new vibrissa geometry) is observed in the streamwise response of a 2D vibrating new vibrissa geometry at an angle of attack of 90° . However, with reference to figure 7.24, secondary dominant streamwise response frequency is found to occur only for $U^* \leq 6$. A further division is observed upon determination of the primary and secondary dominant streamwise response frequencies. For $U^* \geq 4$, the primary streamwise response frequency is found to lie on the lower response frequency branch. For $U^* \leq 3.5$, the primary response streamwise response frequency is found to lie on the upper response frequency branch.

Understand from previous studies on rectangular cylinder suggested that the wake topology of the new vibrissa geometry at an angle of attack of 90° and resonance ($U^* = 4$) would result in the occurrence of strong vortex tubes with strong braids. Such occurrence is widely expected due to the large amplitude of response, which is a direct consequence of the shedding of strong vortex tubes. Such strong vortex tubes would undergo greater instability (due to the high energy content in them) as they travel downstream. Hence, regular and abundant vortical braids are also expected. Figure 7.25 illustrates the wake topology of the 1D and 2D vibrating new vibrissa geometry at an angle of attack

Figure 7.24: Characteristic response of the old vibrissa geometries at an angle of attack of 90° with freely vibrating in crossflow and freely vibrating in streamwise and crossflow setups. Top : amplitude of response ; bottom : dominant response frequency.

of 90° and $U^* = 4$ during their stable resonant response. Similar to previous comparison between the wake topology of a 1D vibrating and 2D vibrating geometry, a more regular wake topology is observed in the wake of the 1D vibrating new vibrissa geometry at an angle of attack of 90° . Strong occurrence of vortex tubes are observed in both setup of the new vibrissa geometry. However, due to the addition of the streamwise motion response in 2D vibrating new vibrissa geometry, larger circulation of the vortex tubes required high kinetic energy which resulted in unstable configuration. Consequently, the vortex tubes are observed to be broken down after the near-field wake region. In

contrast, more persistent vortex tubes and braids are observed in the wake of a 1D vibrating new vibrissa geometry at an angle of attack of 90° . Spanwise periodic shedding is observed in the wake of the 1D and 2D vibrating new vibrissa geometry with observed repeatable feature on each period of undulations. However, due to the more durable vortex tubes, the consistency and repeatability of the vortical braids (for each period of undulations) are seen to be more consistent in the wake of a 1D vibrating new vibrissa geometry at an angle of attack of 90° .

Through comparison between the maximum crossflow amplitudes of response of the 1D and 2D vibrating new and old vibrissa geometry (refer to figure 7.13, 7.14, 7.19, 7.24), similarly directional response are observed. As the angle of attack of the new and old vibrissa geometries is increased, increase in the crossflow amplitudes of response is also observed. Despite the significantly smaller crossflow amplitude of response at an angle of attack of 0° , similar crossflow amplitudes are observed between the responses of the new and old vibrissa geometries at an angle of attack of 45° & 90° . Hence, this indicated a more distinct crossflow amplitudes of response of the new vibrissa geometry to various angle of attacks. However, from the corresponding crossflow responses of the new and old vibrissa geometry, larger variation of the crossflow amplitudes are observed for the new vibrissa geometry, which indicated a greater sensitivity to its boundary conditions. Similar dominant response frequencies are also observed between the 1D vibrating and 2D vibrating new and old vibrissa geometry with the exception of the streamwise response frequency of the new and old vibrissa geometry at an angle of attack of 90° . Hence, it has been shown that the new and old vibrissa geometry showed strong directionality (angle of attack) in their crossflow amplitudes of response with a slightly greater sensitivity observed for the new vibrissa geometry during their resonance. The new vibrissa geometry is also observed to possess higher sensitivity to its boundary condition (as compared to the old vibrissa geometry).

Figure 7.25: Depiction of vortex cores (identification based on λ_2 criterion) of flow around 1D vibrating (left) and 2D vibrating (right) new vibrissa geometry at an angle of attack of 90° at $Re = 500$, $U^* = 4$. Top : isometric view ; middle : side view ; bottom : top view.

Chapter 8

Sensor-*Kármán* Street Interaction at Zero Relative Velocity

Previous studies evaluated the motion response of the vibrissa-like geometries, old and new vibrissa geometries under singly-towed simulations in open water. However, as the harbour seals use their vibrissae for sensing, exposure of their vibrissae to significant perturbations are expected. Despite the knowledge on the response of the new and old vibrissa geometries at various natural frequency of the system in open water towing, no evaluation are yet performed on the real sensing ability of the new and old vibrissa geometry to perturbation. Hence, no comparison between the sensing ability of the vibrissa geometries and elliptical cylinder as well as circular cylinder are performed. In this chapter, sensitivity of sensory structures in the wake of a circular cylinder is analysed. The tested sensory structures (in the wake of a wake-generator circular cylinder) are circular cylinder, elliptical cylinder, old and/or new vibrissa geometries. Two different setups are configured for sensory structures, i.e. stationary and freely vibrating in crossflow. However, despite the stronger similarity to the setup of the vibrissae on the muzzle of the harbour seals, it is not possible to perform simulations with freely vibrating in streamwise and crossflow sensory structure. This is due to the incomparable relative placement of the streamwise spacing between the wake-generator circular cylinder and the various sensor structures due to different constant displacement and amplitudes of oscillation of the sensory structures. Hence, no simulations with freely vibrating in streamwise and crossflow setup of the sensory structures are performed. A circular cylinder is chosen as the wake generator due to the well-known regularity in its wake (von *Kármán* street). In order to ensure use of simpler setup (less

possible occurrence of errors and/or variability of wake-generator setup), the wake-generator circular cylinder is set to be stationary. A known wake (von Kármán street of a stationary circular cylinder) is chosen as the perturbation on the sensory structure for easier understanding on the response of the sensory structure. A von Kármán street wake is also known for its periodic yet unstable structure, hence, this may serve to represent the wakes of a fish (which is known as Inverted von Kármán street) [88]. Details on the sizing of the grid are presented in chapter 5, hence, no repetition on the grid sizing will be presented here. For all simulations, a constant distance of $4D_1$ is applied with $Re = 500$ (based on the characteristic diameter of the sensory structure), unless otherwise stated for freely vibrating in crossflow sensory structure simulations. Studies on the effect of Reynolds Number are also performed for stationary sensory structure. For freely vibrating in crossflow (1D vibrating) sensory structure simulations, evaluations are performed on the resonance response of the structure (natural frequency of system is set to be equivalent to the frequency of vortex shedding of a single towed sensory structure). Studies on the effect of the natural frequency of the system is performed for the new and old vibrissa geometry due to their greater relevance to the sensing of the harbour seals with their vibrissae. Studies on the variation of the relative size effects between the wake-generator circular cylinder and the sensory structures are also conducted. The following sections on stationary and 1D vibrating sensory structures would provide greater details of the simulations and their corresponding results.

8.1 Stationary sensory structure

In this section, simulations on stationary sensory structures are described. Studies on the effects of Reynolds Number on the responses of the sensory structures are also evaluated. In these simulations, the characteristic diameter of the sensory structure is set to be equal to the characteristic diameter of the wake-generator circular cylinder and the Reynolds Number is 500. Constant inline spacing of $4D$ is applied between the trailing edge of the wake-generator circular cylinder and the leading edge of the sensory structure. Various sideways (offset) spacing is studied in order to determine the positional variation of the force response of each sensory structure as well as their sensitivity to such sideways variation. With reference to previous studies on wake stiffness, it is found that determination of the sensitivity of the responses of the sensory structures to various sideways displacement would be

strongly relevant to the understanding of the underwater sensing ability of the harbour seal vibrissae (where the seals sideways position resulted in significant variation in the relative position of the vibrissae to the wake generator) [4]. Figure 8.1 shows the mean drag coefficient (C_D) and mean lift coefficient (C_L) of a circular cylinder in the wake of a wake-generator circular cylinder of equivalent size at $Re = 9600, 14500$ and 19200 [4]. With reference to figure 8.1, the C_D and C_L are found to be relatively invariant for the experimented Reynolds Number. The C_D of the trailing stationary circular cylinder is found to mimic a parabolic profile with variation of sideways displacement for $-2.0 \leq y_0/D \leq 2.0$. Such profile is expected due to its relevance to the pressure field in the near-field wake of the wake-generator circular cylinder. The perfectly aligned position ('zero' sideways displacement) is found to result in the smallest drag coefficient due to the smallest pressure difference between the leading edge and trailing edge of the 'sensor' circular cylinder. This is due to the lower pressure in the leading edge of the circular cylinder as the trailing circular cylinder is centred about the line of lowest pressure (in the given plane) in the wake of the wake-generator circular cylinder for the given inlined spacing. Higher drag coefficient are observed for $|y_0/D| \leq 2.0$ due to the placement of the 'sensor' circular cylinder outside of the wake-affected zone of the wake-generator circular cylinder. A more interesting note on figure 8.1 is the near-linear variation of the mean lift coefficient (C_L) with the increase in the sideways displacement for $-1.0 \leq y_0/D \leq 1.0$. Such near-linear trend indicated the proportionality of the lift forcing on the trailing circular cylinder due to the wake of the wake-generator circular cylinder and its corresponding relative sideways positioning. Such trend is observed despite the unstable nature of the von *Kármán* street in the wake of both circular cylinders. It should be noted that the experimental results in figure 8.1 are based on experiments with inline spacing of $3D$ (measured from trailing edge of wake-generator circular cylinder to the leading-edge of the trailing circular cylinder).

In order to assess the validity and relative comparison of our simulation setup, simulations of tandem circular cylinder with inline spacing of $4D$ at $Re = 500, 5000, 10000, 20000$ and 30000 are performed. Figure 8.2 shows the mean drag coefficient and mean lift coefficient of the trailing circular cylinder in the wake of a wake-generator circular cylinder at various sideways displacement (y_0/D) and Reynolds Number. Despite the difference in inline spacing ($4D$ - current simulation and $3D$ - previous experiments), similar parabolic profiles and drag coefficients are observed. From figure 8.2, it can also be observed that the effects of Reynolds Number on the drag coefficient of

Figure 8.1: Mean Drag coefficient (left) and mean Lift coefficient (right) on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 9600$, 14500 , and 19200 [4].

the trailing circular cylinder are minute with similar drag coefficients amongst $Re = 500$, 10000 and amongst $Re = 5000$, 20000 , 30000 . The effects of Reynolds Number are more observed in the variation of the mean lift coefficients with sideways displacement. However, a near-linear profile is still observed on the variation of the mean lift coefficient with the sideways displacement for various Reynolds Number and $-0.9 \leq y_0/D \leq 0.9$. The effect of the Reynolds Number is found to significantly reduce the gradient of the near-linear variation of the mean lift coefficient with sideways displacement. However, such reduction is only observed for $Re = 500$ with insignificant effects observed among simulations at $Re = 5000$, 10000 , 20000 and 30000 . While the evaluations of the sensory circular cylinder are currently focused on the effects of Reynolds Number on the mean and amplitude of the force coefficients, it is also important to analyse the dominant forcing frequencies of the sensor circular cylinder. While the sensor circular cylinder will shed its own von Kármán street, it will simultaneously be exposed to a von Kármán street at various relative positioning. Figure 8.3 and figure 8.4 illustrate the frequency responses on the drag and lift forcing on the sensor circular cylinder, respectively. With reference to figure 8.3, two dominant response frequencies are observed for most Reynolds Number with exception of simulations at $Re = 500$ and $20,000$. These response frequencies are found to be in the vicinity of $0.17 - 0.2$ and $0.35 - 0.4$, which corresponds to the vortex shedding frequency and the dominant drag forcing frequency of the circular cylinders (same size), respectively. For $-0.15 \leq y_0/D \leq 0.15$, first dominant response frequency of drag coefficient on sensor circular cylinder is observed at $0.35 \leq f^* \leq 0.38$. For regions outside of such sideways displacement, switching between primary and secondary dominant response frequency of the drag coefficients are observed. In such cases, the primary dominant response frequencies are observed

Figure 8.2: Mean Drag coefficient (left) and mean Lift coefficient (right) on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at various Reynolds Number. Legend : blue circle - $Re = 500$; red square - $Re = 5000$; black diamond - $Re = 10,000$; magenta inverted triangle - $Re = 20,000$; green triangle - $Re = 30,000$.

in the vicinity of $0.17 \leq f^* \leq 0.2$. From figure 8.3, the strongest effect of Reynolds Number on the first dominant response frequency could be observed at $y_0/D = 1.2$, which represents the limit of the wake-affected zone. Simultaneously, a switch of secondary response frequencies from 0.17 - 0.2 to 0.35 - 0.4 is observed. Interestingly, for $Re = 500$, three dominant response frequency peaks are observed with most of the second and third response frequencies observed around 0.38 to 0.4. A third response frequency of drag forcing is found to always be in the vicinity of the primary or secondary response frequency. This is due to the similarity of sizes (and therefore, vortex shedding and drag forcing frequencies) of the wake-generator circular cylinder and sensor circular cylinder. As such, slightly-modified vortex shedding frequency of the sensor circular cylinder is found to result in tertiary response frequency. It should be noted that within a spacing of $4D$, the forcing frequency of a von *Kármán* street (in this case, from a wake-generator circular cylinder) on a structure should maintain the similar frequency (spacing) to the frequency of its shedding. Additionally, the response frequency in the vicinity of $0.17 \leq f^* \leq 0.2$ is due to the exposure of the sensor circular cylinder to only one of the major vortex tubes (instead of both vortex tubes, which is the case for perfectly inline configuration of the sensor circular cylinder).

In comparison, at most, only two main response frequencies are observed on the lift forcing of the corresponding setup of sensor circular cylinder (refer to figure 8.3). However, unlike the primary response frequencies of the drag forcing, the primary response frequencies of the lift forcing are found in the vicinity of 0.17 - 0.2. This frequency is found to correspond to the vortex shedding

frequency of the wake-generator circular cylinder. The secondary response frequency of lift forcing is found to coincide with the frequency of drag forcing with exception of simulations at $Re = 500$. It is also interesting to note that secondary frequency of lift forcing is only observed for sensor circular cylinder at higher Reynolds Number and $|y_0/D| \leq 0.45$. The occurrence of secondary frequency of lift forcing is found to increase with higher Reynolds Number. All the power corresponding to the

Figure 8.3: Response frequencies of the drag forcing on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at various Reynolds Number. Legend : blue circle - $Re = 500$; red square - $Re = 5000$; black diamond - $Re = 10,000$; magenta inverted triangle - $Re = 20,000$; green triangle - $Re = 30,000$.

secondary frequency of lift forcing is found to be significantly smaller than the primary frequency of lift forcing. Both these facts combined to explain the occurrence of such secondary frequency peak of lift forcing. For $|y_0/D| \leq 0.45$, vortex tubes in the wake of the wake-generator circular cylinder resulted in minuscule lift forcing due to the breaking of the vortex tubes as they collided with the sensor circular cylinder and restructured. However, for $|y_0/D| \leq 0.45$, no head-on collision occurred between the centre of the vortex tubes and the centroid of the sensor circular cylinder. As such, suction forces of the vortex tubes are found to result in the occurrence of such secondary lift forcing. In general, with reference to figure 8.4, the primary frequency of lift forcing on the sensor circular cylinder is observed to vary minutely at various sideways displacement. At higher Re , slightly lower primary frequency of lift forcing is observed on the sensor circular cylinder. Similar to the drag forcing on the sensor circular cylinder, the greatest effect of the Reynolds Number on the primary frequency of lift forcing is observed at $y_0/D = 1.2$, which corresponded to the border of the wake-affected zone of the wake of the upstream circular cylinder.

