

UNIVERSIDAD POLITÉCNICA DE MADRID
Escuela Técnica Superior de Ingeniería Aeronáutica y del Espacio



Modelling and Simulation of Vortex-Induced Vibrations in Wind Turbines

DOCTORAL THESIS

Submitted for the degree of Doctor by:

Ricardo Fernández de Aldama
MSc in Aeronautical Engineering

Madrid, 2025



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“Practicing mutual aid is the surest means for giving each other and to all the greatest safety, the best guarantee of existence and progress, bodily, intellectual and moral.”

– Peter Kropotkin, *Mutual Aid: A Factor of Evolution*

To my loved ones, near and far, and to the cause of universal public education.

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Abstract

This thesis investigates vortex-induced vibrations (VIV) in wind turbine towers and blades. It addresses the challenges posed by this complex aeroelastic phenomenon through two complementary approaches: the application of engineering models to wind turbine towers and high-fidelity simulations of wind turbine airfoil sections. The research aims to enhance our understanding and predictive capabilities regarding VIV, which is critical for ensuring the structural integrity and safe installation and maintenance of wind turbines.

The first part of the thesis focuses on the application of time-domain engineering models to the NREL 5-MW wind turbine tower. Three types of models are analysed: Type A (forced system), Type B (fluidelastic system), and Type C (coupled system). The Type B model, which incorporates both stochastic lift forces and aeroelastic damping terms, is found to be effective for predicting the VIV response of both highly and lowly damped structures. The Type C model, while more complex due to the presence of an extra differential equation for the wake dynamics, offers the potential for a more physically grounded representation of the system, involving fewer semi-empirical approximations. The Type C model is especially promising in its application to wind turbine blades, given its inherent ability to capture the aeroelastic phenomena involved in VIV and the lack of experimental and full-scale data available to tune a Type B semi-empirical model. However, further research is needed to develop and validate a Type C model that is suited to the blade case.

The second part of the thesis extends the investigation to wind turbine blades using high-fidelity fluid-structure interaction simulations. Three cases are considered: a 2D airfoil using the Reynolds-averaged equations (URANS) turbulence-modelling approach, and two 3D cases using a hybrid approach between URANS and large-eddy simulations (DDES), for two blade aspect ratios. The simulations reveal critical insights into vortex shedding behaviour, including intermittent shedding regimes and the challenges of defining a single Strouhal number for 3D cases. The VIV response is characterized using a novel growth rate metric, which effectively identifies the fluid-structure lock-in range even in short simulations. A strong agreement is found between 2D and 3D results within the lock-in range, highlighting the potential of 2D simulations for preliminary investigations.

In conclusion, this thesis advances the understanding of VIV in wind turbines by analysing time-domain engineering models and high-fidelity simulations. The research provides a foundation for future work aimed at refining VIV models, incorporating turbulence and 3D effects, and validating findings through experimental data. These advancements contribute to the design of safer, more efficient wind turbines, supporting the global transition to sustainable energy.

Resumen

Esta tesis investiga las vibraciones inducidas por torbellinos (VIV) en torres y palas de aerogenerador. El desafío que representa predecir este complejo fenómeno aeroelástico se aborda mediante dos enfoques complementarios: la aplicación de modelos ingenieriles a torres de aerogeneradores y la simulación de alta fidelidad de secciones de pala de aerogeneradores. El objetivo de la investigación es mejorar nuestra comprensión y capacidad predictiva en relación con las VIV, lo cual es fundamental para garantizar la integridad estructural y la seguridad durante la instalación y mantenimiento de los aerogeneradores.

La primera parte de la tesis se centra en la aplicación de modelos ingenieriles en el dominio del tiempo a la torre del aerogenerador NREL 5-MW. Se analizan tres tipos de modelos: Tipo A (sistema forzado), Tipo B (sistema fluidoelástico) y Tipo C (sistema acoplado). El modelo Tipo B, que incorpora tanto fuerzas de sustentación estocásticas como términos de amortiguamiento fluidoelástico, demuestra ser eficaz para predecir la respuesta a las VIV en estructuras tanto con bajo como con alto amortiguamiento. El modelo Tipo C, aunque más complejo debido a la incorporación de una ecuación diferencial adicional que representa la dinámica de la estela, ofrece el potencial de una representación más fundamentada físicamente del sistema, implicando menos aproximaciones semiempíricas. El modelo Tipo C resulta especialmente prometedor para su aplicación a palas de aerogeneradores, dado su capacidad inherente para capturar los fenómenos fluidoelásticos asociados a las VIV y la escasa disponibilidad de datos experimentales y mediciones en campo necesarios para ajustar un modelo Tipo B semiempírico. Sin embargo, queda como trabajo futuro el desarrollar y validar un modelo Tipo C adaptado específicamente al caso de las palas.

La segunda parte de la tesis amplía la investigación a palas de aerogeneradores mediante simulaciones de interacción fluido-estructura de alta fidelidad. Se consideran tres casos: un perfil aerodinámico 2D utilizando las ecuaciones promediadas de Reynolds (URANS), y dos secciones 3D con diferente relación de aspecto, utilizando un método híbrido entre URANS y la resolución de grandes torbellinos (DDES). Las simulaciones revelan información crítica sobre el comportamiento del desprendimiento de torbellinos, incluida la aparición de regímenes intermitentes de desprendimiento y los desafíos que supone definir un único número de Strouhal en estos casos. La respuesta a las VIV se caracteriza de manera novedosa mediante una métrica de ritmo de crecimiento, que identifica de forma efectiva el rango de sincronización fluido-estructura incluso en simulaciones de corta duración. Se observa una fuerte concordancia entre los resultados 2D y 3D dentro del rango de sincronización, lo que pone de relieve el potencial de las simulaciones 2D para estudios preliminares.

En conclusión, esta tesis contribuye al avance en la comprensión de las VIV en aerogeneradores mediante el análisis de modelos ingenieriles en el dominio del tiempo y simulaciones de alta fidelidad. La investigación sienta las bases para futuros trabajos orientados a refinar los modelos ingenieriles de VIV mediante simulaciones de alta fidelidad, incorporando efectos tridimensionales y debidos a la turbulencia, y validando los resultados mediante datos experimentales. Estos avances contribuyen al diseño de aerogeneradores más seguros y eficientes, apoyando la transición energética.

Table of Contents

Acknowledgement	v
Abstract	vi
Resumen	vii
List of Figures	x
List of Tables	xvi
Abbreviations and acronyms	xix
Nomenclature	xxvi
1 Introduction	1
1.1 Background and motivation	2
1.2 Research question and objectives	7
1.3 Scope and organization of the thesis	8
2 State of the art	9
2.1 Physical phenomena	9
2.1.1 Basic phenomena - vortex shedding	10
2.1.2 Basic phenomena - vortex-induced vibrations (VIV)	11
2.1.3 Knowledge gaps	11
2.2 Mathematical modelling	12
2.3 Numerical Simulations	17
3 Tower vortex-induced vibrations: engineering models comparison	21
3.1 Methodology	21
3.1.1 Structural model	22
3.1.2 VIV models	22
Type A	23
Type B	25
Type C	28
Summary	30
3.2 Type B model validation: Aachen chimney	32
3.2.1 Sources of uncertainty in validation	32
3.2.2 Aachen chimney VIV response	34
3.3 Case study: NREL 5-MW tower	35
3.3.1 Simulations overview	35
3.3.2 VIV models parameters	38

3.3.3	Results comparison	45
3.4	Conclusions and future work	49
4	Blade vortex-induced vibrations: high-fidelity numerical simulations	53
4.1	Methodology	54
4.1.1	Simulations overview	54
4.1.2	CFD setup	56
4.1.3	Structural dynamics system setup	60
4.2	Results and discussion	62
4.2.1	Verification	62
	Numerical setup sensitivity analysis	62
	Comparison to published data	63
4.2.2	Computational cost	64
4.2.3	Static airfoil case	66
	Lift coefficient frequency content	66
	Flow regimes	68
	Strouhal number analysis	74
4.2.4	Free to move airfoil case	81
	Initial displacement condition for the 3D-21c case	81
	Frequency ratio definition	82
	Amplitude growth rate	82
	Aerodynamic coefficients characterization	87
4.2.5	Flow field analyses	96
4.3	Conclusions and future work	100
5	Discussion	103
5.1	Type A	104
5.2	Type B	106
5.3	Type C	108
5.4	Preliminary application of engineering models to the NREL 5-MW blade . .	111
5.5	Closing remarks	116
6	Conclusions	119
6.1	Conclusions	119
6.2	Future work	122
6.3	Final remarks	123
	Bibliography	125
A	Tower structural equation of motion derivation	141

List of Figures

1.1	Climate stripes representing the global increase in temperature from 1850 to 2024. Reprinted from #ShowYourStripes (last accessed March 2025), created by Professor Ed Hawkins (University of Reading).	2
1.2	Share of world electricity production by source, with wind energy highlighted, from 2000 to 2023. Data source: Ember (2024); Energy Institute - Statistical Review of World Energy (2024) - with major processing by Our World in Data. Retrieved on March 14, 2025 from Our World in Data	3
1.3	Von Kármán vortex streets behind Madeira island, as captured by NASA's Terra satellite. This image is a cropped version of the original. Source: NASA Earth Observatory (archived). NASA image by Jeff Schmaltz, LANCE/EOSDIS Rapid Response.	4
1.4	Schematic of the vortex-induced vibration problem in wind turbines and its treatment in the present thesis. (a) Depiction of a wind turbine, with inflow coming from one side at velocity U . Two arbitrary spanwise sections are marked, one at the blade (A-A'), and one at the tower (B-B'). The vortex-induced motion is depicted as $x(t)$ for the blade and $w(t)$ for the tower. (b) Detail of the vortex-induced vibration of the blade at station A-A', showing the blade motion and the vortex shedding in the wake. The black vertical line represents the undeformed blade position, while the grey curves represent the blade deflection. (c) Detail of the vortex-induced vibration of the tower at station B-B', showing the tower motion and the vortex shedding in the wake. The black vertical line represents the undeformed tower position, while the grey curves represent the tower deflection. (d) Example of the high-fidelity simulations performed of the flow around a blade section, with a schematic representation of the one degree-of-freedom elastic system coupled to the fluid dynamics solver. (e) Schematic representation of the application of an engineering model to the tower, where a sheared and turbulent inflow, $U(z, t)$, results in a vortex-induced force, represented by a nonlinear function of the motion, $\mathcal{F}[w(z, t)]$, which is a function of height, z	6

2.1	Schematic representation of the three main types of VIV mathematical models. The structural motion is denoted $w(t)$. (a) In the Type A model (forced system), the fluid forces acting on the structure are modelled as a time-dependent force $F(t)$. (b) In the Type B model (fluidelastic system), the fluid force is a nonlinear function of the structural response, $\mathcal{F}[w(t), t]$. (c) In the Type C model (coupled system), a fluid variable, $q(t)$, is also modelled, such that the fluid force on the structure is a nonlinear function of this variable, $\mathcal{F}[q(t), t]$, and the fluid variable is coupled to the structural motion by a nonlinear term, $\mathcal{Q}[w(t), t]$	13
3.1	Turbulence intensity profiles, assuming that the reported value of $I_u = 0.23$ corresponds to a height of $z_{I_u} = 10$ m or of $z_{I_u} = 26$ m.	33
3.2	Type B modal linear aerodynamic damping ratio for mode 1, $\zeta_{a,1}$, as a function of the velocity ratio, for both assumed heights for the reported turbulence intensity value, z_{I_u} , together with the structural damping ratio, $\zeta_{s,1}$	34
3.3	Maximum dimensionless chimney top displacement, $\max(\tilde{w}_t)$, as a function of the ratio between the chimney top inflow wind speed and the first mode critical wind speed, $U_{\text{ratio},t,1} = U(H)/U_{\text{crit},1}$. Type B VIV model results, for both assumed heights for the reported turbulence intensity value, z_{I_u} , are shown as continuous lines. Experimental data for the three test runs presented in Galemann and Ruscheweyh (1992) are shown as individual points. The Type B results shown correspond to the ensemble average over 12 realizations of the stochastic process.	35
3.4	Recommended damping ratios for the first structural mode of steel wind turbine towers or chimneys, according to different sources. “Jonkman et al. (2009)” shows the value proposed for the NREL 5-MW tower. “Manolas et al. (2022)” shows the value proposed for the NREL 5-MW tower, adjusted to state-of-the-art levels. “IEC 61400-6 Ed.1 (2020)” shows recommended values for steel tubular towers, where “max”: maximum value without further justification, and “calc”: calculated value as a function of the natural frequency for the NREL 5-MW tower. “DS 472, 1992” shows recommended values for “w”: welded and “b”: bolted steel structures (data from Burton et al. (2011)). “EN 1991-1-4:2005+A1 (2010)” shows recommended values for “w”: welded steel stacks, “b,h”: lattice steel towers with high resistance bolts, and “b,o”: lattice steel towers with ordinary bolts (data presented also in Burton et al. (2011)). “Blevins (2001)” data corresponds to statistical quantities derived from measurements of 21 steel towers, where “avg”: average value, “m”: median value, and “Q1”: lower quartile value. The colours of the bar graph are only used to differentiate the cases, and do not imply any further meaning. . . .	37
3.5	(a) Value of the function $\Psi_1(z)D^4(z)$ along the NREL 5-MW tower height. The maximum of this function is used to define the reference height z_{ref} , marked with a horizontal line. (b) NREL 5-MW tower diameter along its height, with the reference diameter $D_{\text{ref}} = D(z_{\text{ref}})$ marked with a vertical line.	38

3.6	Strouhal number of circular cylinders as a function of Reynolds number. Circles and crosses correspond to experimental data as presented in Schewe (1983) and Milford (1982), respectively. Full, dotted and dash-dotted lines correspond to empirical approximations proposed in the standards EN 1991-1-4:2005+A1 (2010), prEN 1991-1-4:2019 (2020) and National Building Code of Canada (1977), respectively. The Reynolds-defined flow regimes are labelled, and their limits marked with vertical dashed lines, as per the definitions given in Sumer and Fredsøe (2006).	39
3.7	Reynolds number along the NREL 5-MW tower height for the range of velocity ratios studied. Marked in a dashed horizontal line is the reference height used for the definition of $U_{\text{ratio,ref},1}$	40
3.8	Strouhal number along the NREL 5-MW tower height for the range of velocity ratios studied. Marked in a dashed horizontal line is the reference height used for the definition of $U_{\text{ratio,ref},1}$	41
3.9	Dimensionless PSD of the lift coefficient according to the Type A model, evaluated at the reference height z_{ref} . The analytical expression from B. J. Vickery and Clark (1972) is compared to the PSD obtained for N_α synthetic realizations of the c_l stochastic process.	42
3.10	Type B damping parameters, specifically: linear negative aerodynamic modal damping parameter, $k_{a,j}$, and cubic positive aerodynamic modal damping parameter, $G_{a,j}$, for mode $j = 1$, as a function of the velocity ratio.	43
3.11	Type B modal linear aerodynamic damping ratio for mode 1, $\zeta_{a,1} = k_{a,1}/(2m_1\omega_1)$, as a function of the velocity ratio, together with the two representative values for the structural damping ratio, $\zeta_{s,1}$	43
3.12	Maximum dimensionless tower top displacement as a function of the ratio between the reference inflow and critical wind speeds, comparing the three VIV models, for (top) low structural damping and (bottom) high structural damping cases. For Type A and Type B, the ensemble average of the maximum dimensionless displacement is shown.	47
3.13	Standard deviation of the dimensionless tower top displacement as a function of the ratio between the reference inflow and critical wind speeds, comparing the three VIV models, for (top) low structural damping and (bottom) high structural damping cases. For Type A and Type B, the ensemble average of the dimensionless displacement standard deviation is shown.	48
3.14	PSD of the dimensionless tower top displacement, as a function of the ratio between frequency and the natural frequency of the first tower mode. Results are shown for the first mode velocity ratio $U_{\text{ratio,ref},1} = 1.28$, comparing the three VIV models, for (top) low structural damping and (bottom) high structural damping cases.	49
4.1	Sketch of the computational domain for the 3D simulations, and axes system. Not to scale.	56

4.2	Visualization of the CFD meshes used in the simulations. (a) Close-up view of the mesh around the airfoil for the 2D configuration. (b) Close-up view of the surface mesh and a cross-sectional slice at one end of the mesh for the 3D-1c (aspect ratio one) configuration. (c) Close-up view of the surface mesh and a cross-sectional slice at one end of the mesh for the 3D-21c (aspect ratio twenty-one) configuration.	58
4.3	Dynamic system representation. The inflow velocity U_∞ is shown for reference.	61
4.4	Mesh sensitivity analysis of the 2D simulations. (a) Time series segments of the airfoil dimensionless displacement (top), lift coefficient (middle) and drag coefficient (bottom), for the three setups considered. (b) Percentage difference of dimensionless maximum displacement, standard deviation of the lift coefficient and mean drag coefficient, for the coarse and medium meshes with respect to the fine mesh.	63
4.5	Comparison of results from the literature replicated with the present implementation. For the comparison with Skrzypiński et al. (2014) (left), the maximum amplitudes are shown normalized with their highest value for each case, after the dimensionless simulation times depicted, as a function of the reduced velocity. In the Lian et al. (2022) case (right), the dimensionless maximum amplitudes are shown as a function of the inflow wind speed.	64
4.6	Normalized dimensionless PSD of the lift coefficient fluctuations, for the static airfoil at Reynolds number $Re = 2 \times 10^6$. The left y -axis corresponds to the 3D-1c case and the right y -axis to the 2D and 3D-21c cases.	67
4.7	Detail of the (a) lift coefficient and (b) drag coefficient as a function of dimensionless time, for the three simulated aspect ratios of the DU96-W-180 airfoil.	69
4.8	2D static case. (Top) lift coefficient time series, and (bottom) dimensionless scalogram of the lift coefficient. In the scalogram, the blue solid line (—) marks the frequency bin with the highest magnitude at each time bin, and the dashed horizontal line (--) indicates the Strouhal number obtained from the 2D static case.	71
4.9	3D-1c static case. (Top) lift coefficient time series, and (bottom) dimensionless scalogram of the lift coefficient. In the scalogram, the blue solid line (—) marks the frequency bin with the highest magnitude at each time bin, and the dashed horizontal line (--) indicates the Strouhal number obtained from the 2D static case.	72
4.10	3D-21c static case. (Top) lift coefficient time series, and (bottom) dimensionless scalogram of the lift coefficient. In the scalogram, the blue solid line (—) marks the frequency bin with the highest magnitude at each time bin, and the dashed horizontal line (--) indicates the Strouhal number obtained from the 2D static case.	73
4.11	Strouhal number estimated as the highest peak in the PSD of the lift coefficient fluctuations, and from the Gaussian fit to the PSD, as a function of the total time used to calculate the PSD. (Top) for the 2D configuration, (middle) for the 3D-1c configuration, and (bottom) for the 3D-21c configuration.	75

4.12	2D static airfoil case. Normalized PSD of the lift coefficient fluctuations together with a Gaussian fit, in (a) linear axes, (b) logarithmic y-axis and (c) log-log axes and for a wider range of frequencies. The central frequency of the Gaussian fit is marked with a vertical line.	78
4.13	3D-1c static airfoil case. Normalized PSD of the lift coefficient fluctuations together with a Gaussian fit, in (a) linear axes, (b) logarithmic y-axis and (c) log-log axes and for a wider range of frequencies. The central frequency of the Gaussian fit is marked with a vertical line.	79
4.14	3D-21c static airfoil case. Normalized PSD of the lift coefficient fluctuations together with a Gaussian fit, in (a) linear axes, (b) logarithmic y-axis and (c) log-log axes and for a wider range of frequencies. The central frequency of the Gaussian fit is marked with a vertical line.	80
4.15	(a) Time series of the dimensionless edgewise displacement, $\tilde{x}$, along with, in a thicker line, the envelope of the time series calculated with the Hilbert transform, $h_{\tilde{x}}$. Additionally, the limits of the time interval selected for the linear fit to $h_{\tilde{x}}$ are marked with vertical dashed lines. (b) Time series of the dimensionless edgewise displacement $\tilde{x}$ within the time interval selected for the linear fit, along with the estimated amplitude $h_{\tilde{x}}$, and the straight line fitted to $h_{\tilde{x}}$ in this portion of envelope. All results are for a 2D FSI simulation at $f_{\text{St},2\text{D}}/f_s = 0.98$	84
4.16	Same as Figure 4.15 but for a 3D-1c FSI simulation at $f_{\text{St},3\text{D-1c}}/f_s = 1.02$. . .	85
4.17	Same as Figure 4.15 but for a 3D-21c FSI simulation at $f_{\text{St},3\text{D-21c}}/f_s = 1.05$. .	86
4.18	Dimensionless displacement amplitude growth rate, $r_{\tilde{x}}$ as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches.	87
4.19	Illustration of the lift coefficient growth rate calculation procedure. See Figure 4.15 for details. All results are for a 2D FSI simulation at $f_{\text{St},2\text{D}}/f_s = 0.98$	89
4.20	Lift coefficient amplitude growth rate, r_{c_l} as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches.	89
4.21	Illustration of the mean drag coefficient growth rate calculation procedure. See Figure 4.15 for details. In this case, the linear fit is made to a time-moving average of c_d . The window used for the moving average is $T_{\text{movmean}} = 3/\text{St}_{2\text{D}}$, which is approximately three vortex shedding periods. All results are for a 2D FSI simulation at $f_{\text{St},2\text{D}}/f_s = 0.98$	91
4.22	Mean drag coefficient growth rate, $r_{\overline{c_d}}$ as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches.	92
4.23	Lift coefficient standard deviation (top) and mean drag coefficient (bottom) as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches. In the 2D and 3D-1c cases the time series used to calculate these statistical parameters are limited to the dimensionless time interval $\tilde{t} \in [150, 300]$, and in the 3D-21c case it is limited to $\tilde{t} \in [50, 200]$	93

4.24	Time series segment of the airfoil (a) dimensionless displacement, (b) lift coefficient and (c) drag coefficient, for the 2D and 3D-1c approaches, at a ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency close to one (inside the lock-in range).	94
4.25	Phase plane of lift coefficient versus (a) dimensionless displacement and (b) dimensionless velocity, for simulation time between $t = 7$ and $t = 10$ s, for the 2D and 3D-1c approaches, at a similar ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency inside the lock-in range. The quadrants with the + sign correspond to segments of the time series where the lift coefficient is adding energy to the motion, and the quadrants with the – sign correspond to those where energy is extracted from the motion.	95
4.26	(a) Instantaneous pressure coefficient field, $c_p = (p - p_\infty) / (0.5\rho_\infty U_\infty^2)$, at a slice in spanwise position $z = 0.5c$, and (b) instantaneous Q-criterion isosurfaces, for $\tilde{Q} = Qc^2/U_\infty^2 = 1$, coloured by the dimensionless spanwise velocity field, $\tilde{U}_z = U_z/U_\infty$, and superimposed to the instantaneous dimensionless spanwise velocity field at the $z = 0.5c$ slice. The $\tilde{U}_z$ colour range is limited to ± 0.2 for improved visualization. In both panels four different cases are shown, from top to bottom: (i) 3D-21c static simulation, (ii) a time instant in the L regime of the 3D-1c static simulation, (iii) a time instant in the H regime of the 3D-1c static simulation, and (iv) 3D-1c free to move simulation with frequency ratio $f_{\text{St},3\text{D-1c}}/f_s = 1.08$	98
4.27	(a) Instantaneous dimensionless velocity magnitude $ \tilde{U} $, (b) instantaneous dimensionless lateral velocity $\tilde{U}_x$ and (c) instantaneous dimensionless longitudinal velocity $\tilde{U}_y$. In the three panels, four different cases are shown, from top to bottom: (i) 3D-21c simulation, (ii) a time instant in the L regime of the 3D-1c simulation, (iii) a time instant in the H regime of the 3D-1c simulation, and (iv) 2D simulation. All panels correspond to the static simulations. For the 3D cases, the results are shown at a slice in spanwise position $z = 0.5c$	99
5.1	Type B modal linear aerodynamic damping ratio for the first NREL 5-MW blade edgewise mode, as a function of the velocity ratio, together with two representative values for the structural damping ratio, $\zeta_{s,1}$	113
5.2	Maximum dimensionless blade tip displacement as a function of the ratio between the reference inflow and critical wind speeds, for the three VIV models. For Type A and Type B, the ensemble average of the maximum dimensionless displacement is shown.	116

List of Tables

2.2	Published research on the application of VIV models to wind turbine (WT) towers and blades or airfoils.	16
3.1	Comparison of the computed structural properties of the validation model with the data reported in Ruscheweyh and Galemann (1996), for the first natural vibration mode.	33
3.2	Type C model parameters chosen to match the response of the Type B model, for each structural damping ratio value considered.	45
4.1	List of simulations performed. For each configuration there is one static CFD simulation and the rest are FSI simulations at different inflow wind speeds. .	55
4.2	Details on the computational cost of the simulations performed. N_{steps} refers to the number of time steps run, N_{CPUs} is the number of CPUs used in parallel via the Message Passing Interface (MPI), $T_{\text{wall-clock}}$ is the approximate wall-clock time for a single simulation, and N_{sims} is the number of simulations of each type (i.e., the number of different inflow wind speeds). The total cost is expressed in CPU hours.	65
4.3	List of Strouhal number values reported for wind turbine airfoils at $\alpha = 90^\circ$ using URANS CFD models (top part of the table) and DES-like models (bottom part of the table). For each CFD model, the studies are sorted in increasing aspect ratio (AR) order. Note that the “2D” entries in the aspect ratio column mean that the simulation is two-dimensional. In the case “Present work, 3D-1c”, a range of Strouhal numbers is given instead of a single value.	77
5.1	Model parameters of the three engineering models, as applied to the blade, and to the tower cases with a structural damping ratio value of $\zeta_{s,1} = 0.25\%$. Note that all Type A parameters are also employed in the Type B model, and that the Strouhal number, St , is also present in the Type C model.	114

Abbreviations and acronyms

AR	aspect ratio
BEM	blade element momentum
CFD	computational fluid dynamics
CPSD	cross-power spectral density
CWT	continuous wavelet transform
DDES	delayed detached-eddy simulation
DES	detached-eddy simulation
DNS	direct numerical simulations
FEM	finite element method
FSI	fluid-structure interaction
IEA	International Energy Agency
IEC	International Electrotechnical Commission
IPCC	Intergovernmental Panel on Climate Change
LES	large eddy simulations
MICIU	Ministerio de Ciencia, Innovación y Universidades
NACA	National Advisory Committee for Aeronautics
NREL	National Renewable Energy Laboratory
PSD	power spectral density
SST	shear stress transport

UPM Universidad Politécnica de Madrid

URANS unsteady Reynolds-averaged Navier-Stokes

VIV vortex-induced vibrations

Nomenclature

Greek symbols

α	Realization number of a stochastic process	$[-]$
β	Normalized wake-structure velocity coupling parameter	$[s^{-1}]$
Δ	LES length scale	$[m]$
δ_{ij}	Kronecker delta	$[-]$
$\delta_{s,j}$	Modal structural logarithmic decrement of mode j	$[-]$
ε	Strength of the van der Pol nonlinear damping term	$[-]$
μ	Fluid dynamic viscosity	$[kg\ m^{-1}\ s^{-1}]$
ω_j	Natural angular frequency of mode j	$[rad\ s^{-1}]$
Φ_j	Wake variable mode shape for mode j	$[-]$
Ψ_j	Tower mode shape of mode j	$[-]$
ρ, ρ_∞	Fluid density	$[kg\ m^{-3}]$
$\zeta_{a,j}$	Modal negative aerodynamic damping ratio of mode j	$[-]$
$\zeta_{s,j}$	Modal structural damping ratio of mode j	$[-]$

Latin symbols

a_L	Dimensionless limiting displacement amplitude	$[-]$
B	Bandwidth parameter	$[-]$
b	Wake-structure velocity coupling parameter	$[m^{-1}\ s^{-1}]$
B_0	Bandwidth parameter in smooth flow	$[-]$
c	Chord length of the airfoil section	$[m]$

c_1	Exponential decay coefficient of the lift coefficient CPSD coherence function	$[-]$
c_2	Sinusoidal decay coefficient of the lift coefficient CPSD coherence function	$[-]$
c_d	Drag coefficient	$[-]$
c_l	Lift coefficient	$[-]$
$c_{l,0}$	Stationary body vortex-shedding lift coefficient amplitude	$[-]$
c_p	Pressure coefficient	$[-]$
D	Tower diameter	$[m]$
D_{ref}	Tower diameter at the tower reference height	$[m]$
$\overline{EI}$	Sectional beam bending stiffness	$[N\ m^3]$
F	Force per unit length	$[N\ m^{-1}]$
f	Frequency	$[Hz]$
F_j	Modal force for mode j	$[N]$
f_s	Structural natural frequency of the airfoil section	$[Hz]$
$f_{s,j}$	Natural frequency of mode j	$[Hz]$
f_{St}	Vortex-shedding frequency according to Strouhal law	$[Hz]$
f_{Stref}	Vortex-shedding frequency according to Strouhal law, at the tower reference height	$[Hz]$
f_w	Instantaneous motion frequency	$[Hz]$
G	Nonlinear positive aerodynamic damping factor	$[kg\ s\ m^{-3}]$
$G_{a,j}$	Uncoupled nonlinear positive aerodynamic modal damping factor	$[kg\ s\ m^{-2}]$
$G_{a,jklm}$	Coupled nonlinear positive aerodynamic modal damping tensor	$[kg\ s\ m^{-2}]$
H	Tower top height	$[m]$
H_1	Stall parameter	$[-]$
I_u	Longitudinal turbulence intensity	$[-]$
I^*	Effective turbulence intensity	$[-]$
j	Mode number	$[-]$
k	Stiffness constant per unit length of the airfoil section	$[N\ m^{-2}]$

K_a	Dimensionless linear negative aerodynamic damping factor	$[-]$
k_a	Linear negative aerodynamic damping factor	$[\text{kg m}^{-1} \text{s}^{-1}]$
$k_{a,j}$	Uncoupled linear negative aerodynamic modal damping factor	$[\text{kg s}^{-1}]$
$k_{a,jk}$	Coupled linear negative aerodynamic modal damping matrix	$[\text{kg s}^{-1}]$
K_{I_u}	Dimensionless linear negative aerodynamic damping turbulence contributing factor	$[-]$
k_p	Peak factor	$[-]$
K_{Re}	Dimensionless linear negative aerodynamic damping Reynolds contributing factor	$[-]$
$K_{U_{\text{ratio},j}}$	Dimensionless linear negative aerodynamic damping resonance contributing factor	$[-]$
L	Spanwise length of the airfoil section	$[\text{m}]$
l	Lift force per unit length of the airfoil section	$[\text{N m}^{-1}]$
l_j	Wake variable generalized coordinate of mode j	$[-]$
L_u	Longitudinal turbulence length scale	$[\text{m}]$
m	Mass per unit length	$[\text{kg m}^{-1}]$
$m_{\text{eq},j}$	Equivalent mass for mode j	$[\text{kg m}^{-1}]$
m_j	Modal mass, of mode j	$[\text{kg}]$
N_α	Number of stochastic realizations	$[-]$
N_{CPUs}	Number of CPUs used in parallel	$[-]$
n_m	Number of modes	$[-]$
N_{sims}	Number of simulations	$[-]$
N_{steps}	Number of time steps	$[-]$
n_w	Number of wake modes	$[-]$
n_z	Number of tower height levels	$[-]$
p	Pressure field	$[\text{Pa}]$
p_∞	Pressure at the inlet	$[\text{Pa}]$
p_j	Tower lateral deflection generalized coordinate of mode j	$[\text{m}]$
Q	Second invariant of the velocity gradient tensor	$[\text{s}^{-2}]$

q	Wake-dynamics related variable	$[-]$
q_0	Wake-dynamics related variable amplitude for the stationary body condition	$[-]$
$R(z_1, z_2)$	Coherence function of the lift coefficient CPSD between tower heights z_1 and z_2	$[-]$
$r(z_1, z_2)$	Normalized distance along the tower for the lift coefficient CPSD coherence function	$[-]$
Re	Reynolds number	$[-]$
$S_{c_l}(z_1, z_2, f)$	Cross-power spectral density of the lift coefficient, between tower heights z_1 and z_2	$[\text{Hz}^{-1}]$
Sc_j	Scruton number for mode j	$[-]$
St	Strouhal number	$[-]$
St_{ref}	Strouhal number at the tower reference height	$[-]$
T	Time period	$[\text{s}]$
t	Time	$[\text{s}]$
Δt	Simulation time step	$[\text{s}]$
t_{sd}	Structural damping constant	$[\text{s}]$
t_{sim}	Simulated time	$[\text{s}]$
t_{trans}	Transient time removed for results post-processing	$[\text{s}]$
$T_{\text{wall-clock}}$	Wall-clock time	$[\text{h}]$
U, U_{∞}	Inflow velocity	$[\text{m s}^{-1}]$
$U_{\text{crit},j}$	Critical inflow velocity of mode j	$[\text{m s}^{-1}]$
$U_{\text{crit,ref},j}$	Critical inflow velocity at the tower reference height, for mode j	$[\text{m s}^{-1}]$
$U_{\text{ratio},j}$	Ratio between inflow velocity and critical velocity of mode j	$[-]$
$U_{\text{ratio,ref},j}$	Ratio between inflow wind speed and critical velocity, at the tower reference height, for mode j	$[-]$
$U_{\text{ratio},t,j}$	Ratio between inflow wind speed and critical velocity, at the tower top, for mode j	$[-]$
U_{ref}	Inflow velocity at the tower reference height	$[\text{m s}^{-1}]$
$\tilde{u}_x$	Dimensionless velocity of the airfoil section	$[-]$

U_x	Flow velocity component in the zx (lateral) direction	$[\text{m s}^{-1}]$
U_y	Flow velocity component in the y (longitudinal) direction	$[\text{m s}^{-1}]$
U_z	Flow velocity component in the z (spanwise) direction	$[\text{m s}^{-1}]$
w	Tower lateral deflection	$[\text{m}]$
w_t	Tower top lateral deflection	$[\text{m}]$
x	Lateral (chordwise) displacement of the airfoil section	$[\text{m}]$
x_0	Initial displacement condition of the airfoil section	$[\text{m}]$
$[x, y, z]$	Coordinate system employed in the airfoil section high-fidelity simulations. x is the lateral (chordwise) direction, pointing towards the trailing edge, y is the downstream direction, and z is the spanwise direction	$[\text{m}]$
$[\Delta x, \Delta y, \Delta z]$	Mesh cell lengths	$[\text{m}]$
y^+	Dimensionless wall distance	$[-]$
z	Tower height from ground, or blade spanwise coordinate	$[\text{m}]$
z_{I_u}	Height at which the longitudinal turbulence intensity is defined	$[\text{m}]$
z_{ref}	Reference height of the tower for engineering models	$[\text{m}]$

Operators

$\tilde{\square}$	Dimensionless form of variable $\square$
$\dot{\square}$	Time derivative of $\square$
$\overline{\square}$	Time average of $\square$
$\sigma_{\square}$	Standard deviation of $\square$
$A_{\square}$	Amplitude of $\square$
$d\square$	Differential operator
$\mathcal{F}[\square]$	Generic VIV force function
$h_{\square}$	Magnitude of the Hilbert transform of $\square$
$\mathcal{L}[\square]$	Linear operator for beam bending stiffness
$\max(\square)$	Maximum value of $\square$
$\mathcal{Q}[\square]$	Generic VIV wake-structure coupling function

$r_{\square}$ Growth rate of the amplitude of $\square$

$S_{\square}$ Power spectral density of $\square$

Chapter 1

Introduction

“We were wrong on peak oil. There’s enough to fry us all.”

– George Monbiot, *The Guardian* (2012)

Scientific evidence shows that human activities, particularly through greenhouse gases emissions, have lead to significant global warming, and climate change in general IPCC (2023). An effective way to visualize the global increase in temperature is through the so-called climate stripes, which represent the global temperature anomaly for each year from 1850 to the present. As shown in Figure 1.1, the stripes reveal a clear warming trend over the past century, underscoring the urgency of addressing climate change. International agreements and efforts, such as the Paris Agreement and the IPCC reports, highlight the global commitment to tackling climate change. These initiatives set ambitious targets for reducing greenhouse gas emissions and promote the adoption of renewable energy sources. The Paris Agreement, for instance, aims to limit global warming to well below 2°C above pre-industrial levels, with efforts to limit the temperature increase to 1.5°C. Achieving these goals requires a significant transformation of the global energy system, with renewable energy, including wind energy, playing a crucial role in this transition (IEA, 2021).

As a renewable and abundant resource, wind energy has become a key enabler of the reduction of greenhouse gases. One of the main sources of greenhouse gases is the burning of fossil fuels for electricity production, and it is here that wind energy has become not just a much cleaner alternative, but also a cost-competitive one. This reality has been made possible by the rapid advancement of engineering solutions to a great number of multidisciplinary challenges. This is highlighted by the increasing adoption of wind energy worldwide, as shown in Figure 1.2.

The future development of wind turbines is not only expected to see a large growth in terms of units installed, but also a growth in their size and flexibility, and the realization of innovative solutions such as their installation in floating platforms. And with this new paradigm come new challenges. One such challenge is vortex-induced vibrations (VIV). VIV is a complex aeroelastic phenomenon that occurs when a bluff body interacts with the wind inflow, leading to the generation of vortices in the wake downstream and the subsequent oscillation of the structure. This applies to wind turbine towers, as well as to wind turbine blades facing

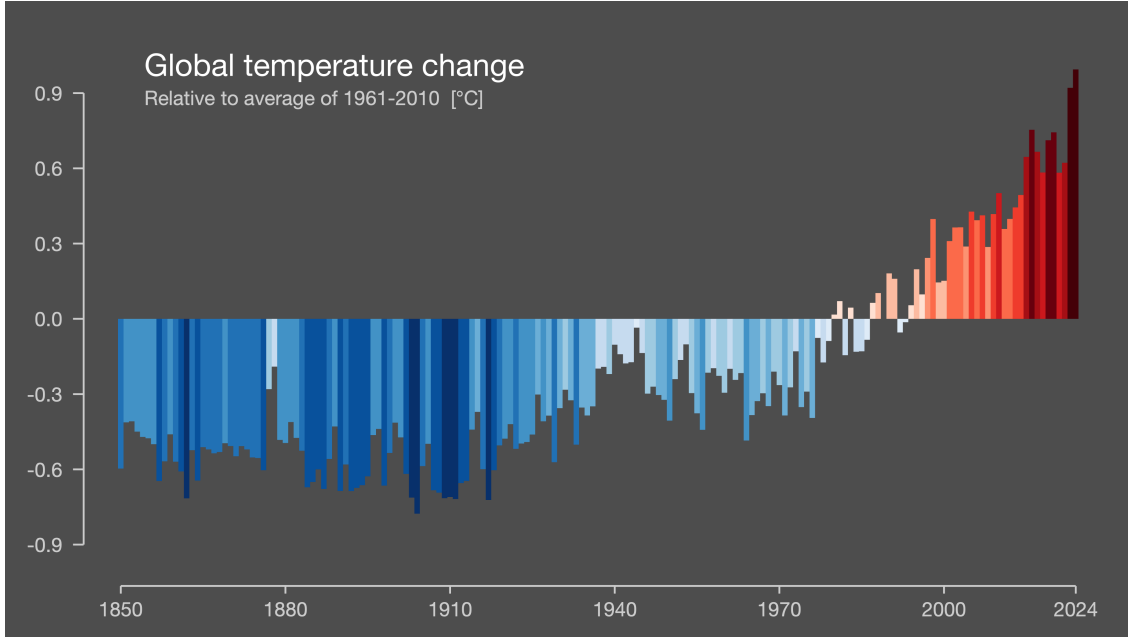


Figure 1.1: Climate stripes representing the global increase in temperature from 1850 to 2024. Reprinted from [#ShowYourStripes](#) (last accessed March 2025), created by Professor Ed Hawkins (University of Reading).

broadside to the wind. The VIV phenomenon can significantly impact the structural integrity of wind turbine towers and blades, the safety of operations, and may become a design limiter on future turbines if not addressed timely and with sufficient accuracy (P. Veers et al., 2023).

Despite decades of research, the prediction and mitigation of VIV remain an open question in engineering, particularly for the next generation of wind turbines with their unprecedented scales, inflow characteristics and design requirements. The difficulties of facing this challenge include a lack of thorough understanding of the aeroelastic phenomena involved as well as a lack of appropriate mitigating solutions, simplified models for application at design and certification levels, sufficient simulation capabilities, and experimental data at both scaled and full-scale. This thesis investigates the modelling of VIV in wind turbine towers and blades, with a focus on advanced techniques that integrate structural dynamics and aerodynamic interactions in the time-domain.

1.1 Background and motivation

Wind turbines are designed to withstand the structural loads that will presumably occur during their lifetime. These loads may be obtained from aeroelastic simulations of the wind turbine, which account for aerodynamic, inertial and gravitational loads (IEC 61400-1 Ed.3.1, 2010). But there is one type of aerodynamic loading which is not commonly considered in these aeroelastic calculations: vortex-induced loading.

Vortex-induced loading, and the ensuing vortex-induced vibrations, have been an issue of industrial concern since at least the mid-20th century (Hartog, 1954). VIV occurs as a

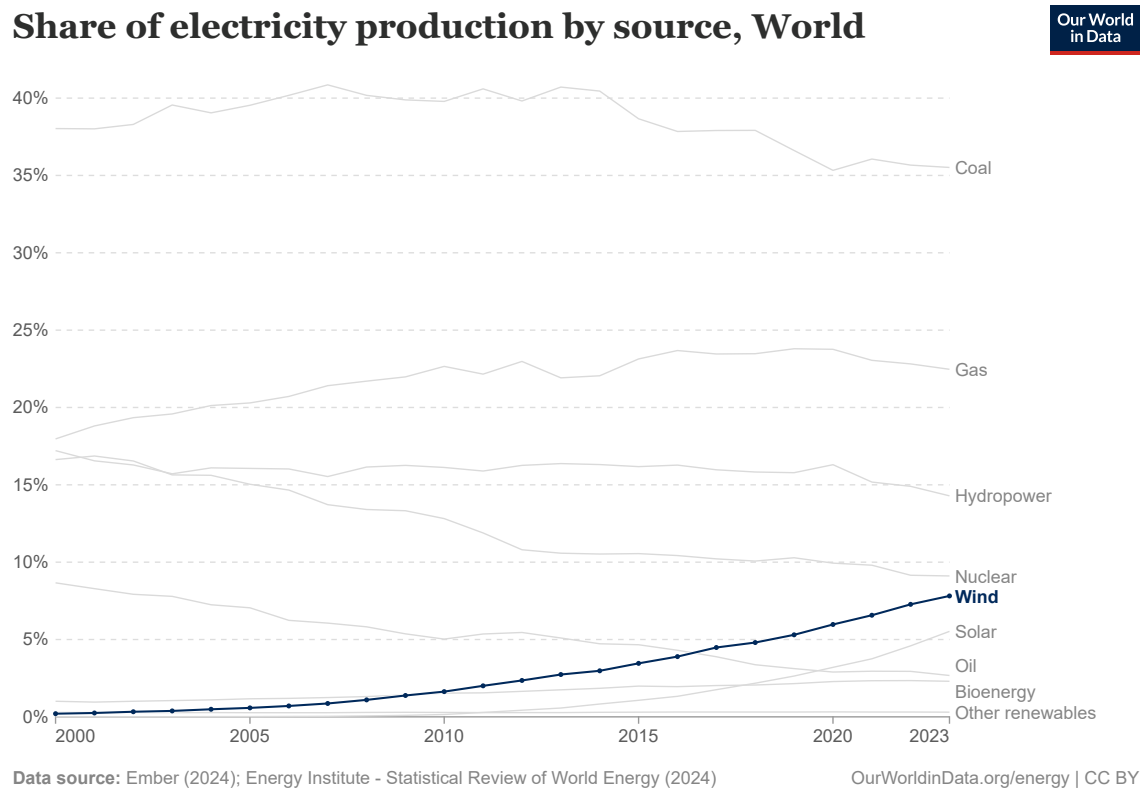


Figure 1.2: Share of world electricity production by source, with wind energy highlighted, from 2000 to 2023. Data source: Ember (2024); Energy Institute - Statistical Review of World Energy (2024) - with major processing by Our World in Data. Retrieved on March 14, 2025 from [Our World in Data](https://OurWorldinData.org).

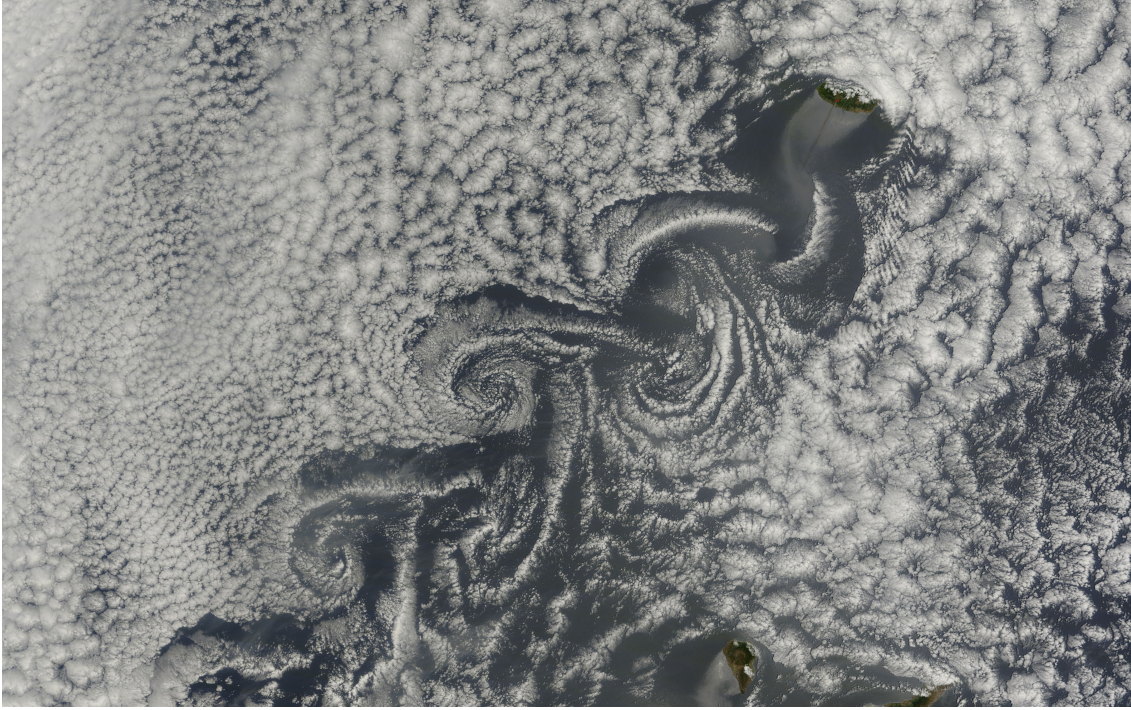


Figure 1.3: Von Kármán vortex streets behind Madeira island, as captured by NASA’s Terra satellite. This image is a cropped version of the original. Source: [NASA Earth Observatory](#) (archived). NASA image by Jeff Schmaltz, LANCE/EOSDIS Rapid Response.

consequence of the unstable character of the wake behind a bluff body. This instability leads to the wake becoming a street of vortices travelling downstream, called a von Kármán vortex street. An example of such a von Kármán street can be seen in the satellite photograph in Figure 1.3, where the island of Madeira is causing the formation of a vortex street in the atmosphere, made visible by the cloud cover. In this image, the wind is flowing from the top right corner of the image towards the bottom left corner. The vortices are shed from the island and travel downstream, forming a pattern of alternating clockwise and anticlockwise rotating vortices. Vortex shedding may occur at many scales; from very large scales, like the island shown in the image, to bridges (Astiz, 2010), buildings (Irwin, 2010; Zheng et al., 2019) and wind turbines (Kurniawati et al., 2025), all the way down to microscale energy-harvesting devices (Y. J. Lee et al., 2019).

In most cases, the vortices are shed behind the body in an alternating and periodic manner, meaning that after a clockwise rotating vortex is shed from one side of the body, an anti-clockwise vortex will be shed from the other side of the body. As a vortex is formed and shed, the pressure field around the body changes. This periodic change in the pressure field results in a likewise periodic force acting on the body. This on its own constitutes a source of loading to the wind turbine which is not easy to calculate in the design process. The intensity and periodicity of the vortex-shedding loading have been extensively studied, and have been found to be affected by numerous parameters. One explanation for the difficulty in modelling vortex-shedding loading is that it is determined by the boundary layer separation

over the body surface, which is itself a very complicated phenomenon to model. The problem is complicated even further by the fact that there can be interaction between the vortex shedding and the motion of the body. This interaction can transform a problem of obtaining an aerodynamic force to a problem of solving for an aeroelastic unsteady equilibrium, adding another layer of complexity.

Despite the complexity of the problem, engineers have developed a number of models to predict the vortex-induced vibrations of structures. Using the knowledge obtained from fundamental research, including data from wind tunnel experiments and field measurements, and some still debated assumptions, two main engineering models for the prediction of VIV are currently used for design purposes of civil structures subject to the wind, including wind turbine towers, as coded into design standards, for instance in the Eurocode EN 1991-1-4:2005+A1 (2010). A description of these approaches will be given in Section 2.2.

Thus, the study of VIV in wind turbine towers is supported by a well-established body of research and engineering practices. However, recent research efforts in both academia and industry have highlighted a previously overlooked scenario: the blades of stopped or idling wind turbines can also experience VIV. Despite being a rare event, as it requires the conjunction of unfavourable environmental and operational conditions or control malfunction, a VIV event can lead to large oscillating loads on the wind turbine blade, consuming a significant amount of its fatigue lifetime. VIV in wind turbine blades differs from classical VIV cases due to their complex three-dimensional geometries and considerable flexibility. Additionally, these structures encounter very high Reynolds number flows, which introduces further challenges in obtaining data to inform and calibrate the modelling of the phenomenon.

Currently, the application of VIV analytic engineering models is mandatory for the calculation of fatigue life and extreme loading of wind turbine towers, as per the tower design standard IEC 61400-6 Ed.1 (2020). To the best of the author's knowledge, no specific consideration of VIV is required for wind turbine blade design. For the design of the complete wind turbine, time-marching aeroelastic simulations are used to calculate the loads on all wind turbine components. Some of these aeroelastic simulations could be enhanced by incorporating vortex-induced loading, as it can significantly affect the overall dynamic behaviour of the wind turbine. For example, the design standard IEC 61400-1 Ed.3.1 (2010) mandates the evaluation of the wind turbine's response under scenarios such as yaw system failure, which can induce VIV in the blades due to potential misalignment between the inflow and the blades. Additionally, the standard requires consideration of transport, assembly, maintenance, and repair conditions; situations where both the blades and the tower are particularly susceptible to VIV. Other wind turbine components that may induce VIV include the nacelle, and, for offshore wind turbines, underwater components such as monopiles, spar-buoys, mooring lines and tension legs.

The present thesis aims to address the challenges posed by VIV in wind turbines by investigating the modelling of VIV in wind turbine towers and blades. Two main approaches are considered: the application of well-known engineering models to wind turbine towers, and the execution of high-fidelity simulations of the fluid-structure interaction of wind turbine airfoil sections. The results obtained with these two approaches are then compared and contrasted to assess the applicability of the engineering models to the wind turbine blade

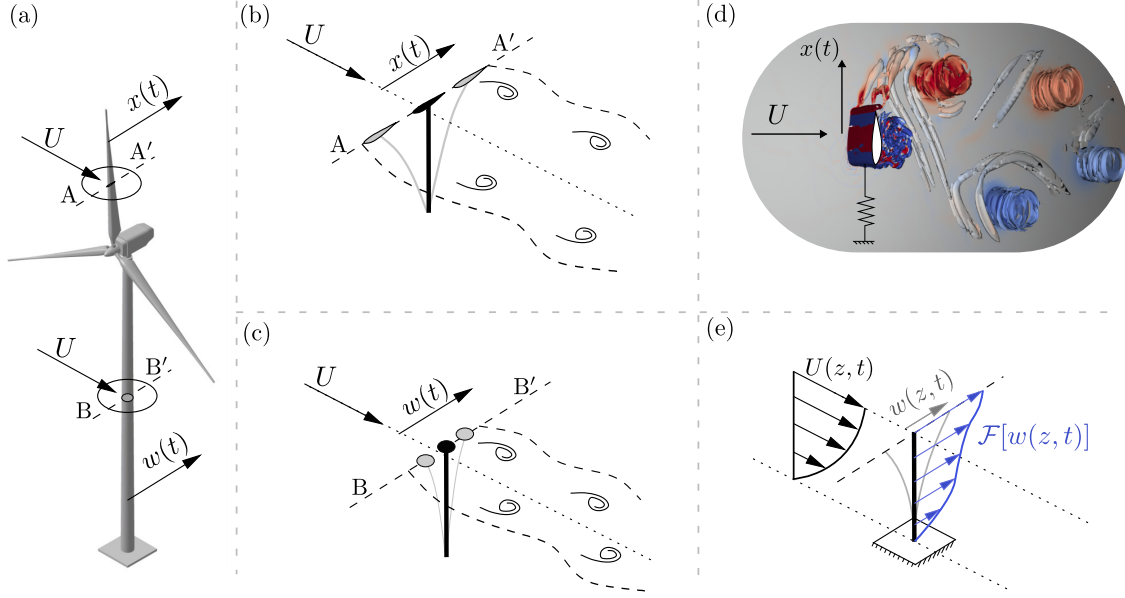


Figure 1.4: Schematic of the vortex-induced vibration problem in wind turbines and its treatment in the present thesis. (a) Depiction of a wind turbine, with inflow coming from one side at velocity U . Two arbitrary spanwise sections are marked, one at the blade (A-A'), and one at the tower (B-B'). The vortex-induced motion is depicted as $x(t)$ for the blade and $w(t)$ for the tower. (b) Detail of the vortex-induced vibration of the blade at station A-A', showing the blade motion and the vortex shedding in the wake. The black vertical line represents the undeformed blade position, while the grey curves represent the blade deflection. (c) Detail of the vortex-induced vibration of the tower at station B-B', showing the tower motion and the vortex shedding in the wake. The black vertical line represents the undeformed tower position, while the grey curves represent the tower deflection. (d) Example of the high-fidelity simulations performed of the flow around a blade section, with a schematic representation of the one degree-of-freedom elastic system coupled to the fluid dynamics solver. (e) Schematic representation of the application of an engineering model to the tower, where a sheared and turbulent inflow, $U(z, t)$, results in a vortex-induced force, represented by a nonlinear function of the motion, $\mathcal{F}[w(z, t)]$, which is a function of height, z .

case. A schematic of the vortex-induced vibration problem in wind turbines, along with both treatments considered in the present thesis, is shown in Figure 1.4. In panel (a), the schematic depicts a wind turbine with a high yaw misalignment and a stopped or idling rotor. In this situation, some blade sections will experience very large angles of attack, and thus act as a bluff body. The depicted vortex shedding behind arbitrary sections of the (b) blade and (c) tower may induce vibrations in these components. To analyse these vibrations, in panel (d) an example is shown of the result of a high-fidelity simulation performed on the blade section, where the structural model is reduced to a mass-spring system. In contrast, panel (e) represents the application of an engineering model applied to the tower, where the structure is modelled as an elastic cantilevered beam, and the vortex-induced force is modelled as a nonlinear function of the tower motion, $\mathcal{F}[w(z, t)]$, using different approaches.

Bridging the gap between existing knowledge on vortex-induced vibrations and the unique

challenges posed by wind turbine applications will lead to more cost-effective designs and safer operations on wind turbines, contributing to the global transition to sustainable energy sources.

1.2 Research question and objectives

The primary motivation of this thesis, as well as many related research efforts, is to address the following ambitious question: How can we improve the time-domain modelling of vortex-induced vibrations (VIV) for wind turbine towers and blades?

Answering this question requires overcoming two significant challenges. First, the phenomenon of VIV in wind turbines has been scarcely explored in the literature, with limited full-scale, experimental and numerical data available. Second, while VIV has been extensively studied for a wide variety of engineering structures, energy harvesting devices, and even biological systems (e.g. for fish swimming dynamics and coral feeding strategies), its modelling remains a challenging problem in all areas.

This thesis seeks to improve our understanding of VIV time-domain modelling and its application to wind turbines, by addressing three key questions:

- What are the differences in the response predictions obtained with two families of well-established VIV engineering models, when applied in the time-domain to wind turbine towers?
- What are the differences in the response predictions obtained with commonly used high-fidelity VIV simulation strategies, when applied to wind turbine airfoil sections?
- How well-suited are the different time-domain VIV engineering models to the wind turbine blade case, based on insights from high-fidelity simulation results?

The first two questions involve two distinct modelling approaches (engineering models and high-fidelity simulations) and two different geometries (circular cross-section towers and airfoil sections), and are linked via the analysis performed to answer the third question, where the results obtained with the engineering models and high-fidelity simulations are compared and contrasted.

The reason for the different approaches chosen for the tower and blade cases is that for the wind turbine tower case this research builds on a solid foundation of prior work. Cylindrical structures with circular cross-sections have been analysed extensively, with an abundance of available experimental and full-scale data, leading to the development of practical engineering models for predicting VIV responses (although they are still not quite robust in many cases). These models, while not originally designed for time-domain analyses of wind turbine towers, offer a strong starting point for assessing their applicability to this specific case. Thus, the first objective of this thesis is to critically evaluate the results obtained when applying these well-known engineering models to the prediction of the time-domain VIV response of wind turbine towers. In contrast, the case of wind turbine blades presents a far less developed field. Unlike circular cross-sections, there are no well-established engineering models for the VIV of airfoil-shaped structures. Consequently, the second objective of this thesis is to

generate and analyse new data on the VIV response of an airfoil section as a means towards developing time-domain engineering models to predict wind turbine blade VIV. Finally, the combination of these two objectives leads to the third objective: to bridge the gap between the well-established understanding of VIV in cylindrical structures and the emerging challenges of modelling wind turbine blades VIV. By incorporating insights from both geometries and both modelling approaches, this research aims to contribute to a clearer and richer framework for understanding and predicting time-domain VIV in wind turbines.

1.3 Scope and organization of the thesis

After this introduction, the thesis is divided into five further chapters.

Chapter 2 deals with the state of the art of research in vortex shedding and VIV, as relevant for wind turbine applications. This includes a summary of insights gained of the relevant physical phenomena involved, the identification of knowledge gaps, and a review on its mathematical modelling and numerical simulation.

Chapter 3 presents the development of three types of engineering models for the prediction of the VIV response of wind turbine towers. One of these models is validated against full-scale measurements. Then, the three models are applied to the NREL 5-MW wind turbine tower, and the differences in the response predictions obtained with these models are compared and discussed.

Chapter 4 presents high-fidelity simulations of the vortex shedding and VIV of wind turbine airfoil sections at a 90° angle of attack. Firstly, the numerical setup is described, including descriptions of the computational fluid dynamics (CFD) solver and the structural dynamics solver. The setup is verified with a comparison to previously published numerical data and a mesh sensitivity analysis. Then, the results are presented and discussed, including an analysis of the vortex shedding behind static airfoils, and of the VIV response of free to move airfoils. The discussion is strengthened by means of flow field visualizations.

Chapter 5 serves as a global discussion on the mathematical modelling and simulation of VIV for wind turbines. It discusses the differences between the response of wind turbine towers and blades, as discerned from the previous chapters, assessing the suitability of the engineering models for the wind turbine blade case. This chapter also proposes potential parametrizations and modifications to these models, and assesses the role of high-fidelity simulations in supporting the development of improved engineering models.

Finally, Chapter 6 presents the global conclusions of the thesis, summarizing the main findings and contributions, and suggesting future research directions.

Chapter 2

State of the art

In this chapter, the state of the art in the field of vortex shedding and vortex-induced vibrations (VIV) is reviewed, with a special focus on wind turbine applications. Section 2.1 provides an overview of the physical phenomena of vortex shedding and VIV, including their basic concepts and mechanisms, with reference to experimental and numerical observations. The second section (Section 2.2) discusses the mathematical characterization of VIV for engineering purposes, dividing the models into different categories, summarizing canonical works and recent contributions, and commenting on examples of their application to wind turbines. Finally, the third section (Section 2.3) presents the current state of research in vortex shedding and VIV by means of numerical simulations based on computational fluid dynamics (CFD).

2.1 Physical phenomena

Vortex shedding is a fluid dynamic phenomenon where alternating low-pressure vortices are created behind a bluff body, thereby causing periodic forces on the body. This phenomenon is particularly relevant in engineering applications such as wind turbines, where these alternating forces can produce oscillations which lead to structural fatigue or even failure. The coupling between vortex shedding and structural response is known as vortex-induced vibrations.

The study of vortex shedding and VIV relies on observations from wind tunnel tests, field measurements, and numerical simulations. Wind tunnel tests provide controlled environments to study the flow patterns and forces on scaled models. Field studies offer insights into the much more complex real-world conditions encountered by full-scale structures. Numerical simulations, such as computational fluid dynamics (CFD), allow for detailed analysis of the flow and structural response under various conditions.

Several comprehensive reviews on vortex shedding and VIV have been published, providing valuable insights into these complex phenomena. For instance, Sarpkaya (1979) and Bearman (1984) offer early overviews of fundamental aspects and results on the circular cylinder case. More recent reviews, such as Sarpkaya (2004) and Williamson and Govardhan (2004), provide further insight and include numerical simulations in their reviews to complement experimental

results. A recent review focused on vortex-induced motions of floating offshore wind turbines is given in Yin et al. (2022), including research in VIV at very high Reynolds numbers.

Certainly, there are also a number of books that provide a comprehensive overview of the field of fluid-structure interaction and on the effects of wind on structures, which devote specific sections to the vortex shedding and VIV phenomena. Some of the most notable books in this area include Blevins (2001), Dyrbye and Hansen (1996), Païdoussis et al. (2010), and Sumer and Fredsøe (2006). These books are essential references for researchers and engineers working in the field of fluid-structure interaction and wind engineering.

To provide a comprehensive overview of the physical phenomena of vortex shedding and VIV, the following sections will delve into the basic characteristics of these phenomena, highlighting their fundamental aspects and the knowledge gaps that remain to be addressed.

2.1.1 Basic phenomena - vortex shedding

The purpose of this section is to introduce and explain the fundamental phenomenon of vortex shedding behind static bodies, emphasizing its characteristics and governing parameters. This includes discussing the historical foundations of vortex shedding, the physical mechanisms behind it, the influence of flow conditions, and advanced observations. By understanding these aspects, we can better appreciate the complexities of vortex shedding and its implications for engineering applications, particularly in the context of wind turbines.

The systematic study of vortex shedding commenced as early as the 19th century. Strouhal (1878) studied the sound generated by wires in airflow due to vortex shedding, and discovered a relationship between the frequency of vortex shedding (f), the flow velocity (U), and the wire diameter (D). This relationship later became known as the Strouhal number ($St = fL/D$), a dimensionless parameter used to characterize the shedding frequency. It was thus established that vortex shedding occurs at specific and predictable frequencies that depend on geometry and flow conditions. Strouhal's discovery laid the groundwork for understanding vortex shedding, but it was von Kármán (v. Kármán, 1911) who provided a theoretical explanation. Von Kármán's stability analysis showed that the alternating vortices form a stable pattern later known as the Kármán vortex street. This pattern arises because the fluid cannot smoothly flow around the bluff body, leading to an unstable shear layer in the wake, which rolls up periodically in a stable series of vortex formation and shedding events.

It was soon found that the characteristics of this vortex street are highly dependent on the flow conditions, particularly the Reynolds number, as notably characterized by Roshko (1961) and Schewe (1983), who performed experiments on circular cylinders up to very high (transcritical) Reynolds numbers, identifying distinct regimes of wake behaviour as the Reynolds number increased. Another insightful set of experiments was made on the vortex shedding behind inclined flat plates by Fage and Johansen (1927), later complemented by the effect of inflow turbulence studied by Bearman (1971). Additionally, Achenbach and Heinecke (1981) investigated the very significant influence of the cylinder surface roughness and three-dimensional effects on the vortex shedding characteristics. These insights are crucial for understanding the complexities of vortex shedding and their implications for engineering applications.

The complexity in vortex shedding behaviour poses a challenge for the accurate prediction of the forces acting on structures. The inherent connection between vortex shedding and structural response leads to the phenomenon of vortex-induced vibrations (VIV). In the following section, the basic characteristics of VIV will be described, exploring how the periodic forces generated by vortex shedding interact with structures, leading to potentially significant oscillations and therefore structural damage.

2.1.2 Basic phenomena - vortex-induced vibrations (VIV)

The VIV phenomenon of bluff bodies is well known and has been studied extensively both experimentally and numerically. VIV may be described as a resonance with nonlinear feedback phenomenon, where the structural motion affects the vortex shedding and vice versa (de Langre, 2006). A critical aspect of this feedback is that it can result in a frequency lock-in, if the unperturbed vortex-shedding frequency is close to the structural natural frequency (Griffin et al., 1973). Moreover, the VIV response is self-limiting, meaning that the amplitude of the structural motion reaches a maximum value due to the nonlinear interaction between the fluid forces and the structure (Blevins, 2001). This self-limiting behaviour is a key characteristic of VIV, as it prevents the structure from undergoing unbounded oscillations. Another significant effect of VIV is an increase in the mean drag experienced by the bluff body (Bishop et al., 1964b). VIV is typically characterized by the maximum amplitudes reached by the body, as well as by the lock-in range, i.e. the range of inflow velocities for which large vibrations build up, as a consequence of the synchronization between the vortex-shedding frequency and the structural natural frequency.

The most commonly analysed geometry in VIV studies is the circular cylinder. This geometry has been widely analysed experimentally from very early on. The first detailed account of the VIV response obtained experimentally is given by Feng (1968), who described the influence of the structural damping on several characteristics of the response, such as the phase angle between force and motion, the position within the lock-in range of the maximum amplitude, or the spanwise correlation length. In Jauvtis and Williamson (2003) and Cagney and Balabani (2014) experimental evidence suggested that constraining the structural motion of the cylinder to its transverse direction, rather than allowing for transverse and in-line motion, has very little effect on the transverse VIV response and the wake topology. Full-scale studies of VIV of towers with circular sections are also common (Christensen & Askegaard, 1978; Galemann & Ruscheweyh, 1992), including recent efforts to characterize the vortex-shedding loads on wind turbine towers (Kurniawati et al., 2024, 2025).

2.1.3 Knowledge gaps

The study of vortex shedding and vortex-induced vibrations (VIV) has advanced significantly, yet several knowledge gaps remain that hinder the complete understanding and accurate prediction of these phenomena. Furthermore, the research on VIV applied to wind turbine towers, and especially blades, is relatively novel. Tailored experimental and numerical studies are needed to address the specific challenges posed by these structures. Addressing these knowledge gaps is crucial for improving the design and safe operation of wind turbines.

One of the primary gaps is the limited amount of analyses at the very high Reynolds numbers which are relevant to full-scale applications, as experimental setups start to become impractical. Most studies have focused on lower Reynolds numbers, leaving significant gaps in understanding the behaviour of vortex shedding and VIV at the higher Reynolds numbers encountered by wind turbines.

Another area that requires further research is the interaction between turbulence and vortex shedding. The detailed effects of turbulence length scales and turbulence intensity on vortex shedding are not well understood, and more experimental and numerical studies are needed to elucidate these interactions. This understanding is important for wind turbine applications, which are subjected to the atmospheric boundary layer.

Research on VIV for non-cylindrical shapes, such as airfoil sections of wind turbine blades, is also limited. Most studies have focused on circular cylinders, and the unique characteristics of VIV in non-cylindrical shapes need to be explored to develop accurate predictive models for wind turbine blades.

The interaction between the tower and blades of a wind turbine, both aerodynamically and structurally, adds another layer of complexity to the VIV phenomenon. This interaction can be particularly significant in downwind turbines, where the rotor operates in the wake of the tower.

In summary, addressing these knowledge gaps requires robust theoretical and computational tools, as well as targeted experimental studies. The next sections will review the historical development and current state of mathematical models and numerical simulations for VIV, discussing some of the efforts made to overcome these difficulties, which are essential for advancing our understanding and prediction capabilities.

2.2 Mathematical modelling

There are several families of VIV models used in engineering practice, each with its own assumptions and simplifications. These models range from simple analytical expressions to more complex time-dependent differential equations, each offering different levels of accuracy, complexity and computational cost. The choice of model depends on the specific application and the desired level of detail in the prediction of VIV, but also on the quantity and quality of the available data to calibrate the models.

Engineering models of VIV can be broadly classified into three categories, following the nomenclature given by Païdoussis et al. (2010): Type A or forced system models, Type B or fluidelastic system models and Type C or coupled system models. A schematic representation of such models can be seen in Figure 2.1, where the interaction between the fluid force and the structural dynamics is depicted. Three panels are shown, corresponding to each of the VIV model types. In all panels, the structural motion is denoted by $w(t)$. For Type A models, the fluid forces acting on the structure are modelled as a time-dependent force $F(t)$. For Type B models, the fluid force is a nonlinear function of the structural response. The nonlinear function of the structural response is here denoted as $\mathcal{F}[w(t), t]$, but note that it can be a function of the time derivatives of $w(t)$. For Type C models, a hidden variable related to the

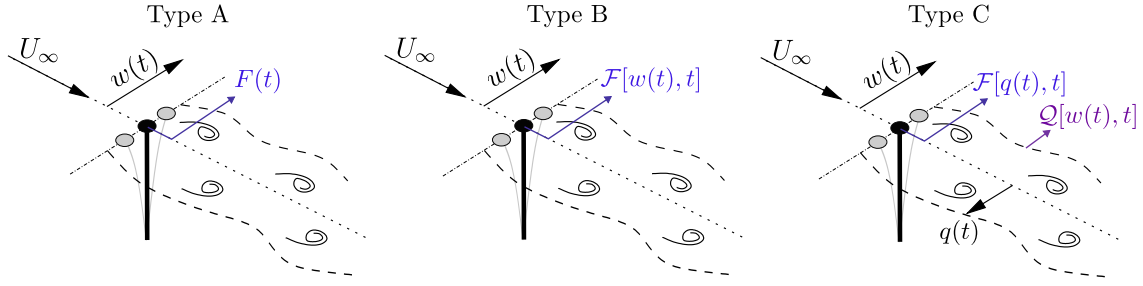


Figure 2.1: Schematic representation of the three main types of VIV mathematical models. The structural motion is denoted $w(t)$. (a) In the Type A model (forced system), the fluid forces acting on the structure are modelled as a time-dependent force $F(t)$. (b) In the Type B model (fluidelastic system), the fluid force is a nonlinear function of the structural response, $\mathcal{F}[w(t), t]$. (c) In the Type C model (coupled system), a fluid variable, $q(t)$, is also modelled, such that the fluid force on the structure is a nonlinear function of this variable, $\mathcal{F}[q(t), t]$, and the fluid variable is coupled to the structural motion by a nonlinear term, $\mathcal{Q}[w(t), t]$.

wake dynamics, $q(t)$, is also modelled. In this case, the fluid force acting on the structure is a nonlinear function of this fluid variable, $\mathcal{F}[q(t), t]$, and the fluid variable is coupled to the structural motion by a nonlinear term, $\mathcal{Q}[w(t), t]$. Again, the coupling terms between the structure and the fluid may be functions of the time derivatives of $q(t)$ and $w(t)$. Thus, the models range in complexity, with Type A models being the simplest and Type C models being the most complex.