Observation on the response of the sensor circular cylinder from previous experiments and current simulations showed a near-linear gradient variation of mean lift force coefficient with sideways

displacement (for $|y_0/D| \leq 1.0$). The gradient of such near-linear trend (termed 'linear' line in subsequent emergence for convenience) provide strong indication of the 'sensed' lift forcing on the sensory structure, which allowed for better deduction of relative sideways (offset) positioning relative to target (in this case, wake-generator circular cylinder). Hence, evaluation on the relative lift forcing on the sensory geometries are performed with circular cylinder, elliptical cylinder, and old vibrissa geometry as sensory structures. A wake-generator circular cylinder is used to generate regular but unstable wakes. As aforementioned, due to limited time and resources, priority is given to freely vibrating in crossflow simulations due to its greater proximity to 'real' sensing situation of the harbour seal vibrissae. As such, the new vibrissa geometry is simulated in tandem with the wake-generator circular cylinder on in freely vibrating in crossflow setup. Such result will be presented in the next section. Figure 8.5 shows the mean drag and lift coefficients of the stationary trailing sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry) in the wake of a stationary circular cylinder of equivalent characteristic diameter at $Re = 500$ and 5000 . It should be noted that a higher Reynolds Number ($Re = 5000$) is simulated due to the mildness of the linear-trend of the mean lift coefficient of the sensor circular cylinder at $Re = 500$. Hence, in order to better assess the possibly geometry dependent sideways-positioning sensitivity, higher Re is simulated. From figure 8.2, it can be observed that the results indicated the consistency of parabolic (near-parabolic) profiles of the mean drag coefficients for various sideways positioning. Higher mean drag coefficient is observed for circular cylinder as compared to elliptical cylinder and old vibrissa geometry. This is expected due to the more streamlined geometries of elliptical cylinder and old vibrissa geometry as compared to circular cylinder. Hence, a lower pressure difference between the leading and trailing edge are observed. These mean drag coefficients of elliptical cylinder and old vibrissa geometry are found to be approximately half of the mean drag coefficients of the circular cylinder. Slightly higher mean drag coefficients are observed for elliptical cylinder at $Re = 500$ as compared to mean drag coefficients of elliptical cylinder at $Re = 5000$ (and therefore, old vibrissa geometry), which indicated strong Reynolds Number effects on the drag forcing of the elliptical cylinder. Weak effects of Reynolds Number on the drag coefficients are observed for circular cylinder and old vibrissa geometry with perceived consistency of their mean drag coefficients at various sideways positioning (refer to figure 8.5). Correspondingly, linear trends are observed on the mean lift coefficient of the elliptical cylinder and old vibrissa geometry for various sideways displacement relative to the wake-generator

circular cylinder. At $Re = 500$, similar peak of mean (absolute) lift coefficient are observed between circular cylinder and old vibrissa geometry with maximum lift coefficient occurring at $|y_0/D| = 0.9$. However, it is important to note that consistent mean lift coefficient of the old vibrissa geometry is observed between $0.3 \leq |y_0/D| \leq 0.9$. This is found to be due to the modification of the vortices in the wake of the circular cylinder due to the presence of the old vibrissa geometry. Longer transient period of the wake is observed for old vibrissa geometry with no sideways displacement relative to the wake-generator circular cylinder. At $Re = 500$, the gradient of the linear trend of the mean lift coefficient of the circular cylinder is found to be $(\Delta C_L / \Delta(y_0/D) = -0.1)$. In comparison, the mean lift coefficient of a sensor elliptical cylinder is found to vary similarly to the circular cylinder at various sideways displacement (peak mean lift coefficient at $|y_0/D| = 0.9$). However, the gradient of the linear trend of the elliptical cylinder $(\Delta C_L / \Delta(y_0/D) = -0.2)$ is observed to be more than twice to the corresponding gradient for circular cylinder. As a result, for $Re = 500$, it can be observed that the sensor elliptical cylinder has better sensitivity (based on mean lift forcing) to variability in its sideways positioning. However, such sensitivity is limited to the range of $|y_0/D| = 0.9$. As aforementioned, consistent mean lift coefficient is observed on the response of the old vibrissa geometry . In comparison, at $Re = 5000$ (refer to figure 8.5), larger mean lift coefficients are observed for all the geometries as compared to the corresponding coefficients at $Re = 500$. This is as expected due to the shedding of higher vorticity vortex tubes in the wake of the wake-generator circular cylinder as well as the trailing sensor structures. Consistency of linear and non-linear trend with sideways displacement are found on the mean lift coefficients of the circular and elliptical cylinders (as compared to $Re = 500$). Similar turning point can also be observed on the old vibrissa geometry at $|y_0/D| = 1.0$. Larger gradients of the variation of the mean lift coefficients with sideways displacement are observed for elliptical cylinder $(\Delta C_L / \Delta(y_0/D) = -0.4)$ and circular cylinder $(\Delta C_L / \Delta(y_0/D) = -0.26)$ at $Re = 5000$ (as compared to $Re = 500$). However, despite the lower gradients of mean lift coefficients on the old vibrissa geometry $(\Delta C_L / \Delta(y_0/D) = -0.18)$, a more consistent (with the linear trend) variation of mean lift coefficient is observed with sideways displacement. Interestingly, for the old vibrissa geometry, a sudden increase in the mean lift coefficient is observed at $|y_0/D| = 1.5$.

Similar to previous analysis on the forcing on the circular cylinder at various Re , analysis on the frequency peaks of the drag and lift forcing are performed for these sensor structures at $Re = 500$ and 5000 . Figure 8.6 illustrates the response frequencies of the drag forcing on the stationary

sensor structures for $Re = 500$ and 5000 . Similar to previous findings on the response frequencies, from figure 8.6, switching between primary and secondary response frequency of drag forcing on the sensor structures are observed with relevance to the sideways displacement. For $|y_0/D| \leq 0.15$, the primary response frequencies of all the sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry) are found to be $0.35 \leq f^* \leq 0.4$. However, beyond this region, the primary response frequencies are observed to switch with the secondary response frequencies (to the region of $0.17 \leq f^* \leq 0.22$). Interestingly, for $|y_0/D| \geq 0.3$, the frequencies of response of the trailing old vibrissa geometry at $Re = 500$ are found to lie in the midst of the primary and secondary frequencies of response of drag forcing on trailing circular cylinder and elliptical cylinder. Beyond $|y_0/D| = 0.15$, consistent primary response frequency of drag forcing on the trailing sensor structures are observed for various sideways displacement. The effect of Reynolds Number is found to be insignificant on the variability of the primary response frequency of drag forcing of the sensor structures. However, this could not be said of the secondary and tertiary response frequencies of the sensor structures. At $Re = 500$, the trailing old vibrissa geometry is found to consist of no secondary and tertiary response frequencies (as compared to $Re = 5000$). Secondary frequencies of the trailing old vibrissa geometry at $Re = 5000$ are commonly found in the region of $0.35 \leq f^* \leq 0.38$ for $|y_0/D| \geq 0.5$. As aforementioned, at $|y_0/D| = 0.15$, switching of region of secondary response frequency is observed. Similarly, at $Re = 5000$, the secondary response frequency of the trailing circular cylinder and elliptical cylinder is found to perform the frequency switching with dependence on the sideways displacement with minimal variability beyond $|y_0/D| = 0.3$. In comparison, greater variability of the secondary response frequency are observed at $Re = 500$ for trailing circular cylinder and elliptical cylinder. With reference to figure 8.6, the secondary response frequencies of the drag forcing on the trailing circular are found to switch between the region of $0.16 \leq f^* \leq 0.18$ and $0.35 \leq f^* \leq 0.43$ in an alternating pattern as the sideways displacement is increased. From figure 8.6, scarce occurrence of tertiary response frequencies are observed with consistent prevalence observed only at $Re = 500$ for trailing circular cylinder. At $Re = 5000$, no tertiary response frequency are observed for trailing circular cylinder. Tertiary response frequencies of the trailing elliptical cylinder and old vibrissa geometry is found to occur at $0.35 \leq f^* \leq 0.39$ at certain sideways displacement. At $Re = 500$, a perfectly inlined old vibrissa geometry exhibited a tertiary response frequency at $f^* = 0.35$. However, trailing circular cylinder and elliptical cylinder is found to exhibit

tertiary frequencies with predominant occurrence at $0.34 \leq f^* \leq 0.4$ for $|y_0/D| \geq 0.3$. Similar to analysis on the forcing of the trailing circular cylinder at various Re , three response frequencies are observed on the drag forcing and two response frequencies are observed on the lift forcing of the trailing sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry). Figure 8.7 shows the response frequencies of the lift forcing on the stationary sensor structures at $Re = 500$ and 5000 . With reference to figure 8.7, the primary response frequencies of the lift forcing of the trailing circular cylinder and elliptical cylinder are observed in the region of $0.17 \leq f^* \leq 0.22$. The effects of Reynolds Number are found to be negligible on the lift forcing of the trailing circular cylinder and elliptical cylinder. However, this is found to be untrue on the primary response frequency of the old vibrissa. Significantly larger primary response frequency ($f^* \approx 0.3$) is observed on the primary response frequency of lift forcing on the trailing old vibrissa geometry at $Re = 500$. At $Re = 5000$, such primary response frequency of lift forcing is observed to remain in similar region of frequency as the corresponding frequency of trailing circular cylinder and elliptical cylinder. For all the simulated sensor structures, only slight variations are observed on the primary response frequencies of the lift forcing on the trailing sensor structures (with exception of old vibrissa geometry, where increase in primary frequency is observed as the trailing old vibrissa geometry moved away from the centreline of the wake-generator circular cylinder). Secondary frequencies of lift forcing on the trailing sensor structures, none are observed at $Re = 5000$ and at $Re = 500$ for old vibrissa geometry. Only trailing circular cylinder and elliptical cylinder at $Re = 500$ are observed to signify the presence of secondary response frequencies.

While Reynolds Number is found to play significant role in determining the forcing and dominant response frequencies, certain inherent sensitivity are observed in the trailing sensor structures. Such sensitivity is observed through the variation of the mean lift coefficient and mean drag coefficient for various sideways displacement of the sensor structures relative to the wake-generator circular cylinder. Despite the current findings on the sensitivity of the sensor structures at various simulated Re , conclusive deductions are yet to be achieved due to the necessity to assess the boundary conditions effect on the response of the sensor structures. As this section assessed the response of stationary sensor structures in the wake of an equivalently-sized stationary circular cylinder, the next section shall evaluate the response of freely vibrating in crossflow sensor structures in the wake of a stationary circular cylinder.

Figure 8.4: Response frequencies of the lift forcing on the stationary circular cylinder in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at various Reynolds Number. Legend : blue circle - Re = 500 ; red square - Re = 5000 ; black diamond - Re = 10,000 ; magenta inverted triangle - Re = 20,000 ; green triangle - Re = 30,000.

Figure 8.5: Mean Drag coefficient (left) and mean Lift coefficient (right) on the stationary sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry) in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 500$ and 5000 .

Figure 8.6: Response frequencies of the drag forcing on the stationary sensor structures (type of structure indicated by legend) in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 500$ and 5000 . Legend : blue circle - circular cylinder $Re = 500$; red square - elliptical cylinder at $Re = 500$; black diamond - old vibrissa geometry at $Re = 500$; green inverted triangle - circular cylinder at $Re = 5000$; teal triangle - elliptical cylinder at $Re = 5000$; yellow circle - old vibrissa geometry at $Re = 5000$.

Figure 8.7: Response frequencies of the lift forcing on the stationary sensor structures (type of structure indicated by legend) in the wake of a wake-generator circular cylinder with equivalent characteristic diameter at $Re = 500$ and 5000 . Legend : blue circle - circular cylinder $Re = 500$; red square - elliptical cylinder at $Re = 500$; black diamond - old vibrissa geometry at $Re = 500$; green inverted triangle - circular cylinder at $Re = 5000$; teal triangle - elliptical cylinder at $Re = 5000$; yellow circle - old vibrissa geometry at $Re = 5000$.

8.2 Freely vibrating in crossflow sensory structure

In this section, the trailing sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry) are allowed to freely vibrating in crossflow direction at their respective resonance (refer to chapter 6). Such evaluations at their respective resonance would provide indication of the 'worst' possible performance (due to strongest effect of self-induced vibration) of each respective sensor structure in sensing the perturbations. Similar simulation setup to the stationary trailing sensor structures are applied (inline spacing $4D_1$, $Re = 500$ (based on characteristic diameter of trailing sensor structures), where D_1 is the characteristic diameter of the wake-generator circular cylinder). However, in this section, simulations are performed with various sizes of wake-generator circular cylinder. With reference to chapter 5, a Characteristic Dimension Ratio (CDR) is defined as the ratio of the characteristic diameter of the wake-generator circular cylinder (D_1) to the characteristic diameter of the sensor structures. Through variation in sizes of the wake-generator circular cylinder (and therefore, the size of the simulation domain refer to chapter 5), the CDR of simulations presented in this section are allowed to vary between 0.6 to 3.0. Due to limitation of time and resources, strategic selection of CDR (1.0, 1.5, 2.0, 2.5) are performed for simulations on trailing new vibrissa geometry. Hence, in addition to the prior sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry), a new vibrissa geometry setup is added to the simulations as the sensor structures. In order to assess the role of natural frequency of the system (spring-mass system), variation of reduced velocity ($U/f_n D$) is conducted for new and old vibrissa geometry at $CDR = 1.0, 1.5, 2.0, 2.5$ and 3.0. A swift assessment on the effects of inline spacing is also performed on the trailing new vibrissa geometry at inline spacing of $3D$ and $4D$, respectively.

Initial assessment on the geometrical effect on the forcing response of the sensor structures in the wake of a wake-generator circular cylinder are dedicated for studies on the identification of various sizes of wake-generator relative to the sensor structures. Figure 8.8 shows the amplitude of oscillation of the trailing sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry and new vibrissa geometry) in the wake of a circular cylinder at various relative sizes ($CDR = 0.6$ to 3.0) at $Re = 500$ (based on the characteristic diameter of the trailing sensor structures) and inline spacing of $4D$ with no sideways displacement. With reference to figure 8.8, significant variation of the amplitudes of response is observed for various sensor structures. While the amplitude of response of the trailing circular cylinder is found to be largest at $CDR = 1.2$, the greatest amplitude of response of the trailing

elliptical cylinder is observed at $CDR = 1.4$. In the subsequence of these occurrence of largest amplitude of response, the amplitudes of response of the trailing circular cylinder and elliptical cylinder are found to steadily decrease with increase in the characteristic diameter of the wake-generator circular cylinder. However, prior to their peak responses, the trailing circular cylinder and elliptical cylinder are found to respond with increasingly larger amplitude of oscillation as the characteristic diameter of the wake-generator circular cylinder is increased (from $CDR = 0.6$ to respective peak amplitude of response for trailing circular cylinder and elliptical cylinder). Such behaviour is unexpected as increase in the amplitude of oscillation is expected with increase in the size of the wake-generator circular cylinder due to the stronger vortex tubes in its wake. Hence, a more detailed investigation is necessary in order to determine the main reason for such limitation in amplitude of oscillation. Figure 8.9 depicts the vortex cores in the wake of a 1D vibrating trailing circular cylinder at $Re = 500$, $CDR = 1.0$ under Wake-Induced Vibration (WIV). For conclusive assessment, analysis on the wake topology and WIV of a 1D vibrating trailing circular cylinder for $CDR = 0.6$ and 2.6 is also presented here. Details of the wake topology for 1D vibrating trailing circular cylinder at $CDR = 0.6$ and 2.6 can be found in appendix F. With reference to figures 8.9, F.1, F.2, it can be observed that for $CDR = 1.0$, the motion of the circular cylinder lagged behind the vortex tubes (and their corresponding forcing). The lagging occurred at certain perfect phase angle, which did not result in head on collision between vortex tubes and the trailing circular cylinder. In comparison, for $CDR = 2.6$, double oscillation amplitude could be observed. With reference to figure F.2, it can be observed that the first vortex tube encountered by the trailing circular cylinder (counter-clockwise rotation) resulted in large amplitude of oscillation (refer to figure F.2 for $t^* = 272 - 280$). This oscillation is found to occur at a certain phase angle such that no collision occurred between the first vortex tube encountered and the trailing circular cylinder. However, due to this large amplitude of oscillation, large recovery force (towards neutral axis or centreline of the wake-generator circular cylinder) resulted in large acceleration (and therefore, larger terminal velocity). However, due to the domination of such recovery force over the forcing due to the vortex tubes, head on collision occurred between the subsequent vortex tube (clockwise rotation) and the circular cylinder (refer to figure F.2 for $t^* = 281 - 287$). From figure F.2, the subsequent progression of the vortex tube after collision showed breaking of the vortex tube, which is found to weaken the vortical braids (due to loss of source of energy/vorticity). Such weakening of vortical braids is evident in the reduction of the size of the subsequent vortex tube

(counter-clockwise rotation). This subsequent vortex tube is found to evidently decrease its vorticity (energy) due to the sudden surge of energy-request from the vortical braids as a consequence of the weakening (loss) of the collided vortex tube. Hence, a full oscillatory response of a trailing 1D vibrating circular cylinder for $CDR = 2.6$ shows one large oscillation and one (or two) small oscillation(s). From figure 8.9, it can be observed that the amplitude of oscillation of the trailing circular cylinder is equivalent to the amplitude of oscillation of a towed 1D vibrating circular cylinder at the same reduced velocity (refer to chapter 6). This is due to the weaker influence of the With reference to figure F.1, it can be observed that the 1D vibrating trailing circular cylinder shed its own von *Kármán* street. This is observable through the different arrangement of vortical tubes between the wakes of the leading and trailing circular cylinder as well as the similarity of the wake topology (and shedding mode 'P+S') to the towed 1D vibrating circular cylinder at the same reduced velocity. The main difference on the comparison between the wake topologies is the absence of vortical braids in the wake topology of the a trailing 1D vibrating circular cylinder at $CDR = 0.6$. The comparison of the motion of a trailing 1D vibrating circular cylinder and their corresponding frequency for $CDR = 0.6, 1.0$ and 2.6 showed significant variation with smaller dominant frequency of response observed for larger wake-generator circular cylinder. This is mainly due to the effect of relativity of the sizes of the structures. Suppose, f_1^* is the reduced vortex shedding frequency of the wake-generator circular cylinder with characteristic dimension D_1 and f_2 is the corresponding reduced frequency detected and response frequency of the trailing structure with characteristic dimension D_2 and f is the absolute vortex shedding frequency. Then, the following relation ship shall stand true for a trailing freely vibrating in crossflow circular cylinder which responded in correspondence to vortices shed by the wake-generator circular cylinder:

$$\frac{f_1^* U_1}{D_1} = f = \frac{f_2^* U_2}{D_2} \quad (8.1)$$

For $U_1 = U_2$, re-arranging equation 8.1, the author have:

$$f_2^* = \frac{f_1^* D_2}{D_1} \quad (8.2)$$

It should be noted that equation 8.2 will only hold true if the vortex tubes (vortices) in the wake of the wake-generator dominates the response of the trailing structure in its wake. Hence, from equa-

Figure 8.8: RMS amplitude of oscillation of trailing freely vibrating in crossflow sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry, new vibrissa geometry) at $Re = 500$ (based on characteristic diameter of trailing sensor structures) for inline spacing of $4D$ and no offset positioning. Natural frequency of each simulation setting is tuned to the resonance frequency of the respective sensor structure. Legend for trailing sensor structure: blue cross - circular cylinder; red square - elliptical cylinder ; black diamond - old vibrissa geometry at $U^* = 12$; green circle - new vibrissa geometry at $U^* = 5$; dark yellow inverted triangle - old vibrissa geometry at $U^* = 5$.

tion 8.2, it can be deduced that the dominant response frequency of the trailing structure is inversely proportional to the characteristic dimension ratio ($CDR = D_1/D_2$) of the structures. Figure 8.10 shows the response frequencies of the trailing freely vibrating in crossflow sensor structures at their resonance corresponding to setup described in figure 8.8. In relevance to earlier description on the response of trailing circular cylinder at $CDR = 2.6$, from figure 8.10, a secondary response frequency could be observed at a frequency ratio of ≈ 0.74 . Secondary ($f_2 = f_{r2}/f_n = 0.6$) and tertiary ($f_3 = f_{r3}/f_n = 1.5$) frequency peaks are observed in the response of trailing circular cylinder at $CDR = 0.6$. For $1.0 \leq CDR \leq 2.0$, a single dominant response frequency is observed where the trailing circular cylinder is observed to lock-in to the vortices in the wake of the wake-generator circular cylinder (instead of shedding its own vortices). For larger wake-generator circular cylinder ($CDR \geq 2.2$), secondary response frequency is found to prevail at a consistent frequency ratio indicating the independence of these frequencies to the size of the wake-generator circular cylinder. For this regime of $CDR (\geq 2.2)$, the primary response frequency of the trailing circular cylinder is found

Figure 8.9: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von *Kármán* street) on the interaction between the von *Kármán* street and a trailing circular cylinder at inline spacing of $4D$ and $CDR = 1.0$. Direction of view: left to right and top to bottom (refer to dimensionless time located at the bottom of each picture, for reference).

to coincide with vortex shedding frequency of the wake-generator circular cylinder (refer to equation 8.2). With reference to figure 8.10, for $CDR \leq 0.9$, secondary and tertiary (except $CDR = 0.9$) of response frequencies are observed on the response of the trailing circular cylinder. For $CDR \leq 0.9$, the primary response frequencies of the trailing circular cylinder is found to coincide with the vortex shedding frequencies of the trailing circular cylinder. This is expected as the vortex shedding of the trailing circular cylinder dominates over the forcing due to vortices from the smaller wake-generator circular cylinder. However, the secondary and tertiary response frequencies of the trailing circular cylinder clearly indicated the presence of the vortex shedding frequency of the wake-generator circular cylinder. For $CDR \leq 0.8$, correlation is found to exist among the primary, secondary and tertiary response frequencies of the trailing circular cylinder. It is found that the primary frequency is the average of the secondary and tertiary response frequencies of the trailing circular cylinder at for a given $CDR (\leq 0.8)$. While the frequency scaling relative to the trailing structure is performed in equations 8.1, 8.2, frequency scaling relative to the wake-generator circular cylinder is yet to be performed. Such frequency would indicate the frequencies of response of the trailing structure relative to the normalised vortex shedding frequency of the wake-generator circular cylinder. Similar to earlier definition of variables in equations 8.1, 8.2, the scaled frequency relative to the vortex shedding frequency of the wake-generator circular cylinder can be defined as follows:

$$\frac{f_1 D_1}{U_1} = St = \frac{f_2 D_2}{U_2} \quad (8.3)$$

For $U_1 = U_2$, re-arranging equation 8.1, the author have:

$$f_2 = \frac{f_1 D_1}{D_2} \quad (8.4)$$

where St is Strouhal number (non-dimensional vortex shedding frequency). Hence, in order to scale the response frequencies of the trailing circular cylinder relative to the vortex shedding frequency of the wake-generator circular cylinder. Figure 8.11 shows the scaled vortex shedding frequency ratio (relative to the wake-generator circular cylinder) corresponding to figure 8.10. From figure 8.11, it is conclusive that the primary response frequencies of the trailing circular cylinder for $CDR \geq 1.0$ are locked-in to the vortex shedding frequency of the wake-generator circular cylinder. Such observation is also found to be true on the primary response frequency of a trailing elliptical

cylinder. From figure 8.11, strong similarity between the primary, secondary and tertiary response frequencies of the trailing circular cylinder and elliptical cylinder are observed with slightly lower response frequencies observed for elliptical cylinder. Locked-in between the vortex shedding frequency of the wake-generator circular cylinder and the primary response frequency of the trailing elliptical cylinder is also observed. Interestingly, despite their weaker VIV (as compared to circular

Figure 8.10: Response frequency ratios of oscillation of trailing freely vibrating in crossflow sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry, new vibrissa geometry) at $Re = 500$ (based on characteristic diameter of trailing sensor structures) for inline spacing of $4D$ and no offset positioning. Natural frequency of each simulation setting is tuned to the resonance frequency of the respective sensor structure. Legend for trailing sensor structure: blue cross - circular cylinder; red square - elliptical cylinder ; black diamond - old vibrissa geometry at $U^* = 12$; green inverted triangle - new vibrissa geometry at $U^* = 5$; violet triangle - old vibrissa geometry at $U^* = 5$.

cylinder), the primary response frequencies (or secondary response frequencies) of the trailing elliptical cylinder for $CDR \leq 0.9$ are found to be within the vicinity of its vortex shedding frequency (instead of the vortex shedding frequency of the wake-generator circular cylinder). Such response is found to occur not due to the change in the nature of vortex shedding of the flow around 1D vibrating elliptical cylinder (refer to figure F.3 in appendix F). This response is found to be mainly due to the larger added mass coefficient of the elliptical cylinder in the crossflow direction. Hence, despite the smaller amplitude of VIV of the trailing elliptical cylinder, the pressure force of the vortices in the wake of the smaller wake-generator circular cylinder resulted in minor effect on the VIV response (VIV dominates WIV), as observed through the occurrence of the WIV frequency in the secondary and tertiary response frequencies of the trailing elliptical cylinder. Figure 8.12 illustrates the vortex cores in the wake of a stationary wake-generator circular cylinder and a trailing freely vibrating in crossflow elliptical cylinder at $CDR = 1.4$ (peak amplitude of response of trailing elliptical cylinder) for one cycle of oscillation of the trailing elliptical cylinder and $Re = 500$ (based on characteristic

diameter of trailing elliptical cylinder). For conclusive description, it is necessary to compare the response of the elliptical cylinder at various regime of response i.e. singular response frequency regime (represented by $CDR = 1.4$), dual response frequencies regime (represented by $CDR = 2.8$), and triple response frequencies regime (represented by $CDR = 0.6$). Similar to previous finding on trailing circular cylinder, for singular response frequency (refer to figure 8.12), the trailing elliptical cylinder is found to move with certain phase lag to the vortex tubes in the wake of the wake-generator circular cylinder. The dodging vortex tubes motion occurred perfectly with no occurrence of 'reset'. 'Reset' is signified by a consistent move of the trailing structure in perfectly 'dodging' (phase lagging) the vortex tubes through crossflow motion along the vortical braids with occasional occurrence of swift small amplitude motion (resulting in collision with vortex tubes). Upon completion of the swift small amplitude motion, the trailing elliptical cylinder is found to return to the state of perfectly dodging the vortex tubes. The occurrence of 'reset' is unlike the consistent switch in amplitudes of motion observed in the case of trailing circular cylinder. The occurrence of 'reset' is observed in the response of the trailing elliptical cylinder for $CDR = 2.8$ (refer to figure F.4), where dual response frequencies are identified.

With reference to figure 8.8, the amplitudes of response of the new and old vibrissa geometries at $U^* = 5$ (in the vicinity of their resonance) are found to be very similar for the various CDR simulated. Slightly greater variability with CDR is observed on the response of the old vibrissa geometry. However, from figures 8.10, 8.11, it can be observed that no secondary and tertiary frequencies are observed in the response of the new and old vibrissa geometry at $U^* = 5$ for $CDR \geq 1.0$. The primary response frequencies of the new and old vibrissa geometry at $U^* = 5$ are found to be equivalent to the primary response frequencies of the circular and elliptical cylinder, which indicates the dominance of the forcing by the vortices in the wake of the wake-generator circular cylinder. The absence of secondary and tertiary frequencies in the response of the new and old vibrissa geometries is found to reveal the nature of these frequencies and the intricacy of eliminating these frequencies. While similar 'dodging' response are observed on the motion response of the new and old vibrissa geometry for single response frequency cases, this is not true for larger CDR . Figure 8.13 shows the vortex cores in the wake of the wake-generator circular cylinder and the WIV response of the trailing new vibrissa geometry for $Re = 500$ (based on D_2), $U^* = 5$ and $CDR = 3.0$. As the motion response of the new and old vibrissa geometries are found to be relatively similar, priority analysis

will be performed on the new vibrissa geometry due to their greater relevance to the real vibrissae of seals. With reference to figure 8.13, it is observed that, unlike the response of the trailing circular cylinder and elliptical cylinder at $CDR \geq 2.6$, the motion of the new vibrissa geometry is found to be very sinusoidal-like. The motion of new vibrissa geometry is also observed to be well synchronised with the wake of the wake-generator circular cylinder with minimal deformation in the wake

Figure 8.11: Scaled response frequency ratios of the oscillation of trailing freely vibrating in crossflow sensor structures (circular cylinder, elliptical cylinder, old vibrissa geometry, new vibrissa geometry) at $Re = 500$ (based on characteristic diameter of trailing sensor structures) for inline spacing of $4D$ and no offset positioning. Natural frequency of each simulation setting is tuned to the resonance frequency of the respective sensor structure. Legend for trailing sensor structure: blue cross - circular cylinder; red square - elliptical cylinder ; black diamond - old vibrissa geometry at $U^* = 12$; green inverted triangle - new vibrissa geometry at $U^* = 5$; violet triangle - old vibrissa geometry at $U^* = 5$.

subsequent to the new vibrissa geometry observed. However, simulations with trailing old vibrissa geometry $U^* = 12$ showed significantly different amplitudes of response and response frequencies to the von *Kármán* street of the wake-generator circular cylinder at various CDR. From figure 8.8, a consistently increasing amplitude of response of the old vibrissa geometry is observed with increase in the characteristic diameter of the wake-generator circular cylinder. This indicated a strong influence of the natural frequency of the system on the amplitude of the response of any elliptical-based sensor structure (due to the directionality of the added mass coefficient). From figure 8.11, it is found that the primary response frequencies of the old vibrissa geometry at $U^* = 12$ followed the consistent trend of decreasing frequency ratio with increasing size of the wake-generator circular cylinder. This indicated that the response of the old vibrissa geometry at $U^* = 12$ is dominated by Wake-Induced Vibration (WIV). Interestingly, the primary response frequencies of the old vibrissa geometry at $U^* = 12$ and $CDR \leq 0.9$ are found to correspond to the vortex shedding frequency of the wake-generator circular cylinder. However, due to the significantly lowered natural frequency of

Figure 8.12: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Kármán street) on the interaction between the von Kármán street and a trailing elliptical cylinder at inline spacing of $4D$ and $CDR = 1.4$. Direction of view: left to right and top to bottom (refer to dimensionless time located at the bottom of each picture, for reference).

the system, the response frequency ratio is found to be significantly higher than the corresponding frequencies of the trailing circular cylinder, elliptical cylinder, new and old vibrissa geometry at $U^* = 5$. However, with reference to figure 8.11, the consistency of the scaled response frequency indicating the occurrence of WIV can be observed in the response of the old vibrissa geometry at $U^* = 12$ for $CDR \geq 0.7$. Through lowering of the natural frequency of the system, the old vibrissa geometry is found to respond with singular frequency peaks for $CDR \geq 1.0$. However, for $CDR \leq 0.9$, two or more response frequencies are identified in the response of the old vibrissa geometry with five response frequencies observed in the PSD of the response of the old vibrissa geometry for $CDR = 0.6$.

The current finding indicated the significance of the natural frequency of the system for flow sensing purpose. As such, it is necessary for us to evaluate the effects of variation in the natural frequency of the system on the motion and frequency responses of the trailing sensor structures.

8.2.1 Effects of Natural Frequency of Sensor Structure System

In this section, the motion and frequency responses of trailing new and old vibrissa geometries at $U^* = 3, 4, 5, 6$ will be performed. An additional simulation on trailing elliptical cylinder at $U^* = 12$ is also performed in order to compare and confirm the response of a trailing elliptical-based geometry at the given reduced velocity (natural frequency). All the simulations are performed with an inline spacing of $4D_1$ between the leading edge of the trailing structure to the trailing edge of the wake-generator circular cylinder. Figure 8.14 shows the RMS amplitudes of oscillation of the trailing freely vibrating in crossflow new and old vibrissa geometries at various U^* , $Re = 500$, $m^* = 1.04$ in the wake of a wake-generator circular cylinder ($CDR = 0.6$ to 3.0). With exception of $CDR = 1.0$ and 3.0 , the amplitudes of oscillations of the new and old vibrissa geometries are found to increase with increasing U^* . For $U^* = 6$ (resonance condition), the amplitudes of oscillation of the new vibrissa geometry are found to increase considerably at various CDR . Comparison between the amplitude of response of the old vibrissa geometry at $U^* = 12$ and other U^* showed increasing disparity between their corresponding amplitudes of oscillation with doubled (or greater) amplitudes of oscillation observed at $CDR = 2.5$. Hence, from figure 8.14, it can be concluded that, for $CDR \geq 1.5$, the amplitudes of oscillation of the new and old vibrissa geometries will increase with lower natural frequency of the system (higher reduced velocity). Comparison between the amplitudes of response

of the new and old vibrissa geometry at their respective reduced velocities showed slightly larger amplitude of oscillation of the old vibrissa geometry as compared to the corresponding amplitude of the new vibrissa geometry. This showed the insignificant effect of the variation in the wavelength of undulatory elliptical cylinder on the amplitudes of response of the structures to vortex tubes. Through our additional simulation on a trailing freely vibrating in crossflow elliptical cylinder at $U^* = 12$ in the wake of a wake-generator circular cylinder at $CDR = 2.8$, similar amplitude of oscillation and dominant response frequency are achieved. This confirmed the strong effects of the natural frequency of the system on the amplitudes of oscillation of the structure.