Type A models can be computed without solving for the structural motion, which makes them cheap and simple to compute. Nonetheless, this lack of coupling indicates that they may only be suitable for static or highly damped structures, as they do not reproduce the self-exciting and self-limiting behaviour of VIV. This kind of model may be used for preliminary design, and it may also serve as basis for other Type B and Type C models. Some approaches to characterize the load in Type A models are the harmonic load model, the wake oscillator model and the stochastic load model. The harmonic load model approximates the lift force as a purely sinusoidal function. This approach provides conservative estimates of the response of lightly damped structures and allows for very simple analytical solutions (Blevins, 2001).

The wake oscillator model represents the lift force through a differential equation, often as a van der Pol or Rayleigh oscillator. This equation is associated with the dynamics of the wake, typically involving a hidden variable (Païdoussis et al., 2010), or a physical characteristic such as the rotation of the wake (Tamura, 2020). The properties of wake oscillator models, including three-dimensional effects, have been extensively studied by Facchinetti et al. (2002). The resulting load from this model is quasi-harmonic and captures some of the nonlinear characteristics of vortex shedding.

The stochastic load model characterizes the lift force as a stochastic process, reflecting observations from wind tunnel experiments where vortex-induced forces exhibit stochastic behaviour at certain Reynolds numbers (Schewe, 1983). Additionally, turbulent structures in the flow can further contribute to the stochastic nature of the forces. A notable example is the model proposed by B. J. Vickery and Clark (1972), which defines the power spectral density

function of the lift force based on experimental data. This model is particularly suitable for capturing the randomness introduced by turbulence.

Type B models are more complex than Type A models, as they incorporate the dependency of the vortex-induced force on the structural motion. This interaction allows Type B models to capture the self-exciting and self-limiting characteristics of VIV, making them more accurate for predicting the structural response.

One Type B model is the correlation length method. It is used in several standards such as the Eurocode (EN 1991-1-4:2005+A1, 2010), where it is presented as “Approach 1”. A detailed description of this model can be found in Ruscheweyh (1990). The correlation length model is based on the harmonic load formulation, enhanced with a semi-empirical correction for the vortex-induced load, based on the notion that correlation of the forces along the structure increases with motion amplitude. While this modified harmonic load model provides a practical approach for estimating the VIV response, it essentially functions as a resonance model, and, as a result, may yield overly conservative predictions for structures with low damping, as it does not account for the self-limiting behaviour of VIV. Another adaptation of the harmonic model is that by Blevins and Burton (1976), where the magnitude of the lift coefficient is expressed as a quadratic function of the amplitude of motion.

Another approach to Type B models is to express the vortex-induced loading as nonlinear aerodynamic damping force. An early example of such a model is that of B. J. Vickery and Basu (1983), who proposed a model for the lift force that includes a linear negative damping term and a cubic positive damping term. This model is based on the assumption that the lift force is in phase with the structure’s velocity for small amplitudes, but it becomes nonlinear for large amplitudes, leading to a self-limiting effect. This model has been widely used in the literature and is the basis of the “Approach 2” model in EN 1991-1-4:2005+A1 (2010). Nevertheless, the particular form of the nonlinearity is a matter of debate. Wind tunnel experiments on structures forced to vibrate allow researchers to obtain an equivalent damping coefficient, which is a nonlinear function of the vibration amplitude and frequency. This has led to several semi-empirical formulations of the aerodynamic damping as a function of the structural vibration, such as those presented by Cheung and Melbourne (1983a) and Scruton and Flint (1964) and more recently by Lupi et al. (2017b, 2018). This last model has been applied with some success (Lupi et al., 2017a, 2021) to predict the VIV response of circular cylinders and full-scale chimneys.

A typical approach to Type B models is to combine the nonlinear motion-dependent force with a stochastic load model (Type A), as proposed by Blevins and Burton (1976) and B. J. Vickery and Basu (1983). These formulations allow for the development of analytical solutions for the VIV response (Guo et al., 2022; Pagnini et al., 2020; B. J. Vickery & Basu, 1983). For engineering purposes, closed-form or analytic solutions are often sought. This type of solution can help in gaining understanding of the studied phenomenon by making the functional dependencies easier to grasp, and they also allow for quicker application of the model, making them more practical for parametric dependency analyses, optimization and routine calculations. Nevertheless, this comes at the cost of risking accuracy due to assumptions and simplifications made, and the loss of the detailed time-evolution of the model solution.

Type C models are the most complex of the three types, as they involve solving for an additional flow variable with its own dynamics. This also makes them potentially the most accurate, as they can account for the full coupling between the fluid and structural dynamics. This coupling can lead to complex interactions between the fluid and structure, resulting in a rich variety of behaviours.

Several Type C models have been proposed based on observations of the wake behaviour (Hartlen & Currie, 1970; Iwan & Blevins, 1974), or on physical considerations (Tamura, 1981). These models are typically based on the notion of a wake oscillator system, coupled to the structural dynamic system. In these wake oscillator models, the vortex-induced vibration is a result of the interaction between the structural modes and the wake modes, in what has been identified as coupled-mode flutter by de Langre (2006). This mode coupling is able to capture the self-exciting and self-limiting characteristics of VIV, as well as the synchronization, or lock-in, between the vortex-shedding frequency and the structural natural frequency. An insightful review on wake oscillator models has been recently done by Tamura (2020).

A different methodology has been applied to the wind turbine tower case by Arnold et al. (2024), where the vortex-induced force is modelled as a harmonic oscillator, coupled to the tower motion through semi-empirical dependencies for the force amplitude and instantaneous phase. This model lies between the Type B and Type C classification. While it does not involve a differential equation for a hidden flow variable, obtaining the oscillator phase angle requires a time-marching phase controller and time integration, making it fundamentally distinct from typical Type B nonlinear damping models. Additionally, this model cannot be simplified into a closed-form solution.

The application of VIV models to wind turbines is a relatively novel field, with most of the research focused on the VIV response of wind turbine towers. Table 2.2 provides an overview of the published research on the application of VIV models to wind turbine towers and blades or airfoils. The stochastic force from the Type A model of B. J. Vickery and Clark (1972) has been considered to predict the response of a wind turbine tower by Huo and Tong (2020). A novel model for the VIV of wind turbine tower and blades was proposed by Manolas et al. (2022), based on a Type B model. This model is a combination of the “Approach 1” and “Approach 2” models in EN 1991-1-4:2005+A1 (2010), where the stochastic load in “Approach 2” is substituted by the harmonic load in “Approach 1”. This allows for greater flexibility regarding its application to different structural mode shapes, including coupled modes of the complete wind turbine. The canonical Type B model by B. J. Vickery and Basu (1983) has been applied to predict the VIV response of a wind turbine tower at different stages of installation by Nunez-Casado et al. (2017), and it was employed by Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al. (2024) to compare the predicted VIV response of a wind turbine tower against that obtained with a wake oscillator Type C model. The alternative analytical formulation developed by Lupi et al. (2017a) has been extended and applied to the wind turbine tower case by Livanos (2018) and Livanos et al. (2024). In addition to the aforementioned self-adaptive harmonic oscillator model by Arnold et al. (2024) for wind turbine towers, C. Chen et al. (2023, 2025) recently applied a Type C van der Pol oscillator model to investigate the VIV of wind turbine towers, and contrasted their findings with on-site experimental results. To the best of the author’s knowledge, the only application of a

Type C model to the transverse VIV of wind turbine blades is presented in Pellegrino (2015), where the model was employed to predict the VIV response of a 2D wind turbine airfoil section and compared with high-fidelity numerical simulations. Notably, Type C models have also been applied to other airfoil VIV cases, particularly in turbomachinery, for example by Hoskoti et al. (2020) and Wang et al. (2016a, 2016b). To the best of the author’s knowledge, the present thesis is the only example of a one-to-one comparison of a Type B and a Type C model for wind turbine VIV. These studies demonstrate the potential of VIV models to predict the response of wind turbine towers and blades, providing valuable insights into the behaviour of these structures under vortex-induced vibrations. However, further research is needed to reduce uncertainties, and to refine these models for a wide range of wind turbine configurations and operating conditions, such as inclined blades and turbulence considerations.

VIV model	WT tower	WT blade/airfoil
Type A	Huo and Tong (2020)	–
Type B	Manolas et al. (2022) Nunez-Casado et al. (2017) Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al. (2024) Livanos (2018) Livanos et al. (2024) Present thesis	Manolas et al. (2022)
Type C	Arnold et al. (2024) C. Chen et al. (2023) C. Chen et al. (2025) Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al. (2024) Present thesis	Pellegrino (2015)

Table 2.2: Published research on the application of VIV models to wind turbine (WT) towers and blades or airfoils.

One of the main challenges in VIV mathematical modelling is the accurate representation of the fluid forces acting on the structure. The lift forces generated by vortex shedding are highly nonlinear and depend on a wide range of parameters (Sarpkaya, 2004), including the geometry of the structure, the structural properties, and the flow conditions, as evidenced by the fundamental dependency of the vortex-shedding topology on the Reynolds number regime (Schewe, 1983; Sumer & Fredsøe, 2006). Developing accurate models for these forces is essential for predicting the occurrence of VIV and the response of the wind turbine components, and for designing mitigating solutions.

2.3 Numerical Simulations

Recent computational advancements have favoured the use of high-fidelity computational fluid dynamics (CFD) models, coupled with structural models, to characterize the fluid-structure interaction (FSI) associated with VIV. The still large computational cost of this approach limits the extent to which realistic scenarios can be simulated (e.g. full wind turbine blades and towers immersed in the atmospheric boundary layer at high Reynolds numbers) and also limits the VIV characterization throughout the extensive parameter space.

The flow behind bluff bodies are canonical cases often used for CFD code validation, development of new techniques and benchmarking (Norberg, 2003), for example in the Benchmark on the Aerodynamics of a Rectangular 5:1 Cylinder (Bruno et al., 2014) or the Bluff Body Benchmark initiative (Breitkopf et al., 2022; Nieto et al., 2024).

The prediction of vortex-induced vibrations requires solving the fluid-structure interaction, not solely CFD. The interaction between the fluid and the structure necessitates either a coupled simulation approach, integrating both CFD and structural dynamics, or a forced vibration approach. Forced vibration cases are simpler to implement as they involve one-way coupling. In these scenarios, the structure's motion is prescribed, and the fluid flow is computed based on this motion. This simplification reduces computational complexity and allows for a more straightforward analysis of the fluid dynamics.

CFD simulations allow for a detailed analysis of the wake structure. By using CFD, researchers can visualize and describe the complex flow patterns in the wake of a body, providing insights that are difficult to obtain through experimental methods alone. Additionally, CFD may reduce the need for expensive and time-consuming physical experiments.

Simulating vortex shedding in CFD is still challenging due to, among other things, the sensitivity of the results to boundary layer separation and domain boundary conditions. Accurate simulation of vortex shedding requires precise modelling of the boundary layer, which can significantly affect the flow separation and vortex formation. Additionally, the choice of boundary conditions can greatly influence the simulation results, making it a critical aspect of CFD modelling. For example, the effect of the spanwise domain size on the vortex shedding behind a flatback wind turbine airfoil has been analysed in Manolesos and Papadakis (2021).

For the wind turbine case, large Reynolds numbers and atmospheric turbulence present additional challenges. Wind turbines operate in high Reynolds number regimes, and immersed in the atmospheric boundary layer, where the flow is highly turbulent and complex. Simulating these conditions realistically would require advanced turbulence models and significant computational resources. For example, the effect of free-stream turbulence on the vortex shedding of a cylinder has been simulated in Tutar et al. (2006), for a Reynolds number $Re = 1.4 \times 10^5$.

Apart from the ubiquitous CFD simulations of circular cylinder, other geometries are typically investigated, and the insights obtained from their study may be applied to the wind turbine blade case. One such geometry are prisms with square cross-sections, both with a face normal to the flow and rotated 45° . FSI simulations have shown that the diamond geometry is

susceptible to VIV (Leontini & Thompson, 2013; Sourav et al., 2020) but always for relatively small Reynolds numbers. It was found by Leontini and Thompson (2013) that the VIV response is very sensitive to the sharpness of the corners pointing in the cross-flow direction. Additionally, experimental results showing the VIV response of square prisms with several different orientations can be found for example in Nemes et al. (2012) and Zhao et al. (2018), which were also limited to small Reynolds numbers.

Another extensively studied geometry is that of a flat plate at a 90° angle of attack. However, the present author has not found VIV analyses of the flat plate undergoing linear displacements. Numerical analyses of the vortex shedding behind a static flat plate provide valuable information about the flow topology in the wake. In this regard, in Najjar and Balachandar (1998) and Hemmati et al. (2016) different intermittent vortex-shedding regimes were identified. Experimental results of vortex-shedding characteristics behind a static flat plate can be found in the classical work by Fage and Johansen (1927), and a more recent review can be found in Teimourian et al. (2018).

Analyses of the VIV response of other geometries include the experimental study performed by Nakamura and Hirata (1991) for a prism of a rectangular cross-section, where the lower end of the lock-in region is said to be associated with low-speed galloping behaviour, where galloping refers to the canonical cross-flow aeroelastic instability (not related to vortex shedding), with low-speed galloping being a special case of galloping occurring at inflow wind speeds below the VIV critical velocity and potentially interacting with it. Lo et al. (2023) experimentally obtained the VIV response of a prism with a thin elliptical cross-shape, and signs of combined VIV and galloping were identified.

Recent contributions can also be found focused on the problem of vortex shedding behind wind turbine towers (Mousavi & Kamali, 2020), and its VIV response (Viré et al., 2020; Vlastos et al., 2024). The effect on blade loads of the vortex shedding emanating from the tower of a downwind wind turbine has been simulated in Zahle et al. (2009).

The airfoil geometry at a 90° angle of attack is not one of the shapes canonically studied in the VIV literature. One of the first VIV studies that focused on airfoils is that of Skrzypiński et al. (2014), where the VIV response of a wind turbine airfoil at an angle of attack of 90° and a high Reynolds number was obtained with FSI simulations and analysed, coupling CFD simulations to structural motion simulations with 1 degree of freedom, considering chordwise displacements, and 3 degrees of freedom, considering chordwise and flatwise displacements and torsional rotation. The FSI simulations were performed with the unsteady Reynolds-averaged Navier-Stokes (URANS) turbulence-modelling approach on a two-dimensional (2D) airfoil model and with the delayed detached-eddy simulation (DDES) turbulence-modelling approach on a three-dimensional (3D) airfoil extruding one chord length. As a main conclusion, the authors found negative aerodynamic damping with both approaches at certain wind speed intervals, indicating the possibility of lock-in behaviour in airfoils. For both cases, it was found that, when resonance occurred, the edgewise vibrations dominated over the flatwise and torsional vibrations. For the DDES case, the lock-in range obtained did not agree well with the Strouhal number obtained from a static simulation. Lian et al. (2023b) extended the 1 degree of freedom URANS 2D airfoil FSI analysis of Skrzypiński et al. (2014) to study the effect of the structural damping value on the VIV response. More recently, the same authors

continued this line of work by including results from 3D detached-eddy simulation (DES) results of the moving airfoil, with different damping values (Lian et al., 2023a). A limited number of experimental results of the VIV response of airfoils have also been performed, such as those by Benner et al. (2019) and Iyer (2023), but at Reynolds numbers much smaller than those typical of wind turbine airfoil flows.

The FSI response of a full wind turbine blade has been widely investigated through numerical simulations (Dose et al., 2016; Grinderslev et al., 2023; Heinz et al., 2016; Horcas, Sørensen, et al., 2022), showing that the full blade is also susceptible to VIV, at least under constant inflow conditions. The response obtained with an FSI blade simulation was compared with that obtained using the Blade Element Momentum (BEM) theory typically used for wind turbine design by Rahimi et al. (2016), showing the degree to which the BEM theory underestimates the edgewise vibrations of the blade. The differences on the VIV response of a wind turbine blade with different tip geometries have been analysed in Horcas et al. (2020). The influence of including a trailing edge flap to mitigate blade VIV has been studied in Horcas, Madsen, et al. (2022). A very significant dependency of the VIV response on the inflow angle with respect to the blade pitch axis has been found in Horcas, Sørensen, et al. (2022). The effect on the response amplitude of initial displacement conditions, and the predictive potential of forced-motion simulations were investigated by Grinderslev et al. (2022). To the best of the author’s knowledge, no numerical studies on the VIV response of wind turbine blades or airfoils have been made with a turbulent inflow velocity field, nor have any experimental results of full wind turbine blades undergoing VIV been reported. Despite the extensive existing literature on VIV, characterizing the VIV response of a stopped wind turbine blade remains a challenging task. A large gap exists between the insight obtained from analyses of the airfoil VIV response and from simulations of the full wind turbine blade (Horcas, Sørensen, et al., 2022).

The parameters affecting blade VIV include, among others, the Reynolds number, inflow turbulence characteristics, spatial heterogeneity (wind shear, blade twist, chord length, mass and stiffness distributions, flow relative inclination, etc.), cross-sectional shape (airfoil type, angle of attack, surface roughness, etc.), structural properties (natural frequencies, structural damping, etc.), and the presence of other concurrent forces or initial vibration conditions (due to the existence of multiple stable limit-cycles). The effect of some of these parameters may be obtained from analyses of sectional models, whereas others will require analyses of the full blade. An additional layer of complexity is added when considering the interaction between more than one aeroelastic body, as would be the case of a wind turbine. The blade dynamics can interact with the tower dynamics, and the blades can interact with each other both structurally and aerodynamically. The multi-body problem has been studied extensively for multiple cylinders (Yu et al., 2024) and only recently for a whole wind turbine rotor (Pirrung et al., 2024) and for rotor–tower interactions (Vicente-Ludlam et al., 2024). Thus, characterizing the effects on blade VIV of this large parameter space, whether to understand the phenomena or to define accurate engineering models, will involve a combination of simulation approaches with different levels of fidelity.

CFD simulations can be categorized into different levels of fidelity, each offering a trade-off between computational cost and accuracy. At the highest level of fidelity, direct numerical

simulations (DNS) resolve all scales of turbulence without relying on turbulence models, but their computational cost is prohibitive for most practical applications. Some applications of DNS to the vortex shedding problem include flat-plate vortex shedding in Hemmati et al. (2016), and VIV of a diamond cross-section cylinder in Leontini and Thompson (2013). Large eddy simulations (LES) provide a compromise by resolving large turbulent structures while modelling smaller scales, making them suitable for high-fidelity studies of complex flows. An example of GPU-accelerated LES for the vortex shedding behind wind turbine airfoils can be found in Chakrabarti et al. (2024). Unsteady Reynolds-averaged Navier-Stokes (URANS) simulations, while less accurate for capturing transient flow features, are computationally cheaper and widely used for industrial applications. Detached eddy simulations (DES) and delayed detached eddy simulations (DDES) combine URANS in regions of flow close to the solid boundaries (i.e. within the boundary layer) with LES in regions far from these boundaries, offering a balance between accuracy and computational efficiency. A comparison of LES, DES and URANS for the VIV prediction of a 5:1 rectangular cylinder can be found in P. Zou et al. (2025). Finally, free-vortex models, which have lower fidelity and cost than CFD, have also been employed to simulate vortex shedding and VIV of cylinders and airfoils (Ramos-García et al., 2017; F. Zou et al., 2015).

Chapter 3

Tower vortex-induced vibrations: engineering models comparison

This chapter applies different vortex-induced vibration (VIV) engineering models to wind turbine towers, specifically analysing the NREL 5-MW tower. The results provide insight for future modelling and assessment of VIV phenomena in wind turbines. As presented in Chapter 2, numerous engineering models for the VIV phenomenon of tower-like structures have been proposed, and the matter is far from settled, while published full-scale measurements of VIV events in wind turbine towers are scarce. This limits the ability to propose, refine, and validate mathematical models for VIV, making the process difficult and uncertain. However, some insights can be gained on the suitability and lines of improvement of these models by analysing the results they produce. In this chapter, three engineering models are applied in the time domain in order to obtain a detailed structural response, retaining some nonlinearities of the VIV phenomena. Time-domain VIV models are also good candidates to be included as part of the hydro-servo-elastic load calculation simulations of a wind turbine required for its certification, as per IEC 61400-1 Ed.3.1 (2010). The present work does not intend by any means to cover the full range of existing VIV modelling options, but it represents a first step towards applying time-marching VIV engineering models to the wind turbine certification process.

The content presented in this chapter is an extension of the author's previously published work Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al. (2024) and Fernandez-Aldama et al. (2021), with additional details and cases studied.

3.1 Methodology

This section deals with the specific methodologies followed in the present work to obtain the VIV response of a wind turbine tower. The tower is modelled using the classical modal decomposition, as presented in Section 3.1.1, and thus the VIV loading is presented in terms of modal forces in Section 3.1.2, where three different VIV models are described in detail.

3.1.1 Structural model

A typical VIV calculation aims to predict either the structure's displacement and acceleration (e.g., for occupant comfort) or the stresses it experiences (e.g., for wall thickness design). In the case of a wind turbine tower, the tower top displacement can be critical during nacelle and blade installation, and during transportation. Additionally, the stresses due to VIV can consume a significant part of the fatigue lifetime, or even be design driving in terms of ultimate loads analysis. Thus, the modelling of the structure is an essential part of any VIV calculation. Structural models range from simple rigid body models, to flexible beam models, up until highly detailed finite element method (FEM) 3D models, among others. In this work, the flexible beam modal description is used, for the sake of simplicity and reduced computational time resulting from mode truncation.

The wind turbine tower is modelled as a cantilevered Euler-Bernoulli beam and simplified using classical modal superposition. The derivation of the modal differential equation, as obtained from the Euler-Bernoulli beam formulation, is presented in Appendix A. The tower motion and the vortex-induced load may be considered generally as stochastic processes, which are denoted here by including a functional dependency with the realization number, α , as per the notation in Wyngaard (2010). The following set of modal differential equations, for modes $j = 1 : n_m$, is thus defined:

$$m_j \ddot{p}_j(t; \alpha) + 2\zeta_{s,j} m_j \omega_j \dot{p}_j(t; \alpha) + m_j \omega_j^2 p_j(t; \alpha) = F_j(t; \alpha), \quad (3.1)$$

where m_j is the modal mass, $\zeta_{s,j}$ is the structural modal damping ratio, $\omega_j = 2\pi f_{s,j}$ where $f_{s,j}$ is the undamped modal natural frequency, t is time, and z is the height from the ground. The structural response is here defined by the variable $p_j(t; \alpha)$, which is the modal coordinate of the tower lateral (or cross wind) deflection, such that the lateral deflection $w(z, t; \alpha)$ is obtained using the following modal superposition,

$$w(z, t; \alpha) = \sum_{k=1}^{n_m} p_k(t; \alpha) \Psi_k(z), \quad (3.2)$$

where $\Psi_k(z)$ is the modal shape normalized by its value at the tower top, $z = H$, such that $\Psi_k(H) = 1$. The VIV load corresponds to the variable $F_j(t; \alpha)$, which is the modal projection over mode j of the general external force per unit length, $F(z, t; \alpha)$, defined as

$$F_j(t; \alpha) = \int_0^H F(z, t; \alpha) \Psi_j(z) dz. \quad (3.3)$$

The VIV models presented in the following section deal mainly with the definition of the external modal forces $F_j(t; \alpha)$.

3.1.2 VIV models

This section provides a detailed description of the VIV models used in this thesis for their application to wind turbine towers. Three different models are here described: a forced system model named Type A, a fluidelastic system model named Type B, and a coupled

system model named Type C, following the nomenclature given in Païdoussis et al. (2010). A description of the nature of these models is detailed in Section 2.2.

The three VIV models are implemented and solved here in the time domain. Modelling VIV in the time domain allows for a deeper analysis of the modelled response. Time-domain simulations can be more readily extended to more complex situations (e.g. including vortex shedding from the nacelle or blades or including modal coupling) and transient events (e.g. wind gusts), making them a requisite for most load calculations as per IEC 61400-1 Ed.3.1 (2010). Most importantly, by solving the equations in the time domain, nonlinear phenomena can be directly modelled. Additionally, the slow development of VIV oscillations, together with the turbulent nature of the atmospheric wind, may result in the need to model VIV in the time domain for fatigue estimations. The main drawback of time-domain simulations is the computational cost, which can be significant for long-duration simulations or when a large number of realizations is required. The specific implementation of the three time-domain models employed in this work is now described.

Type A

Type A or forced system models are defined solely by a vortex shedding force, represented here by its lift coefficient $c_l(z, t; \alpha)$, which is independent of the structural motion. In the present implementation, the formulation for the lift coefficient c_l is based on a spectral description of said load, i.e. its PSD $S_{c_l}(z, f)$. Spectral models were first proposed by B. J. Vickery and Clark (1972), and have since then been the basis of other important implementations, such as the VIV model of B. J. Vickery and Basu (1983) and the formulation present in EN 1991-1-4:2005+A1 (2010). This is the model employed in this work.

Specifically, the equation defined in B. J. Vickery and Clark (1972) to approximate the lift coefficient power spectral density is

$$\frac{f S_{c_l}}{\sigma_{c_l}^2} = \frac{f}{B \pi f_{St}} \exp \left[- \left(\frac{1 - \frac{f}{f_{St}}}{B} \right)^2 \right], \quad (3.4)$$

where f denotes frequency; $\sigma_{c_l}(z)$ is the standard deviation of the lift coefficient; $B(z)$ is a bandwidth parameter and $f_{St}(z) = St(z)U(z)/D(z)$ is the characteristic vortex shedding frequency according to the so-called Strouhal law. The parameters σ_{c_l} and B have been approximated from experimental results as a function of the turbulence characteristics in a number of different ways. In this work, the definitions used will be that of B. J. Vickery (1995) for σ_{c_l} and that of B. J. Vickery and Basu (1983) for B . These expressions are

$$\sigma_{c_l} = (0.15 + 0.55I^*) - (0.09 + 0.55I^*) \exp(-20I^*)^3, \quad (3.5)$$

and

$$B^2 = B_0^2 + 2I_u^2, \quad (3.6)$$

being $I^*(z) = I_u(z) (D(z)/L_u(z))^{1/3}$ and effective turbulence intensity, where I_u is the longitudinal turbulence intensity and L_u is the longitudinal turbulent length scale. The turbulence length scale, L_u may be obtained for a specific case for example from the standard IEC

61400-1 Ed.3.1 (2010). The bandwidth value in smooth flow, B_0 , is evaluated here as $B_0 = 0.1$ as per B. J. Vickery (1998).

The formulation presented up to this point refers to the PSD at a single tower height z (also named tower section or station, indistinctly). But it is well known that the vortex shedding is significantly correlated along adjacent sections. Thus, a coherence function, $R(z_1, z_2)$, between any two heights z_1 and z_2 is typically employed to define the two-point cross-power spectral density (CPSD) of the lift coefficient, $S_{c_l}(z_1, z_2, f)$, such that

$$S_{c_l}(z_1, z_2, f) = R(z_1, z_2) \sqrt{S_{c_l}(z_1, f)} \sqrt{S_{c_l}(z_2, f)}. \quad (3.7)$$

The coherence function employed in this work is of the form of that proposed in B. J. Vickery and Clark (1972), which is

$$R(z_1, z_2) = \cos(c_2 r) \exp(-c_1 r^2), \quad (3.8)$$

where

$$r(z_1, z_2) = \frac{2|z_1 - z_2|}{D(z_1) + D(z_2)},$$

and c_1 and c_2 are constants. In this work, the values assigned to these constants are those proposed in Hansen (1981), which are derived from measured full-scale chimney data. The values are $c_1 = 0.12$ and $c_2 = 0.13$. Note that coherence functions are typically a function of frequency, $R(z_1, z_2, f)$, but, because the vortex shedding process is narrowband, it is assumed in this model that the coherence is the same for every frequency.

To apply this model to time-marching simulations, numerical generation of synthetic time series of the lift coefficient, $c_l(z, t; \alpha)$, is required. Given the stochastic nature of the model, multiple simulations are needed to characterize the VIV response sufficiently well. In this work, the c_l time series are generated using the method by P. S. Veers (1988), originally developed for the generation of synthetic wind velocity time series, and combined with the harmonic superposition method, see Shinozuka and Deodatis (1991).

Having defined the lift coefficient c_l , the lift force is, as usual, defined as

$$F_A(z, t; \alpha) = \frac{1}{2} \rho U^2(z) D(z) c_l(z, t; \alpha), \quad (3.9)$$

where ρ is the fluid density, and the subscript A denotes that this is the force of the Type A model. Ultimately, the modal load applied to the modal dynamic differential equation for the Type A model, F_{jA} , is, for $j = 1 : n_m$,

$$F_{Aj}(t; \alpha) = \int_0^H F_A(z, t; \alpha) \Psi_j(z) dz. \quad (3.10)$$

By their definition, Type A or forced system models, like the one presented here, do not account for the effect of the structural motion on the vortex-induced loads. Therefore, this type of model may fail to accurately predict the response of a structure when its amplitude of motion is not small, and is thus only suitable for structures with high enough structural damping. For this reason, more advanced Type B and Type C models are also considered in this thesis and presented next.

Type B

As defined in Section 2.2, the characteristic of Type B or fluidelastic models is that the fluid force is modelled as an explicit function of the structural motion. In this implementation, the fluid force is defined as the sum of a stochastic force, the same as in the Type A model just presented, plus a motion dependent force.

The motion-dependent force is modelled as a nonlinear damping, consisting of a negative linear damping term and positive cubic damping term, as proposed in Marris (1964), and most notably applied in B. J. Vickery and Basu (1983). This model successfully accounts for the widely observed self-excited and self-limited characteristics of VIV. This modelling choice also serves as basis to the EN 1991-1-4:2005+A1 (2010) simplified engineering model, among others. The equation defined in B. J. Vickery and Basu (1983) to approximate the motion-dependent force, F_{B-A} , is presented in Equation (3.11). The subscript $B - A$ is used to denote that this is the force of the Type B model minus that of the Type A model, such that $F_B = F_{B-A} + F_A$. Thus, the expression for F_{B-A} is

$$F_{B-A}(z, t; \alpha) = k_a (\dot{w} - G\dot{w}^3), \quad (3.11)$$

where $k_a(z, t; \alpha)$ and $G(z, t; \alpha)$ are the linear negative aerodynamic damping factor and the nonlinear positive aerodynamic damping factor, respectively.

To transform the load in Equation (3.11) into a modal load, the displacement, w , is substituted by its modal representation, see Equation (3.2), which results in

$$F_{B-A}(z, t; \alpha) = k_a \left(\sum_{k=1}^{n_m} \Psi_k \dot{p}_k - G \sum_{k=1}^{n_m} \sum_{l=1}^{n_m} \sum_{m=1}^{n_m} \Psi_k \Psi_l \Psi_m \dot{p}_k \dot{p}_l \dot{p}_m \right), \quad (3.12)$$

Next, the force is projected over mode j (see Equation (3.3)), giving, for $j = 1 : n_m$,

$$F_{B-A_j}(t; \alpha) = \sum_{k=1}^{n_m} k_{a,jk} \dot{p}_k - \sum_{k=1}^{n_m} \sum_{l=1}^{n_m} \sum_{m=1}^{n_m} G_{a,jklm} \dot{p}_k \dot{p}_l \dot{p}_m, \quad (3.13)$$

where the coupled modal linear aerodynamic damping parameter matrix, $k_{a,jk}(t; \alpha)$, and the coupled modal nonlinear positive aerodynamic damping parameter fourth order tensor, $G_{a,jklm}(t; \alpha)$, are defined as

$$k_{a,jk} = \int_0^H k_a \Psi_k \Psi_j dz, \quad (3.14)$$

$$G_{a,jklm} = \int_0^H G k_a \Psi_k \Psi_l \Psi_m \Psi_j dz. \quad (3.15)$$

In this work, it is assumed that there is no coupling between the modes, such that

$$k_{a,jk} = \begin{cases} k_{a,j}, & \text{if } j = k \\ 0, & \text{if } j \neq k, \end{cases}$$

and

$$G_{a,jklm} = \begin{cases} G_{a,j}, & \text{if } j = k = l = m \\ 0, & \text{otherwise.} \end{cases}$$

This simplification results in the following modal force equation, for $j = 1 : n_m$,

$$F_{B-A_j}(t; \alpha) = k_{a,j} \dot{p}_j - G_{a,j} \dot{p}_j^3. \quad (3.16)$$

If one considered the aerodynamic damping, k_a , to be independent of the structural motion, $k_a = k_a(z)$, Equation (3.11) would only be valid for theoretical resonant conditions (i.e. when the structural motion frequency is very close to the natural vortex shedding frequency), as this expression implies that the same negative damping appears as a result of any structural vibration regardless of its frequency, whereas it is well known that VIV occurs for a range of velocities around the resonant velocity, due to the lock-in phenomenon (Feng, 1968). Therefore, some dependence of k_a with the motion frequency and natural vortex shedding frequency is required. In general, this aerodynamic damping could be expressed as $k_a = k_a \left(\frac{f_{St}}{f_w} \right)$, where $f_{St} = StU/D$ is the Strouhal-defined vortex shedding frequency and f_w is the instantaneous motion frequency. Some authors have proposed models in this direction, such as those by Arnold et al. (2024), D’Asdia and Noè (1998), and Thorsen et al. (2017). Experimental evidence of the dependence of the aerodynamic damping with the frequency ratio is gathered in B. J. Vickery and Basu (1983). In the present implementation, it is assumed that the modal aerodynamic damping matrix diagonal terms $k_{a,j}$ have a parametric modal dependency with a velocity ratio, $U_{ratio,j} = \frac{U}{U_{crit,j}} = \frac{f_{St}}{f_{s,j}}$, as proposed in Hansen (1998), where $U_{crit,j} = f_{s,j}D/St$ is the critical or resonant velocity. This parametric relationship defines the lock-in range extent, which may be different for each mode.

Note that in simplified approaches to the VIV problem, the dependence of the vortex-induced damping with the velocity ratio is typically avoided, being the result of the model the maximum displacement expected at the critical velocity, or close to it (Cheung & Melbourne, 1983a; Lupi et al., 2017a; B. J. Vickery & Basu, 1983). This limitation becomes relevant when it is necessary to capture the lock-in range extent, for example to calculate the fatigue damage given a distribution of wind speeds. While the present implementation captures the lock-in range behaviour, it assumes that the motion occurs at the structural natural frequencies. The model will not correctly account for negative damping induced by vibrations at other frequencies, which may occur if the structure undergoes forced motion.

The aerodynamic damping factor k_a can be expressed in dimensionless form as proposed in Hansen (1998) and B. J. Vickery and Basu (1983). This dimensionless parameter, denoted as $K_a(z)$, is formulated as

$$K_a = \frac{k_a}{4\pi f_{s,j} \rho D^2}, \quad (3.17)$$

resulting in the following expression for its modal counterpart $k_{a,j}$

$$k_{a,j}(z) = 4\pi f_{s,j} \rho \int_0^H K_a D^2 \Psi_j^2 dz. \quad (3.18)$$

The aerodynamic damping parameter K_a is one of the most important parameters of the model by B. J. Vickery and Basu (1983). Its importance resides in that, depending on its value, it determines whether the total damping becomes positive or negative. In general, the aerodynamic damping parameter has a very complex functional dependency on several dimensionless parameters, e.g. $K_a = K_a(U_{\text{ratio},j}, \text{Re}, \frac{A_w}{D}, I_u, L_u, \dots)$. The most common starting point to define such functional dependence consists on establishing a multiplicative dependency on the local velocity ratio, $U_{\text{ratio},j}(z)$, the local Reynolds number, $\text{Re}(z) = \rho U(z) D(z) / \mu$, and the local turbulence intensity, $I_u(z) = \sigma_u / U(z)$, as proposed in Hansen (1998) and also employed by Nunez-Casado et al. (2017). This is expressed as $K_a = K_a(U_{\text{ratio},j}, \text{Re}, I_u) = K_{U_{\text{ratio},j}} K_{\text{Re}} K_{I_u}$. Note that in this approach the dependence of the aerodynamic damping with the amplitude of motion, A_w , is accounted for by the nonlinear expression of the force. The expression for each of the damping contributing factors is

$$K_{U_{\text{ratio},j}}(z) = \begin{cases} 0, & \text{if } U_{\text{ratio},j}(z) \leq 1 \\ 2 - U_{\text{ratio},j}(z), & \text{if } 1 < U_{\text{ratio},j}(z) < 2 \\ 0, & \text{if } U_{\text{ratio},j}(z) \geq 2, \end{cases} \quad (3.19)$$

$$K_{\text{Re}}(z) = \begin{cases} 2, & \text{if } \text{Re}(z) \leq 10^5 \\ 2 - \frac{1.5}{\ln(5)} \ln\left(\frac{\text{Re}(z)}{1 \times 10^5}\right), & \text{if } 10^5 < \text{Re}(z) < 5 \times 10^5 \\ 0.5 + \frac{0.5}{\ln(2)} \ln\left(\frac{\text{Re}(z)}{5 \times 10^5}\right), & \text{if } 5 \times 10^5 < \text{Re}(z) < 10^6 \\ 1, & \text{if } \text{Re}(z) \geq 10^6, \end{cases} \quad (3.20)$$

$$K_{I_u}(z) = \begin{cases} 1 - 3I_u(z), & \text{if } 0 \leq I_u < 0.25 \\ 0.25, & \text{if } I_u \geq 0.25. \end{cases} \quad (3.21)$$

Note that the expression for K_{Re} is the same as that proposed in EN 1991-1-4:2005+A1 (2010), and the expression for K_{I_u} has been recently added to prEN 1991-1-4:2019 (2020). The dependency with the velocity ratio is not typically considered in design standards (Verboom & van Koten, 2010), which aim to provide an estimate of the maximum response only.

Similarly to the linear negative damping modal factor, $k_{a,j}$, the nonlinear positive damping modal factor, $G_{a,j}$ can be expressed in terms of G and K_a as

$$G_{a,j}(z) = 4\pi f_{s,j} \rho \int_0^H G K_a D^2 \Psi_j^4 dz. \quad (3.22)$$

In this work, the nonlinear term $G(z)$ is defined, as per B. J. Vickery and Basu (1983), as a function of a dimensionless limiting amplitude constant, a_L , the local diameter, $D(z)$, and the mode frequency, $f_{s,j}$. The expression for G is derived by B. J. Vickery and Basu (1983), assuming that the response is harmonic when the limit cycle is reached in the absence of external forces and structural damping, and therefore the amount of energy dissipated per cycle by the linear and nonlinear damping terms in Equation (3.11) have to be equal. This results in the following expression for G ,

$$G = \left(\frac{1}{a_L D \pi f_{s,j} \sqrt{3}} \right)^2. \quad (3.23)$$

The limiting amplitude factor is set in this work to $a_L = 0.4$, which is the value proposed in EN 1991-1-4:2005+A1 (2010).

It is noticeable that the Type B formulation relies on several semi-empirical approximations for the aerodynamic damping values (negative linear damping and positive nonlinear damping), which are not easily derived from first principles. The coupled system Type C models, presented next, are based on a more physical approach and therefore have the potential to reduce reliance on these approximations.

Type C

Type C models are characterized by having a separate but coupled system of equations for the structural and wake dynamics. In this implementation, the fluid force on the structural system is calculated as a damping force or stall term, and an external force generated by the wake dynamics and introduced by a lift coefficient, c_l , as proposed in Skop and Balasubramanian (1997).

$$F_C(z, t) = \frac{1}{2} \rho U^2 D \left(c_l + H_1 \frac{\dot{w}}{U} \right), \quad (3.24)$$

where $H_1(z)$ is the stall parameter. A lift coefficient $c_l(z, t)$ is introduced, which is related to a hidden wake variable $q(z, t)$ by

$$c_l = q \frac{c_{l,0}}{2}, \quad (3.25)$$

where $c_{l,0}$ is the vortex-shedding lift coefficient amplitude for the stationary body, which is here evaluated as $c_{l,0} = 0.3$ in accordance with Facchinetti et al. (2004). The wake variable q can then be thought of as a lift amplification factor.

The wake variable is here represented by its modal approximation, such that

$$q(z, t) = \sum_{k=1}^{n_w} \Phi_k(z) l_k(t), \quad (3.26)$$

where $\Phi_k(z)$ is the wake variable mode shape for mode k , and $l_k(t)$ is the temporal evolution of the wake generalized coordinate. A number of wake modes n_w is considered, which need not be equal to the number of tower modes n_m . Several approximations have been proposed for the mode shapes Φ_k . The formulation is here presented for an arbitrary shape.

Substituting the lift coefficient by the wake variable to Equation (3.24), and applying the modal decomposition of $q(z, t)$ and $w(z, t)$, we obtain

$$F_C(z, t) = \frac{1}{2} \rho U^2 D \left(\frac{c_{l,0}}{2} \sum_{k=1}^{n_w} \Phi_k l_k + \frac{H_1}{U} \sum_{i=1}^{n_m} \Psi_i \dot{p}_i \right). \quad (3.27)$$

And projecting over mode j of the structural motion, and grouping terms, the modal force is, for $j = 1 : n_m$,

$$F_{Cj}(t) = \sum_{k=1}^{n_w} L_{jk} l_k + \sum_{i=1}^{n_m} S_{ji} \dot{p}_i, \quad (3.28)$$

where

$$L_{jk} = \frac{1}{4}\rho \int_0^H U^2 D c_{l,0} \Phi_k \Psi_j dz, \quad (3.29)$$

and

$$S_{ji} = \frac{1}{2}\rho \int_0^H U D H_1 \Psi_i \Psi_j dz. \quad (3.30)$$

This formulation allows us to naturally retain the mode couplings, as there is no need to model the wake variable explicitly for each tower mode in order to characterize the lock-in range, as was done for the Type B case damping factor.

It can be seen that it is required to evaluate the wake variable $l_k(t)$ in order to compute the Type C force on the structure. As mentioned earlier, the variable $q(z, t)$, or its modal counterpart $l_k(t)$, is the solution to a dynamical equation for the wake. In this work, the wake dynamics is modelled as a van der Pol oscillator and is coupled to the structural motion by a force proportional to the structural velocity, $\dot{w}(z, t)$. The velocity coupling is argued by K. Xu et al. (2015) to be the most suitable for wind engineering applications, due to the much smaller mass ratios of these structures when compared to hydrodynamic applications, for which an acceleration coupling is often considered.

The wake oscillator is here presented in its general form, and the modal decomposition is applied to obtain the modal representation of the wake dynamics, as required to solve the modal structural modal equations of motion. The wake oscillator equation is given by

$$\ddot{q} + 2\pi f_{St} \varepsilon (q^2 - 1) \dot{q} + (2\pi f_{St})^2 q = b \dot{w}, \quad (3.31)$$

where ε is the strength of the van der Pol nonlinear damping term, $f_{St}(z) = St(z)U(z)/D(z)$ is the Strouhal-defined frequency, as previously defined for the Type A model, and $b(z) = \beta/D(z)$, being $b(z)$ and β structural coupling parameters.

The relationship between q and c_l , presented earlier in Equation (3.25), is given by the homogeneous solution to Equation (3.31), which provides a stable quasi-harmonic oscillation of finite amplitude $q_0 = 2$ at frequency f_{St} (Facchinetti et al., 2004), such that, in the static tower case, the lift coefficient is $c_l = q_0 \frac{c_{l,0}}{2} = c_{l,0}$.

Applying the modal decomposition of $w(z, t)$, Equation (3.2), and of $q(z, t)$, Equation (3.26), to the wake oscillator Equation (3.31), it is obtained that

$$\sum_{k=1}^{n_w} \Phi_k \ddot{l}_k + 2\pi f_{St} \varepsilon \left(\sum_{l=1}^{n_w} \sum_{m=1}^{n_w} \Phi_l \Phi_m l_l l_m - 1 \right) \sum_{k=1}^{n_w} \Phi_k \dot{l}_k + (2\pi f_{St})^2 \sum_{k=1}^{n_w} \Phi_k l_k = b \sum_{i=1}^{n_m} \Psi_i \dot{p}_i. \quad (3.32)$$

To obtain the modal representation of Equation (3.32), the whole equation is projected over the wake mode j , such that, for $j = 1 : n_w$,

$$\sum_{k=1}^{n_w} n_{jk} \ddot{l}_k + \sum_{k=1}^{n_w} \sum_{l=1}^{n_w} \sum_{m=1}^{n_w} a_{jklm} \dot{l}_k l_l l_m - \sum_{k=1}^{n_w} d_{jk} \dot{l}_k + \sum_{k=1}^{n_w} h_{jk} l_k = \sum_{i=1}^{n_m} B_{ji} \dot{p}_i, \quad (3.33)$$

where:

$$\begin{aligned}
n_{jk} &= \int_0^H \Phi_k \Phi_j dz, \\
a_{jklm} &= 2\pi\varepsilon \int_0^H f_{St} \Phi_k \Phi_l \Phi_m \Phi_j dz, \\
d_{jk} &= 2\pi\varepsilon \int_0^H f_{St} \Phi_k \Phi_j dz, \\
h_{jk} &= 4\pi^2 \int_0^H f_{St}^2 \Phi_k \Phi_j dz, \\
B_{ji} &= \int_0^H b\Psi_i \Phi_j dz.
\end{aligned}$$

Even though there are attempts at providing physical interpretations of the parameters involved in these equations, namely ε , H_1 and β , some experimental results are still needed to evaluate these parameters.

The Type C model is based on a more physical approach than the Type B model, as it does not rely on semi-empirical approximations for the aerodynamic damping lock-in range behaviour. Instead, the lock-in range emerges naturally from the interaction between the tower and wake dynamics, identified as coupled-mode flutter in de Langre (2006). On the other hand, the Type C model presented does not account for the effect of inflow turbulence. Recent efforts in this regard have been made by Denoël (2020), Mannini (2020), and Rigo et al. (2022a).

Summary

The full system of differential equations for each VIV model are summarized below. For the Type A and Type B models, a single ordinary differential equation is solved for each mode. In contrast, the Type C model results in a coupled system of equations, comprising two ordinary differential equations per mode: one for the generalized tower deflection and another for the generalized wake variable.

 Type A

$$m_j \ddot{p}_j + 2\zeta_{s,j} m_j \omega_j \dot{p}_j + m_j \omega_j^2 p_j = F_{A_j},$$

$$F_{A_j}(t; \alpha) = \frac{1}{2} \rho \int_0^H U^2 D_{C_l} \Psi_j dz.$$

Type B

$$m_j \ddot{p}_j + 2\zeta_{s,j} m_j \omega_j \dot{p}_j + m_j \omega_j^2 p_j = F_{B_j},$$

$$F_{B_j}(t; \alpha) = F_{A_j} + k_{a,j} \dot{p}_j - G_{a,j} \dot{p}_j^3.$$

Type C

$$m_j \ddot{p}_j + 2\zeta_{s,j} m_j \omega_j \dot{p}_j + m_j \omega_j^2 p_j = F_{C_j},$$

$$\sum_{k=1}^{n_w} n_{jk} \ddot{l}_k + \sum_{k=1}^{n_w} \sum_{l=1}^{n_w} \sum_{m=1}^{n_w} a_{jklm} \dot{l}_k \dot{l}_l \dot{l}_m - \sum_{k=1}^{n_w} d_{jk} \dot{l}_k + \sum_{k=1}^{n_w} h_{jk} l_k = Q_{C_j},$$

$$F_{C_j}(t) = \sum_{k=1}^{n_w} L_{jk} l_k + \sum_{i=1}^{n_m} S_{ji} \dot{p}_i,$$

$$Q_{C_j}(t) = \sum_{i=1}^{n_m} B_{ji} \dot{p}_i.$$

From these equations, the differences between the three models become more evident. The force of the Type A model, F_{A_j} , consists of the modal projection of the synthesized stochastic lift force. The Type B load, F_{B_j} , is the sum of the stochastic Type A force, a negative damping force and a cubic positive damping force; where these damping coefficients are defined as the modal projection of semi-empirical and phenomenological sectional damping models. The Type C load, F_{C_j} , consists of two terms; one being the modal projection of the lift force introduced through the wake oscillator generalized coordinate, l_k , which introduces a coupling between the tower modes and the wake modes. The second term is an aerodynamic damping term, which introduces a coupling between the tower modes. Thus, this force F_{C_j} depends on the solution of the generalized coordinate of all wake modes and all tower modes, making the problem highly coupled.

The wake generalized coordinates, l_k , are obtained by solving a van der Pol oscillator equation for each mode. These equations are coupled to the tower motion through the term Q_{C_j} , which is proportional to the tower generalized velocities $\dot{p}_i$. Consequently, each wake mode is coupled to all tower modes. Thus, the Type C formulation introduces not only coupling between the tower and wake modes but also intermodal coupling among the tower modes and among the wake modes, resulting in a fully coupled system of equations.

3.2 Type B model validation: Aachen chimney

As a previous step to the application of the VIV models described in Section 3.1.2 to the wind turbine tower case, which will be done in Section 3.3, a validation of the Type B VIV engineering model is presented here. The validation is performed using data from a full-scale test case of a steel chimney, as presented in Galemann and Ruscheweyh (1992) and Ruscheweyh and Galemann (1996), where measurements of the vortex-induced cross-wind vibrations of the full-scale chimney are documented. The chimney is a 28 m high, 0.91 m diameter, steel structure located in a test field at the Technical University of Aachen, Germany.

To ensure consistency with the application of the Type B model to the NREL 5-MW tower case study presented later in Section 3.3, the same choice of parameters is adopted here. Following the methodology outlined in Section 3.1, all parameters related to the Type B VIV model –namely the lift coefficient PSD definition, the negative damping parameter and the limiting amplitude– can be expressed in terms of the Strouhal number, inflow conditions and geometric properties, which are all obtained from the data given in Galemann and Ruscheweyh (1992). Additionally, for a comprehensive description of the numerical method and parameters used to compute the response, the reader is referred to Section 3.3.3.

3.2.1 Sources of uncertainty in validation

There are two main sources of uncertainty regarding the reproducibility of the results presented in Galemann and Ruscheweyh (1992) and Ruscheweyh and Galemann (1996); the Strouhal number and the turbulence intensity values.

Regarding the Strouhal number, it is reported in Ruscheweyh and Galemann (1996) that “The Strouhal number varies in the range of 0.16-0.26 with a frequent value of $St = 0.21$ ”. Being a parameter of the model, the value $St = 0.21$ has been chosen for the calculations.

Regarding the turbulence intensity values, Ruscheweyh and Galemann (1996) report a single value of the site turbulence intensity, I_u , without specifying the height at which it was measured. However, it is well established that turbulence intensity varies significantly with height (see, for example, EN 1991-1-4:2005+A1 (2010) and IEC 61400-1 Ed.3.1 (2010)). To overcome this uncertainty, a sensitivity analysis to the assumed height at which I_u is referred to is performed, resulting in two different results assuming two different heights: one is $z_{I_u} = 10$ m which corresponds to the lowest anemometer placed on the meteorological mast, and the other one is $z_{I_u} = 26$ m which corresponds to the highest anemometer placed on the meteorological mast (mast data is given in Ruscheweyh and Galemann (1996)). A vertical profile of the turbulence intensity is assumed, following its definition $I_u(z) = \sigma_u/U(z)$, being σ_u the standard deviation of the wind velocity fluctuations, which is considered constant with height as defined in EN 1991-1-4:2005+A1 (2010) and IEC 61400-1 Ed.3.1 (2010). The two different turbulence intensity profiles thus employed are shown in Figure 3.1, where z represents the height above ground. It can be seen that for the case $z_{I_u} = 10$ m the turbulence intensity is smaller at all chimney heights. This can have an effect on the stochastic loading introduced in the model, by increasing the bandwidth of the lift coefficient PSD as well as its standard deviation (see Section 3.1.2). Additionally, the turbulence intensity reduces the aerodynamic damping parameter (see Section 3.1.2).

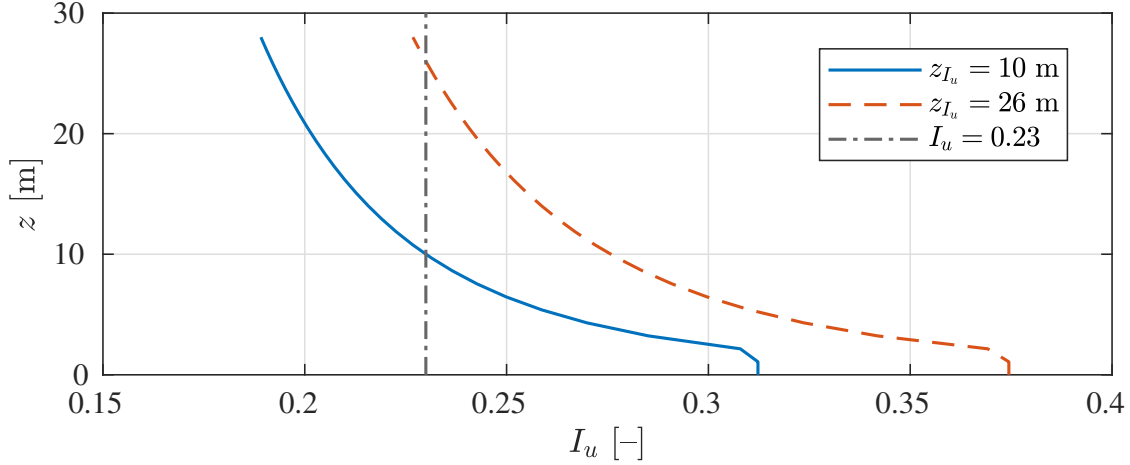


Figure 3.1: Turbulence intensity profiles, assuming that the reported value of $I_u = 0.23$ corresponds to a height of $z_{I_u} = 10$ m or of $z_{I_u} = 26$ m.

Additionally, when computing the modal transformation of the Euler-Bernoulli beam to be used for the structural definition, it was detected that some mass other than the pure steel material are probably present (e.g. bolts, welds, flanges or paint), as the computed natural frequencies and equivalent mass, $m_{eq,1}$ as per Equation (3.34), differ noticeably (15.9 % and -19.3 % error, respectively) from the values reported in Ruscheweyh and Galemann (1996). Therefore, a mass adjustment is made, assuming the missing mass to be uniformly distributed and equal to a 25 % of the total mass. This adjustment value is chosen manually to have a good fit of the frequencies and equivalent mass. The results of the first natural frequency and equivalent mass using the corrected mass are shown in Table 3.1. The Scruton number for the first mode, Sc_1 , which is a function of the equivalent mass and structural damping ratio, as shown in Equation (3.35), is also tabulated for completeness. The structural damping ratio is evaluated as $\zeta_{s,1} = (1 + 4\pi^2\delta_{s,1}^{-2})^{-1/2} = 0.191$ %, where $\delta_{s,1} = 1.2$ % is the logarithmic decrement as given in Galemann and Ruscheweyh (1992).

$$m_{eq,1} = \frac{\int_0^H m(z) \Psi_1^2(z) dz}{\int_0^H \Psi_1^2(z) dz}. \quad (3.34)$$

$$Sc_1 = \frac{4\pi\zeta_{s,1}m_{eq,1}}{\rho D^2}, \quad (3.35)$$

Parameter	Reported	Computed	Error
Natural frequency, f_1	1.7 Hz	1.76 Hz	3.66 %
Equivalent mass, $m_{eq,1}$	87 kg/m	93.1 kg/m	7.05 %
Scruton number, Sc_1	2.1	2.2	4.92 %

Table 3.1: Comparison of the computed structural properties of the validation model with the data reported in Ruscheweyh and Galemann (1996), for the first natural vibration mode.

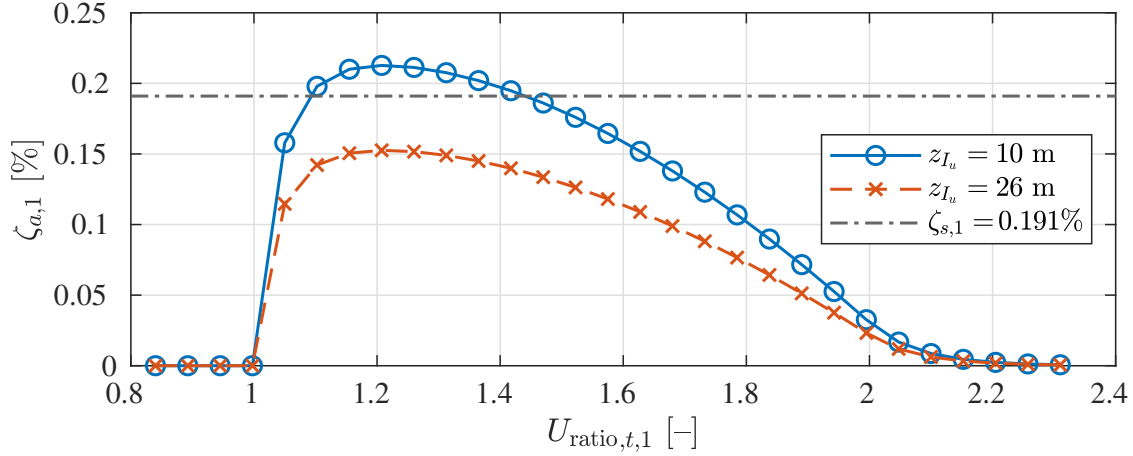


Figure 3.2: Type B modal linear aerodynamic damping ratio for mode 1, $\zeta_{a,1}$, as a function of the velocity ratio, for both assumed heights for the reported turbulence intensity value, z_{I_u} , together with the structural damping ratio, $\zeta_{s,1}$.

3.2.2 Aachen chimney VIV response

As a previous step to the dynamic calculations, the response can be characterized as being in either the “forced vibration” or the “lock-in” regime, based on whether the negative aerodynamic damping, $\zeta_{a,1}$, surpasses the structural damping, $\zeta_{s,1}$, (Pagnini et al., 2020), where the modal aerodynamic damping ratio is defined as $\zeta_{a,j} = \frac{k_{a,j}}{2m_j\omega_j}$. In Figure 3.2, the computed aerodynamic damping values for both reference heights considered (i.e. different turbulence intensity values), are shown as a function of the velocity ratio at the tower top, $U_{ratio,t,1}$. The Aachen chimney structural damping value is also shown as a horizontal line. It can be seen that the chimney may only be in “lock-in” regime for the lowest turbulence intensity case, and for a range of velocity ratios between approximately $1.1 \leq U_{ratio,t,1} \leq 1.4$, thus notably larger vibrations may be expected for this turbulence intensity case and these wind speeds.

The maximum lateral displacement results obtained using the Type B model are shown in Figure 3.3, where they are compared with the experimental measurements as presented in Galemann and Ruscheweyh (1992). It is assumed that the inflow wind speed reported in the experimental data corresponds to that at the chimney top. There is no information in Galemann and Ruscheweyh (1992) or Ruscheweyh and Galemann (1996) about any differences between the test runs performed and reproduced in the figure. The Type B model results are shown for both assumed heights for the reported turbulence intensity value, z_{I_u} , and correspond to the ensemble average over 12 realizations of the stochastic process. The computed maximum displacement shows a reasonable agreement with the experimental data, particularly for the low turbulence intensity case. The $z_{I_u} = 10$ m case exhibits larger maximum displacements, consistent with the lower turbulence intensity at the chimney top for this scenario, as predicted by the results in Figure 3.2. Despite the limitations of the model, uncertainties in the Strouhal number and other structural and environmental parameters, and the dispersion in the experimental measurements, the Type B model demonstrates satisfactory

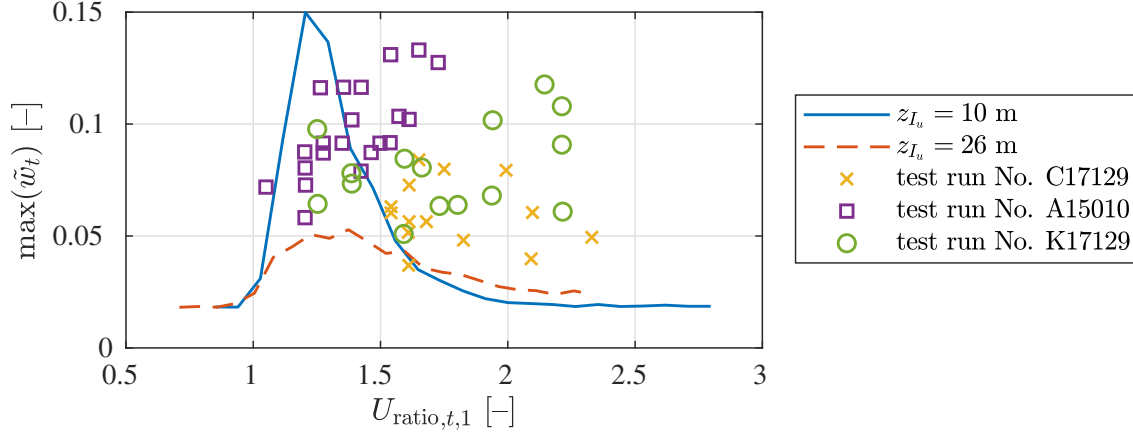


Figure 3.3: Maximum dimensionless chimney top displacement, $\max(\tilde{w}_t)$, as a function of the ratio between the chimney top inflow wind speed and the first mode critical wind speed, $U_{\text{ratio},t,1} = U(H)/U_{\text{crit},1}$. Type B VIV model results, for both assumed heights for the reported turbulence intensity value, z_{I_u} , are shown as continuous lines. Experimental data for the three test runs presented in Galemann and Ruscheweyh (1992) are shown as individual points. The Type B results shown correspond to the ensemble average over 12 realizations of the stochastic process.

accuracy in predicting the maximum displacement of the chimney top. The onset of the lock-in range is well captured, although its extent is underestimated, as the experimental data indicate a broader range of higher velocities where lock-in persists.

The Type B VIV model has been validated against the full-scale measurements of the Aachen chimney, and it is shown to be able to reproduce the main features of the VIV response. Some differences exist between the Aachen chimney and the NREL 5-MW wind turbine tower, such as the larger diameter of the tower and its variation with height, which result, among other things, in notably higher Reynolds numbers being encountered by the thicker wind turbine tower, and in a different distribution of the vortex-induced forces along the tower height, thus affecting the overall VIV response. Nonetheless, the model is considered adequate for its application to the NREL 5-MW wind turbine tower in the following section.

3.3 Case study: NREL 5-MW tower

In this section, the methodology presented in Section 3.1 is applied to the NREL 5-MW baseline wind turbine tower, defined in Jonkman et al. (2009). The VIV response is obtained using the three different vortex-induced vibrations (VIV) models previously described in Section 3.1.2.

3.3.1 Simulations overview

The response is limited to the first mode of vibration, as the natural frequencies of higher modes of the isolated NREL 5-MW tower would only be directly excited at wind speeds well above the design wind speed of the wind turbine. The inflow characteristics are defined

according to the wind turbine design standard IEC 61400-1 Ed.3.1 (2010), with the exception that the turbulence intensity is set to zero. This adjustment is made to facilitate a direct comparison with the Type C VIV model, which does not account for inflow turbulence.

To showcase the qualitative differences between the three engineering models studied, the tower response is here simulated using two different structural damping values. The two damping values chosen are such that, for the lower damping value, the response is self-limited and driven by the tower motion-induced forces, whereas, for the higher damping value, the response is driven mainly by the natural vortex shedding force. To contextualize the damping values commonly employed for wind turbine towers and related structures, a collection of such values is presented in Figure 3.4. The dispersion in the proposed damping values shown in Figure 3.4 is quite considerable. In all accounts, determining the structural damping of a full-scale structure is a complex task (Blevins, 2001; Scruton & Flint, 1964). The two modal damping values employed in this analysis are $\zeta_{s,1} = 0.0025 = 0.25\%$, for the low damping case, and $\zeta_{s,1} = 0.01 = 1\%$, for the high damping case. The value $\zeta_{s,1} = 0.25\%$ is typically used for wind turbine steel towers, as it is the proposed value in Annex P of IEC 61400-6 Ed.1 (2020), while $\zeta_{s,1} = 1\%$ is the value originally given for the NREL 5-MW tower in Jonkman et al. (2009). Both values are present in Figure 3.4.

As the velocity, tower diameter, and other derived quantities vary along the tower height, reference values are defined at a reference height z_{ref} , so that $\square_{\text{ref}} \equiv \square(z = z_{\text{ref}})$. The reference height is chosen following the criterion of B. J. Vickery and Clark (1972), which defines it as the location at which the response is expected to be maximum, given an analytical simplified analysis, which results in z_{ref} being the section at which the expression $\Psi_j(z)D^4(z)$ reaches its maximum value. The application of this definition to the first mode of the NREL 5-MW tower results in the following reference values $z_{\text{ref}} = 74.1$ m, $D_{\text{ref}} = 4.2$ m, as can be seen in Figure 3.5 (a), which illustrates the distribution of $\Psi_j(z)D^4(z)$ along the tower height and highlights the corresponding reference height. In Figure 3.5 (b) the tower diameter along the height is shown, marking the reference height and diameter for the first structural mode.

To cover the lock-in range extent, the response is obtained for different inflow velocities. To characterize the response over the lock-in range, results will be given as a function of the velocity ratio at the reference height, which is defined for each structural mode as $U_{\text{ratio,ref},j} = U_{\text{ref}}/U_{\text{crit,ref},j}$, with $U_{\text{crit,ref},j} = f_{s,j}D_{\text{ref}}/St_{\text{ref}}$ being the critical velocity of mode j at the reference height. This velocity ratio gives an approximate idea of the distance of the inflow velocity to the most critical velocity for each mode. The velocity ratio range employed in this work consists of 27 linearly spaced velocities between $0.7 \leq U_{\text{ratio,ref},1} \leq 2.3$. This range covers velocities where lock-in might occur at different tower stations due to wind shear and diameter changes. Note that if multiple modes were used, different reference heights (and all derived quantities) could be employed for each mode, to make the maximum response of each mode occur at a velocity ratio close to one.

In the description of the NREL 5-MW tower given in Jonkman et al. (2009), the distributed tower properties over 11 stations are documented. To obtain the tower's dynamic response, a modal description of this tower is employed in this work. Therefore, a modal transformation based on these distributed properties is performed. First, the distributed properties are linearly interpolated to a larger number of stations, n_z , namely from 11 to $n_z = 27$, to enable

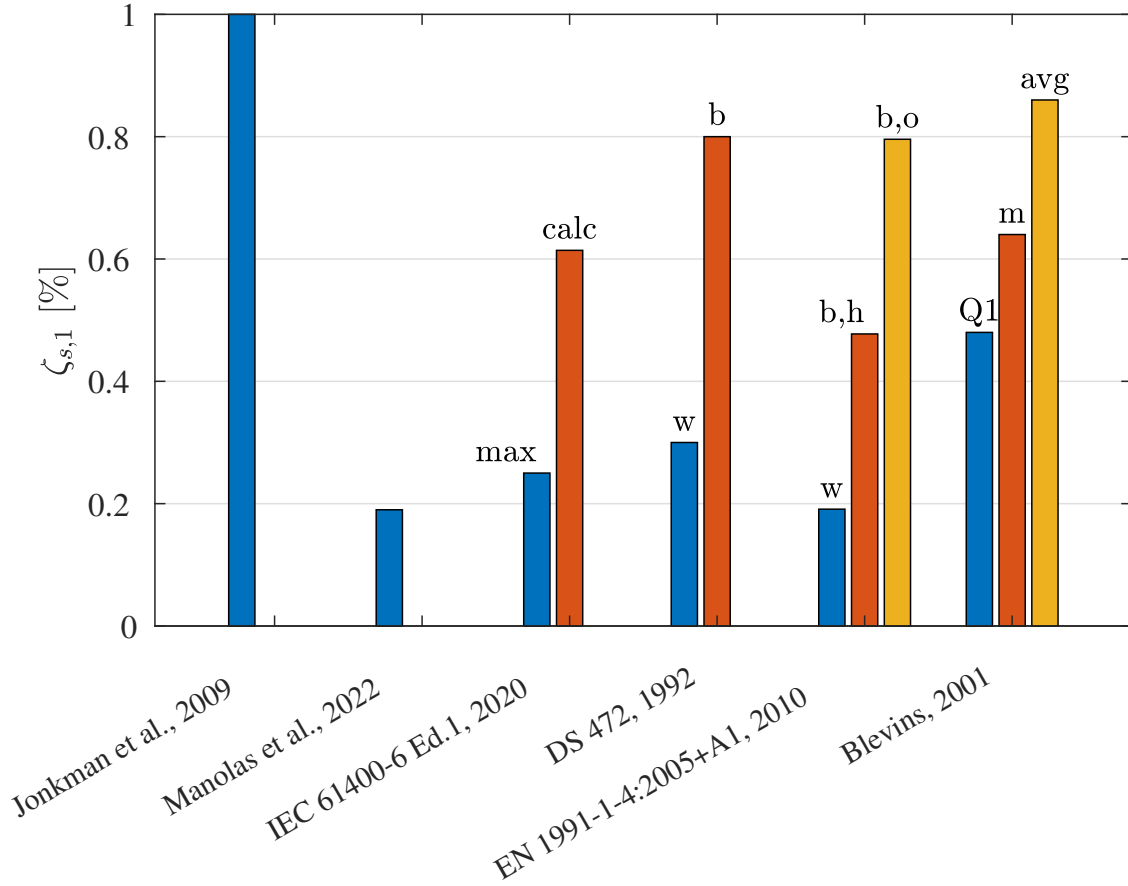


Figure 3.4: Recommended damping ratios for the first structural mode of steel wind turbine towers or chimneys, according to different sources. “Jonkman et al. (2009)” shows the value proposed for the NREL 5-MW tower. “Manolas et al. (2022)” shows the value proposed for the NREL 5-MW tower, adjusted to state-of-the art levels. “IEC 61400-6 Ed.1 (2020)” shows recommended values for steel tubular towers, where “max”: maximum value without further justification, and “calc”: calculated value as a function of the natural frequency for the NREL 5-MW tower. “DS 472, 1992” shows recommended values for “w”: welded and “b”: bolted steel structures (data from Burton et al. (2011)). “EN 1991-1-4:2005+A1 (2010)” shows recommended values for “w”: welded steel stacks, “b,h”: lattice steel towers with high resistance bolts, and “b,o”: lattice steel towers with ordinary bolts (data presented also in Burton et al. (2011)). “Blevins (2001)” data corresponds to statistical quantities derived from measurements of 21 steel towers, where “avg”: average value, “m”: median value, and “Q1”: lower quartile value. The colours of the bar graph are only used to differentiate the cases, and do not imply any further meaning.