While lower natural frequency of the system allowed for greater distinctive amplitudes of response of the trailing sensor structures, their influence on the response frequencies of the trailing structures are yet to be analysed. Figure 8.15 illustrates the corresponding response frequencies of the trailing sensor structures presented in figure 8.14. Observations on figure 8.15 revealed similar trend and response frequency ratios of the new and old vibrissa geometries at their respective U^* . This indicated the WIV response of the new and old vibrissa geometries, which represents the domination of the von Kármán street (scaled to the size of the wake-generator circular cylinder) on the response of the new and old vibrissa geometries. With increase in U^* , the response frequency ratio of the new and old vibrissa geometries are found to increase correspondingly. Interestingly, the response frequency ratio of the new and old vibrissa geometries are found to collapse together with larger wake-generator circular cylinder. From figure 8.15, it is also observed that no secondary frequencies are observed for $U^* = 5$ and 6, regardless of CDR . Similar to previous findings, for $U^* = 3$ and 4, secondary frequencies on the response of the old and new vibrissa geometries are observed. These frequencies are found to be independent of CDR for old vibrissa geometries and partially dependent on CDR for new vibrissa geometries (no secondary response frequency for $CDR = 1.0$ and 1.5 with consistent occurrence and value for $CDR = 2.0, 2.5$, and 3.0). Figure 8.16 shows the scaled frequency of response (refer to equation 8.4) of the various trailing sensor structures corresponding to the response frequencies in figure 8.15. From figure 8.16, it can be observed that the primary response frequencies of the trailing new and old vibrissa geometries are proportionately scaled to their frequency ratios. This further confirms the domination of the von Kármán street of the wake-generator circular cylinder with independent response frequencies to the natural frequency of the system. However, the secondary response frequencies of the system is found to be a func-

tion of the natural frequency of the system and their occurrence is found to be correlated to CDR. Hence, through lowering of the natural frequency of the system, proportional increase in amplitude of oscillation with CDR could be achieved. Simultaneously, the range of sensitivity of the primary response frequency of the geometry could also be increased. The significant disadvantage of the lowered natural frequency of the system is its proneness to capturing perturbations due to background flow (noise). Due to the absence of such noise (perturbations) in simulations, distinct advantage of lower natural frequency of the system is evident. However, when deployed in natural environment, delicate balance between sensitivity of the sensor structure and their impulse to accrue noise needs to be well-calibrated. Hence, the author believe that the natural frequency of the system of the sensor structure should be calibrated to tailor to the individual trials of the sensors.

8.2.2 Effect of Inline Spacing Variation In Near-Field Wake

In order to swiftly assess the significance of inline spacing, slight variation of the inline spacing between the wake-generator circular cylinder and trailing freely vibrating in crossflow new vibrissa geometry are assessed at inline spacing of $3D_1$ and $4D_1$ for $CDR = 1.5$. Such test is performed due to the known regime of flow response of tandem circular cylinders at the given inline spacings [110]. For tandem circular cylinders at inline spacing of $3D_1$, a single von *Kármán* street would be shed. While for tandem circular cylinders at inline spacing of $4D_1$, dual von *Kármán* streets would be shed with each cylinder shedding their own von *Kármán* street. Hence, the current assessment would evaluate the occurrence of single and dual von *Kármán* streets based on their respective inline spacing. Figure 8.17 shows the amplitudes of response and response frequency ratios of the trailing new vibrissa geometry at $U^* = 3, 4$ and 5 , $Re = 500$ and $m^* = 1.04$ for inline spacing of $3D_1$ and $4D_1$ in the wake of a wake-generator circular cylinder with $CDR = 1.5$. From figure 8.17, for $U^* = 3$, significantly lower amplitude of response of the trailing new vibrissa geometry is observed for both inline spacing configurations with slightly larger amplitude observed for $S = 4D_1$, where S is the inline spacing between trailing edge of the wake-generator and the leading edge of the trailing sensor structure. For $U^* = 4$ and 5 , alternating larger amplitudes between trailing new vibrissa geometry at $S = 3D_1$ and $4D_1$ are observed. From figure 8.17, proportionately scaled primary response frequencies to the natural frequency of the system (U^*) is observed with secondary and tertiary frequency peaks only observed for trailing new vibrissa geometry at $S = 3D_1$. However, significant distinction in

the primary response frequencies of the trailing new vibrissa geometry at $S = 3D_1$ and $4D_1$ are evident. For all the simulated U^* , the response frequencies of the trailing new vibrissa geometry at $S = 4D_1$ indicated lock-in to the frequency of vortex shedding of the wake-generator circular cylinder. For the trailing new vibrissa geometry at $S = 3D_1$ at all U^* , the primary response frequencies are found to lock-in at 1.5 times the frequency of vortex shedding of the wake-generator circular cylinder at $CDR = 1.5$ with $S = 4D_1$. Interestingly, the primary response frequency ($f_1 = 0.42$) of the trailing new vibrissa geometry at $S = 3D_1$ and $U^* = 3$ is found to be the average of its secondary ($f_2 = 0.64$) and tertiary response ($f_3 = 0.28$) frequencies indicating the strong correlation of all the response frequencies. Such distinction between the primary response frequencies for the new vibrissa geometry at slightly differing inline spacing is due to the different form of vortex shedding observed in the wakes of each structure. Figure 8.18 shows the isometric view of the vortex cores in the wake of the wake-generator circular cylinder and trailing new vibrissa geometry for $S = 3D_1$ and $4D_1$, $U^* = 5$ and $Re = 500$. From figure 8.18, it can be observed that similar shedding mode to the case of tandem circular cylinders are observed in the vortex shedding of the wake-generator circular cylinder and the response of the trailing new vibrissa geometry. For trailing new vibrissa geometry at $S = 3D_1$, single von Kármán street are observed with significantly reduced vortical braids observed upon shedding from the trailing new vibrissa geometry (as compared to wake topology for $S = 4D_1$). For trailing new vibrissa geometry at $S = 4D_1$, a 'dual' von Kármán street configuration is observed. However, in actuality, only a single von Kármán street is observed as the trailing new vibrissa geometry is found to respond to Wake-Induced Vibration resulting in its motion which is found to cause minimal changes to the wake topology of the wake-generator circular cylinder. Hence, despite the similarity in its appearance to the cases of tandem circular cylinders, distinctions are observed in the wake topology as the trailing structure is replaced from circular cylinder to a new vibrissa geometry (more streamlined body) resulting in a single von Kármán street with little to no modification of the wake topology due to the trailing new vibrissa geometry.

The results presented in this chapter indicated similar amplitudes of response between the circular cylinder, elliptical cylinder, old and new vibrissa geometries in a near-field sensing. Such result is not applicable to far field sensing where the advantage of new (and/or old) vibrissa geometry(s) is prevalent in their natural habitat, where the sensitivity of the seal vibrissae is found to perceive minute (yet regular) perturbations. Such evaluation would require simulations in an extensive domain where

significant amount of inline spacing ($\geq 30D_1$) is introduced between the wake-generator and trailing structure.

Figure 8.13: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von *Kármán* street) on the interaction between the von *Kármán* street and a trailing new vibrissa geometry at $U^* = 5$, inline spacing of $4D$ and $CDR = 3.0$. Direction of view: left to right and top to bottom (refer to dimensionless time located at the bottom of each picture, for reference).

Figure 8.14: RMS amplitude of oscillation of trailing freely vibrating in crossflow new and old vibrissa geometry at $Re = 500$ (based on characteristic diameter of trailing sensor structure) for various reduced velocities (U^*). Legend for trailing sensor structures: 'New' indicates new vibrissa geometry ; 'Old' indicates old vibrissa geometry ; blue cross - new vibrissa at $U^* = 3$; red square - new vibrissa at $U^* = 4$; black diamond - new vibrissa at $U^* = 5$; green inverted triangle - new vibrissa at $U^* = 6$; teal triangle - old vibrissa at $U^* = 3$; dark yellow circle - old vibrissa at $U^* = 4$; violet plus - old vibrissa at $U^* = 5$; grey asterisk - old vibrissa at $U^* = 12$.

Figure 8.15: Response frequency ratios of oscillation of trailing freely vibrating in crossflow new and old vibrissa geometry at $Re = 500$ (based on characteristic diameter of trailing sensor structure) for various reduced velocities (U^*). Legend for trailing sensor structures: 'New' indicates new vibrissa geometry ; 'Old' indicates old vibrissa geometry ; blue cross - new vibrissa at $U^* = 3$; red square - new vibrissa at $U^* = 4$; black diamond - new vibrissa at $U^* = 5$; green inverted triangle - new vibrissa at $U^* = 6$; teal triangle - old vibrissa at $U^* = 3$; dark yellow circle - old vibrissa at $U^* = 4$; violet left-oriented-triangle - old vibrissa at $U^* = 5$; grey right-oriented-triangle - old vibrissa at $U^* = 12$.

Figure 8.16: Scaled response frequency ratios of oscillation of trailing freely vibrating in crossflow new and old vibrissa geometry at $Re = 500$ (based on characteristic diameter of trailing sensor structure) for various reduced velocities (U^*). Legend for trailing sensor structures: 'New' indicates new vibrissa geometry ; 'Old' indicates old vibrissa geometry ; blue cross - new vibrissa at $U^* = 3$; red square - new vibrissa at $U^* = 4$; black diamond - new vibrissa at $U^* = 5$; green inverted triangle - new vibrissa at $U^* = 6$; teal triangle - old vibrissa at $U^* = 3$; dark yellow circle - old vibrissa at $U^* = 4$; violet left-oriented-triangle - old vibrissa at $U^* = 5$; grey right-oriented-triangle - old vibrissa at $U^* = 12$.

Figure 8.17: Amplitudes of oscillation (left) and response frequency ratios (right) of the trailing new vibrissa geometry at $U^* = 3, 4$, and 5 , $Re = 500$, $m^* = 1.04$ for inline spacing (S) of $3D_1$ and $4D_1$ in the wake of a wake-generator circular cylinder with $CDR = 1.5$. Inline spacing measured between trailing edge of wake generator to leading edge of trailing structure. Legend : $S = 3D_1$ - inline spacing of $3D_1$; $S = 4D_1$ - inline spacing of $4D_1$; 1st- $S = 3D_1$ - primary response frequency for inline spacing of $3D_1$; 1st- $S = 4D_1$ - primary response frequency for inline spacing of $4D_1$; 2nd- $S = 3D_1$ - secondary response frequency for inline spacing of $3D_1$; 3rd- $S = 3D_1$ - tertiary response frequency for inline spacing of $3D_1$.

Figure 8.18: Isometric view of the vortex cores in the wake of the wake-generator circular cylinder and trailing freely vibrating in crossflow new vibrissa geometry at inline spacing of $3D_1$ (left) and $4D_1$ (right), $Re = 500$ and $U^* = 5$.

Chapter 9

Sensor-Karman(St) Interaction at Relative Velocity

This chapter focuses on the interaction between von *Kármán* Street in the wake of a wake-generator circular cylinder and the trailing sensor structure. Due to limitation in time and resources, only circular cylinder, elliptical cylinder and old vibrissa geometries are simulated as the trailing sensor structure. Such interaction shall represent studies on the vibrations on the vibrissae at the muzzle of harbour seals during their pursuit of preys. A wake-generator circular cylinder is used to represent the regular pattern and dynamic nature of the wake generated by the preys of the harbour seals. As the harbour seals pursued their preys, relative velocities between the harbour seals (specifically their vibrissae) and their preys are existent with positive relative velocity indicating instances of approaching and negative relative velocity indicating instances of distancing between the harbour seals and their preys. Hence, in this study, the trailing sensor structures are allowed to move relative to the wake-generator circular cylinder. Other simulation parameters are maintained at consistent settings to previous chapter (chapter 8) on interaction between von *Kármán* street and trailing sensor structures at zero relative velocity. Hence, for simulations with equal sized wake-generator and trailing sensor structure, the Reynolds Number based on the characteristic diameter of the wake generator is 500 with a freely vibrating in crossflow setup applied on the trailing sensor structure. The wake-generator circular cylinder is defined to be stationary with incoming freestream flow. The mass ratio of the trailing sensor structure is maintained at 1.04 and the natural frequency of the trailing sensor structure is designed to correspond to their stationary vortex shedding frequency. The setup of old

vibrissa geometry is conducted as an exception with the natural frequency of the system setup to correspond to $U^* = 12$ (not its true resonance frequency) in order to simultaneously investigate the effect of the natural frequency of the system and the geometry. This is done because of the small nature of self-induced vibration of the old vibrissa geometry with small absolute amplitude variability despite their variation in natural frequency of the system (refer to chapter 6). In order to investigate a greater extent of the effects of the size of wake-generator circular cylinder on the response of trailing sensor structures at non-zero relative velocity, simulations on trailing circular cylinder at CDR = 3.0 is performed. It should be noted that the initial inline spacing for approaching simulations and final inline spacing of distancing simulations (measured from the trailing edge of wake generator to the leading edge of trailing sensor structure) of all the simulations are maintained at $23.0D_1$, where D_1 is the characteristic diameter of the wake-generator circular cylinder. The inline spacing on stopping position for approaching simulations and initial position for distancing simulations is defined as $0.5D_1$. Constant streamwise motion acceleration of the trailing sensor structure is defined to be $0.4 D/(\text{solutiontime})^2$. Various streamwise velocities are simulated for equal-sized and three times-sized wake generator to trailing sensor ratio.

9.1 Equivalently-sized wake-generator and trailing sensor structure

This section describes the simulations and their corresponding results on trailing sensor structures in the wake of an equivalently-sized circular cylinder (as wake generator) at 1D/s and 2D/s relative velocity, where s is solution time (not 'seconds' of physical world). Simulations on approaching and distancing trailing sensor structure relative to the wake-generator circular cylinder are performed with various initial sideways displacement ($G/D_1 = 0.0$ to 1.5 at an interval of 0.25). This will allow for assessment on the effects of velocity of approach (or distance) as well as effects of sideways positioning on the response of the trailing sensor structures. Further assessment on the effects of velocity on the response of the trailing sensor will be performed in the next section where the significantly larger initial inline spacing for approaching simulations (or final for distancing simulations) is designated due to the larger wake-generator circular cylinder (CDR = 3.0), $S = 23D_1 = 69D_2$. For this section, circular cylinder, elliptical cylinder and old vibrissa geometry will be simulated as the trailing sensor

structure. For all the simulations conducted in this chapter, the simulation is run for $t^* = 180 * CDR$ before the streamwise motion of the trailing sensor structure is initiated. This is to ensure the full development of the von *Kármán* street before the motion of the trailing sensor structures. Hence, such condition would similarly mimic the event of the real sensing by the harbour seal vibrissae, which are exposed to regular occurrence of vortices of their preys (such as fishes, jellyfishes and etc). Due to the existence of relative velocity between the wake-generator circular cylinder and trailing sensor structure, it is also necessary to include the Doppler effect on the response frequency of the trailing sensor structure. However, evaluating the traveling speed of the vortex tubes (in the von *Kármán* street) requires delicate methodology. Intuitive method of comparing the motion of the trailing sensor structure to the motion of the vortex tubes is found to fail due to the variation of the arrangement of the vortex tubes (including vortex tubes further downstream) as the trailing sensor structure approaches them. Figure 9.1 illustrates the comparison of the vortex cores in the wake of the tandem circular cylinders before and after the motion of the trailing circular cylinder towards the vortex tubes at 1D/s relative to the stationary wake-generator circular cylinder (freestream flow velocity is 1D/s). From figure 9.1, it can be observed that only vortex tubes are observed in the wake of the tandem circular cylinder, regardless of the motion of the trailing circular cylinder (no vortical braids). Distinction in the arrangement of the vortex tubes in the von *Kármán* street are found with smaller inline spacing and shorter distance to convergence of vortex tubes observed as the trailing circular cylinder approaches the vortex tubes. Measurements on the vortex shedding and spacing between vortex tubes on stationary or freely vibrating in crossflow circular cylinder would fail due to the physical difference of the nature of vortex shedding. Additionally, the absence of vortical braids in the wake of the current tandem circular cylinders (refer to figure 9.1) serves to emphasise the different nature of the vortex shedding (as compared to stationary or 1D vibrating circular cylinder). Hence, assessment on the traveling speed of the vortex tubes in the von *Kármán* street is determined from the ratio between the centre-to-centre inline distance between two vortex tubes of similar polarity in the middle of the streamwise simulation domain (where the vortical arrangements are highly regular) and the propagation time for the vortex tube to reach the position of the subsequent downstream vortex tube of similar polarity. Such ratio is found to be 0.6D/s (u_w defined as velocity of the wake) for distancing circular cylinder, hence, the Doppler effect can be defined as follows:

Figure 9.1: Top view of vortex cores in the wake of tandem circular cylinders with stationary upstream circular cylinder and freely vibrating in crossflow trailing circular cylinder at an inline spacing of $0.5D$, freestream flow velocity (U) = $1D/s$ and $Re = 500$. Top : before motion of trailing circular cylinder ($t^* = 180$) ; bottom : after streamwise motion of trailing circular cylinder away from upstream circular cylinder at $u = 1D/s$ (relative to the upstream stationary circular cylinder) at $t^* = 187.4$.

$$f_{dop} = \left(1 + \frac{u}{u_w}\right)f_0 \quad (9.1)$$

where f_{dop} is the detected frequency by the trailing sensor structure indicated through its primary response frequency (often, its only response frequency), f_0 is the vortex shedding frequency of the wake-generator circular cylinder, u is the relative velocity of the trailing sensor structure to the stationary wake-generator circular cylinder and u_w is the travelling velocity of the vortex tubes relative to the stationary wake-generator circular cylinder. For approaching circular cylinder simulations, the velocity of the wake is found to be approximately $0.8D/s$ [87]. Such study on approaching and distancing trailing structure relative to a wake-generator circular cylinder is a novel approach and no published studies are found to the date of submission of this thesis. Hence, careful analyses on the relative positioning of the trailing sensor structure to the wake (specifically vortex tubes due to their

pressure forcing effect) of the wake-generator circular cylinder is necessary. Figure 9.2 shows the RMS amplitudes of oscillation of the approaching and distancing sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) at $CDR = 1.0$ for various sideways displacement and $u = 1D/s$ and $2D/s$. From figure 9.2, distinct response amplitudes are observed for various sideways displacement of the approaching sensor structure. However, the amplitudes of response of the old vibrissa geometry is found to be significantly more distinctive as compared to the response of the circular cylinder and elliptical cylinder. Strong indication of the differentiability of the response of the old vibrissa geometry to various sideways displacement relative to the wake-generator circular cylinder are observed. Largest amplitude of response of the old vibrissa geometry is observed at $G/D = 1.0$ which represents the approximate borderline of the wake affected zone. At approaching speed of $1D/s$, less patterned response are observed on the response of the trailing circular cylinder and elliptical cylinder to sideways displacement. Interestingly, the amplitude of response of the circular cylinder is found to be larger than the amplitudes of oscillation of the old vibrissa geometry despite the significantly lower natural frequency of the system (lower spring stiffness). With reference to figure 9.2, the amplitudes of response of the elliptical cylinder is observed to be significantly smaller as compared to the corresponding responses of the circular cylinder and old vibrissa geometry. This is found to be mainly due to the effects of the higher natural frequency of the system (as compared to the old vibrissa geometry) and higher added mass coefficient of the elliptical cylinder (as compared to the circular cylinder). However, regardless of the approaching velocity, the amplitudes of oscillation of the old vibrissa geometry at $U^* = 12$ is found to clearly distinguish the respective sideways positioning of the old vibrissa geometry. Interestingly, at perfectly inline position between the wake-generator circular cylinder and trailing sensor structure, the amplitudes of response of the approaching trailing sensor structure are found to be smallest (or one of the smallest) amplitude of response of the structure across various sideways displacement and for $u = 1D/s$ and $2D/s$. With reference to figure 9.2, for distancing trailing sensor structure, no pattern regularity is observed in the amplitudes of response of the circular cylinder, elliptical cylinder and old vibrissa geometry. However, unlike the approaching trailing sensor structure results, the amplitudes of response of the distancing trailing elliptical cylinder and old vibrissa geometry are found to be one of the larger amplitudes for various sideways displacements. In essence, a trailing sensor system with lower natural frequency of the system is found to respond with larger amplitude of os-

cillation as well as greater sensitivity (or regularity) of response to sideways displacement. Despite its increased directional sensitivity, the use of elliptical-based geometry as sensor structure increases the added mass coefficient in the crossflow direction which resulted in lower amplitude of response of the structure. However, the response of the elliptical cylinder is observed to peak when it is positioned at sideways displacement of 0.5. The amplitudes of response of the elliptical cylinder and old vibrissa geometry at distancing velocity of $2D/s$ setup are found to be an inverted profile of the corresponding response at approaching velocity of $2D/s$ setup, especially in the wake-affected zone ($-1.0 \leq G/D \leq 1.0$). Interestingly, it can be observed that the amplitudes of response of distancing sensor structure are almost always higher than the corresponding amplitude of approaching sensor structure. This is mainly due to the lengthen duration of exposure of the distancing sensor structure to the vortex cores (tubes), as their relative velocities are much lower (as compared to approaching sensor structure). Such exposure led to increase in the net pressure force on the sensor structure towards the vortex tubes, which led to the larger amplitude of response. In contrast, approaching sensor structure is much more briefly exposed to the pressure forcing by the vortex tubes, which could also be observed from the doppler effect on the response frequency of the approaching sensor structure (nearly doubled frequency indicating doubling of the velocity amplitude of oscillation).

While Dykes indicated the ability of the seal in sensing vibratory response, he also noted the ability of the seal vibrissae in sensing static displacement [29]. As such, observations on the variation of the mean displacement (static displacement) of the trailing sensor structure with variation in sideways displacement is necessary. Figure 9.3 shows the mean displacements of oscillation of the approaching and distancing sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) at $CDR = 1.0$ for various sideways displacement and $u = 1D/s$ and $2D/s$. Strong linear line with a gradient of 0.3 is observed in the variation of the mean displacement of the old vibrissa geometry with sideways displacement for $-1.0 \leq G/D \leq 1.0$ at approaching velocity of $1D/s$. Although an increasing mean displacement trend is still observed with larger sideways displacement for $-1.0 \leq G/D \leq 1.0$, the linear line trend is not longer observed on the profile of mean displacement response of the old vibrissa geometry with sideways displacement at approaching velocity of $2D/s$. Interestingly, the maximum mean displacement of response of the old vibrissa geometry is found to occur at the similar amplitude and sideways displacement to the corresponding amplitude of response for approaching velocity of $1D/s$ old vibrissa geometry. Such trend is not

Figure 9.2: RMS amplitude ratios of the approaching and distancing sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) relative to a wake-generator circular cylinder at $CDR = 1.0$ for various sideways displacement and $Re = 500$ (based on wake-generator circular cylinder). Top left : approaching sensor at $1D/s$; top right : approaching sensor at $2D/s$; bottom left : distancing sensor at $1D/s$; bottom right : distancing sensor at $2D/s$.

observed on the response of the trailing circular cylinder and elliptical cylinder, where 50% (circular cylinder) and 100% (elliptical cylinder) increase in mean displacements are observed due to doubling of the approach velocity. However, their response shows a peak of mean displacement at $-G/D = 0.75$, where no head-on collision between the sensor structure and vortex tubes are observed. No significant profile of mean displacement variation with sideways displacement are observed for distancing sensor structures. However, it is noted that the amplitude of oscillation of the old vibrissa geometry peaked at $G/D = 1.2$, which corresponded to borderline region of the wake-affected zone. Interestingly, a consistent mean displacement towards the centreline of the wake are observed on all the trailing sensor structure at all approaching velocities and sideways displacement. This indicated the generally dominant inward forcing (into the wake-affected zone) on a trailing structure into the wake-affected zone (specifically, the centreline of the wake). However, such trend is not observed in

Figure 9.3: Mean displacement ratios of the approaching and distancing sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) relative to a wake-generator circular cylinder at $CDR = 1.0$ for various sideways displacement and $Re = 500$ (based on wake-generator circular cylinder). Top left : approaching sensor at $1D/s$; top right : approaching sensor at $2D/s$; bottom left : distancing sensor at $1D/s$; bottom right : distancing sensor at $2D/s$.

distancing sensor structure cases where more random variation of mean displacement with sideways displacement are observed. Unlike the profile of variation of RMS amplitudes with sideways displacement, no consistent occurrence of larger mean displacement are observed for distancing sensor structure as compared to approaching sensor structure.

While the amplitudes of response and mean displacement of the trailing sensor structure are found to be strong dependent on the natural frequency of the system in identification of its sideways displacement, different variation of frequency response is observed on the sensor structures. While no dominant response frequency is observed on the distancing sensor structures (circular cylinder, elliptical cylinder and old vibrissa geometry), some dominant response frequency could be observed for approaching cases. It should be noted that the absence of detectable dominant response frequency in distancing sensor structure is due to the large period of oscillation of the sensor structure as a

consequence of the small relative velocity between the vortex tubes and sensor structure). Additionally, a 'runway' of $23D$ is found to produce insufficient number of oscillation for distancing cases due to their large period of oscillation. Hence, this finding is found to reinforce the frequency-based sensing principle with harbour seal vibrissae. Figure 9.4 illustrates the dominant response frequency (detected frequency, not frequency of encounter) of the approaching sensor structure for relative velocity (u) of $1D/s$ and $2D/s$ to the wake-generator circular cylinder. With reference to figure 9.4, for approaching velocity of $2D/s$, only circular cylinder as sensor structure is observed to detect a dominant response frequency. However, regardless of the type of sensor structure and speed of approach, no dominant response frequency is observed at $G/D = 0.75$, which indicated the lack of a regular oscillation (and perturbation). For circular cylinder as sensor structure with approaching velocity of $1D/s$, consistent dominant response frequency of ≈ 0.2 are observed for $|G/D| = 0, 0.25, 0.5$ and 1.5 . Despite, the significant difference in sideways displacement, similar source of response frequency are found on the oscillatory response of the trailing circular cylinder. This can be observed in figures 9.5, 9.6, 9.7 which depicts the vortex cores in the wake of the wake-generator circular cylinder and WIV on the trailing circular cylinder for $G/D = 0, 1.5$ and 1.0 , respectively. From figures 9.5, 9.6, it can be observed that the motion response of the approaching circular cylinder involved head-on collision with the clockwise-rotating and counter-clockwise-rotating vortex tubes. In comparison to figure 9.7, head-on collision is found to occur only between the approaching circular cylinder and clockwise-rotating vortex tubes. No collision is observed between the approaching circular cylinder at initial sideways displacement of $G/D = 1.0$ and counter-clockwise-rotating vortex tubes. This is found to be peculiar due to the larger initial sideways displacement of the approaching circular cylinder with $G/D = 1.5$ as compared to the corresponding cylinder at $G/D = 1.0$. Observation on the amplitude of oscillation and mean displacement indicated similarity of response between the approaching circular cylinder at $G/D = 1.5$ and $G/D = 1.0$. Hence, coincidence between the crossflow position of the approaching circular cylinder and the counter-clockwise-rotating vortex tubes could be excluded from reasoning.

From analysis between the lift forcing and the motion of the structure, it is found that the phase angle between the motion and the forcing underlies such differing response. Figure 9.8 illustrates the time variation of the scaled lift forcing on the approaching circular cylinder and its corresponding motion response for sideways displacement of $0.0D, 1.0D$ and $1.5D$. It is important to note that the

Figure 9.4: Dominant response frequency (detected frequency, not frequency of encounter) of the approaching sensor structure (circular cylinder at $U^* = 5.5$, elliptical cylinder at $U^* = 5.0$, and old vibrissa geometry at $U^* = 12$) relative to a wake-generator circular cylinder at $CDR = 1.0$ for various sideways displacement and $Re = 500$ (based on wake-generator circular cylinder).

significance of figure 9.8 does not lie on the absolute value of the scale in the y-axis, hence, emphasis should not be made on the absolute value of the lift forcing relative to the crossflow motion response of the structure. From figure 9.8, it is found that consistently out of phase motion response and lift forcing occurred on the approaching circular cylinders with primary response frequency. An irregular phase angle between the motion response and lift forcing is observed on the approaching circular cylinder with no response frequency ($|G/D| = 1.0$), which conclusively explained the reason of absence of a dominant response frequency. No dominant response frequencies are observed on the response of the approaching circular cylinder at a velocity of 1D/s and $|G/D| = 0.75$ to 1.25 due to the aforementioned findings on the variation of the phase angle between the lift forcing and the motion response of the approaching circular cylinder. For approaching velocity of 2D/s, it is observed that similar dominant response frequency pattern are observed to the corresponding response of approaching circular cylinder at a velocity of 1D/s. Interestingly, slight modification on the frequency of response of the approaching circular cylinder a velocity of 2D/s is observed as compared to the corresponding frequency of the approaching circular cylinder at a velocity of 1D/s. This is due to the modification on the vortical arrangements (specifically, vortex tubes) in the wake of the wake-

Figure 9.5: Vortex cores in the wake of the wake-generator circular cylinder at $CDR = 1.0$ and Wake-Induced Vibration (WIV) on trailing circular cylinder with $G/D = 0.0$ and approaching velocity of $1D/s$. Top left : $t^* = 186.6$ - before encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; top right : $t^* = 187.4$ - encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; bottom left : $t^* = 188.8$ - encounter of approaching circular with counter-clockwise-rotating vortex tubes ; bottom right : $t^* = 189.4$ - aftermath of the encounter of approaching circular cylinder with counter-clockwise rotating vortex tubes.

generator circular cylinder as the trailing circular cylinder approaches. Earlier in figure 9.1, such modification has been observed on the *Kármán* street of the wake-generator circular cylinder due to the downstream (distancing) motion of the trailing circular cylinder. However, a strong upstream effect of an approaching circular cylinder in a *Kármán* street has yet to be studied. As such, the modification of this *Kármán* street due to pressure-bow effect is not foreseen. Figure 9.9 shows the

Figure 9.6: Vortex cores in the wake of the wake-generator circular cylinder at $CDR = 1.0$ and Wake-Induced Vibration (WIV) on trailing circular cylinder with $G/D = 1.5$ and approaching velocity of $1D/s$. Top left : $t^* = 186$ - before encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; top right : $t^* = 187$ - encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; bottom left : $t^* = 188.4$ - encounter of approaching circular with counter-clockwise-rotating vortex tubes ; bottom right : $t^* = 189.6$ - aftermath of the encounter of approaching circular cylinder with counter-clockwise rotating vortex tubes.

pressure profiles in the wake of the wake-generator circular cylinder and around the approaching circular cylinder at a velocity of $2D/s$ with initial sideways displacement of $0.5D$. It is important to note that a planar-pressure profile introduced midway to emphasise pressure colouring. With reference to figure 9.9, a large upstream positive pressure field can be observed in the pressure profile around the approaching circular cylinder. Such influence on the pressure field due to the relative motion of the

Figure 9.7: Vortex cores in the wake of the wake-generator circular cylinder at $CDR = 1.0$ and Wake-Induced Vibration (WIV) on trailing circular cylinder with $G/D = 1.0$ and approaching velocity of $1D/s$. Top left : $t^* = 186.2$ - before encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; top right : $t^* = 187$ - encounter of approaching circular cylinder with clockwise-rotating vortex tubes ; bottom left : $t^* = 188.4$ - close-encounter of approaching circular with counter-clockwise-rotating vortex tubes ; bottom right : $t^* = 189.2$ - aftermath of the close-encounter of approaching circular cylinder with counter-clockwise rotating vortex tubes.

trailing circular cylinder is found to result in a positive upstream pressure field of approximate radius of $3D$. Such positive pressure field is found to significantly modify (and/or eliminate) the vortex tubes in the wake of the wake-generator circular cylinder. As the flow is observed to move from the high pressure in the leading edge of the approaching circular cylinder to the low pressure region of the vortex tubes (in contrast, low pressure region around the trailing cylinder and the vortex tubes is

found to repel each other in near proximity). Hence, this result in the variation of the detected pressure forcing by the *Kármán* street on the approaching circular cylinder at 1D/s and 2D/s. In contrast, both the approaching elliptical cylinder and old vibrissa geometry at a velocity of 2D/s are found to respond with no dominant response frequency of motion. However, at approaching velocity of 1D/s, consistent emergence of dominant response frequency are observed (similar to circular cylinder) for old vibrissa geometry while such scarce emergence of dominant response frequency are observed for elliptical cylinder. This scarcity is found to be due to the large added mass coefficient of the elliptical cylinder which resulted in smaller amplitude of oscillation and slightly less regular oscillation. This is due to the variation of the phase angle between the motion of the elliptical cylinder and the lift forcing from the vortex tubes during the approaching period of the elliptical cylinder. However, for $|G/D| = 1.0$ and 1.25, consistent phase angle between the motion response of the elliptical cylinder and the lift forcing of the vortex tubes on the elliptical cylinder is observed which is found to result in a dominant response frequency of the elliptical cylinder.