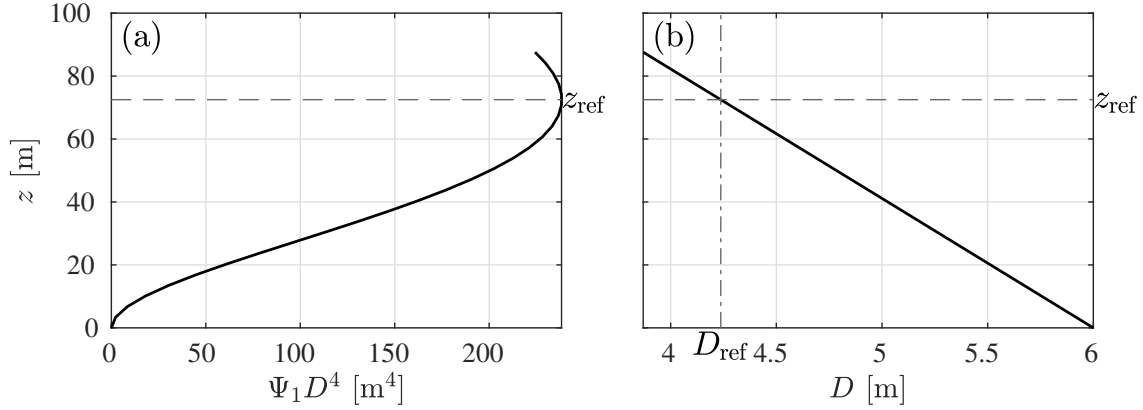


Figure 3.5: (a) Value of the function $\Psi_1(z)D^4(z)$ along the NREL 5-MW tower height. The maximum of this function is used to define the reference height z_{ref} , marked with a horizontal line. (b) NREL 5-MW tower diameter along its height, with the reference diameter $D_{\text{ref}} = D(z_{\text{ref}})$ marked with a vertical line.

a smoother modal description. These refined distributed properties are used to solve the eigenvalues and eigenvectors problem, as presented in Section 3.1.1, thus obtaining the tower modal definition. For the first tower mode analysed here, the relevant parameters are the natural frequency, $f_{s,1} = 0.891$ Hz, the modal mass, $m_1 = 6.11 \times 10^4$ kg, and the modal damping, which was previously discussed.

3.3.2 VIV models parameters

The natural vortex shedding and VIV response can be qualitatively different depending on the Reynolds number at which it occurs. Typically, vortex shedding is described in terms of flow regimes for different Reynolds number intervals (Sumer & Fredsøe, 2006), where different regimes imply different boundary layer separation points and wake characteristics. The flow regime dictates, among others, the mean drag coefficient and the standard deviation of the lift coefficient, and the Strouhal number of the shedding (Schewe, 1983).

The Strouhal number, which defines the vortex shedding frequency in the case where there is no synchronization with the structural motion, is one of the main parameters of VIV engineering models, being shared by the three models employed here. The Strouhal number is highly dependent upon the Reynolds number of the flow (Schewe, 1983). A compilation of some canonical measurements of the Strouhal number for circular cylinders, together with the recommended values by different engineering standards are shown in Figure 3.6. The Strouhal number is shown as a function of the Reynolds number. Symbols are used for experiments and full-scale measurements, and lines for engineering models. Note that the data reproduced from Milford (1982) originally corresponds to five different data sets. The Reynolds number regimes are labelled in the figure, and their limits marked with vertical dashed line. The regimes of interest for wind turbine towers are the supercritical, upper-transition and transcritical regimes. Some experimental data show a notable increase in Strouhal values in the supercritical regimes, to values over $St = 0.4$. These values are not reported for full-scale structures, probably due to the effect of surface roughness and/or inflow

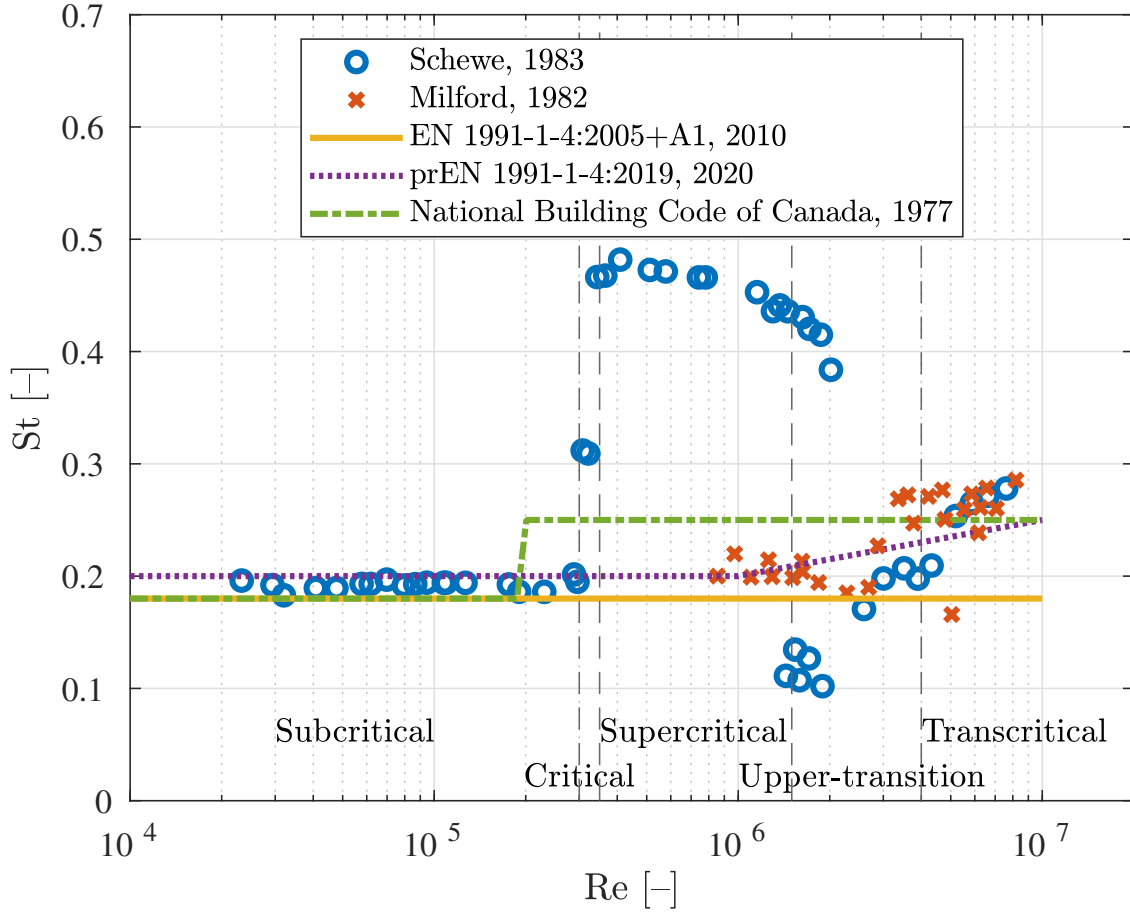


Figure 3.6: Strouhal number of circular cylinders as a function of Reynolds number. Circles and crosses correspond to experimental data as presented in Schewe (1983) and Milford (1982), respectively. Full, dotted and dash-dotted lines correspond to empirical approximations proposed in the standards EN 1991-1-4:2005+A1 (2010), prEN 1991-1-4:2019 (2020) and National Building Code of Canada (1977), respectively. The Reynolds-defined flow regimes are labelled, and their limits marked with vertical dashed lines, as per the definitions given in Sumer and Fredsøe (2006).

turbulence, both of which reduce the Strouhal number to about 0.2 in the supercritical regime, as shown by Achenbach and Heinecke (1981) and Cheung and Melbourne (1983b). None of the engineering standards presented recommend using these high values in the supercritical regime. Experimental results shown for the transcritical regime hint at an increasing Strouhal with Reynolds number. The increase in St for very high Re is reflected in the draft standard prEN 1991-1-4:2019 (2020). For the high Reynolds numbers relevant for the VIV response of wind turbine towers, there is limited information about the Strouhal number and the effect of roughness and turbulence, due to the difficulty and cost of performing wind tunnel experiments and full-scale tests. Note that Reynolds numbers over $Re = 10^7$ may be experienced by wind turbine towers.

The Reynolds number experienced by the tower is a function of the local wind speed and diameter at each tower station. Given the considered wind shear and that the tower is

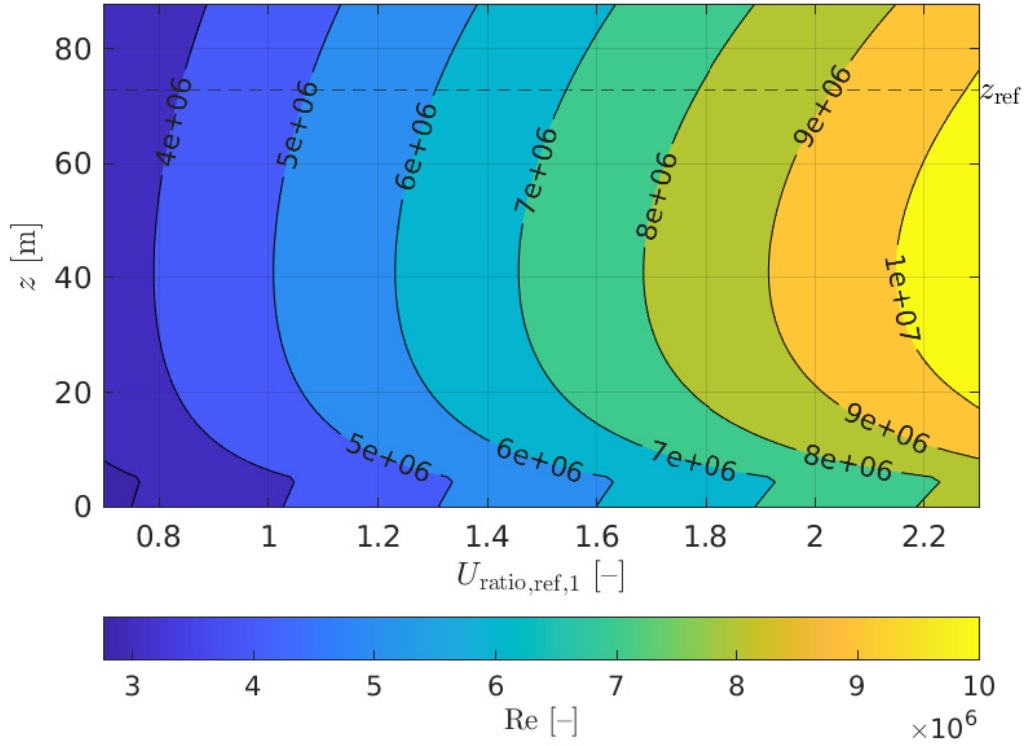


Figure 3.7: Reynolds number along the NREL 5-MW tower height for the range of velocity ratios studied. Marked in a dashed horizontal line is the reference height used for the definition of $U_{\text{ratio,ref},1}$.

tapered, the Reynolds number varies along the tower height. In Figure 3.7, the flow Reynolds number calculated at every station of the NREL 5-MW tower, and for all the velocity ratios considered, is shown. It can be observed that the Reynolds number ranges approximately between $\text{Re} = 3 \times 10^6$ and $\text{Re} = 1.1 \times 10^7$, which correspond to the upper-transition and transcritical regimes. As expected, the Reynolds number increases with the velocity ratio. For a given velocity ratio, the Reynolds is higher towards the centre of the tower, as a result of the combined effect of wind shear and tower diameter.

Similarly, Figure 3.8 presents the Strouhal number at every station of the NREL 5-MW tower, for all velocity ratios considered. The Strouhal number is defined here as a function of the Reynolds number, following the definition given in prEN 1991-1-4:2019 (2020), where it is considered to be $\text{St} = 0.2$ for $\text{Re} < 1 \times 10^6$, and then to linearly increase with the logarithm of the Reynolds number, up until $\text{St} = 0.25$ at $\text{Re} = 1 \times 10^7$, as depicted in Figure 3.6. For Reynolds numbers higher than $\text{Re} = 1 \times 10^7$, which are relevant for this tower as shown in Figure 3.7, the formulation is here extended such that $\text{St} = 0.25$ for $\text{Re} \geq 1 \times 10^7$. As expected, the behaviour of the Strouhal number with the velocity ratio and tower height is very similar to that of the Reynolds number shown in Figure 3.7. The Strouhal number ranges approximately between 0.23 and 0.25, showing variations of almost 10 % due to changes in

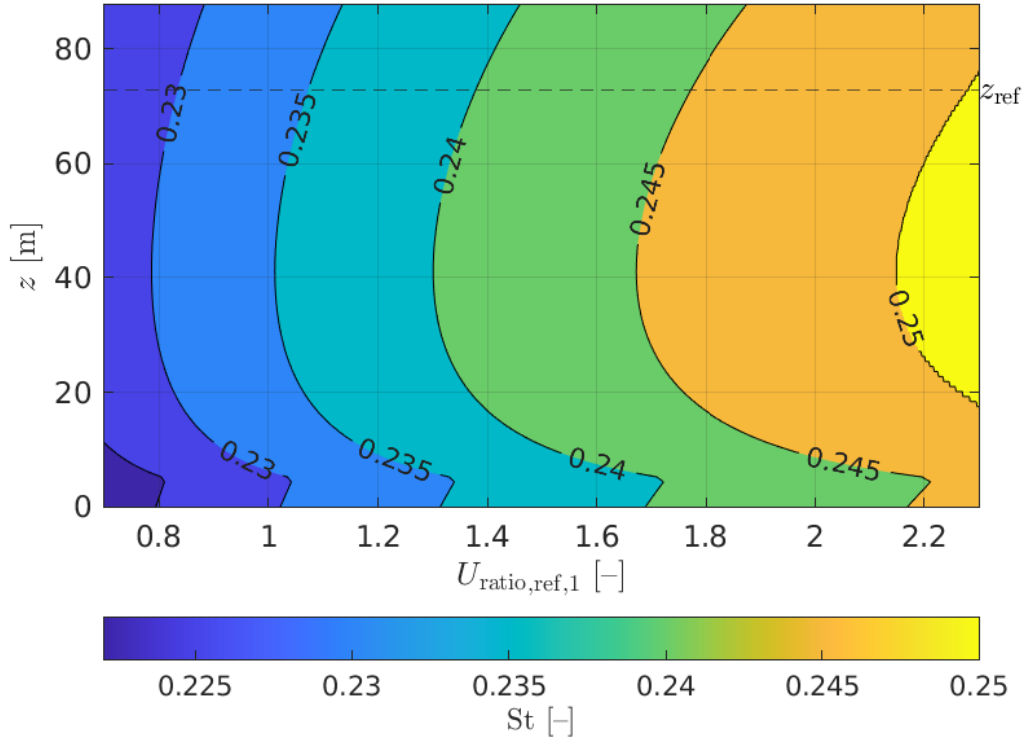


Figure 3.8: Strouhal number along the NREL 5-MW tower height for the range of velocity ratios studied. Marked in a dashed horizontal line is the reference height used for the definition of $U_{\text{ratio,ref},1}$.

the Reynolds number.

Regarding the model-specific parameters, the Type A vortex-induced load formulation was presented in Section 3.1.2, where it is seen to depend exclusively on the synthetic lift coefficient time series $c_l(z, t; \alpha)$. The synthesis is performed as defined in Section 3.1.2, based on the presented formulation of the lift coefficient cross-power spectral density (CPSD). All the parameters relevant to said CPSD were established from relevant literature considerations. The case specific parameters were the Strouhal number, just presented above, the lift coefficient standard deviation, which for the case of zero turbulence intensity employed here is $\sigma_{c_l} = 0.06$, and the bandwidth parameter, which is $B = B_0 = 0.1$ for $I_u = 0$. Once these parameters are defined, the c_l time series can be generated, and the modal force computed. An example of the one-point lift coefficient PSD at the reference height, $S_{c_l}(z_{\text{ref}})$, is shown in Figure 3.9. This PSD is obtained in two ways; using the analytical formulation by B. J. Vickery and Clark (1972), and obtained as the ensemble average of the PSDs calculated from the generated synthetic lift coefficients. The PSD of the lift coefficient time series has been calculated with Welch's method, using 8 segments with 50 % overlap, windowed with a Hamming function. After a preliminary sensitivity analysis to the number of realizations of the stochastic process, a value of $N_\alpha = 12$ has been chosen. The agreement is seen to be good, validating the lift

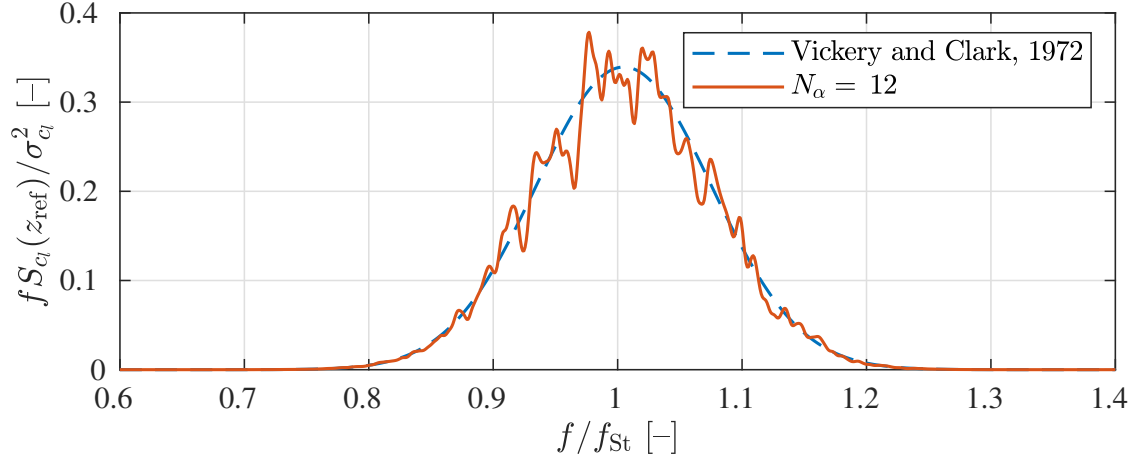


Figure 3.9: Dimensionless PSD of the lift coefficient according to the Type A model, evaluated at the reference height z_{ref} . The analytical expression from B. J. Vickery and Clark (1972) is compared to the PSD obtained for N_{α} synthetic realizations of the c_l stochastic process.

coefficient synthesis employed.

The Type B vortex-induced loading formulation, summarized in Section 3.1.2, has three parts: a stochastic load (the same as in Type A), a linear negative damping term, and a nonlinear positive damping term. The aerodynamic damping depends on $k_{a,j}$ and $G_{a,j}$, defined in Section 3.1.2, which are a function of the reference modal velocity ratio $U_{\text{ratio},j}$ (and therefore the Strouhal number), Reynolds number, turbulence intensity, tower geometry and tower modal properties. All of these parameters have been defined in the previous sections, and therefore the aerodynamic damping parameters can be evaluated.

The aerodynamic modal damping parameters, evaluated for all the studied velocities, are shown in Figure 3.10. Although, by its definition, the sectional aerodynamic damping, $k_a(z)$, is maximum at a local velocity ratio of one, its modal counterpart, $k_{a,1}$, is seen to have its maximum at approximately $U_{\text{ratio,ref},1} = 1.2$. This is a consequence of the different contributions to the modal aerodynamic damping of different sections, due to the wind shear, tower taper and mode shape. The definition of $G_{a,j}$ was shown in Section 3.1.2 to be analogous to $k_{a,j}$, except for an additional multiplication by $\Psi_j^2(z)$ inside the mode-projection integral along the tower height, and another multiplication by the parameter G , which is considered a constant for each mode. Thus, $G_{a,j}$ is expected to have a similar shape to $k_{a,j}$, but with a different amplitude.

A good indicator of whether the structure will undergo large-amplitude vibrations is to compare the modal structural damping ratio, $\zeta_{s,j}$, with the modal negative aerodynamic damping ratio, defined as $\zeta_{a,j} = \frac{k_{a,j}}{2m_j\omega_j}$. This comparison is shown in Figure 3.11, where both of the chosen structural damping values are shown. The modal aerodynamic damping ratio increases in the explicitly predefined range, although the shape is smoothed around its peak and tails due to the lengthwise integrals involved in obtaining the parameters. In the velocity range where the structural damping is lower than the aerodynamic damping, the structure

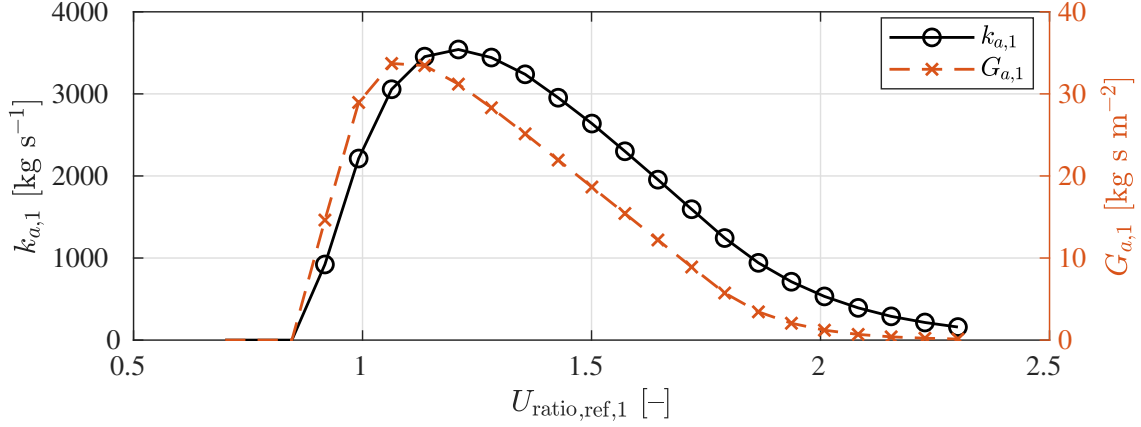


Figure 3.10: Type B damping parameters, specifically: linear negative aerodynamic modal damping parameter, $k_{a,j}$, and cubic positive aerodynamic modal damping parameter, $G_{a,j}$, for mode $j = 1$, as a function of the velocity ratio.

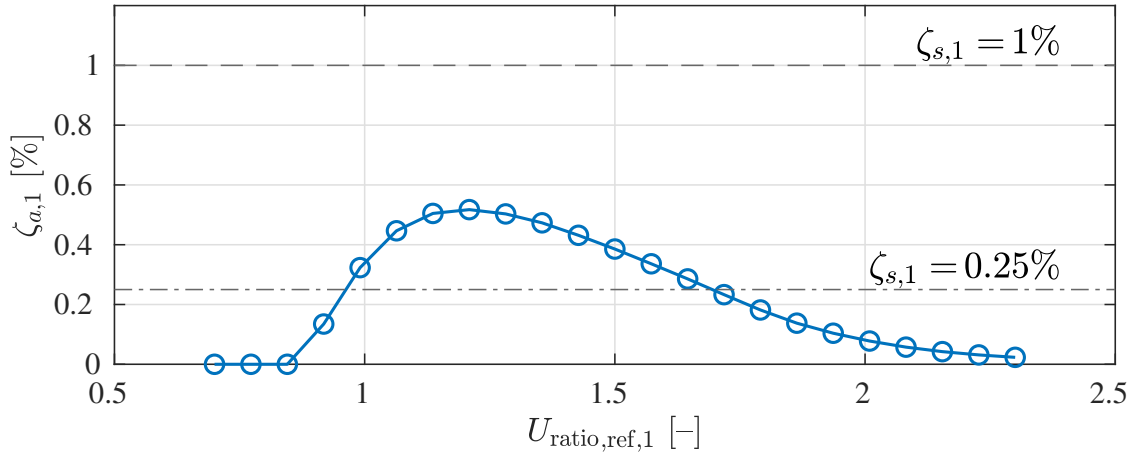


Figure 3.11: Type B modal linear aerodynamic damping ratio for mode 1, $\zeta_{a,1} = k_{a,1}/(2m_1\omega_1)$, as a function of the velocity ratio, together with the two representative values for the structural damping ratio, $\zeta_{s,1}$.

is expected to undergo large-amplitude vibrations. Note that a positive $\zeta_{a,1}$ value implies a negative aerodynamic damping. It can also be inferred that for the high damping case the structure is not expected to undergo large-amplitude vibrations, as the structural damping is notably higher than the aerodynamic damping.

For the Type C model, the modal loading was shown in Section 3.1.2 to depend upon two matrices and the wake generalized coordinate. The matrices are functions of the mode shape, tower geometry, and inflow properties, and of two parameters, the vortex-shedding lift coefficient for the stationary tower, $c_{l,0}$, and the stall parameter value, H_1 . To obtain the modal wake variable, l_1 , an additional differential equation has to be solved, coupled to the structural modal equation. The parameters involved in this wake oscillator equation are the Strouhal number, the van der Pol damping parameter, ε , and the velocity coupling constant, β .

In this application, the wake variable mode shape is prescribed to be the same as the modal shape of the tower, $\Phi_1(z) = \Psi_1(z)$, as proposed, among others, by Keber (2012), Skop and Balasubramanian (1997), Skop and Griffin (1975), and Wang et al. (2016a). The stationary lift coefficient is here evaluated as $c_{l,0} = 0.3$, as commonly used in the literature (EN 1991-1-4:2005+A1, 2010; Facchinetti et al., 2004). This value is based on experimental data for circular cylinders at high Reynolds numbers (Blevins, 2001). The Strouhal number is evaluated as discussed earlier in this section. Thus, the parameters remaining to be evaluated are H_1 , ε and β .

The three free parameters are selected to closely approximate the response obtained with the Type B model across the studied velocity range. Parameter fitting can be performed using optimization techniques, where the objective function to be minimized may be the difference between models in terms of maximum displacement or standard deviation. This can be done at the velocity of maximum response or across all velocities using a least-squares approach. Similar methodologies have been employed to calibrate wake oscillator models to experimental data, as shown by Kurushina et al. (2018), Rigo et al. (2022b), and K. Xu et al. (2015). It is important to acknowledge that due to the qualitative differences between the models, achieving an exact match between the Type B and Type C models is neither feasible nor necessary. For instance, even if the maximum displacement aligns well, the time series of the two models may differ significantly, particularly because the Type C model is deterministic, while the Type B model incorporates stochastic forcing. In this study, the parameters are manually selected for simplicity and computational efficiency. The selection process focuses on achieving a reasonable match in maximum displacement and capturing the overall response shape within the lock-in range. Preliminary sensitivity analyses of the Type C parameters' influence on the response were conducted, as detailed in Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al. (2024), where the three parameters are varied and the results compared with Type A and Type B results for a high damping case.

The shape of the maximum structural response as a function of the velocity ratio predicted by the Type C model can be described with three geometrical indicators (Rigo et al., 2022b); an indicator of the lock-in range width, an indicator of the magnitude of the maximum response, and an indicator of the position of the maximum response within the lock-in range. The effect of each of the three free parameters on the response shape is as follows. The stall parameter, H_1 , acts as a damping constant, which adds damping when it takes negative values. Thus, the smaller the value of H_1 , the smaller the lock-in range width and the smaller the magnitude of the response. The effect of H_1 on the relative position of the maximum response within the lock-in range is not apparent to the authors. The van der Pol damping parameter, ε , controls the nonlinearity of the wake oscillator. Larger values of ε were seen in Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al. (2024) to decrease the magnitude of the response and slightly move the position of its maximum towards higher velocity ratios, and had no effect on the lock-in range width. Regarding the velocity coupling constant, β , determines the coupling between the structural motion and the wake oscillator. Larger values of β therefore increase the width of the lock-in range. As a consequence, the maximum displacement also increases if its position is skewed towards the high velocity end of the lock-in range.

In the present fitting of the Type C parameters to the Type B response, the focus is put on

matching the magnitude of the maximum response, as well as keeping a balance between matching the position of the maximum response and the lock-in range width, as these two considerations are found to be mutually exclusive.

As the response is evaluated for two structural damping values, two distinct sets of parameters for the Type C model are selected. These parameters are calibrated to match the response of the Type B model separately for each damping value. The parameter thus obtained are shown in Table 3.2. While earlier implementations of wake oscillator models, such as those by Skop and Balasubramanian (1997) and Srinil et al. (2009), have included dependencies of the wake oscillator parameters on structural damping, it is preferable for the model to be independent of such structural properties. Ideally, the wake oscillator parameters should depend solely on the tower geometry and inflow conditions, ensuring a more general and robust formulation.

	$\zeta_{s,1} = 0.25 \%$	$\zeta_{s,1} = 1 \%$
$H_1 [-]$	-0.555	-33.3
$\varepsilon [-]$	0.075	0.3
$\beta [s^{-1}]$	99.5	298

Table 3.2: Type C model parameters chosen to match the response of the Type B model, for each structural damping ratio value considered.

The results obtained with the Type C model fitted parameters are presented next, in Section 3.3.3, alongside those obtained with the Type A and Type B models.

3.3.3 Results comparison

The structural modal equation, and the wake oscillator equation for the Type C model, are numerically solved in time using the `ode45` Matlab® function (Shampine & Reichelt, 1997). The simulation time is set to $t_{\text{sim}} = 1200$ s, which is twice the minimum required duration for wind turbine load calculations for design and certification according to IEC 61400-1 Ed.3.1 (2010). Long time series are necessary due to the slow growth of vibrations in some cases. For the Type A and Type B models, which include a stochastic component, twelve realizations of the stochastic process are performed. This is done to improve the statistical characterization of the response, which is computed as the ensemble average of the realizations. The time step, Δt , used in the simulations should be small enough to accurately capture the fastest oscillation of the response. Since only the first mode is considered, the fastest oscillation is of the order of the first modal frequency of the structure, $f_{s,1}$. According to the Nyquist–Shannon sampling theorem, at least two steps per cycle are required. Therefore, the time step should be $\Delta t < 1/(2f_{s,1}) = 0.561$ s. In this work, a time step of $\Delta t = 0.1$ s is chosen. This is sufficient to capture even the third harmonics of the excitation at approximately $3f_{s,1}$, which can be introduced by the nonlinearity of the vortex-induced load, although it will be shown that these harmonics do not significantly affect the response. For the calculation of statistical values (maximum values, standard deviations, and PSD functions), the first $t_{\text{trans}} = 600$ s of the time series are removed to avoid the transient behaviour, which is related to the initial conditions

and VIV development. For Type A and Type B, these statistical values are obtained as their ensemble average over the $N_\alpha = 12$ realizations. The initial condition is set as a dimensionless modal displacement, $\tilde{p}_j(t=0) = \tilde{p}_{j,0} = 0.02$, where $\tilde{p}_j = p_j/D_{\text{ref}}$.

The tower vibration response obtained with the three VIV models is compared next. In Figure 3.12, the maximum dimensionless tower top displacement, $\max(\tilde{w}_t)$, where $\tilde{w}_t = w_t/D_{\text{ref}}$ and $w_t(t) = w(H, t)$, is shown as a function of the reference velocity ratio for the first mode, $U_{\text{ratio,ref},1}$. The results for the low and high structural damping cases are shown in the top and bottom panels respectively. The results are shown for the three VIV models, and for Type A and Type B, the result corresponds to the ensemble average of the $N_\alpha = 12$ realizations of the stochastic process simulated. The Type C model parameters were chosen to obtain a good match between the maximum amplitude reached by the Type B model, and it can be seen that the match is indeed good for both damping values. Nevertheless, the overall shape of the curve differs significantly; for the low damping case, the Type B model predicts higher amplitudes within the whole lock-in range, whereas for Type C the curve increases more slowly for low velocities, and the lock-in range ends at a lower velocity. For the high damping case, the behaviour for low velocity ratios is almost identical between Type B and Type C, until the maximum value is reached, but then for Type C the response drops at a lower velocity and more abruptly. The velocity ratio at which the maximum response is reached is about $U_{\text{ratio,ref},1} = 1.3$ for the low damping case, and about $U_{\text{ratio,ref},1} = 1.1$ for the high damping case, for both Type B and Type C models. With the low damping value, the displacement inside the lock-in range is about 50 times higher than with the high damping value. Regarding the Type A results, for the low damping case the maximum amplitude is negligible compared to that of the other models, as in this case the negative aerodynamic damping dominates the response. On the other hand, for the high damping case, the response is quite similar in shape and magnitude to that of the Type B model, while having a maximum response about 30 % lower.

Similarly, in Figure 3.13, the standard deviation of the dimensionless tower top displacement, $\sigma_{\tilde{w}_t}$, is shown as a function of the velocity ratio. For the low damping case, the behaviour of the different models is almost identical to that of the maximum displacement shown in Figure 3.12. On the other hand, for the high damping case, it can be readily observed that the good agreement of the Type C model maximum displacement to the Type B case does not extend to the standard deviation of the response, being the standard deviation of the Type C case notably higher than that of the Type B case. The velocity ratios at which the maximum standard deviation is reached are approximately the same as for the maximum displacement presented in Figure 3.12, for both damping ratio values and both Type B and Type C models. With the Type C model, the maximum standard deviation inside the lock-in range is about 50 times higher for the low damping case than for the high damping case, as was the case for the maximum displacement. On the other hand, for the Type B model this difference increases to about 100 times. The relationship between the maximum displacement and standard deviation can be characterized by the peak factor, $k_p = \frac{\max(w)}{\sigma_w}$, which takes smaller values for harmonic time series, and higher values for stochastic time series. This is reflected in the large mismatch in standard deviation between the Type C and Type B models for the high damping case, as the Type B model behaves in a much more stochastic

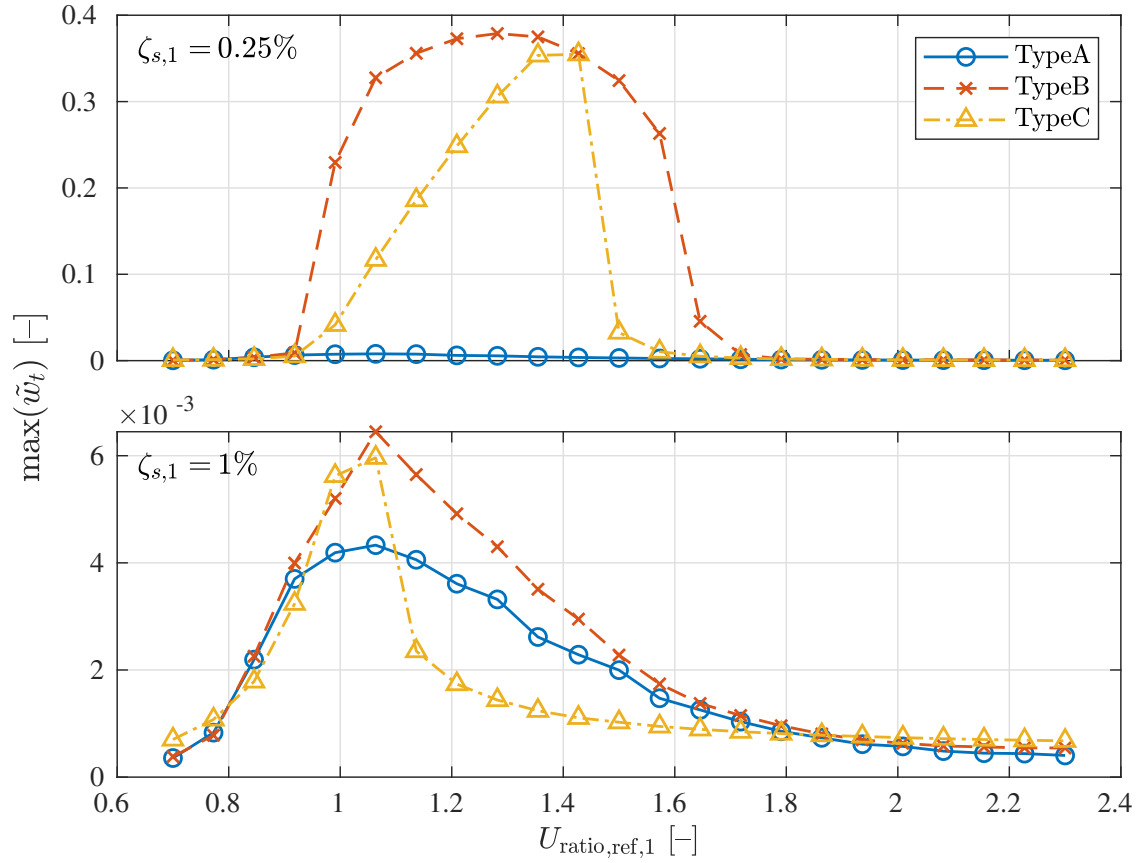


Figure 3.12: Maximum dimensionless tower top displacement as a function of the ratio between the reference inflow and critical wind speeds, comparing the three VIV models, for (top) low structural damping and (bottom) high structural damping cases. For Type A and Type B, the ensemble average of the maximum dimensionless displacement is shown.