Despite the almost similarly large added mass coefficient of the old vibrissa to the elliptical cylinder, consistent emergence of dominant response frequency are observed due to its significantly lowered natural frequency of the system. This allowed for greater sensitivity of the old vibrissa geometry to perturbation by weaker vortex tubes. However, disadvantages on lowering the natural frequency of the system are found when deploying the sensor-system into flow with background noise. Due to its lower natural frequency, such sensor-system would respond with many frequencies, which will result in difficulty of identifying the target frequency. Hence, similar to one of the conclusion in chapter 6, delicate balance is necessary between lowering the natural frequency of the system for sensitivity as well as its corresponding tendency on responding to background flows. For sideways displacements where a dominant response frequency is observed on the approaching elliptical cylinder and old vibrissa at a velocity of 1D/s, higher dominant response frequencies are observed as compared to the corresponding response frequency of a trailing circular cylinder. The response frequencies of the trailing elliptical cylinder and old vibrissa geometry (≈ 0.17) at an approaching velocity of 1D/s are observed to better represent the vortex shedding frequency of the wake-generator circular cylinder (≈ 0.19) as compared to the corresponding response of approaching circular cylinder at the same velocity. This is due to the lower stagnation pressure at the leading edge of the old vibrissa geometry (and elliptical cylinder). This is found to result in smaller modification of the *Kármán* street (target

Figure 9.8: Scaled (to its respective one-sided maximum value) lift forcing and motion response of the approaching circular cylinder at a velocity of $1D/s$ for various sideways displacement (G/D). Top left : $G/D = 0.0$; top right: $G/D = 1.0$; bottom : $G/D = 1.5$.

wake) due to their interactions with the pressure field around the body, as shown in figure 9.10.

Hence, in spite of its lower crossflow added mass coefficient, accurate detection on the target frequency is found to be elusive due to the high stagnation pressure (bluffer configuration) of a circular cylinder as a sensor. However, a trailing streamlined body encountered large crossflow added mass coefficient, which resulted in lower sensitivity to perturbation. In order to rectify such added mass coefficient problem, a lower natural frequency of the system could be applied on the setup of the sensor-system. However, careful assessment is necessary in order to achieve balance between greater sensitivity to perturbations and acquisition of background flows. In reality, the seals are known to be able to adjust the boundary conditions (and therefore, the natural frequency of the vibrissa-sensor system) at the base of their vibrissae. This will allow conditional adjustment of the boundary conditions by harbour seals in relevance to their proximity to the wake target. As the harbour seal vibrissae benefited from sensitivity to weak perturbations (ability to identify small perturbations for the nat-

Figure 9.9: Pressure profiles in the wake of the wake-generator circular cylinder and WIV on approaching circular cylinder at a velocity of $2D/s$ and initial sideways displacement of $0.5D$. A plane slice is located in the midspan of the simulation domain in order to strengthen the coloration of the pressure profiles.

Figure 9.10: Pressure profiles in the wake of the wake-generator circular cylinder and WIV on approaching old vibrissa geometry at a velocity of $1D/s$ and initial sideways displacement of $0.5D$. Left : pressure profile in saddle plane of major axis ; right : pressure profile in nodal plane of major axis

ural frequency of the system), adjustments of the boundary conditions at the base of the vibrissae would greatly compliment the underwater sensing ability of the harbour seals (hardening of muscles at the base of vibrissae when multiple frequencies are detected and vice versa when no frequency is detected).

9.2 Effects of CDR and relative velocity on the response of trailing structure

While previous section focuses on the response of three different approaching and distancing sensor structure to the wake of an equivalently-sized circular cylinder at various sideways displacement, less extensive studies on the effects of CDR and relative velocity are performed. In this section, extensive study is performed on the effects of triple-sized wake-generator circular cylinder as well as the effects of relative approaching and distancing velocity of the sensor structure to the wake-generator. Three sideways displacements of $0.0D_1$, $0.5D_1$ and $1.0D_1$ are investigated for each relative velocity and $CDR = 3.0$ with certain exceptions due to limited resources. Additionally, the current study is only performed on trailing circular cylinder as sensor structure as the main objective is to assess the effects of relative velocity and CDR on the trailing response of the structure. Figure 9.11 illustrates the RMS amplitude, mean displacement and dominant response frequency of trailing circular cylinder for various sideways displacements and relative velocities in the *Kármán* street of a wake-generator circular cylinder at $CDR = 3.0$. On figure 9.11, it is important to note that not all sideways displacements are simulated uniformly for each relative velocities. Hence, reference to figure 9.11a (on RMS amplitude ratio) would provide useful information on simulated structures only as no zero RMS amplitude are observed on the response of the trailing circular cylinder. With reference to figure 9.11, linearly decreasing trend is observed on the variation of the RMS amplitude with greater distancing velocity for $-6 \leq u \leq -2D_2/s$ with no initial sideways displacement. However, no such trend is observed on the amplitude response of the trailing circular cylinder for approaching cases (although linear trends are observed for $2 \leq u \leq 4D_2/s$ for no initial sideways spacing and $3 \leq u \leq 5D_2/s$ for trailing cylinder with initial sideways displacement of $0.5D_1$). However, decreasing amplitude of response is observed with increasing relative velocity of the trailing circular cylinder to the wake generator for $G/D_1 = 0$ and 0.5 . This is observed to be due to the shorter duration of exposure of the trailing circular cylinder to vortex tubes as the relative velocity is increased. At $G/D = 1.0$, the circular cylinder is found to be in the border of wake-affected zone. Hence, a less direct interaction between the vortex tubes (and braids) in the wake and the trailing circular cylinder is observed, which is found to result in a less characteristic response (as compared to responses at other G/D). Larger amplitude of oscillation is observed on distancing circular cylinder as compared to its corresponding

response of an equivalent speed approaching circular cylinder due to the longer duration of exposure of the trailing circular cylinder to the vortex tubes in the *Kármán* street (smaller relative velocity between wake and trailing circular cylinder). Interestingly, from figure 9.11, increasing mean displacement of the trailing circular cylinder is observed for increase in the relative velocity. This is found to be true regardless of the sideways displacement (no data for $u = 2$ and $4D_2/s$ at $G/D_1 = 1.0$). No such trend is observed on the mean displacement of distancing circular cylinder. Interestingly, consistent observation of dominant response frequency is found on the response of the approaching circular cylinder at $0 \leq u \leq 2D_2/s$ and $G/D_1 = 0.0$ and 0.5 , which is observed to be equivalent to the vortex shedding frequency of the wake-generator circular cylinder (when properly scaled, these frequencies correspond to ≈ 0.21 [Strouhal number of wake-generator circular cylinder]). At higher approaching velocity, due to greater upstream effect of the pressure profile on the leading edge of the trailing circular cylinder, modification of the vortical arrangements in the wake of the wake-generator circular cylinder is found to occur. This resulted in higher response frequency of the trailing circular cylinder as compared to the vortex shedding frequency of the wake-generator circular cylinder. For distancing circular cylinder, no dominant response frequency is observed for $-6 \leq u \leq -3D_2/s$. At distancing velocity of $2D_2/s$, the trailing circular cylinder is observed to respond with a relatively similar frequency to the frequency of vortex shedding. This is found to occur due to the similarity in the relative velocity of the trailing circular cylinder and the vortex tubes to the velocity of the wake propagation. At distancing velocity of $1D/s$, the response of the trailing circular cylinder is found to increase profoundly due to the small relative velocity between the vortex tubes and the distancing circular cylinder.

The current findings suggested the limit of pursuit velocity based on wake tracking with the harbour seal vibrissae-like sensor. Further confirmation on frequency-based sensing (instead of amplitude-based sensing only) of the harbour seal vibrissae is found, which is observed to strongly rely on the relative velocity between the wake and the vibrissae (harbour seals). This is due to the strong upstream pressure influence effect of the vibrissae and harbour seals, which are observed to deform (or eliminate) the vortices of the target wake. Such deformation led to modification of the detected frequency, which could lead to false understanding on the characteristics (size and shape) of the target geometry. However, with the right natural frequency setup and vibrissa geometry, highly sensitive underwater sensor (with adjustable natural frequency) will represent strongly robust passive

9.2. EFFECTS OF CDR AND RELATIVE VELOCITY ON THE RESPONSE OF TRAILING STRUCTURE

sensing principle. However, caution should be taken on the relative velocity of the vibrissae (and its carrier) to the wake. This is due to the modification (and/or elimination) of vortices by the high pressure stagnation point on the vibrissae (or its carrier), which may lead to disappearance of the hydrodynamic trail of the target.

Figure 9.11: RMS amplitude ratio relative to D_2 (top), mean Displacement relative to D_2 (middle) and dominant response frequencies (bottom) of the trailing circular cylinder at various sideways displacements and relative velocities (to stationary wake-generator circular cylinder) in the *Kármán* street of a wake-generator circular cylinder at $CDR = 3.0$ and $Re = 1,500$ (based on characteristic diameter of wake-generator circular cylinder, D_1). Relative velocity is based on D_2/s while sideways displacement is based on D_1 .

Chapter 10

Conclusion & Future Work

10.0.1 Conclusion

The current study focuses on understanding the underwater sensing principle of harbour seal vibrissae with the ultimate objective of possibility of field implementation and mimicking of such principle in real world. In order to assess various aspects of the underwater sensing principle, various methodology and steps are applied for greater understanding of this underwater sensing principle. The material properties of the harbour seal vibrissae are investigated given their novelty. Variation of their modulus of elasticity along the spanwise of the vibrissae is found with significant variation of bending stiffness observed (as compared to the bending stiffness of rat vibrissae). Subsequent step for understanding this sensing methodology is to conclusively prove the reduction of Vortex-Induced Vibration (VIV) of the harbour seal vibrissae. Due to the importance of such VIV reduction properties, the assessment is performed experimentally and through numerical simulations. The numerical study involve simulations on stationary, freely vibrating in crossflow (1D vibrating) and freely vibrating in stream-wise and crossflow (2D vibrating) harbour seal vibrissae geometry (new vibrissa geometry). These results are compared with the corresponding response of similarly set-up experiments and simulations on circular cylinder and elliptical cylinder. From these results, it is proven beyond reasonable doubt that the geometrical features of the harbour seal vibrissae are found to reduce Vortex-Induced Forcing (VIF) and VIV. Further studies on the wake topology of the new vibrissa geometry revealed regularly occurring vortical structures with spanwise and crossflow alternating shedding of vortices, which is found to significantly reduce the amplitude of VIV of the new vibrissa geometry as well as the drag coefficient of the new vibrissa geometry. Evaluation on the effects of the geometrical

features of the harbour seal vibrissa geometry is also performed with the objective of finding the geometrical feature with greatest effect on VIV reduction. The existence of major axis undulations is found to be as critical as the existence of minor axis undulations. However, the axis ratio of elliptical-based geometries is found to be critical in determining the response of the structure for 2D vibrating setup. The VIV reduction is found to be mainly affected by the existence of no-axial undulations, single-axial undulations or dual-axial undulations with decreasing amplitudes as the number of axial undulations increases. Upon conclusion of the VIV reduction properties of the harbour seal vibrissae and the corresponding significance of its geometrical feature, further studies are performed to understand the relative sensitivity of the harbour seal vibrissae geometry to periodic-yet-unstable wake pattern (similar to the wakes of the harbour seal diet, e.g. fish). Experiments of trailing circular cylinder, elliptical cylinder, harbour seal vibrissa geometry in the wake of circular cylinder (10 times larger) revealed the higher sensitivity (captured frequency) of the harbour seal vibrissa geometry (as compared to elliptical cylinder and specifically, as compared to circular cylinder). Simulations on trailing circular cylinder, elliptical cylinder and harbour seal vibrissa in the Karman-street of the wake-generator circular cylinder at various relative sizes and constant inline spacing of $4D_1$ (where D_1 and D_2 is the characteristic diameter of the wake-generator circular cylinder and trailing sensor structure, respectively) revealed equivalent performance in the amplitude and frequency response of the trailing circular cylinder, elliptical cylinder and harbour seal vibrissa. However, at large size ratios ($CDR = D_1/D_2$, investigated up to 3.0), the harbour seal vibrissa geometry is found to respond with no secondary response frequency (unlike circular cylinder and elliptical cylinder). The natural frequency of the sensor structure is also found to be crucial in enhancing the sensitivity of the structure. Larger crossflow added mass coefficient are observed for elliptical cylinder harbour seal vibrissae geometry (as compared to circular cylinder), which is found to reduce their amplitude of oscillations for a given wake. Such mechanism is found to minimise the occurrence of secondary and tertiary response frequencies. The amplitudes of oscillation of the trailing elliptical cylinder and harbour seal vibrissa geometry could be increased through lowering of the natural frequency of the sensor setup. In approaching and distancing cases, the natural frequency of the system is found to play a significant role on the amplitude of response of the structure with wake-generator circular cylinder as the target structure. However, it is found that such amplitude-based sensing alone (of the harbour seal vibrissae sensing principle) is unreliable due to the proneness to variability with sideways displacement. In

contrast, frequency-based sensing of the harbour seal vibrissae sensing principle is found to be very robust, regardless of the sideways displacement. However, limiting approach velocity is found to exist due to the high pressure build-up in the leading edge of the trailing sensor structure, which is found to modify or eliminate the vortices of the wake of the target structure. Streamlined structure of the harbour seal vibrissa geometry is found to be significantly more beneficial due to its lower pressure build-up at the leading edge, which is found to result in significantly smaller modification of the wake of the target structure (hence, more accurate sensing). For distancing cases, the harbour seal vibrissa geometry is found to obtain no dominant response frequency peaks corresponding to the vortex shedding frequency of the target structure.

10.0.2 Future Work

Although this thesis attempted to understand the underwater sensing principle of harbour seals vibrissae as intensively and extensively as possible, a lot more grounds need to be covered for a complete understanding on the underwater sensing of harbour seals vibrissae. The author would like to suggest the few of the following possible studies for the future:

- A 3D PIV of the flow around the seal vibrissae would be beneficial in elucidating the details of the wake structure.
- Detailed empirical study on the holding material of the harbour seal vibrissae model would present the possibility of identifying the most optimal configuration for flow sensing. In essence, such study would allow determination of the 'real' boundary condition at the base of the vibrissae as the seals are sensing their preys.
- While the current study is conducted for $Re = 500$ (with reference speed of $U = 1$ m/s), it is noted in biological observations that the seal is capable of swimming up to a top speed of approximately 4.5 m/s (corresponding relevant $Re = 2250$) [102]. It would be important to understand the flow dynamics (Fluid-structure interaction) between the vibrissa and its surrounding fluid at such Reynolds number, in order to better understand the effect of faster swimming speed relative to the signal on the vibrissae due to wakes from an object.
- A study on with multiple vibrissae trailing a wake generator would certainly bring us a bigger step closer towards understanding the underwater sensing of the harbour seal vibrissae.

→ Last but not least, a study on the flow dynamics of vibrissa-vibrissa interactions in various alignments (staggered or inline in transverse direction and streamwise alignment). Such study would prove to be crucial in understanding the seals' ability to discern large and small object as well as objects in near proximity or relatively distant.

Finally, the author would like to thank the reader for spending time on this thesis. The author hoped that research on the underwater sensing of the harbour seals would intrigue many and, hence, it would provide everyone with an in-depth understanding on the marvel of nature. The author also hopes that such sensing methodology would be developed into a 'real-time' technology, which would be used to the benefit of mankind in the future.