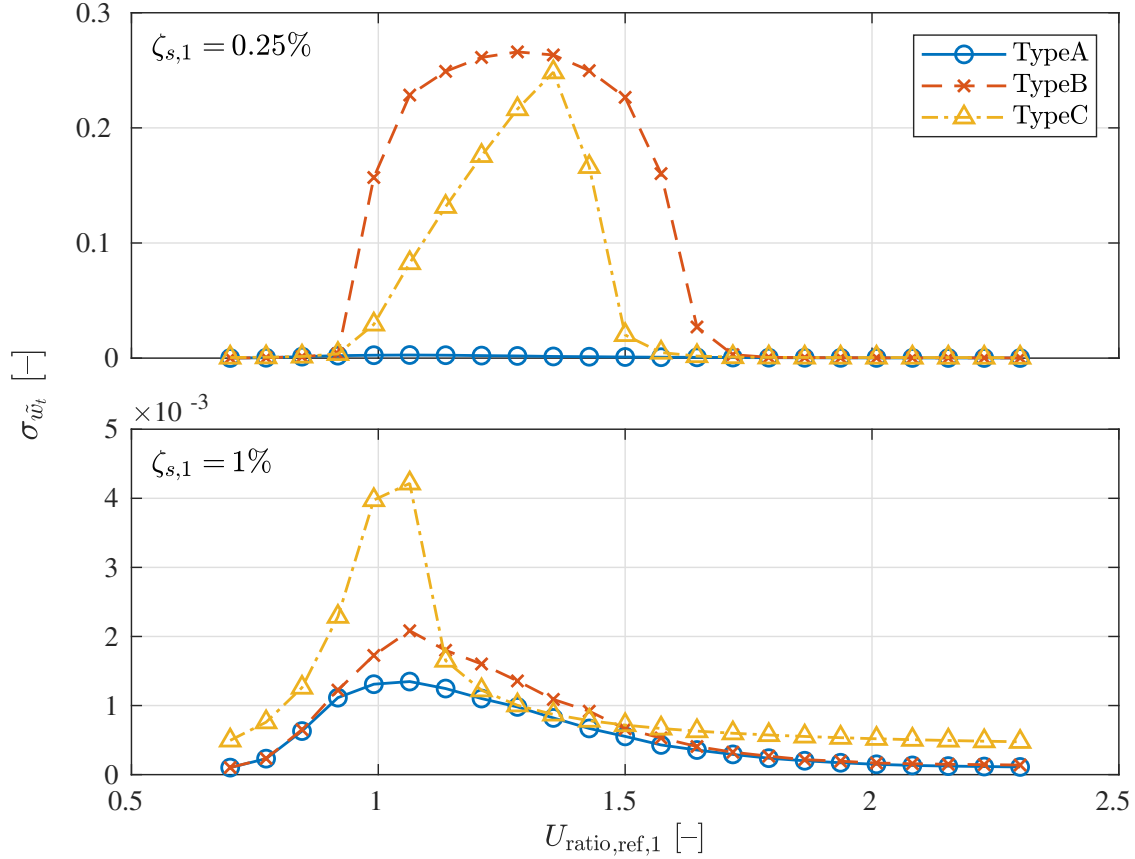


Figure 3.13: Standard deviation of the dimensionless tower top displacement as a function of the ratio between the reference inflow and critical wind speeds, comparing the three VIV models, for (top) low structural damping and (bottom) high structural damping cases. For Type A and Type B, the ensemble average of the dimensionless displacement standard deviation is shown.

way due to the stochastic forcing term. The Type A model behaves as for the maximum displacement shown in Figure 3.12; for the low damping case the standard deviation is much smaller to that of the other models, and for the high damping case, the standard deviation is very similar to that of the Type B model, albeit with a slightly lower maximum value.

The power spectral density (PSD) of the simulated tower top dimensionless displacement, $S_{\tilde{w}_t}$, is shown in Figure 3.14. The PSD is calculated with Welch's method, using 8 segments with 50 % overlap, windowed with a Hamming function. The results are shown for the three VIV models, and for the low (top panel) and high (bottom panel) structural damping cases. All results correspond to the velocity ratio $U_{\text{ratio,ref},1} = 1.28$, which corresponds approximately to the velocity for maximum response for the low damping case, as can be seen in Figure 3.12. For the low damping case (top panel), the PSD of the Type B model is very similar to that of the Type C model, being comprised of a large peak at the resonant frequency $f = f_{s,1}$. On the other hand, for the high damping case (bottom panel), the PSD of the Type B model is now very similar to that of the Type A model, for which the PSD also shows a peak at the resonant frequency, and also some energy spread over a wider range of frequencies around the natural

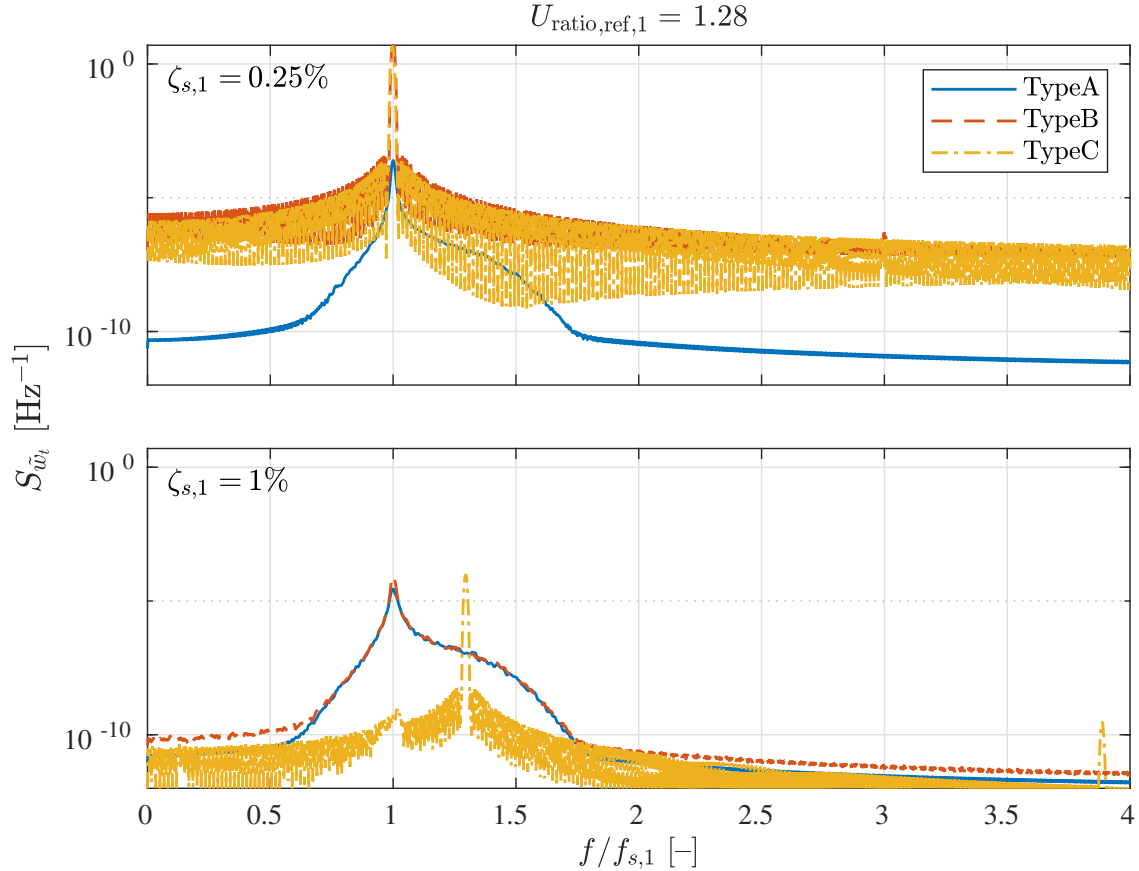


Figure 3.14: PSD of the dimensionless tower top displacement, as a function of the ratio between frequency and the natural frequency of the first tower mode. Results are shown for the first mode velocity ratio $U_{\text{ratio,ref},1} = 1.28$, comparing the three VIV models, for (top) low structural damping and (bottom) high structural damping cases.

vortex shedding frequency, which is, at the reference height, $f_{\text{Stref}}/f_{s,1} = U_{\text{ratio,ref},1} = 1.28$. In the high damping case, the peak of the Type C model PSD is seen to appear prominently at f_{Stref} , with a much smaller peak appearing close to the resonant frequency. A third peak is noticeable in the Type C high damping PSD, at $3f_{\text{Stref}}$, due to the nonlinear loading introduced by the van de Pol wake oscillator.

Overall, the Type B model has been shown to behave similarly to the Type C model for the low structural damping case, and very similarly to the Type A model when the damping value is large. Nevertheless, there are notable differences between the Type B and Type C models regarding the shape and extent of the lock-in range.

3.4 Conclusions and future work

This chapter presented a comparison of three vortex-induced vibration (VIV) time-domain engineering models –a Type A (forced system) model, a Type B (fluidelastic) model, and a Type C (coupled system) model–, applied to the NREL 5-MW wind turbine tower. The

study aimed to evaluate the suitability of these models for predicting VIV responses and to provide insights for their application in wind turbine design and certification processes.

The chapter details the structural modelling of the tower using a modal decomposition approach, where the tower is treated as a cantilevered Euler-Bernoulli beam. Modal quantities such as the mode shape, modal damping ratio, and natural frequency are defined, and the external loading is projected onto the modal coordinates. Below are the key conclusions drawn from this work, for each vortex-induced loading model considered:

Type A (Forced System Model) This model, which relies solely on stochastic vortex shedding forces, is suitable for structures with high structural damping where motion-induced effects are negligible. However, it underestimates the response for low damping scenarios, as it does not account for fluid-structure interactions.

Type B (Fluidelastic System Model) By incorporating motion-dependent forces through linear negative and nonlinear positive damping terms, this model captures the self-excited and self-limited nature of VIV. It shows good agreement with full-scale data (e.g., the Aachen chimney case) and is effective for structures with both high and low structural damping values.

Type C (Coupled System Model) This model goes further by coupling the tower's structural dynamics with wake oscillator dynamics (modelled as a van der Pol oscillator) to capture intermodal interactions. While it can be tuned to match the Type B model in predicting maximum displacements, it exhibits differences in the lock-in range and standard deviation of responses, especially for high damping cases. Its deterministic nature contrasts with the stochastic forcing in Type B, leading to distinct spectral characteristics.

The structural damping ratio significantly influences the VIV response. For low damping ($\zeta_{s,1} = 0.25\%$), the Type B and Type C models predict large-amplitude vibrations, while the Type A model fails to capture this behaviour. For high damping ($\zeta_{s,1} = 1\%$), the Type A and Type B models yield similar results, whereas the Type C model, despite being tuned to match the Type B vibration amplitudes, shows a qualitatively different response, as revealed by the power spectral density (PSD) analysis. As showcased both in the Aachen chimney validation and the application to the NREL 5-MW wind turbine tower, small uncertainties in parameters like turbulence intensity or structural damping can radically change the magnitude and nature of the response. This depends on whether the system reaches a lock-in state, which is related to whether the negative aerodynamic damping outweighs the structural damping. The study underscores the importance of selecting an appropriate VIV model based on the structural damping and the desired level of detail in capturing fluid-structure interactions.

Possible future work could focus on extending the Type A model, to generate synthetic lift coefficient time series that capture the instantaneous characteristics of turbulent wind fields, improving its applicability to the typical time-domain aeroelastic load calculations as per IEC 61400-1 Ed.3.1 (2010). This improvement would also apply to the Type B model stochastic component. Alternatively, the Type C model could be extended to incorporate the effects of both inflow and wake turbulence, building on recent advancements in wake oscillator

formulations (Denoël, 2020; Mannini, 2020; Rigo et al., 2022a).

Additionally, these VIV models could be applied and adapted to other components of the wind turbine, such as the blades or the nacelle, to assess their susceptibility to VIV. Furthermore, both Type B and Type C models could be improved through the characterization of the coupling between multiple structural modes and vortex-induced loads to better capture complex VIV situations in wind turbines, such as when interaction between the components occurs. Full-scale or wind tunnel experiments should be conducted to validate the models under controlled conditions, particularly for high Reynolds numbers and varying turbulence levels.

The long-term fatigue implications of VIV could be explored by integrating time-domain simulations with fatigue damage models, and compared with other sources of fatigue. Such an analysis should include transient events, such as wind gusts or operational changes, to assess their impact on the fatigue life of the wind turbine.

In summary, this chapter provides a foundation for understanding and applying time-domain VIV engineering models to wind turbine towers. While each model has its strengths and limitations, the choice of model should align with the specific requirements of the design process, balancing accuracy, computational cost, and practical applicability. The conclusions point to the need for further refinement of these models, especially in terms of capturing the effects of inflow turbulence. Future research should focus on addressing the identified gaps to further enhance the predictive capabilities of these models. This work lays the groundwork for extending these methodologies to more complex scenarios, such as wind turbine blades or even full wind turbine systems.

Chapter 4

Blade vortex-induced vibrations: high-fidelity numerical simulations

This chapter presents high-fidelity numerical simulations of vortex-induced vibrations (VIV) in wind turbine airfoil sections. Occurrences of VIV in wind turbine blades have been observed in the field, and these events can lead to significant fatigue damage and affect installation and maintenance operations. The simulations presented in this chapter aim to contribute to the understanding of the blade VIV phenomenon and its numerical modelling.

It is crucial to have a good understanding of the capabilities and limitations of the different levels of simulation fidelity in characterizing VIV. To this end, the present work analyses the difference in vortex-shedding flow and VIV response of a wind turbine airfoil between a 2D case using the unsteady Reynolds-averaged Navier-Stokes (URANS) turbulence-modelling approach, and two 3D cases for extruded sections, with a spanwise length of one chord and twenty-one chords, using the delayed detached-eddy simulation (DDES) turbulence-modelling approach. The DDES 3D cases are a higher-fidelity approach but also much more computationally expensive. A similar analysis has been presented in the early work of Skrzypiński et al. (2014), but the short length of the time series they simulated meant that some of the vortex-shedding flow features were not observed, leading to a potentially inaccurate characterization of the Strouhal number for the 3D DDES case and possibly affecting the comparison of the VIV response between the two approaches. These potential shortcomings were replicated in the subsequent work by Lian et al. (2023a). The present work complements and extends those by Skrzypiński et al. (2014) and Lian et al. (2022, 2023a). In this work, the simulation of the static-airfoil vortex shedding for a longer time series and a more robust characterization of the VIV results have allowed for a novel comparison of the VIV response between 2D and 3D simulations. In Skrzypiński et al. (2014) and Lian et al. (2023a), the authors performed static 3D airfoil simulations with a length of 168 and about 260 dimensionless time units, respectively. For the present work, 1200 dimensionless time units are simulated for the static 3D one-chord case. This results in the detection of multiple intermittent flow regimes associated with different Strouhal number values, which was not possible with shorter simulation times. Furthermore, in Skrzypiński et al. (2014) the presented quantitative comparison between their 2D and 3D VIV simulations is based on the

maximum amplitude reached by the airfoil after a dimensionless time of 100 units, which is not sufficient time for the maximum amplitude to be reached, thus making this comparison very sensitive to the initial transient of the simulation. Similarly, in Lian et al. (2022) the VIV amplitude results are shown after 100 dimensionless time units, which again makes it difficult to perform any subsequent comparisons. In the present work, the VIV responses are compared using the initial amplitude growth rate, which is a robust metric of the VIV development even for relatively short time series (300 dimensionless time units in this work). This enables reproducible comparisons by other authors and methods. A notable result is the level of agreement found between these two simulation approaches for the VIV response inside the lock-in range. The results and methodology presented here may serve as a guide for future comparisons and for simulations where the simulation time is considerably limited.

Some of the results in this chapter have been previously published by the author in Fernandez-Aldama et al. (2025) and Fernandez-Aldama, Papadakis, Lopez-Garcia, Avila-Sanchez, and Riziotis (2024). For this thesis, both studies are combined and extended, providing a more complete and detailed analysis of the VIV behaviour of wind turbine airfoil sections using high-fidelity numerical simulations.

4.1 Methodology

In this section, the computational setup of the airfoil fluid-structure interaction (FSI) high-fidelity simulations performed is described. Each FSI simulation consists of two coupled elements, a CFD simulation, described in Section 4.1.2, and a structural simulation, described in Section 4.1.3.

The airfoil chosen for the analysis is the DU96-W-180, which has a relative thickness of 18 %, and was designed for wind turbine applications at the Delft University of Technology (Timmer & van Rooij, 2001). The VIV response of this particular airfoil has been previously analysed by Hu et al. (2021), Lian et al. (2022, 2023a, 2023b, 2024), and Skrzypiński et al. (2014).

The simulation solver employed is MaPFlow, developed at the Laboratory of Aerodynamics of the National Technical University of Athens (NTUA) (Diakakis, 2019; Papadakis, 2014). MaPFlow includes a finite volume compressible CFD solver, offering URANS and DDES turbulence modelling approaches. MaPFlow has been validated in code-to-code comparisons (Prospathopoulos et al., 2021; Sørensen et al., 2016), for the VIV problem of a circular cylinder (Papadakis et al., 2022), and for the vortex shedding behind the trailing edge of flatback airfoils (Manolesos & Papadakis, 2021; Papadakis & Manolesos, 2020; Papadakis et al., 2020). The CFD solver is 2nd order accurate in space and time, uses the Roe approximate Riemann solver for the convective fluxes and is equipped with Low Mach preconditioning.

4.1.1 Simulations overview

Simulations are performed for three configurations: a 2D airfoil, a 3D airfoil extruded one chord length in the spanwise direction (referred to as “3D-1c”), and a 3D airfoil extruded twenty-one chord lengths (referred to as “3D-21c”). All simulations are performed with a constant and uniform velocity inflow, striking the airfoil at an angle of attack of $\alpha = 90^\circ$.

The lock-in behaviour of the airfoil is influenced by the ratio between the inflow velocity and the structural natural frequency. To capture the lock-in range, the inflow velocity is varied, resulting in a Reynolds number range of $Re \in [1.4 \times 10^6, 2.6 \times 10^6]$.

The choice of aspect ratio is driven by the need to balance computational cost and accuracy; different aspect ratios can be used for different purposes, but it is crucial to understand the implications of the choice. Lower-fidelity simulations, such as 2D URANS, are computationally cheaper and can provide valuable insights into the general behaviour of VIV. However, they have limitations in accurately capturing three-dimensional effects and flow separation in some cases. Higher-fidelity simulations, such as 3D DDES, offer a more detailed and accurate representation of the phenomena but at a significantly higher computational cost. The 3D DDES short aspect ratio simulations (case 3D-1c) provide detailed resolution of the flow structures near the airfoil, but these might be affected by the lateral boundaries condition. Analyses of the effect of the spanwise length on the airfoil vortex shedding can be found in Lian et al. (2023a) and Skrzypiński et al. (2014), where in both of these works a one-chord spanwise length is deemed a good compromise and used for subsequent VIV analyses. On the other hand, the long aspect ratio simulations (case 3D-21c) better represent typical full-blade simulations (Grinderslev et al., 2022; Heinz et al., 2016; Horcas, Sørensen, et al., 2022), which may be more relevant for wind turbine applications, and also reduce the influence of the spanwise boundaries condition. The value of twenty-one has been chosen as an approximate value of the NREL 5MW (Jonkman et al., 2009) blade aspect ratio, using as reference chord the one at about the 70 % span position. Note that larger, more modern wind turbine blades may have even higher aspect ratio values. The similarities and differences between the results obtained with the different simulation approaches are analysed to provide insights into the applicability and limitations of each method.

A summary of the simulations performed, categorized by aspect ratio configuration, is provided in Table 4.1. The total number of simulations includes various inflow velocity cases, along with an additional static geometry simulation (CFD-only) for each aspect ratio configuration. The table also includes the number of cells in the CFD meshes, highlighting the difference in computational cost between the 2D and 3D cases, which will be discussed further in Section 4.2.2. Note that the 3D-21c case has the same number of cells as the 3D-1c case, at the expense of a coarser spanwise resolution. The meshes and other details on the CFD setup will be described in detail next, in Section 4.1.2.

	Aspect ratio (AR)	Turbulence modelling approach	Number of mesh cells	Number of simulations
2D	-	URANS	32 768	16
3D-1c	1	DDES	1.26×10^7	8
3D-21c	21	DDES	1.26×10^7	10

Table 4.1: List of simulations performed. For each configuration there is one static CFD simulation and the rest are FSI simulations at different inflow wind speeds.

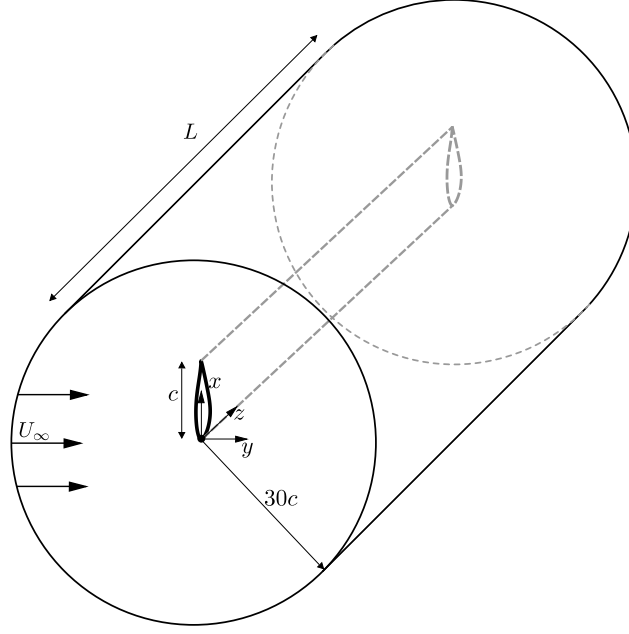


Figure 4.1: Sketch of the computational domain for the 3D simulations, and axes system. Not to scale.

4.1.2 CFD setup

As already mentioned, two different turbulence modelling approaches are employed in the CFD simulations. One approach is using the Unsteady Reynolds-Averaged Navier-Stokes (URANS) equations, with the k - ω SST two-equation turbulence model (Menter, 1994). This is the approach chosen for the 2D simulations. For the 3D simulations (3D-1c and 3D-21c), the Delayed Detached Eddy Simulation (DDES) approach is employed, also with the k - ω SST turbulence model (Gritskevich et al., 2012). The boundary layer is assumed to be fully turbulent in all simulations.

A sketch of the computational domain used in the simulations is shown in Figure 4.1. This computational domain consists of a circular disk in the x - y plane, centred at the airfoil leading edge, and the x -direction aligned with the airfoil chord line, pointing towards the trailing edge. The y -axis is perpendicular to the chord, pointing downstream. The disk has a radius of $30c$, where c is the airfoil chord length. In the case of the 3D simulations, the computational domain is the same but extruded in the spanwise (z) direction, becoming a cylinder of height L in the z -direction, where $L = 1c$ for the 3D-1c case and $L = 21c$ for the 3D-21c case. In the sketch, U_∞ is a representation of the incoming flow velocity, which is perpendicular to the x - z plane, being the airfoil thus at an angle of attack $\alpha = 90^\circ$.

The edge of the disk in the 2D case, and the lateral surface of the cylinder in the 3D cases, both act as mass flow inlet-outlet boundary conditions. The airfoil surface boundary condition is set to no-slip wall. For the 3D cases, a periodic boundary condition is set to the two end faces of the cylindrical domain. It is worth noting that the computational domain should be as big as possible to reduce the influence of the inlet-outlet boundary condition on the results. The effect of extending the outer domain radius to 50 chords and 100 chords has been checked

(with the consequent increase in mesh radial cell count), and no evident differences on the loads or VIV response were found. The domain size of about 30 chords is considered standard practice for airfoils in attached flow conditions (Golmirzaee & Wood, 2024), and is also widely used for bluff body simulations (e.g. Martín-Alcántara et al. (2023) and Skrzypiński et al. (2014)). Note that, as mentioned before, the solver employs the method of Riemann invariants at the far field so that the freestream conditions are more accurately prescribed. The justification for the choice of the domain size derives from the usual compromise between accuracy and computational cost; where the value of 30 chords is deemed appropriate, and is comparable to those in the works by Skrzypiński et al. (2014) and Lian et al. (2023a), which serve as a basis for the present simulation setup. The chosen domain size has been checked to induce a blockage ratio under 2%, which is considered small enough (West & Apelt, 1982).

For the 2D simulations, the domain is discretized using a structured O-type mesh. The number of grid points over the airfoil is 256, being more densely distributed around the leading and trailing edges. In the far field, i.e. at the disk edge, the 256 grid points are evenly distributed. In the radial direction, the mesh is discretized into 128 grid points, concentrating them close to the airfoil surface to obtain a first layer height which ensures a value of $y^+ < 1$. The total number of cells in the mesh of the 2D simulations is thus 32768. For the 3D simulations, the domain is also discretized using a structured O-type mesh, with the same number of mesh cells along the airfoil surface as in the 2D case, that is 256. In the radial direction a finer discretization of 384 grid points is employed in the 3D cases with respect to the 2D case, consequently a first layer height of $y^+ < 1$ is also ensured. In the spanwise direction, the mesh is extruded and discretized into 128 equispaced grid points. The total number of cells in the mesh of the 3D simulations is thus about 12.6 million for both 3D-1c and 3D-21c cases. The 3D-1c mesh is 21 times denser in the spanwise direction than the 3D-21c mesh, having 128 and 6.10 cells per spanwise chord length respectively, as can be observed by comparing Figure 4.2(b) and Figure 4.2(c). The structured meshes files read by MaPFlow have been created with the publicly available code structAirfoilMesher (Diakakis, 2023). A close-up view of the employed meshes is shown in Figure 4.2. Figure 4.2(a) presents a detailed view of the 2D mesh around the airfoil, while Figure 4.2(b) and Figure 4.2(c) display details of the 3D meshes, including an oblique view of the mesh on the airfoil surface and a cross-sectional slice perpendicular to the spanwise direction.

For DDES simulations, it is desirable to have a mesh refinement leading to quasi-isotropic cells all along the LES-resolved wake. This is sometimes done in vortex shedding and VIV simulations, via multiblock or unstructured meshing techniques, but only for low Reynolds number flows, as the cell count quickly explodes. Such wake-refined meshes have been used e.g. by Vlastos et al. (2024) for a Reynolds number of 2.6×10^4 , or by A. H. Lee et al. (2014) for a Reynolds number of 5000. Typically, meshes for vortex shedding and VIV at high Reynolds numbers focus on having a good refinement in the boundary layer to accurately capture the fluctuating separation point, and also in the near wake behind the body to capture the vortex formation and shedding. When using hybrid RANS-LES modelling (like the DDES model in the present work), the resulting meshes are typical RANS meshes in the boundary layer (as for an attached flow condition), and typical LES quasi-isotropic cells in the near wake only (Ebstrup et al., 2024; Lian et al., 2023a; Skrzypiński et al., 2014). When simulating slender bodies, the typical approach is to extrude this kind of mesh in the spanwise direction.

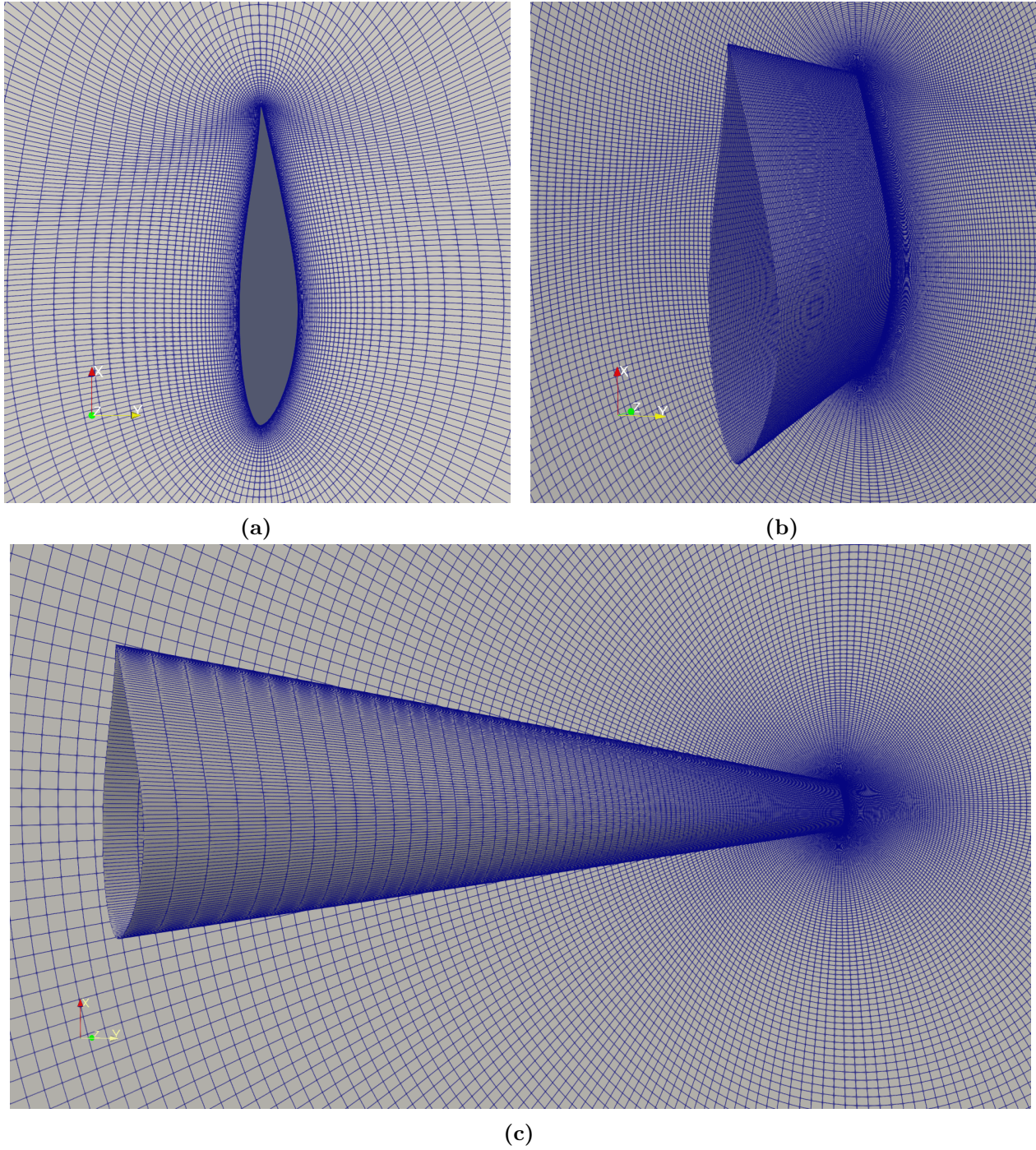


Figure 4.2: Visualization of the CFD meshes used in the simulations. (a) Close-up view of the mesh around the airfoil for the 2D configuration. (b) Close-up view of the surface mesh and a cross-sectional slice at one end of the mesh for the 3D-1c (aspect ratio one) configuration. (c) Close-up view of the surface mesh and a cross-sectional slice at one end of the mesh for the 3D-21c (aspect ratio twenty-one) configuration.

In practice, this often leads to a degradation of the isotropy of the near wake cells due to elongation in the spanwise direction, for the sake of keeping an acceptable the computational cost (Grinderslev et al., 2022; Heinz et al., 2016; Horcas, Sørensen, et al., 2022; Horcas et al., 2020; Horcas, Madsen, et al., 2022; Pirrung et al., 2024). A noteworthy exception can be found in Grinderslev et al. (2023), where much finer spanwise meshes were tested, at a very high computational cost, and significant differences in the results were found for some of the VIV cases studied. Although multiblock or unstructured meshes would probably be preferable in this case, either to improve accuracy or to reduce computational cost, in the present work it was decided to employ a similar mesh topology to those from closely related publications (Lian et al., 2023a; Skrzypiński et al., 2014) for comparability and simplicity. The mesh is refined as much as possible at the near wake considering the computational cost of the simulations.

The present 3D-1c mesh has cells with an aspect ratio under 3 in the near wake (up to one chord length downstream), except very close to the airfoil surface, where RANS modelling is expected. Past the near wake, the aspect ratio of the cells increases due to the coarsening of the mesh in the x - y plane, while the spanwise density remains constant. Having a low resolution in the far wake leads to only the largest eddies being resolved, and the smaller ones being modelled, and ultimately to even the large eddies being dissipated. Nevertheless, it is not expected that this will significantly affect the vortex shedding and VIV mechanisms, as long as the region in which recirculation may reach the airfoil (i.e., the near wake) is well captured, and the transition between the near and far wake provides correct large-scale dynamics (Spalart, 2001).

Regarding the 3D-21c case, it is noted that the spanwise density of this case is similar to that from recent simulations of full-blade geometries (Grinderslev et al., 2022; Horcas, Sørensen, et al., 2022). The 3D-21c mesh quality is quite different to the 3D-1c case; given the coarser spanwise resolution, the aspect ratio of the mesh cells is higher in the near wake. The aspect ratio is bounded between 3 and 30 in the near wake (except very close to the airfoil), and only becomes smaller than 10 after about 0.3 chords downstream. On the other hand, the cells remain quasi-isotropic for most of the far wake. Thus, the cell aspect ratio is small enough to avoid large numerical errors typical of aspect ratios over 100, but it reflects that the mesh is finer than strictly required in the x and y directions from an LES standpoint. Nonetheless, this refinement is necessary to resolve the near-wall region in URANS mode. Additionally, this anisotropy may modify shear production and dissipation, potentially biasing wake vortical structures and unsteady loads. The coarsening of the 3D-21c mesh in the spanwise direction also reduces the LES-resolved content of the wake, promoting an effective URANS behaviour, as the LES length scale is defined by $\Delta = \max(\Delta x, \Delta y, \Delta z)$ and can thus be determined by the spanwise cell size. Results for 3D-21c should therefore be interpreted with these limitations in mind. Note that increasing the spanwise resolution of the 3D-21c mesh to the level of the 3D-1c mesh would require about 21 times as many cells, and thus about 21 times the computational resources and time. Dedicated spanwise grid-sensitivity studies are left for future work, as there are still many uncertainties regarding the impact of spanwise resolution on the results of bluff body simulations, despite recent significant efforts (Grinderslev et al., 2023; Horcas et al., 2020; Wu & Sharma, 2020), and especially for the large Reynolds numbers relevant for wind turbines.

A possible future line of work would be to study the case of an intermediate aspect ratio between the 3D-1c case and the 3D-21c case. This intermediate case would allow for a finer spanwise resolution, while reducing the periodic boundary condition effect expected from short aspect ratio cases. According to Wu and Sharma (2020), for the cylinder case, the spanwise length should exceed 5 times the diameter to avoid the effects of increased spanwise coherence due to the periodic boundary condition. Nevertheless, this study is also limited in its spanwise resolution, having 10 cells per spanwise diameter in their longest cylinder (with a spanwise length of 10 diameters)¹, which is similar to the present 3D-21c case (with about 6 cells per spanwise chord length), and therefore possibly too coarse. The present choice of the spanwise length and mesh density of the 3D-21c is thus driven by the objective of being in-line with those employed in typical full-blade simulations.

Furthermore, as is well-known (Lian et al., 2023a; Sarpkaya, 2004) and will be seen in the results section (Section 4.2), for VIV cases with significant motion the wake becomes more spanwise correlated, relaxing the need to resolve small vortical structures in this direction. Meshes with spanwise densities similar to that of the present 3D-21c case have been shown to be largely insensitive to significant refinements in the spanwise direction when undergoing large VIV motions (Horcas et al., 2020; Horcas, Madsen, et al., 2022).

The equations solved by the CFD and structural solvers are dimensionless. The parameters can be made dimensional by using as reference length the airfoil chord $c = 1$ m, the reference mass ρc^3 , such that the fluid density is $\rho = 1.23$ kg/m³, and the reference time is c/U_∞ , where the inflow velocity value U_∞ depends on the specific simulation. The chord-based Reynolds number, $Re = \rho U_\infty c / \mu$, can be calculated such that the dynamic viscosity is $\mu = 1.81 \times 10^{-5}$ kg/(ms). In the dimensionless equations, the variation of inflow velocity is implemented as a variation of the Reynolds number and of the dimensionless structural frequency, described next in Section 4.1.3.

The dimensionless time step of all CFD simulations is $\Delta \tilde{t} = \Delta t U_\infty / c = 0.01$, which results in about 800 time steps for each vortex shedding period. A maximum of 50 inner iterations per time step are employed in the 2D case, and a maximum of 10 are employed in the 3D cases.

4.1.3 Structural dynamics system setup

For the present FSI simulations, the airfoil is allowed to move freely (i.e. without an imposed motion) along its edgewise direction and without deformation. The CFD solver is tightly coupled to a one degree of freedom lumped-element dynamical system, like the one presented in Figure 4.3. This system, in the absence of structural damping, is described by the following equation:

$$m\ddot{x}(t) + kx(t) = l(t),$$

where m is the mass per spanwise unit length of the airfoil, k is the stiffness constant per unit length, $\ddot{x}$ is the edgewise acceleration of the airfoil and x its displacement, t is time, and l is the lift force per unit length, derived from the edgewise component of the integral of the pressure and viscous stress fields over the airfoil surface, as obtained from the CFD simulation.

¹Note that the mesh employed in Wu and Sharma (2020) is actually defined in Wu et al. (2020).

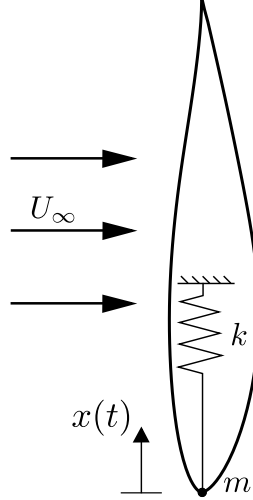


Figure 4.3: Dynamic system representation. The inflow velocity U_∞ is shown for reference.

Only the edgewise motion is considered, as in-line motion has minimal impact on the transverse VIV response (Cagney & Balabani, 2014; Jauvtis & Williamson, 2003), and due to the relatively high aerodynamic damping expected in the flapwise direction. This simplification allows for the use of the present one-dimensional structural model.

In all simulations, the airfoil chord is set to $c = 1$ m. The mass per unit length value is set as $m = 40$ kg/m, and the stiffness constant per unit length is defined as $k = 20808$ N/m². The structural natural frequency of the system is $f_s = \sqrt{k/m}/(2\pi) = 3.63$ Hz, and the mass ratio is $\tilde{m} = m/(\rho_\infty c^2) = 32.7$, where ρ_∞ is the fluid density at the inlet. The dynamic system characteristics are comparable to those employed in previous similar simulations (Lian et al., 2022, 2023a, 2023b; Skrzypiński et al., 2014), and are considered to be representative of a wind turbine blade section. Note that the most relevant parameters for the VIV response are the mass ratio, which for wind turbine blades the authors estimate to be in the range $\tilde{m} \in [10, 30]$, and the ratio between the structural natural frequency and the vortex shedding frequency, which is studied parametrically in the present work. The structural damping is also a relevant parameter in VIV, but it is set here to zero to give conservative results if used to inform structural design engineering models, and to better assess the self-limiting characteristic of VIV. Blade structural damping values and their effect on VIV are discussed in more depth in Chapter 5.

The dynamic equation is solved iteratively for each CFD solver time step using the Newmark-beta method (Newmark, 1959) with a convergence criterion for the residual acceleration between two iterations of 1×10^{-5} . The motion calculated by the structural solver is passed to the CFD solver as a whole grid position update, so no mesh deformation algorithm is required. More details on the FSI two-way coupling implementation in MaPFlow can be found in Papadakis et al. (2022). Note that the time step used by the CFD solver is the same as the one used by the structural solver, as the two solvers are tightly coupled.

4.2 Results and discussion

The solver and procedure employed are evaluated by means of a sensitivity analysis and a comparison with results from the literature (Section 4.2.1). The computational cost of the simulations is presented in Section 4.2.2. The analysis of the simulations is divided into three parts: static airfoil simulations (Section 4.2.3), which investigate the vortex shedding mechanisms; free to move airfoil simulations (Section 4.2.4), which characterize the lock-in response through multiple simulations; and an analysis of the flow fields (Section 4.2.5), which supports the findings from the previous results.

4.2.1 Verification

Numerical setup sensitivity analysis

In addition to the meshes described in Section 4.1.2, two finer 2D meshes were tested to check the sensitivity of the 2D results to the mesh resolution. A mesh called “medium” was created by doubling the resolution over the airfoil and in the radial direction, resulting in approximately 1.3×10^5 cells. The third mesh, called “fine”, was defined by making the mesh three times finer in both directions, resulting in about 3×10^5 cells. The Courant number was preserved in all simulations by reducing the time step accordingly, this was found necessary in preliminary sensitivity analyses not shown here. The 2D mesh presented in Section 4.1.2 and used throughout the rest of this work is named “coarse”.

The inflow velocity for the sensitivity analysis is chosen such that the expected vortex-shedding frequency matches the structural frequency, resulting in large-amplitude oscillations. The variables used for this analysis are the maximum of the airfoil dimensionless motion displacement, $\max(\tilde{x})$, the standard deviation of the lift coefficient σ_{c_l} , and the mean drag coefficient $\overline{c_d}$. These are canonical results of the VIV response (Tanida et al., 1973). The time series length is fifty dimensionless time units, following the wake-development transient period, $\tilde{t}_{\text{trans}}$.

The time series results for the three meshes are plotted in Figure 4.4(a), and the percentage differences between the results obtained with the coarse and medium meshes, with respect to the fine mesh results, are shown in Figure 4.4(b). The results for the three meshes are quite similar for the three quantities analysed.

The use of the coarse mesh setup is considered a good compromise between accuracy and computational cost. A simulation with the medium mesh setup takes about 8 times longer to run than with the coarse mesh setup, and a simulation with the fine mesh takes about 18 times longer. Performing a similar sensitivity analysis for the 3D cases would be significantly more computationally demanding. The mesh used for the 3D simulations is finer than the 2D case mesh, as detailed in Section 4.1.2. For the 3D-1c case, the chosen mesh is consistent with the one previously employed by Skrzypiński et al. (2014). The choice of meshes is further justified in the next section (Section 4.2.1), by comparing the results obtained with the present implementation to those from the literature.

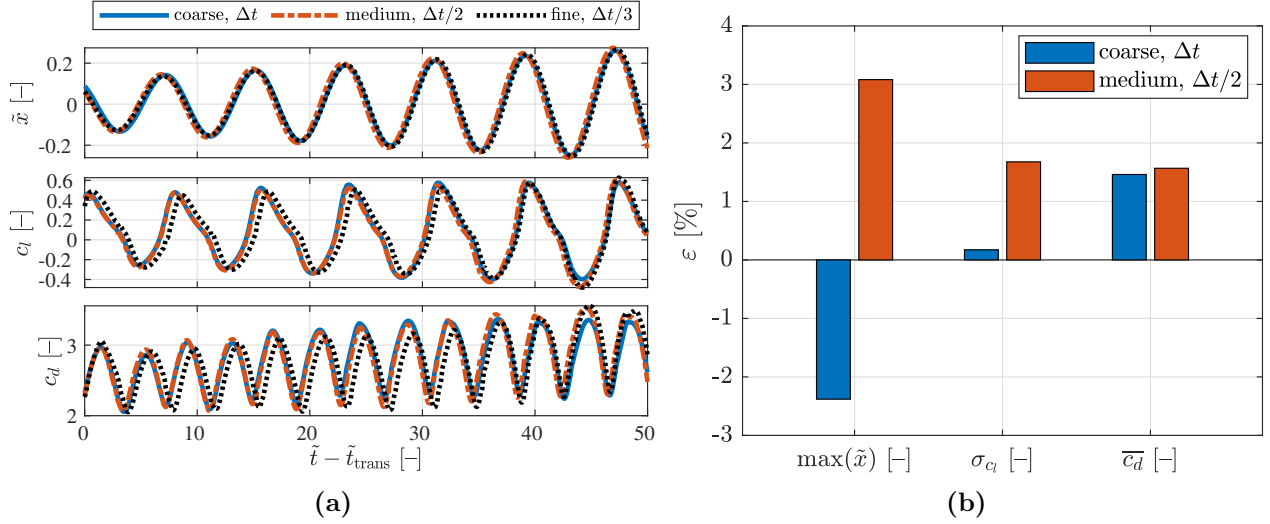


Figure 4.4: Mesh sensitivity analysis of the 2D simulations. (a) Time series segments of the airfoil dimensionless displacement (top), lift coefficient (middle) and drag coefficient (bottom), for the three setups considered. (b) Percentage difference of dimensionless maximum displacement, standard deviation of the lift coefficient and mean drag coefficient, for the coarse and medium meshes with respect to the fine mesh.

Comparison to published data

To further verify the implementation of the FSI solver, results from Skrzypiński et al. (2014) and Lian et al. (2022) are used for comparison. The focus is on 2D URANS VIV cases. The parameter used for the analysis is the maximum displacement of the airfoil, obtained from multiple simulations covering the lock-in range.

In Skrzypiński et al. (2014), these values are provided for 100 dimensionless simulation time units, $\tilde{t} = 100$, and are normalized with the maximum amplitude from all simulations, such that $A_x^*(T^*) = A_x(T^*)/\max(A_x(T^*))$, where $T^* = TU_\infty/c$ is the dimensionless natural oscillation period, which coincides with the widely used reduced velocity $U_{red} = U_\infty/(f_s c)$. The comparison of these published results against those obtained with the present implementation, for an identical setup, are shown in the left panel of Figure 4.5. For the present results, the dimensionless time used is 160 units rather than 100, to compensate for the slightly differences in the VIV transient.

In Lian et al. (2022), the amplitude is normalized with the chord, and the values reported correspond to approximately 240 dimensionless time units. This time is found to be enough to reach a state where the amplitude is growing at a steady rate. However, the actual maximum VIV amplitude is far from being reached, so the comparison is still very sensitive to the transient behaviour. The right panel of Figure 4.5 shows the comparison for this case, where the present results correspond to a dimensionless time of 200 units.

The comparison is fairly good in both cases shown in Figure 4.5, despite the differences mentioned above. It is worth noting that, as per the simulations performed in the present work, the true maximum amplitude is only reached after about 550 dimensionless time units

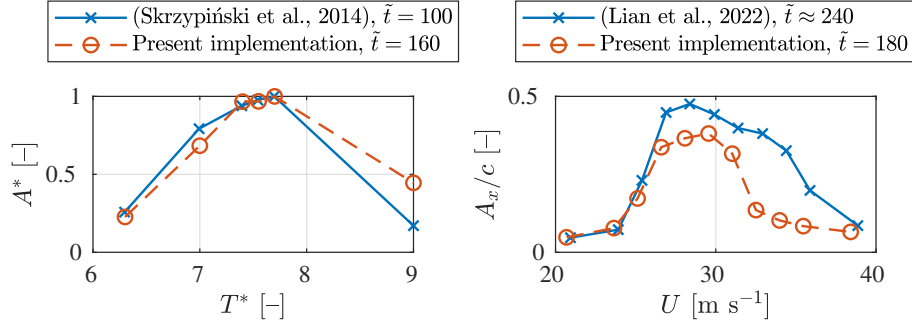


Figure 4.5: Comparison of results from the literature replicated with the present implementation. For the comparison with Skrzypiński et al. (2014) (left), the maximum amplitudes are shown normalized with their highest value for each case, after the dimensionless simulation times depicted, as a function of the reduced velocity. In the Lian et al. (2022) case (right), the dimensionless maximum amplitudes are shown as a function of the inflow wind speed.

and has a value of about 1.5 chords (this is visible in Figure 4.15(a)).

There are some differences between the setups used for this comparison and the setup presented in Section 4.1. In Skrzypiński et al. (2014), the lock-in range is covered by adjusting the stiffness constant in the structural model for each simulation. Conversely, in Lian et al. (2022) and the present work, the inflow wind speed is varied. Consequently, in Skrzypiński et al. (2014), all simulations are conducted at a constant Reynolds number ($\text{Re} = 2 \times 10^6$), whereas in the other two studies, the Reynolds number varies with the inflow velocity. Additionally, in Lian et al. (2022), the dimensionless time step for the VIV simulations is defined as $\Delta \tilde{t} = U_\infty / (100 f_s c)$, which ranges from 0.06 to 0.11 for the simulated velocity range. In Skrzypiński et al. (2014) and the present work, the dimensionless time step is set to $\Delta \tilde{t} = 0.01$ for all simulations. This time step was verified in a preliminary sensitivity analysis not shown here.

It is important to identify robust metrics for comparing results obtained with different tools or modelling choices. This is particularly important for ensuring reproducible results when simulations are too short to capture the limit-cycle or converged behaviour, as is the case with the results used for comparison here. One such metric is introduced later in Section 4.2.4.

In conclusion, the mesh sensitivity analysis indicates that the coarse mesh setup is sufficient for the 2D simulations, providing a good balance between accuracy and computational cost. The comparison with published data shows that the present implementation is reliable and produces results consistent with the literature. This is an important step, especially given the computational cost of each simulation, and the need to perform multiple simulations to characterize the VIV response of the airfoil, as will be shown in the following sections.

4.2.2 Computational cost

The 2D simulations were carried out on the Magerit-3 Supercomputer at CeSViMa (Universidad Politécnica de Madrid), and the 3D simulations were carried out on the MareNostrum 4

Supercomputer at the Barcelona Supercomputing Center. The computational cost of the simulations presented hereafter is detailed in Table 4.2. The computational cost is illustrated by the wall-clock time, which is illustrative of the time passed between launching a simulation and obtaining the results, and the CPU hours, which is a measure of the computational resources employed in a simulation.

These data show, as expected, that the 2D approach is faster and much less resource-intensive than the 3D approaches. For the static simulation case, the total cost in CPU hours of the 3D case is three orders of magnitude larger than for the 2D case. Despite large parallelization and a smaller limit to inner iterations per time step, the wall-clock time of the static 3D cases is still much higher. The reason why the static 3D cases are much more resource-intensive is twofold; on one hand, the time needed to compute a single iteration is much higher, due to the higher mesh cell count (due mainly to the extrusion in the spanwise direction, but also to the denser mesh required to accurately resolve the near-wake with LES), and, on the other hand, longer simulations are needed to obtain a statistically converged solution, especially for the 3D-1c case, for the reasons argued in Section 4.2.3.

To characterize the VIV response of the airfoil, multiple simulations at different inflow velocities may be required and the tight coupling between the fluid and structural solver means that extra evaluations of the fluid solution are performed. This renders fully-converged 3D simulations prohibitively expensive. In this work, the 3D simulations are run for a shorter time, and the results are analysed in Section 4.2.4 with the understanding that the airfoil does not reach its limit-cycle behaviour.

Alternative approaches to reduce the computational cost of the 3D simulations are to introduce an initial condition closer to the limit-cycle (which requires a priori knowledge), or to use a forced-vibrations approach instead of the free to move airfoil approach employed here. Nonetheless, the present approach is chosen to provide insight into the VIV development from rest, and a new metric is proposed which allows for a more robust comparison of the VIV response between the 2D and 3D simulations without the need to reach the limit-cycle behaviour.

	N_{steps} [-]	N_{CPUs} [-]	$T_{\text{wall-clock}}$ [h]	N_{sims} [-]	Total cost [h] (CPU)
2D static	65 000	80	2.5	1	200
3D-1c static	120 000	1 200	315	1	3.8×10^5
3D-21c static	98 000	1 200	288	1	3.5×10^5
2D FSI	130 000	80	12	15	1.5×10^4
3D-1c FSI	30 000	1 200	192	7	1.6×10^6
3D-21c FSI	30 000	1 200	190	9	2.1×10^6

Table 4.2: Details on the computational cost of the simulations performed. N_{steps} refers to the number of time steps run, N_{CPUs} is the number of CPUs used in parallel via the Message Passing Interface (MPI), $T_{\text{wall-clock}}$ is the approximate wall-clock time for a single simulation, and N_{sims} is the number of simulations of each type (i.e., the number of different inflow wind speeds). The total cost is expressed in CPU hours.

Following the analysis of the computational setup and cost, the results of the simulations are presented and discussed in the subsequent sections.

4.2.3 Static airfoil case

The flow field over the airfoil in a static position is simulated to provide information about the vortex-shedding frequency and behaviour. The results from this static case will be used as a reference for the analysis of the VIV response of the moving airfoil. One static simulation is run for each aspect ratio configuration (2D, 3D-1c and 3D-21c). The three simulations are carried out at a Reynolds number $Re = 2 \times 10^6$.

Lift coefficient frequency content

The vortex-shedding frequency can be approximated by the dominant frequency of the lift force fluctuation (Bishop et al., 1964a), which is typically obtained as the frequency at which its power spectral density (PSD) function is maximum. The lift force per spanwise unit length, $l(t)$, is defined in terms of the lift coefficient, $c_l(t)$, as $l = 0.5\rho_\infty U_\infty^2 c c_l$. The lift coefficient fluctuations are obtained as $c_l(t) - \bar{c}_l$, where $\bar{c}_l$ is the time-average of c_l . The mean lift coefficients obtained are $[\bar{c}_{l,2D}, \bar{c}_{l,3D-1c}, \bar{c}_{l,3D-21c}] = [0.055, 0.065, 0.074]$, which are all very small, as expected from an airfoil at 90° angle of attack. The PSD of the zero-mean lift coefficient time series, $S_{c_l}(f)$, where f is frequency, has been calculated using the Blackman–Tukey estimator, with a Hamming window, and with the maximum lag used for the autocovariance function estimation being equal to the time series length minus 1. The dimensionless form of this PSD, $\tilde{S}_{c_l} = S_{c_l} U_\infty / c$, is analysed, and shown in terms of the dimensionless frequency $\tilde{f} = fc / U_\infty$. One-thousand frequency points between $\tilde{f} = 0.1$ and $\tilde{f} = 0.17$ are employed, after checking that almost all the energy is contained within these frequencies. The PSD estimation has been performed using the open-source toolbox mVARbox (Gallego-Castillo et al., 2024). An initial period of the time series associated with the transient wake development, spanning the dimensionless time interval $\tilde{t} \in [0, 100]$, is excluded from all statistical calculations. After the transient period, the total simulation period for the 2D case is 550 dimensionless time units, for the 3D-1c cases it is 1100, and for the 3D-21 case it is 880.

The PSD of the lift coefficient fluctuations for the three modelling cases, normalized with the variance of the lift coefficient of each case $\sigma_{c_l}^2$, are compared in Figure 4.6. Two y -axes are shown in the figure, to accommodate for the different ranges of PSD values; the left axis corresponds to the 3D-1c case, and the right axis to the 2D and 3D-21c cases. Due to the normalization, the area under the three curves is approximately the same. For the 2D and 3D-21c cases, almost all the energy is concentrated on a single peak at the dimensionless frequencies 0.126 and 0.144, respectively. Conversely, the 3D-1c PSD shows multiple peaks at clearly separated frequencies. The Strouhal number for the 2D and 3D-21c cases are thus approximated as $St_{2D} = 0.126$ and $St_{3D-21c} = 0.144$. On the other hand, for the 3D-1c case there is no clear dominant shedding frequency. The energy is spread out over multiple peaks in the range $[0.11, 0.16]$, so it seems problematic to report a single Strouhal number value for this case. Note that, as discussed in Section 4.1, the vortex formation and thus the Strouhal number values might be influenced by the periodic lateral boundary condition, especially for the 3D-1c case, and that the elongated spanwise mesh cells of the 3D-21c case might also

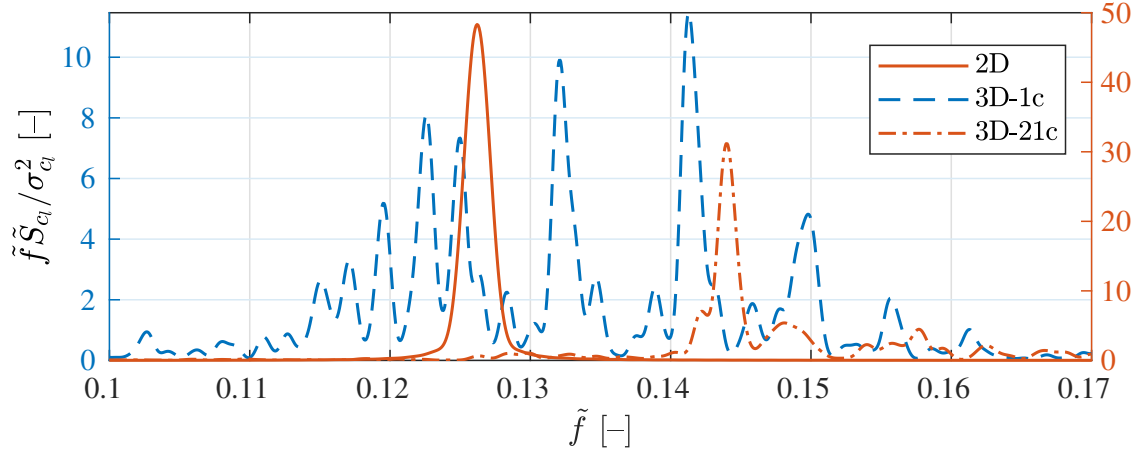


Figure 4.6: Normalized dimensionless PSD of the lift coefficient fluctuations, for the static airfoil at Reynolds number $Re = 2 \times 10^6$. The left y -axis corresponds to the 3D-1c case and the right y -axis to the 2D and 3D-21c cases.

affect the vortex shedding behaviour. Due to the prohibitive cost of higher fidelity simulations, wind tunnel tests are required to obtain precise Strouhal numbers of actual geometries at the Reynolds numbers relevant to wind turbines.

The existence of multiple peaks in the lift coefficient of a body subject to vortex shedding has been reported before under a variety of circumstances. In Schewe (1983), it was found experimentally that for a circular cylinder in the upper transition regime, $Re \in [1 \times 10^6, 5 \times 10^6]$, no clear single vortex-shedding frequency could be identified in the PSD of the lift coefficient fluctuations. It was suggested that, in this regime, the flow experiences multiple unstable configurations, although no evidence could be given of a relationship between the PSD frequency peaks and the lift and drag coefficients behaviour. In Lehmkuhl et al. (2014), LES simulations of a static circular cylinder at different Reynolds numbers were performed, and they observed that in the critical regime ($Re = 3.8 \times 10^5$) there were two different vortex-shedding frequencies in the PSD of the cross-stream velocity fluctuations. They noticed that one of the frequency peaks matched the frequency they obtained in the subcritical regime, and the other peak matched the frequency from the super-critical regime. This duplicity in the vortex-shedding frequency was attributed to a switching between two flow configurations, arising from the asymmetry in the flow characteristic of the critical regime. Recently, in Ellingsen et al. (2022) two different vortex-shedding frequencies were identified in the PSD of the lift coefficient of a circular cylinder, obtained from wind tunnel experiments. These two frequencies were observed in the range of Reynolds numbers $Re \in [8 \times 10^5, 2.17 \times 10^6]$, which belong to the supercritical regime, and they were associated with two different spatial patterns of the cylinder surface pressure. It was not determined whether the two spatial patterns happened simultaneously or were intermittent. To the best of the author's knowledge, the presence of multiple peaks of similar energy in the PSD of the lift coefficient, resulting in ambiguity in the Strouhal number definition, has not been previously reported for the case of an airfoil.

Flow regimes

To compare the time evolution of the aerodynamic forces acting on the airfoil predicted by the different cases, time series of the lift and drag coefficients (c_l and c_d respectively) for all models are shown together in Figure 4.7. The results are shown for a dimensionless time interval $\tilde{t} \in [100, 300]$ to improve the visualization of the fluctuation patterns. It is noticeable that, for the 2D case, the lift and drag time series are highly periodic. For the 3D-1c case, the time patterns of both coefficients are notably irregular; specifically, the lift coefficient predicted with the 3D-1c approach experiences amplitude changes, and the drag coefficient shows simultaneous changes in amplitude and mean value. These low-frequency modulations vary in intensity and duration throughout the entire time series $\tilde{t} \in [100, 1200]$. For the 3D-21c case, the time series exhibit no discernible periodicity and have very small fluctuations compared to the other cases. This can be attributed to the effect of averaging along the span, which is expected to smooth out the fluctuations, and could also be influenced by the coarser spanwise mesh employed in this case.

Similar observations to the ones presented here have been reported in the literature for the case of three-dimensional flat plates. In Lisoski (1993), experiments were performed for flat plates at 90° , with different aspect ratios, and Reynolds numbers between 3000 and 9000, and they reported lift and drag coefficient time patterns similar to those presented in Figure 4.7 for the 3D-1c case. Direct numerical simulations (DNS) were performed in Najjar and Balachandar (1998), for a flat plate at 90° , both in a 2D configuration and a 3D configuration with an aspect ratio of 2π , and for a Reynolds number $Re = 250$ in both cases. The authors presented a drag coefficient time series for their 2D case which is highly periodic, like the 2D case shown in Figure 4.7. Their 3D simulations of the flat plate show a drag and lift time series with irregularities, similar to the ones evidenced in the 3D-1c results presented here. This may imply, leaving out the large difference in Reynolds numbers, that the regular vortex shedding observed for the 2D case is a consequence of the 2D nature of the simulation and not only of the URANS turbulence model. Preliminary results showed that a URANS 3D case was not able to predict the irregular shedding behaviour observed in the 3D DDES cases, so it is considered that both a 3D setup and large-eddy-resolving turbulence modelling are needed to predict the intermittent shedding presented here. Najjar and Balachandar (1998) also explored the drag force behaviour during different low-frequency cycles and identified two main flow regimes: the high-drag regime (H) and the low-drag regime (L), referring to the magnitude of the mean drag. It was found that during the H regime, the shed vortical structures are more coherent. More recently, an additional regime called M was proposed by Hemmati et al. (2016), occurring in the time interval between regimes H and L .

In Figure 4.7 it can be seen that there are some time periods of varying duration where the behaviour of the aerodynamic coefficients predicted by the 2D and the 3D-1c configurations are similar, not only in terms of amplitude and mean values but also frequency, e.g. during dimensionless times 160 to 200, and in the vicinity of $\tilde{t} = 240$ and $\tilde{t} = 290$. These time periods might correspond to the high-drag flow regime, H . Najjar and Balachandar (1998) describe the H regime as one with coherent spanwise vortices, whereas in regime L these vortices are broken apart. It seems reasonable then that the 2D simulation results are more similar to the 3D results during the high-drag regime, as 2D simulations may be considered equivalent to a

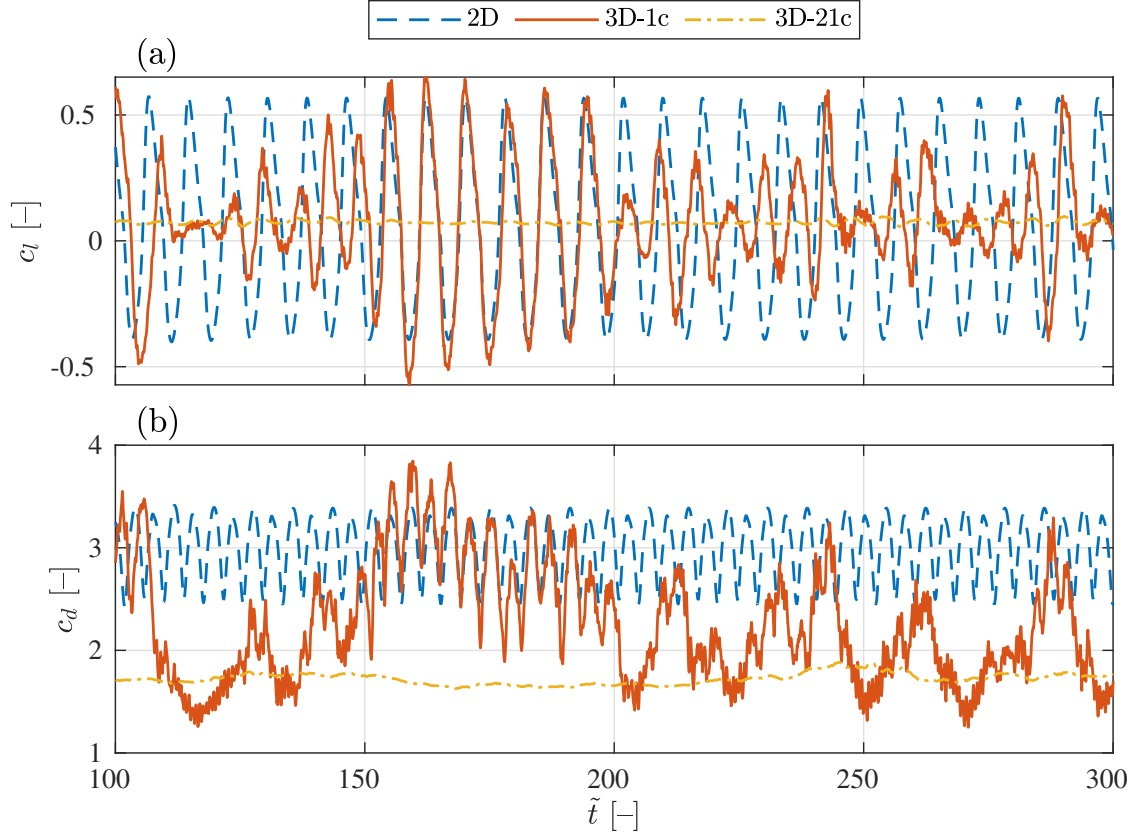


Figure 4.7: Detail of the (a) lift coefficient and (b) drag coefficient as a function of dimensionless time, for the three simulated aspect ratios of the DU96-W-180 airfoil.

spanwise uniform case. A visual confirmation of the presence of largely coherent spanwise structures during these H regime intervals will be given in Section 4.2.5.

One possible explanation for the existence of multiple peaks in the 3D-1c lift coefficient PSD shown in Figure 4.6 is that the vortex-shedding frequency changes as time progresses. This argument is reinforced by the existence of different flow regimes observed in Figure 4.7, each potentially having different characteristic vortex-shedding frequencies. To further explore the idea that the vortex-shedding frequency changes as time progresses, the time evolution of the frequency content of the lift coefficient is quantified using the continuous wavelet transform (CWT) of the lift coefficient fluctuations. To compute the CWT, the bump wavelet with 48 voices per octave in the range $\tilde{f} \in [0.1, 0.17]$ is employed. The use of CWT to inspect the time-frequency content of the aerodynamic forces in a VIV analysis has been previously implemented, for example in Zhao et al. (2022).

The CWT results are visualized in Figures 4.8 to 4.10, corresponding to the 2D, 3D-1c, and 3D-21c cases, respectively. Each figure displays the complete lift coefficient time series in the top panel and its scalogram in the bottom panel. The scalogram represents an estimation of the instantaneous lift coefficient frequency content on the y -axis, against the dimensionless time on the x -axis. Brightness quantifies the magnitude of the CWT, normalized using the L1 or Manhattan norm, indicating the amplitude content of the lift coefficient fluctuations

at each frequency (Lilly, 2017). This panelled layout allows for the association of the time evolution of the lift coefficient with its frequency content. The blue solid line plotted on top of the scalogram marks the time evolution of the most energetic instantaneous frequency.

In Figure 4.8, the scalogram shows that the lift coefficient time series is dominated by a single frequency, which remains constant throughout the simulation time. This frequency is in agreement with the Strouhal number obtained from the PSD of the lift coefficient fluctuations, $St_{2D} = 0.126$, shown in Figure 4.6.

For the 3D-1c case, in contrast, the scalogram in Figure 4.9 shows that the lift coefficient time series is dominated by multiple frequencies, which vary as time progresses. The frequencies are in the range $[0.11, 0.16]$, which is consistent with the multiple peaks observed in the PSD of the lift coefficient fluctuations for the 3D-1c case (Figure 4.6). It is also noticeable that at any given time there is apparently just one dominant shedding frequency. It can be thus concluded that the PSD with multiple peaks observed in Figure 4.6 for the 3D-1c case is the result of individual vortex-shedding frequencies occurring at different time intervals. When comparing the 3D-1c scalogram with its lift coefficient time series, which share the x -axis of Figure 4.9, it is observed that the low amplitude oscillations of c_l occur at a higher frequency than the high amplitude ones. These variations may be attributed to the switching of flow regimes. The H regime, characterized by a highly coherent shedding pattern and large amplitudes of the lift coefficient fluctuations, tends to present lower shedding frequencies than other regimes. It is important to note that the PSD shown earlier in Figure 4.6 exhibits multiple distinct peaks, indicating that there is not a straightforward correspondence between vortex-shedding frequency and flow regime. This lack of one-to-one correspondence is also evident in the scalogram in Figure 4.9; for instance, during the dimensionless time interval $\tilde{t} \in [500, 600]$, identified as the H regime based on the c_d time series (not shown here), the vortex-shedding frequency continuously decreases over time. The dashed line shown in the scalogram corresponds to the Strouhal number obtained with the 2D results, and matches reasonably well with the vortex-shedding frequencies observed during the H regime for the 3D-1c case. This similarity aligns with the observation made in Figure 4.7, where the time series for both configurations matched well qualitatively in the H regime.

For the 3D-21c case, the scalogram in Figure 4.10 shows that the lift coefficient time series presents smaller variations in its frequency content compared to the 3D-1c case, but it is not as constant as for the 2D case. The frequency content is spread out over a range approximately between $\tilde{f} \in [0.13, 0.16]$, the centre of which aligns with the Strouhal number obtained from the PSD of the lift coefficient fluctuations, $St_{3D-21c} = 0.144$, shown in Figure 4.6. It is also noticeable that in the time interval around $\tilde{t} \in [800, 850]$ the amplitude of the lift coefficient has a sharp increase, and that it is associated with an increase in its frequency content at a Strouhal close to the 2D value. This could mean that the flow is briefly transitioning to a high-drag regime (this is supported by the drag coefficient time series, not shown here, where the mean drag is seen to increase noticeably in this time interval).

In conclusion, there is evidence that multiple vortex-shedding flow regimes may exist for the airfoil case. These regimes are associated with significant differences in the vortex-shedding behaviour and thus the aerodynamic loading. The 2D case exhibits a single regime of high-amplitude lift oscillations and high mean drag, regime H . The 3D-1c case changes

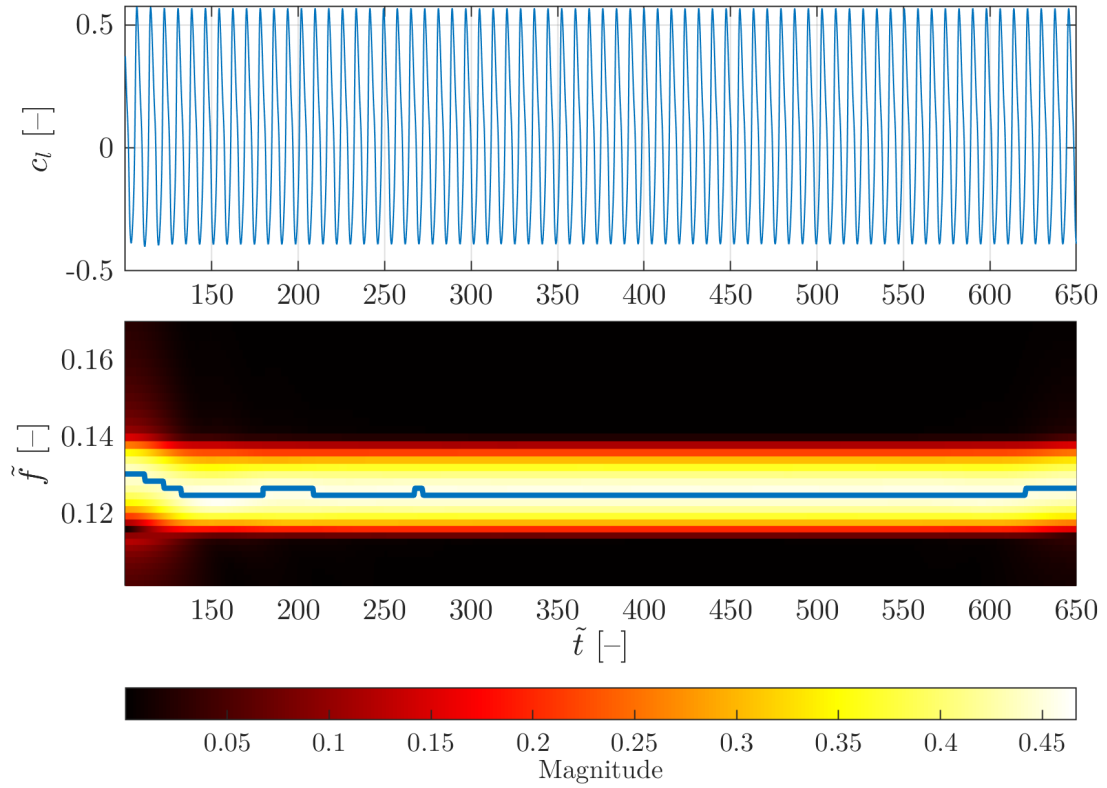


Figure 4.8: 2D static case. (Top) lift coefficient time series, and (bottom) dimensionless scalogram of the lift coefficient. In the scalogram, the blue solid line (—) marks the frequency bin with the highest magnitude at each time bin, and the dashed horizontal line (--) indicates the Strouhal number obtained from the 2D static case.