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Appendix A

Dynamic Scaling of Experimental Setup

Assuming that temperature variation of the experimental setup is not a significant factor, the relevant variables for dimensional analysis are listed as follows:

Control Variables :

- ρ density of fluid (kg/m^3)
- μ dynamic viscosity (kg/ms)
- L_w length of vibrissae (m)
- R diameter of ellipse perpendicular to the free stream flow (m)
- U freestream flow velocity (m/s)
- ε surface roughness of vibrissae (m)
- E modulus of elasticity of vibrissae (Pa)
- ρ_w density of vibrissae (kg/m^3)

Resulting Variables :

- D drag force (N)
- M moment acting on the vibrissae about their spanwise-axis (Nm)

- f frequency of vibration of vibrissae in transverse direction (Hz)
- A amplitude of vibration of vibrissae in transverse direction (m)
- f^* reduced velocity of the vibrissae
- Re Reynolds number

From the list above, it can be observed that the total number of variables involved are 12 (represented as 'n'). The defined basic units are M, L, and T (the number of basic units are represented by variable 'j' = 3). These units represent mass, length and time, respectively, with corresponding units of kg, metre and second. It is also necessary to define basic variables, which consist of the basic units. Such variables are defined to be the density of fluid (ρ), free stream flow velocity (U), and characteristic diameter of the vibrissae (R). As such, the number of π equations required is determined to be 9 (calculated as 'n - j = 12 - 3').

The following are the analysis for each dimensionless constant π for the corresponding nine equations:

For equation π_1

$$\pi_1 = \rho^a U^b R^c \mu \quad (A.1)$$

$$\pi_1 = [ML^{-3}]^a [LT^{-1}]^b [L]^c [ML^{-1}T^{-1}]$$

As π_1 is dimensionless, from equation A.1, it can be deduced that:

$$a = -1; b = -1; c = -1$$

substituting these values to the original equations, we have:

$$\pi_1 = \frac{\mu}{\rho U R} = \frac{1}{Re} \quad (A.2)$$

For equation π_2

$$\pi_2 = \rho^a U^b R^c L_w \quad (A.3)$$

$$\pi_2 = [ML^{-3}]^a [LT^{-1}]^b [L]^c [L]$$

As π_2 is dimensionless, from equation A.3, it can be deduced that:

$$a = 0; b = 0; c = -1$$

substituting these values to the original equations, we have:

$$\pi_2 = \frac{L_w}{R} \quad (\text{A.4})$$

For equation π_3

$$\begin{aligned} \pi_3 &= \rho^a U^b R^c \varepsilon \\ \pi_3 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [L] \end{aligned} \quad (\text{A.5})$$

As π_3 is dimensionless, from equation A.5, it can be deduced that:

$$a = 0; b = 0; c = -1$$

substituting these values to the original equations, we have:

$$\pi_3 = \frac{\varepsilon}{R} \quad (\text{A.6})$$

For equation π_4

$$\begin{aligned} \pi_4 &= \rho^a U^b R^c E \\ \pi_4 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [ML^{-1}T^{-2}] \end{aligned} \quad (\text{A.7})$$

As π_4 is dimensionless, from equation A.7, it can be deduced that:

$$a = -1; b = -2; c = 0$$

substituting these values to the original equations, we have:

$$\pi_4 = \frac{E}{\rho U^2} \quad (\text{A.8})$$

For equation π_5

$$\begin{aligned} \pi_5 &= \rho^a U^b R^c D \\ \pi_5 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [MLT^{-2}] \end{aligned} \quad (\text{A.9})$$

As π_5 is dimensionless, from equation A.9, it can be deduced that:

$$a = -1; b = -2; c = -2$$

substituting these values to the original equations, we have:

$$\pi_5 = \frac{D}{\rho U^2 R^2} \quad (\text{A.10})$$

For equation π_6

$$\begin{aligned} \pi_6 &= \rho^a U^b R^c M \\ \pi_6 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [ML^2T^{-2}] \end{aligned} \quad (\text{A.11})$$

As π_6 is dimensionless, from equation A.11, it can be deduced that:

$$a = -1; b = -2; c = -3$$

substituting these values to the original equations, we have:

$$\pi_6 = \frac{D}{\rho U^2 R^3} \quad (\text{A.12})$$

For equation π_7

$$\begin{aligned} \pi_7 &= \rho^a U^b R^c f \\ \pi_7 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [T^{-1}] \end{aligned} \quad (\text{A.13})$$

As π_7 is dimensionless, from equation A.13, it can be deduced that:

$$a = 0; b = -1; c = 1$$

substituting these values to the original equations, we have:

$$\pi_7 = \frac{fR}{U} = f^* \quad (\text{A.14})$$

For equation π_8

$$\begin{aligned} \pi_8 &= \rho^a U^b R^c \rho_w \\ \pi_8 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [ML^{-3}] \end{aligned} \quad (\text{A.15})$$

As π_8 is dimensionless, from equation A.15, it can be deduced that:

$$a = -1; b = 0; c = 0$$

substituting these values to the original equations, we have:

$$\pi_8 = \frac{\rho_w}{\rho} \quad (\text{A.16})$$

For equation π_9

$$\begin{aligned} \pi_9 &= \rho^a U^b R^c A \\ \pi_9 &= [ML^{-3}]^a [LT^{-1}]^b [L]^c [L] \end{aligned} \quad (\text{A.17})$$

As π_9 is dimensionless, from equation A.17, it can be deduced that:

$$a = 0; b = 0; c = -1$$

substituting these values to the original equations, we have:

$$\pi_9 = \frac{A}{R} \quad (\text{A.18})$$

Appendix B

ILES And QUICK Scheme

In ILES (Implicit Large Eddy Simulation), it is assumed that the large and medium size eddies embody the major part of the flow as compared to the smaller eddies. It is also assumed that the interaction between neighboring length scales are more significant as compared to non-neighboring scale interactions. Lastly, but most importantly, the methodology has to satisfy global conservation of energy, mass and momentum. Through these assumptions, Boris et al found that ILES, at least, qualitatively generates correct results [10]. Further research has also shown the applicability and accuracy of ILES for macro scale flow simulations. The evidence of built-in subgrid scale model are evident through previous studies [10, 12, 11]. For our simulation code, the built-in subgrid scale model is based on the QUICK (Quadratic Upstream Interpolation for Convection Kinematics) integration scheme (Leonard, 1979). QUICK scheme is a convective modeling procedure, which combines the stability of the upstream differencing scheme and the accuracy of central differencing scheme. It also avoids stability problems associated with central differencing scheme and the insufficient accuracy of upstream differencing scheme [63]. QUICK scheme provides a third-order spatial accuracy and a second-order temporal accuracy with stable algorithm with the associated numerical diffusion acting as the subgrid turbulence model. For a known wall values $(\phi_{FL}, \phi_L, \phi_C, \phi_R, \phi_{FR})$ and variable grid spacing, QUICK interpolates the in-grid values (ϕ_l, ϕ_r) using the following method (with reference to B.1):

$$\phi_r = \frac{1}{2}(\phi_C + \phi_R) - \frac{1}{8}(\phi_L + \phi_R - 2\phi_C) \quad (\text{B.1})$$

Figure B.1: Schematic diagram of discretisation with QUICK scheme and its corresponding variable definitions with variable grid spacing.

$$\phi_L = \frac{1}{2}(\phi_L + \phi_C) - \frac{1}{8}(\phi_{FL} + \phi_C - 2\phi_L) \quad (\text{B.2})$$

$$\left(\frac{\delta\phi}{\delta x}\right)_r = \frac{\phi_R - \phi_C}{\Delta x_r} \quad (\text{B.3})$$

$$\left(\frac{\delta\phi}{\delta x}\right)_l = \frac{\phi_C - \phi_L}{\Delta x_l} \quad (\text{B.4})$$

It should be noted that QUICK employs a similar method for handling derivatives with a central differencing scheme. The underlying assumption for 1D QUICK scheme is that the advection in non-streamwise direction is not significantly larger than the diffusion in the corresponding direction (defined by Peclet number $= u\Delta x/\Delta t$). It has been shown that the QUICK scheme is unconditionally stable and computationally inexpensive in comparison to upwind differencing scheme and central differencing scheme [63]. However, for QUICK's convective algorithm to adequately model turbulence, great care should be taken in the design of the simulation grid. This is because the ratio grid-boundary of domain-volume represents the occurring numerical diffusion. An optimal grid size is therefore necessary to ensure the accuracy of the simulation as well as lower computational cost.

B.1 1D Model Problem on Application of QUICK Scheme

$$\frac{\delta\phi}{\delta x} = -\frac{\delta(u\phi)}{\delta x} + \frac{\delta}{\delta x}(\Gamma \frac{\delta\phi}{\delta x}) + S \quad (\text{B.5})$$

where

u convection of dependent variable in fluid

ϕ dependent variable

S source term for dependent variable

t time

x spatial coordinate

Γ diffusivity of dependent variable in fluid

If equation B.5 is discretise into N number of grid points in its axis, the model equation of the domain for time = n+1 at point C is shown as follows (with reference to figure B.1):

$$\begin{aligned} \frac{\phi_C^{n+1} - \phi_C^n}{\Delta t} &= \left[\frac{u_r^n \phi_r^n - u_l^n \phi_l^n}{\Delta x_C} \right] + \frac{1}{\delta x_C} (\Gamma_r (\frac{\delta\phi}{\delta X})_r - \Gamma_l (\frac{\delta\phi}{\delta X})_l) + S \\ \frac{\phi_C^{n+1} - \phi_C^n}{\Delta t} &= \left[\frac{u_r^n \phi_r^n - u_l^n \phi_l^n}{\Delta x_C} \right] + \frac{1}{\delta x_C} (\Gamma_r (\frac{\phi_R - \phi_C}{\Delta x_r}) - \Gamma_l (\frac{\phi_C - \phi_L}{\Delta_l})) + S \end{aligned}$$

$$\phi_C^{n+1} = \phi_C^n - \frac{\Delta t}{\Delta x_C} [u_r^n \phi_l^n - u_r^n \phi_l^n] + \frac{\Delta t}{\delta x_C} (\Gamma_r (\frac{\phi_R - \phi_C}{\Delta x_r}) - \Gamma_l (\frac{\phi_C - \phi_L}{\Delta_l})) + \Delta t S \quad (\text{B.6})$$

The author understands that Direct Numerical Simulation (DNS) would have been deemed appropriate for the single structure simulations presented in this thesis. However, the author believes that DNS would be computationally too expensive for simulations which include tandem structures at large initial separations (hence, large simulation domain). As the author listed the objectives of the project at the early stages, it is evident to the author that a large-scale simulation would be necessary if any understanding on the underwater sensing ability of the harbour seals to be attained. As such, the author consciously opt for ILES which represents the better balance of accuracy and computational resources for our simulations.

Appendix C

Analysis of the coherent structures in the wakes of vibrissa-like geometries

C.1 Analysis of wake topology of geometry 4

Figure C.1: Depiction of coherent structures in the wake of geometry 4 (vibrissa-like geometry with no major axis undulations with axis ratio 0.54) at simulation time, $t^* = 202$, at $x/D = 6.2$ (left) [and 7.2 - (right), details in figure C.2)], where x is measured from the centroid of the geometry. Top left : Side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) denoted by black dotted lines ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.

Figure C.2: Depiction of coherent structures in the wake of geometry 4 (vibrissa-like geometry with no major axis undulations with axis ratio 0.54) at simulation time, $t^* = 202$, at $x/D = 7.2$ (right plane cut - refer to figure C.1), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores in the right plane-cut ; top right : enlarged view of the top half of the vortex cores and tangential velocity vector in the right plane-cut ; bottom : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the right plane-cut. Legend: orange area denotes cross-sectional area of vortex cores.

C.2 Analysis of wake topology of geometry 5

Figure C.3: Depiction of coherent structures in the wake of geometry 5 (vibrissa-like geometry with no minor axis undulations) at simulation time, $t^* = 225$, at $x/D = 5.3$ (left) [6.3 (middle) and 7.05 - (right), details in figures C.4, C.5)], where x is measured from the centroid of the geometry . Top left : Side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) denoted by black dotted lines ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores.

Figure C.4: Depiction of coherent structures in the wake of geometry 5 (vibrissa-like geometry with no minor axis undulations) at simulation time, $t^* = 225$, at $x/D = 6.3$ (middle plane cut - refer to figure C.3), where x is measured from the centroid of the geometry . Top left : overview of the vortex cores in the middle plane-cut ; top right : overview of the vortex cores and tangential velocity vector in the middle plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the middle plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the middle plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores.

Figure C.5: Depiction of coherent structures in the wake of geometry 5 (vibrissa-like geometry with no minor axis undulations) at simulation time, $t^* = 225$, at $x/D = 7.05$ (right plane cut - refer to figure C.3), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores in the right plane-cut ; top right : overview of the vortex cores and tangential velocity vector in the right plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the right plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the right plane-cut. Legend: cyan area denotes cross-sectional area of vortex cores.

C.3 Analysis of wake topology of geometry 9

Figure C.6: Depiction of coherent structures in the wake of geometry 9 (old vibrissa geometry) at simulation time, $t^* = 228$, at $x/D = 6.47$ (left) [and 7.67 - (right), details in figures C.7)], where x is measured from the centroid of the geometry. Top left : Side view of the vortex cores and the location of plane-cuts (perpendicular to streamwise) denoted by black dotted lines ; top right : overview of the vortex cores and tangential velocity vector in the left plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the left plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the left plane-cut. Legend: green area denotes cross-sectional area of vortex cores.

Figure C.7: Depiction of coherent structures in the wake of geometry 9 (old vibrissa geometry) at simulation time, $t^* = 228$, at $x/D = 7.67$ (right plane cut - refer to figure C.6), where x is measured from the centroid of the geometry. Top left : overview of the vortex cores in the right plane-cut ; top right : overview of the vortex cores and tangential velocity vector in the right plane-cut ; bottom left : enlarged view of the top half of the vortex cores and tangential velocity vector in the right plane-cut ; bottom right : enlarged view of the bottom half of the vortex cores and tangential velocity vector in the right plane-cut. Legend: green area denotes cross-sectional area of vortex cores.

Appendix D

Onset of Instability in The Wake of a Circular Cylinder

In this section, the location of onset of the perturbation in the vortex tubes (which subsequently formed the vortical braids) is performed. It should be noted that the onset perturbation of the vortical braids existed long before the appearance of the vortical braids connecting neighbouring vortex tubes. Figure D.1 shows the initial phase perturbation in the spatial evolution of the wake (particularly, the vortex tubes and vortical braids) of the freely vibrating circular cylinder during its stable period of response for $t^* = 224$, $U_* = 5.2$, $Re = 500$ in the vicinity of the circular cylinder.

A closer observation of figure D.1 revealed the onset of the perturbation in the top figures of figure D.1. These figures represented the vortex cores and tangential flow velocity vector at $x/D = 0.35$. From the overview plot, it can be observed that the wake topology of the freely vibrating circular cylinder showed no signs of perturbation. However, from an enlarged view of the near field flow of the circular cylinder in the vicinity of $z/D = 6.15$ (measured from the base of the circular cylinder), a slight introduction of spanwise velocity can be observed along the tangential flow velocity vector at $z/D = 6.15$. In its subsequent evolution at $x/D = 0.4$ shown in figure D.1, the initial very localised perturbation at $z/D = 6.15$ is observed to propagate along the crossflow direction (Y-axis) at the same spanwise location along the circular cylinder. At $x/D = 0.45$ (in figure D.1), no observable deformation is perceivable in the overview of the wake around the circular cylinder. From the enlarged view of the vortex cores and tangential velocity vector at $x/D = 0.5$, it can be observed that the perturbation (in the form of the spanwise flow velocity) propagates to its neighbouring flow.

Figure D.1: Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x) where x is measured from the centroid of the geometry. Top left : overview of vortex cores and tangential flow velocity vectors at $x/D = 0.35$; top right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.35$ in the vicinity of $z/D = 6.15$ (measured from the base of the circular cylinder) ; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.4$ in the vicinity of $z/D = 6.15$; middle right: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.45$; bottom: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.5$ in the vicinity of $z/D = 6.15$. Legend: orange area denotes cross-sectional area of vortex cores.

Figure D.2 illustrates the subsequent spatial propagation of the perturbation in the wake of the freely vibrating in crossflow circular cylinder shown in figure D.1.

As the flow continued downstream, the spanwise flow velocity perturbations are observed to propagate as shown in figure D.2. From figure D.2, variation of the spanwise flow velocity along the transverse direction of the near-field wake is observed to trigger the undulatory shape of the vortex tubes. However, the first undulatory vortex tube occurred on the vortex tube located in the rear of the circular cylinder (region of largest pressure difference, hence, largest amplification of perturbation). Such instability in the vortex tube at the rear of the circular cylinder resulted in propagation of the instability in its neighbouring vortex tubes. However, this instability is observed to quickly propagate to the farther neighbouring (in transverse direction for 'P+S' vortical arrangement in the wake of circular cylinder) vortex tubes. At $x/D = 0.65$, the instability is observed to propagate and affect all the three vortical tubes in the very near-field wake of the body. Figure D.3 depicts the full propagation of the spanwise velocity perturbation on the neighbouring vortex tube as well as the initiation of the perturbation in between the vortex tubes to create a pair of interconnected vortex cores denoted as 'P' in a 'P+S' vortical arrangement in the wake of a freely vibrating circular cylinder.

Detailed observations of figure D.3 show the early stage of perturbation on the neighbouring vortex tubes with haste amplification within the region of $1.1 \leq x/D \leq 1.37$. The instability of the vortex tubes is observed to form undulations along the vortex tubes with the strongest spanwise flow representing the region with region of greatest gradient of the undulations (the location of crossing of the line $y = 0$). Spanwise velocities are found to be absent at the peak and the valley of the instability undulation on the vortex tubes. As the spanwise velocities are further exacerbated, new vortex cores are created in between the neighbouring vortex tubes at $x/D = 1.4$. The creation of these new vortex cores are observed to facilitate the merging between the neighbouring vortex tubes which resulted in the formation of a single-pair of vortex tubes of opposite polarity ('P' vortex formation type). Such formation is observed to reach its conclusion in the annihilation of the weaker vortex tube (vortex tube of lower kinetic energy). Hence, a new vortex tube is formed upon the annihilation of the weaker vortex tube. Figure D.4 shows the further continuation of the development of the instability into vortical braids connecting the vortex tubes (continuation of figures D.2, D.3).

Figure D.2: Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x) where x is measured from the centroid of the geometry. Top left : enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.57$ in the vicinity of $z/D = 6.15$; top right: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.58$; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.58$ in the vicinity of $z/D = 6.15$; middle right: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.6$; bottom left: overview of vortex cores and tangential flow velocity vectors at $x/D = 0.65$; bottom right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 0.65$ in the vicinity of $z/D = 6.15$. Legend: orange area denotes cross-sectional area of vortex cores.

Figure D.3: Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x) where x is measured from the centroid of the geometry. Top left : enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.1$ in the vicinity of $z/D = 6.15$; top right: overview of vortex cores and tangential flow velocity vectors at $x/D = 1.3$; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.37$ in the vicinity of $z/D = 6.15$; middle right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.4$ in the vicinity of $z/D = 6.15$; bottom left: overview of vortex cores and tangential flow velocity vectors at $x/D = 1.4$; bottom right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 1.7$ in the vicinity of $z/D = 6.15$. Legend: orange area denotes cross-sectional area of vortex cores.

Figure D.4: Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at various streamwise location (x), where x is measured from the centroid of the geometry, and in the vicinity of $z/D = 6.15$, where z is measured from the base of the circular cylinder. Top left : enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.1$; top right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.2$; middle left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.3$; middle right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.5$; bottom left: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 2.8$; bottom right: enlarged view of vortex cores and tangential flow velocity vectors at $x/D = 3.0$. Legend: orange area denotes cross-sectional area of vortex cores.

Figure D.5: Depiction of coherent structures in the wake of a freely vibrating in crossflow circular cylinder at $Re = 500$ and $t^* = 224$ at $x/D = 3.3$. Left: Spanwise overview ; right : enlarged spanwise view in the vicinity of $z = 6.15$

From figure D.4, further development of the new vortex tube can be observed. The new vortex tubes is observed to merge further with a larger vortex tubes (of opposite pair) which resulted in annihilation of the new vortex tube. During this annihilation between the two large vortex tubes of opposite polarity, regular occurrences of vortical braids are initiated. Unlike previous annihilation where the kinetic energy of the vortex tubes are significantly different, the current annihilation involved two large vortex tubes of lesser difference in kinetic energy. As such, no chaotic motion of fluid occurred as the kinetic energy of the fluid of the final vortex tube is limited to a larger degree (as compared to the kinetic energy of the fluid in previous newly-minted vortex tubes). The regular occurrences of the vortical braids are observed to draw their kinetic energy from the vortex tubes. Hence, this reduces the kinetic energy of the vortex tubes, which resulted in the lower forcing on the body due to the shedding of the vortex tubes. Figure D.5 shows the spanwise overview of the fully developed vortical braids and its respective tangential velocity vector (to determine polarity of the vortical braids).

The vortical braids in figure D.5 is similar to the vortical braids depicted in figure 6.36 with opposite direction of motion.

Appendix E

Instability of The Wake of A Freely Vibrating Structures

E.1 Freely Vibrating in Crossflow New Vibrissa Geometry

In this section, depiction of vortex cores and tangential velocity vectors in the wake of a freely vibrating new vibrissa geometry will be presented. The tangential velocity vectors are used to determine the polarity of the vortical braids (vortex cores) in the plane-cuts $x/d = 6.6$, 8.9 , and 11.8 (refer to figure 6.60).

Figure E.1: Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the plane-cut (perpendicular to streamwise) at $x/D = 6.6$, $t^* = 194$ and $U^* = 5.97$. Left : overview of bulk of vortical braids for each nodal plane of the minor axis ; right : enlarged view of the upper half of the corresponding (to the left figure) bulk of vortical braids. The bulk of vortical braids at this plane-cut is found to be symmetric about the peak of the nodal plane. Hence, the lower half of the bulk of vortical braids is found to be a mirror-image of the upper half of the vortical braids about the peak of the corresponding nodal plane. Top : vortex cores about the first nodal plane of minor axis (from the top of geometry) ; middle : vortex cores about the second nodal plane of minor axis (mid-span of geometry) ; bottom : vortex cores about the third nodal plane. Legend: orange area denotes vortex cores.

Figure E.2: Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the plane-cut (perpendicular to streamwise) at $x/D = 8.9$, $t^* = 194$ and $U^* = 5.97$. Top left : overview of bulk of vortical braids for the third nodal plane of the minor axis (from the top, for the first and second nodal planes, refer to figure 6.61) ; top right : enlarged view of the upper half of the bulk of vortical braids ; bottom left : enlarged view of the lower half of the bulk of vortical braids ; bottom right : enlarged view of the bulk of vortical braids at the spanwise end (last nodal plane) of the geometry. Legend: orange area denotes vortex cores.

Figure E.3: Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the first nodal plane of the plane-cut (perpendicular to streamwise) at $x/D = 11.8$, $t^* = 194$ and $U^* = 5.97$. Top left : overview of bulk of vortical braids for the first nodal plane of the minor axis (from the top) ; top right : enlarged view of the upper half of the bulk of vortical braids ; bottom : enlarged view of the lower half of the bulk of vortical braids. Legend: orange area denotes vortex cores.

Figure E.4: Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the second nodal plane of the plane-cut (perpendicular to streamwise) at $x/D = 11.8, t^* = 194$ and $U^* = 5.97$. Left : overview of bulk of vortical braids for the first nodal plane of the minor axis (from the top) ; right : enlarged view about the centre of the bulk of vortical braids. Legend: orange area denotes vortex cores.

Figure E.5: Vortex cores and tangential velocity vector of a freely vibrating in crossflow new vibrissa geometry in the plane-cut (perpendicular to streamwise) at $x/D = 11.8$, $t^* = 194$ and $U^* = 5.97$. Top left : overview of bulk of vortical braids for the third nodal plane of the minor axis (from the top) ; top right : enlarged view of the upper half of the bulk of vortical braids ; bottom left : enlarged view of the lower half of the bulk of vortical braids ; bottom right : enlarged view of the bulk of vortical braids at the spanwise end (last nodal plane) of the geometry. Legend: orange area denotes vortex cores.

E.2 Freely Vibrating in Streamwise and Crossflow

Vibrissa-Like Geometry with No Major Axis Undulations and Axis Ratio of 0.44

Figure E.6: Vortex cores and location of plane-cuts in the wake of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) for $Re = 500$, $U^* = 4.8$ and natural frequency ratio (f_{nr}) = 2.0. Top left : side view of the wake and location of plane cuts (from left to right) $x/D = 4.2, 5.6$, and 6.0 for $t^* = 234$; top right : side view of the wake and location of plane cuts (from left to right) $x/D = 4.6, 5.2$, and 6.2 for $t^* = 246$; bottom : side view of the wake and location of plane cuts (from left to right) $x/D = 4.4$ for $t^* = 284$.

Figure E.7: Vortex cores and tangential velocity vector of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) for $Re = 500$, $t^* = 234$ and $U^* = 4.80$. Top left : overview of in the plane-cut at $x/D = 4.2$; top right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 4.2$; middle left : overview of in the plane-cut at $x/D = 5.6$; middle right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 5.6$; bottom left : overview of in the plane-cut at $x/D = 6.0$; bottom right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 6.0$. Legend: orange area denotes vortex cores.

Figure E.8: Vortex cores and tangential velocity vector of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) for $Re = 500$, $t^* = 246$ and $U^* = 4.80$. Top left : overview of in the plane-cut at $x/D = 4.6$; top right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 4.6$; middle left : overview of in the plane-cut at $x/D = 5.2$; middle right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 5.2$; bottom left : overview of in the plane-cut at $x/D = 6.2$; bottom right : enlarged view of a period of vortical braids in the plane-cut at $x/D = 6.2$. Legend: orange area denotes vortex cores.

Figure E.9: Vortex cores and tangential velocity vector of a freely vibrating in streamwise and crossflow geometry 3 (vibrissa-like geometry with no major axis undulations and axis ratio of 0.44) in the plane-cut at $x/D = 4.4$ for $Re = 500$, $t^* = 246$, and $U^* = 4.80$. Left : overview of in the plane-cut ; right : enlarged view of a period of vortical braids in the plane-cut. Legend: orange area denotes vortex cores.

Appendix F

Wake Topology In Wake-Induced Vibration (WIV) Structures

F.1 WIV on Trailing Circular Cylinder

Figure F.1: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing circular cylinder at inline spacing of $4D$ and $CDR = 0.6$. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture)..

Dimensionless Time = 272

Dimensionless Time = 273

Dimensionless Time = 274

Dimensionless Time = 275

Dimensionless Time = 276

Dimensionless Time = 277

Dimensionless Time = 278

Dimensionless Time = 279

Dimensionless Time = 280

Dimensionless Time = 281

Dimensionless Time = 282

Dimensionless Time = 283

Dimensionless Time = 284

Dimensionless Time = 285

Dimensionless Time = 286

Dimensionless Time = 287

Figure F.2: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing circular cylinder at inline spacing of $4D$ and $CDR = 2.6$. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture).

F.2 WIV on Trailing Elliptical Cylinder

Figure F.3: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing elliptical cylinder at inline spacing of $4D$ and $CDR = 0.6$. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture)..

Dimensionless Time = 371

Dimensionless Time = 374

Dimensionless Time = 375

Dimensionless Time = 377

Dimensionless Time = 378

Dimensionless Time = 379

Dimensionless Time = 381

Dimensionless Time = 382

Dimensionless Time = 383

Dimensionless Time = 384

Dimensionless Time = 386

Dimensionless Time = 388

Dimensionless Time = 390

Dimensionless Time = 392

Dimensionless Time = 394

Dimensionless Time = 396

Figure F.4: Depiction of vortex cores and Wake-Induced Vibration (WIV) in a cycle of oscillation (and corresponding to one wavelength of von Karman street) on the interaction between the von Karman street and a trailing elliptical cylinder at inline spacing of $4D$ and $CDR = 2.8$. Direction of view: left to right and top to bottom (refer to dimensionless time for reference, located at the bottom of each picture).

Appendix G

List of Awards and Publications

Lists are chronologically ordered.

G.1 List of Awards and Honorary Appointments

- § Runner-up in World Young Persons' Lecture Competition (Oct 2013),
- § Winner of Young Person's Lecture Singapore Competition (May 2013),
- § Awarded Speaker at World Noble Peace Prize Laureates Summit 2013 (Warsaw, Poland),
- § Winner of Essay Competition for World Noble Peace Prize Laureates Summit 2013,
- § Invited participant to Complexity and Governance Conference (July 2013),
- § Founder Of Marine Technological Society (MTS) Singapore Chapter (2013),
- § Appointed Chair (by HQ-MTS) Of Marine Technological Society Singapore Chapter (2013), and
- § Awarded scholarship from Singapore MIT Alliance for Research and Technology (SMART) for postgraduate studies.

G.2 List of Journal Publications

- § Hans, H., Miao, J.M. and Triantafyllou, M. 2014. **Mechanical Characteristics of Harbour Seal (Phoca Vitulina) Vibrissae under various circumstances and their implications on its underwater sensing methodology.** Bioinspir. Biomim. 9 (2014) 036013.

- § Hans, H., Miao, J.M. and Triantafyllou, M. 2014. **A Cue on The Underwater Sensing Principle of Harbour Seals with Their Vibrissae**. In writing, to be published in Bioinspiration and Biomimetics.
- § Hans, H., Miao, J.M. and Triantafyllou, M. 2014. **Seal Whisker-Like Geometries and Their One and Two-Dimension Vortex-Induced Vibration Properties**. In writing, to be published in Journal of Fluid Mechanics.
- § Hans, H., Miao, J.M. and Triantafyllou, M. 2014. **Responses of Harbour Seal Vibrissae-Inspired Geometry at Various Angle of Attacks**. In writing, to be published in Journal of Fluids and Structures.

G.3 List of Invited Presentations

- § Hans, H., Miao, J.M., and Triantafyllou, M. 2014. **Seal Vibrissa-Like Geometries and Their Secondary Instabilities**. Center for Environmental Sensing and Modelling (CENSAM) Seminar, May 2014.
- § Hans, H. (NTU), Triantafyllou, M. (MIT), Patrikalakis, N. (MIT), Barbastathis, G.(MIT), Miao, J.M. (NTU). 2013. **Biomimetic Flow-Field Sensing for Energy-Efficient AUV Design**. Singapore Advisory Board Review, January 2013.
- § Hans, H., Miao, J.M., Triantafyllou, M. 2013. **Studies on tandem circular cylinders and circular-elliptical cylinder setup through 2D Numerical Simulations**. Department of Fluid Dynamics, IHPC, A*STAR, August 2013.
- § Hans, H., Miao, J.M., and Triantafyllou, M. 2012. . Center for Environmental Sensing and Modelling (CENSAM) Seminar, January 2012.

G.4 List of Conference Publications (Presentations for APS)

- § Hans, H., Miao, J.M. and Triantafyllou, M. 2014. **Vortex-Induced Vibration (VIV) on Circular Cylinder with Offset Angle**. Proceedings of ISOPE 2014, Busan, South Korea.

- § Hans, H., Miao, J.M. and Triantafyllou, M. 2013. **Vortex-Induced Vibration (VIV) Reduction Properties of Seal Whisker-Like Geometries**. Bulletin of American Physical Society 2014, Pittsburgh, USA.
- § Hans, H., Weymouth, G.D., Miao, J.M. and Triantafyllou, M. 2013. **Whisker-Like Geometry And Their Reduction of Vortex-Induced Forces**. Proceedings of OCEANS 2013, Bergen, Norway.
- § Hans, H., Miao, J.M. and Triantafyllou, M. 2012. **Characterisation of von Karman Street with Seal Whisker-Like Sensor**. Proceedings of IEEE SENSOR 2012, Taipei, Taiwan.
- § Hans, H., Miao, J.M., Alvarado, P.V., and Triantafyllou, M. 2011. **Chemical Composition and Physical Features of Harbour Seal (Phoca Vitulina) Vibrissae For Undewater Sensing Application**. Proceedings of IEEE ROBIO 2011, Phuket Island, Thailand.
- § Hans, H., Alvarado, P.V., Thekkoodan, D.J., Miao, J.M. and Triantafyllou, M. 2011. **A Whisker Sensor: The Role of Geometry and Boundary Conditions**. Bulletin of American Physical Society 2011, Baltimore, USA.