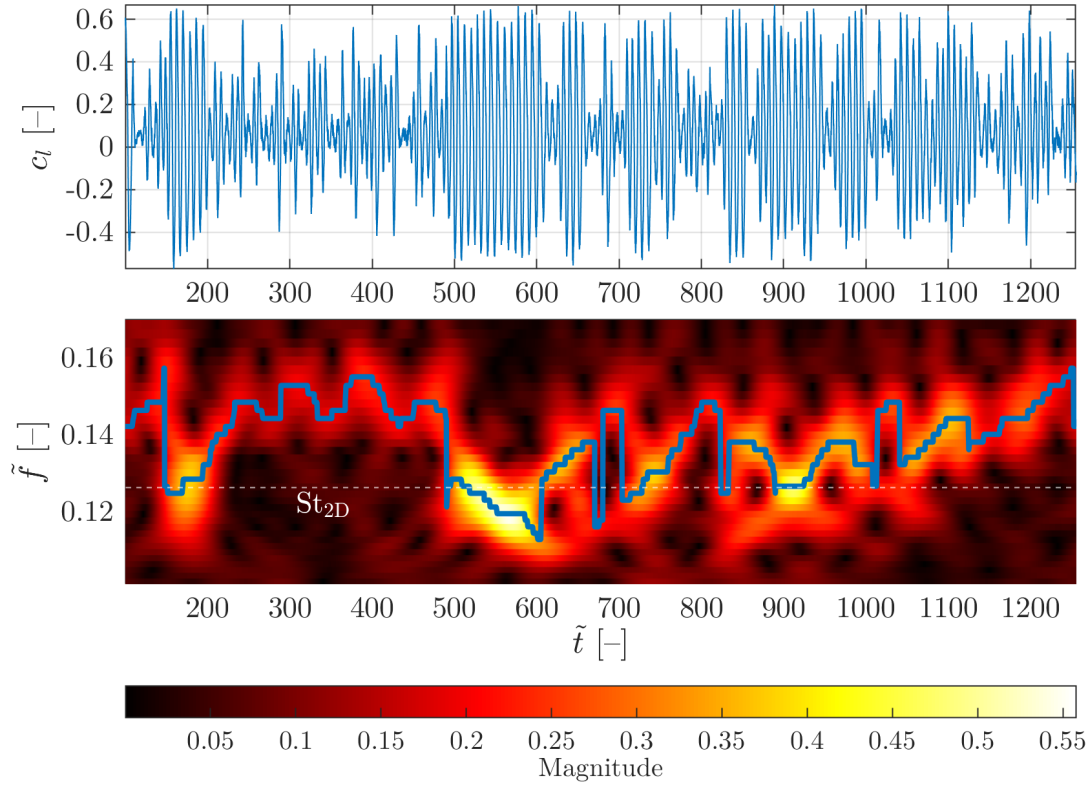


Figure 4.9: 3D-1c static case. (Top) lift coefficient time series, and (bottom) dimensionless scalogram of the lift coefficient. In the scalogram, the blue solid line (—) marks the frequency bin with the highest magnitude at each time bin, and the dashed horizontal line (--) indicates the Strouhal number obtained from the 2D static case.

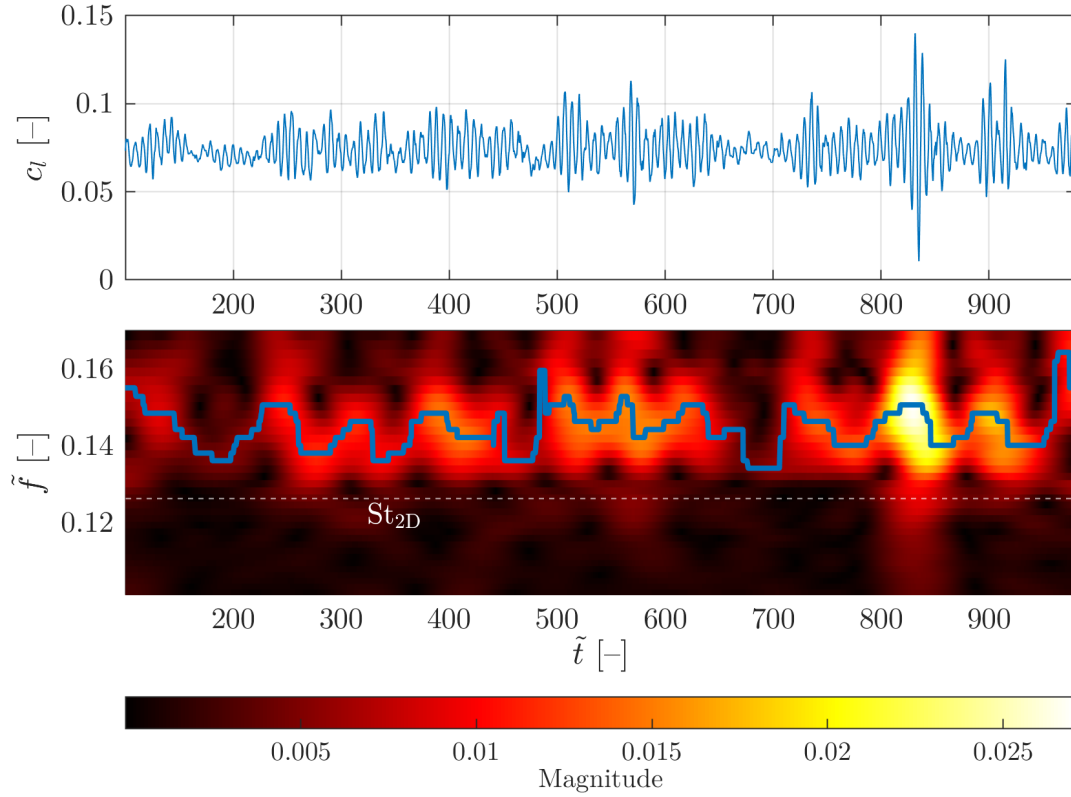


Figure 4.10: 3D-21c static case. (Top) lift coefficient time series, and (bottom) dimensionless scalogram of the lift coefficient. In the scalogram, the blue solid line (—) marks the frequency bin with the highest magnitude at each time bin, and the dashed horizontal line (--) indicates the Strouhal number obtained from the 2D static case.

regime intermittently, from the H regime to a low-amplitude lift oscillations and low mean drag regime, L . The 3D-21c case exhibits a regime L behaviour, with small amplitude lift oscillations and low mean drag. The H regime appears to be associated with lower vortex-shedding frequencies than the L regime. These findings highlight the importance of considering three-dimensional effects and the potential limitations of 2D simulations in accurately capturing the complex vortex-shedding dynamics of an airfoil. Additionally, the multiplicity of regimes puts into question the validity of using a single Strouhal number to describe the vortex-shedding behaviour of the airfoil, as will be explored in the following section.

Strouhal number analysis

To further assess the ambiguity in the Strouhal number definition for the 3D-1c configuration, a sensitivity analysis has been performed with respect to the length of the time series used to calculate the PSD of the lift coefficient fluctuations, S_{c_l} . In Figure 4.11, the Strouhal number corresponding to the highest peak in the PSD of the lift coefficient fluctuations, $\max(S_{c_l})$, is plotted as a function of the dimensionless duration of the time series used to compute this PSD, $\tilde{t}_{\text{final}}$. That is, as $\tilde{t}_{\text{final}}$ increases, a longer part of the time series is used, until for the last value the whole series is employed. Additionally, the Strouhal number obtained from fitting a Gaussian function to the PSD is also shown in the figure, as a function of the same $\tilde{t}_{\text{final}}$. It is observed that, while for the 2D and 3D-21c cases the frequency quickly converges to its final value, for the 3D-1c case there are significant frequency jumps. This is true both for the maximum peak in the PSD and for the Gaussian fit to the PSD, although in the 3D-1c case the Gaussian fit is more stable than the maximum peak frequency. The 3D-1c case shows no sign of convergence for the maximum peak frequency, which oscillates between 0.123 and 0.151, while the Gaussian fit frequency could be said to converge to a value of approximately 0.132. Given the intermittent nature of the vortex-shedding process in the 3D-1c case, as described earlier in this section, it is possible that the Gaussian fit frequency would keep oscillating for indefinitely long simulation times, within a bounded range. The percentage difference in the PSD peak frequency range for the 3D-1c case observed in Figure 4.11, $St_{3D-1c} \in [0.123, 0.151]$, with respect to the converged 2D frequency, $St_{2D} = 0.126$, ranges from -2% to 20% , and with respect to the converged 3D-21c frequency, $St_{3D-21c} = 0.144$, the range is -15% to 5% . Note that the Strouhal number value is typically employed in engineering models for predicting the vortex-induced forces and VIV response of a body.

Table 4.3 presents a list of Strouhal numbers reported in the literature for wind turbine airfoils, at $\alpha = 90^\circ$ and using different CFD modelling options. The turbulence modelling approach is specified under “Method”, being URANS methods shown at the top part of the table, and DES-like methods at the bottom part. Within each part of the table, the studies are sorted in increasing aspect ratio (AR) order. It can be observed that, for the URANS cases, the reported St values become larger as the aspect ratio increases. The Strouhal number obtained in the present work with the 2D (URANS) approach agrees reasonably well with those present in the literature. However, the 3D-1c case shows a range of Strouhal numbers that encompasses all the values reported in the literature; including all aspect ratios, all turbulence modelling methods, and both 2D and 3D cases. When considering the 3D-1c case Gaussian fit central frequency, the St value is quite different to those reported in the literature

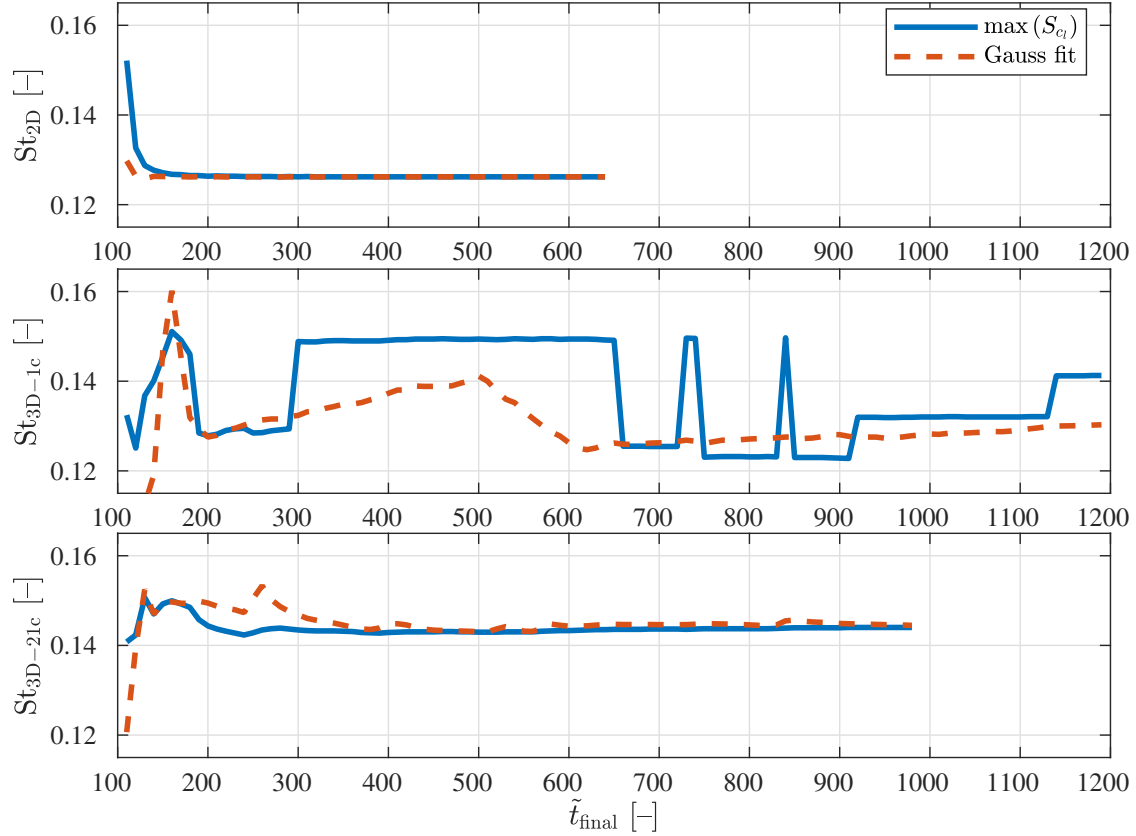


Figure 4.11: Strouhal number estimated as the highest peak in the PSD of the lift coefficient fluctuations, and from the Gaussian fit to the PSD, as a function of the total time used to calculate the PSD. (Top) for the 2D configuration, (middle) for the 3D-1c configuration, and (bottom) for the 3D-21c configuration.

for the same configuration. The Strouhal number for the 3D-21c (DDES) approach is notably close to the value obtained in Skrzypiński et al. (2014) for an aspect ratio of 8 using URANS. To the author’s knowledge, no other Strouhal numbers for high aspect ratio airfoil sections have been reported.

Both in Skrzypiński et al. (2014) and Lian et al. (2023a), it is mentioned that their lift coefficient PSDs for the DDES and DES cases, respectively, show energy content at more than one frequency, but that there is a single dominant peak, which they report as the Strouhal number. The total dimensionless simulation time used to determine the Strouhal value in Skrzypiński et al. (2014) is 168 dimensionless time units, and in Lian et al. (2023a) it is reported as “over 40 vortex shedding periods” where 40 periods would correspond to a dimensionless simulation time of approximately 260 units. Both of these time series might be too short to allow for the detection of the multiple shedding frequencies observed in the present work, judging by the results shown in Figure 4.11 for the 3D-1c case. In Figure 4.11, a value of about $St_{3D-1c} = 0.15$, very close to the Strouhal number 0.153 reported in Table 4.3 for the DDES and DES cases, can be seen prominently in the present 3D-1c results, especially in the dimensionless time interval $\tilde{t}_{\text{final}} \in [300, 600]$. This means that, if the present simulation had been of a length within that range, the highest peak in the PSD would have been found at $St = 0.15$. Furthermore, it has been checked that this peak would have been clearly dominant, if the simulation had been run with a dimensionless final time in the interval $\tilde{t}_{\text{final}} \in [400, 500]$. But, as has been shown throughout this section, such a frequency value would not have been representative of the vortex shedding flow behaviour in the present 3D-1c case.

To further characterize the vortex-shedding process, the Gaussian fit to the complete time series lift coefficient fluctuations PSDs are shown in Figures 4.12 to 4.14. Each figure corresponds to a different configuration: 2D, 3D-1c, and 3D-21c, respectively. For each configuration, the PSD of the lift coefficient fluctuations is shown using three different representations: in panel (a) it is shown in linear axes, in panel (b) in logarithmic y -axis, and in panel (c) in log–log axes. The Gaussian fit is shown as a dash-dotted line, and the central frequency of the Gaussian fit is marked with a vertical dashed line. For the 2D case, Figure 4.12, the fit is seen to be very good, with the Gaussian curve following the PSD closely. The central frequency of the Gaussian fit matches the peak of the PSD. The log–log representation in panel (c) reveals the appearance of harmonics at all multiples of the main frequency, highlighting the non-linear behaviour of the vortex-shedding process, which is not purely sinusoidal in nature. Figure 4.13 reveals that, for the 3D-1c case, the PSD of the vortex-shedding loading can be approximated as a narrowband process with a central frequency around $\tilde{f} = 0.132$. In panel (a), the Gaussian central frequency is seen to be very close to one of the most prominent peaks in the PSD, although the Gaussian fit does not capture the individual peaks. On the other hand, by showing the y axis in logarithmic scale, panel (b) reveals that there is an underlying distribution of the spectral content which is well captured by the Gaussian fit. The narrowband process can be identified even better in panel (c), where a wider range of frequencies is shown in log–log axes; the PSD is seen to grow above its background values for a limited range of frequencies, even if distinct peaks exist within this range. In panel (c), the third harmonic of the vortex shedding can be observed, although less clearly than for the 2D case, and this third harmonic is also distributed over multiple frequencies. Lastly, for the 3D-21c case shown in Figure 4.14, a hybrid behaviour is observed; the PSD shows both

Study	Airfoil	Method	AR	Re	St
Present work, 2D	DU96-W-180	URANS	2D	2×10^6	0.126
Pellegrino and Meskell (2013)	NREL S809	URANS	2D	1×10^6	0.116
Skrzypiński et al. (2014)	DU96-W-180	URANS	2D	2×10^6	0.125
Lian et al. (2022)	DU96-W-180	URANS	2D	2×10^6	0.121
Lian et al. (2023a)	DU96-W-180	URANS	2D	2×10^6	0.119
Skrzypiński et al. (2014)	DU96-W-180	URANS	1/8	2×10^6	0.128
Skrzypiński et al. (2014)	DU96-W-180	URANS	1	2×10^6	0.137
Skrzypiński et al. (2014)	DU96-W-180	URANS	2	2×10^6	0.14
Skrzypiński et al. (2014)	DU96-W-180	URANS	8	2×10^6	0.143
Present work, 3D-1c	DU96-W-180	DDES	1	2×10^6	$\in [0.11, 0.16]$
Present work, 3D-1c, Gaussian fit	DU96-W-180	DDES	1	2×10^6	0.132
Skrzypiński et al. (2014)	DU96-W-180	DDES	1	2×10^6	0.153
Lian et al. (2023a)	DU96-W-180	DES	1	2×10^6	0.153
Skrzypiński et al. (2014)	DU96-W-180	DDES	2	2×10^6	0.159
Present work, 3D-21c	DU96-W-180	DDES	21	2×10^6	0.144

Table 4.3: List of Strouhal number values reported for wind turbine airfoils at $\alpha = 90^\circ$ using URANS CFD models (top part of the table) and DES-like models (bottom part of the table). For each CFD model, the studies are sorted in increasing aspect ratio (AR) order. Note that the “2D” entries in the aspect ratio column mean that the simulation is two-dimensional. In the case “Present work, 3D-1c”, a range of Strouhal numbers is given instead of a single value.

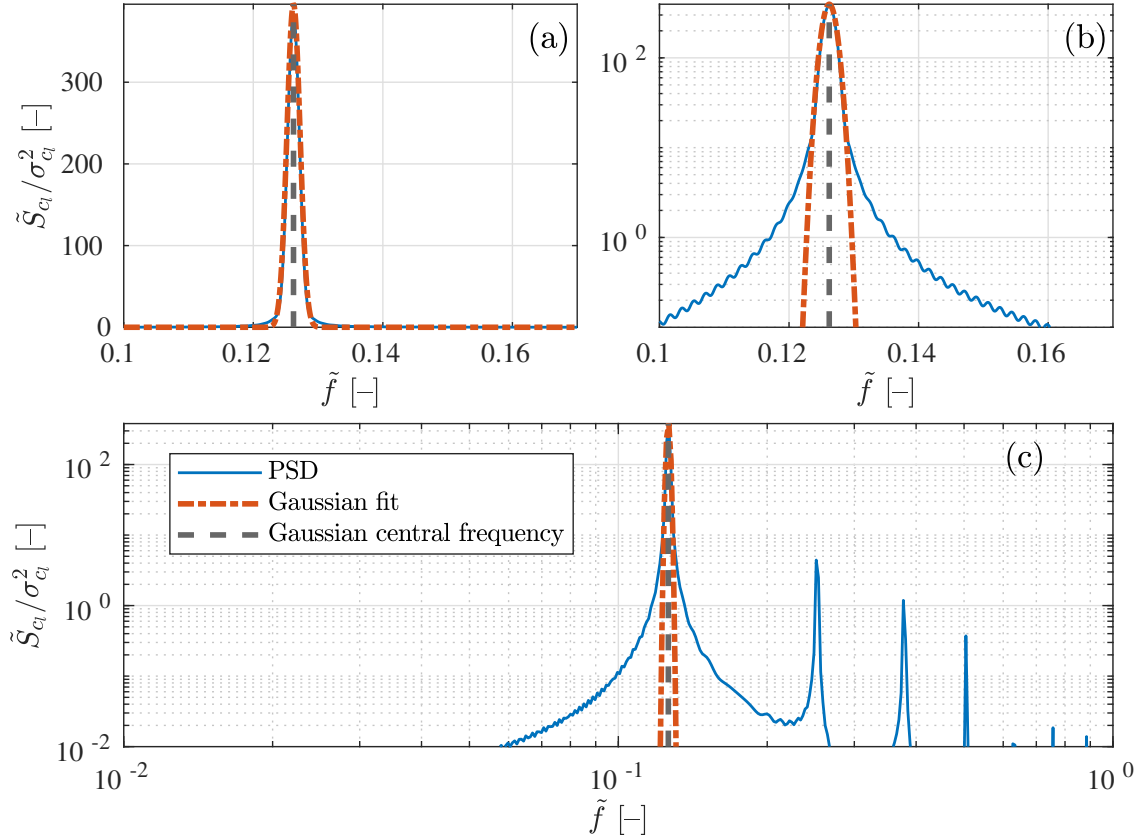


Figure 4.12: 2D static airfoil case. Normalized PSD of the lift coefficient fluctuations together with a Gaussian fit, in (a) linear axes, (b) logarithmic y-axis and (c) log-log axes and for a wider range of frequencies. The central frequency of the Gaussian fit is marked with a vertical line.

a clear peak and a broader underlying distribution of the spectral content. The Gaussian fit does not follow the PSD closely in any of the three representations, but still the central frequency of the Gaussian fit matches well the maximum of the PSD. The harmonics are not clearly visible in this case.

According to the analysis carried out in this work, the central frequency of the narrowband vortex-shedding spectrum (defined by the Gaussian fit) appears to be more significant than the frequency of its highest peak. The Gaussian fit is capable of defining the underlying vortex-shedding process in the 3D-1c case, making it robust to changes in the simulation time, while also capturing a prominent peak when it appears. Note that the simulation should be long enough to ensure convergence of the Gaussian-fit central frequency.

The Gaussian approximation of the vortex-shedding spectral content has been utilized since the early work of B. J. Vickery and Clark (1972) and forms the foundation of spectral engineering models for predicting VIV (Arunachalam & Lakshmanan, 2015; B. J. Vickery & Basu, 1983). This approach was also employed by Kurniawati et al. (2024) to identify the Strouhal number from field experimental data of a wind turbine tower. In these studies, the Strouhal number is defined as the central frequency of the Gaussian fit.

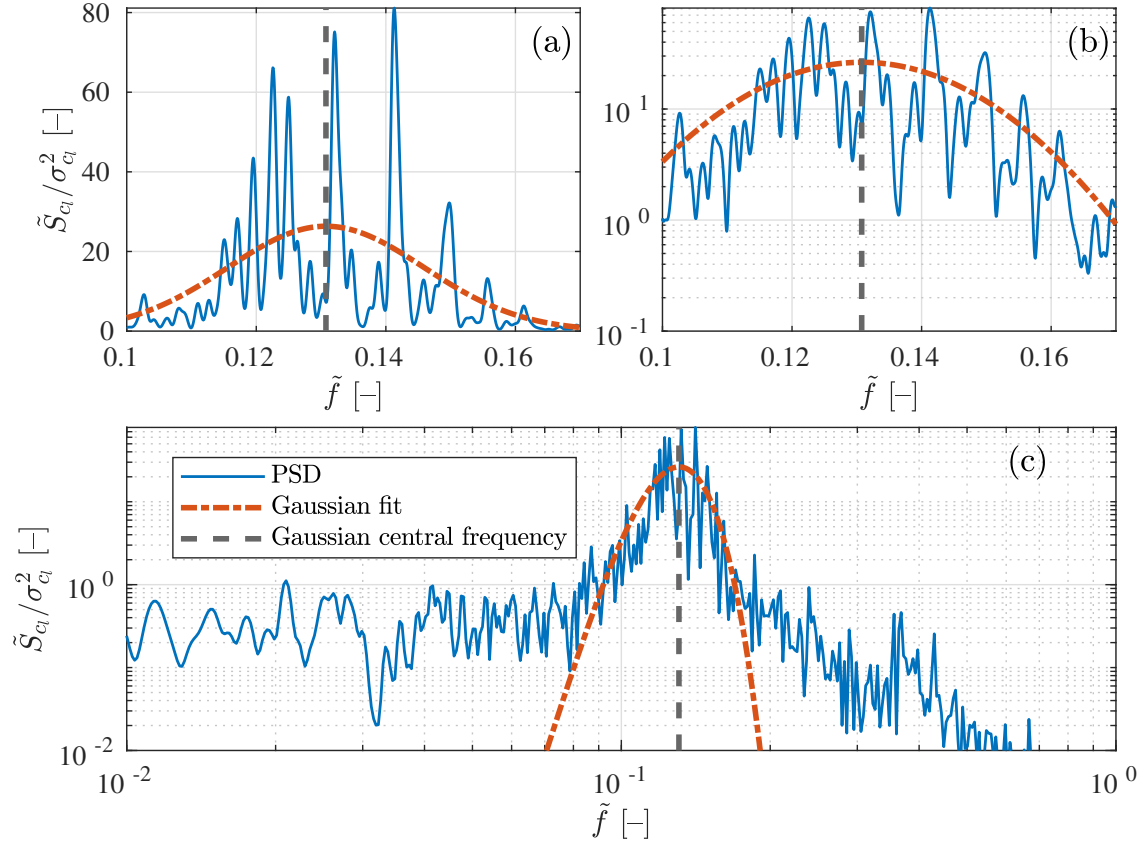


Figure 4.13: 3D-1c static airfoil case. Normalized PSD of the lift coefficient fluctuations together with a Gaussian fit, in (a) linear axes, (b) logarithmic y-axis and (c) log-log axes and for a wider range of frequencies. The central frequency of the Gaussian fit is marked with a vertical line.

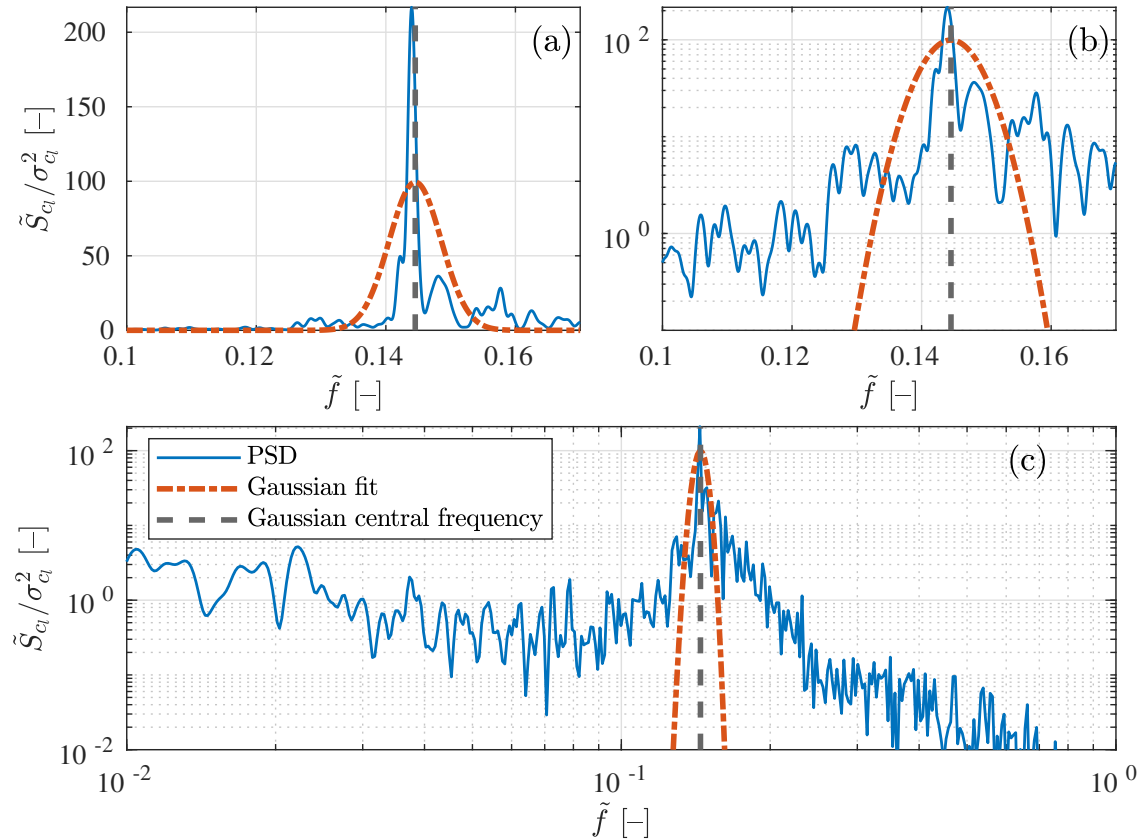


Figure 4.14: 3D-21c static airfoil case. Normalized PSD of the lift coefficient fluctuations together with a Gaussian fit, in (a) linear axes, (b) logarithmic y-axis and (c) log-log axes and for a wider range of frequencies. The central frequency of the Gaussian fit is marked with a vertical line.

In summary, this section has highlighted both similarities and differences in the static vortex shedding behaviour of the 2D, 3D-1c, and 3D-21c cases. The 3D-1c case exhibits intermittent shedding regimes, alternating between a high-drag regime (H) that closely resembles the 2D case, and a low-drag regime (L) that is more similar to the 3D-21c case. The spectral content of the 3D-1c lift coefficient is found to be distributed over a narrow frequency range, with distinct peaks emerging at different times, following approximately a Gaussian distribution. These insights will be instrumental in interpreting the VIV simulation results in the following sections.

4.2.4 Free to move airfoil case

In the results of this section, the fluid-structure interaction (FSI) response of the airfoil is simulated by coupling the CFD solver to the one-degree-of-freedom elastic system solver, as described in Section 4.1, being the airfoil free to move along its edgewise (chordwise) direction. Different simulations are performed varying the inflow velocity in order to capture the airfoil response inside and outside the frequency lock-in range, as in Lian et al. (2022, 2023a, 2023b). The 2D FSI simulations are run for 1300 dimensionless time units each, whereas the 3D simulations, due to computational resources limitations, are run for 300 dimensionless time units each. To avoid the initial simulation numerical transient, during which unrealistically high forces could induce unrealistic displacements, the airfoil is only allowed to move after 100 time steps, i.e. after one dimensionless time unit. From the relatively computationally inexpensive 2D simulations it is obtained that roughly the first 50 to 100 vortex shedding periods correspond to the initial growth transient where the displacement amplitude keeps increasing. After this, the response will become periodic and, depending on the velocity ratio, it may undergo low-frequency amplitude modulations. As the 3D simulations are much more computationally expensive, their final time is limited to about 40 vortex shedding periods, although some simulations are slightly shorter. This is expected to be enough to characterize the VIV transient response.

Initial displacement condition for the 3D-21c case

When running the first 3D-21c VIV simulations, it was observed that, unlike for the 2D and 3D-1c configurations, no vibrations were building up in the cases where resonance was anticipated; i.e., when the vortex-shedding frequency was close to the natural frequency of the spring-mass system. This behaviour can be qualitatively explained by the loss of vortex-shedding correlation expected as the spanwise length increases (Wu & Sharma, 2020). As a result, the 3D-21c case spanwise lift fluctuations are reduced, as was observed in Figure 4.7, since the overall lift is determined by integrating sectional contributions that become increasingly out of phase with greater spanwise separation. Given this observation, to obtain the VIV response for the 3D-21c case, the 3D-21c VIV simulations are run considering an initial non-zero edgewise displacement of the airfoil. It is expected that this initial airfoil motion will trigger some spanwise correlation in the shedding of vortices in the lock-in range—where vortex shedding synchronizes with airfoil motion—thus making the results comparable to those of the other aspect ratio cases. An initial condition of $\tilde{x}(t = 0) = 0.2$ chords in the x -direction was chosen; this value was found to be high enough to trigger the VIV self-excited

behaviour, as will be seen in the rest of this section, but also low enough to allow for a transient build-up of the response which can be compared to the other cases. For brevity, the 3D-21c case with the non-zero initial displacement condition will be referred to simply as “3D-21c” in the following sections.

Frequency ratio definition

To present the results for different inflow velocities in dimensionless form, the ratio between the vortex-shedding frequency defined by the static case Strouhal number and the structural natural frequency, f_{St}/f_s , is often used (Griffin et al., 1973; Lian et al., 2022). Here, $f_{\text{St}} = \text{St}U_\infty/c$, where the Strouhal number St is obtained from static simulations or experiments. A frequency ratio of one means that the vortex-shedding frequency if the body were stationary matches the structural natural frequency. The vortex-shedding frequency obtained from a static case, f_{St} , is expected to approximate well the free to move case outside the lock-in range. In the present work, for the 2D case, the Strouhal number used to determine the frequency ratio is that obtained from the static CFD simulations, $\text{St}_{2\text{D}} = 0.126$, as reported in Section 4.2.3. For the 3D-1c case, it is argued in Section 4.2.3 that there is not a single Strouhal number for this configuration, but it was observed in Figure 4.13 that the central frequency of the Gaussian fit to the PSD of the lift coefficient fluctuations seemed a robust measure of the vortex-shedding frequency. This frequency is now used to define the Strouhal number for the 3D-1c case, $\text{St}_{3\text{D-1c}} = 0.132$. For the 3D-21c case, the Strouhal number used is that obtained from the static simulations, $\text{St}_{3\text{D-21c}} = 0.144$, as commented in Section 4.2.3.

In what follows, different characteristics of the VIV response of the airfoil obtained with the 2D, 3D-1c and 3D-21c simulations will be compared, and the suitability of the value of $\text{St}_{3\text{D-1c}}$ will be discussed. The results will be presented in terms of the frequency ratio $f_{\text{St},i}/f_s$, where $i = \{2\text{D}, 3\text{D-1c}, 3\text{D-21c}\}$, and $f_{\text{St},i} = \text{St}_i U_\infty/c$ is the vortex-shedding frequency calculated using the corresponding Strouhal number.

Amplitude growth rate

One of the results that characterize the vortex-induced vibration response is the maximum amplitude of the airfoil displacement. In the present analysis, due to the large computational cost of the 3D simulations, the simulated time was not long enough to obtain the airfoil displacement maximum amplitude. This computational time limitation is common to other studies. Some authors choose to show the maximum amplitude reached after a certain simulation time has passed, and use this value for comparison between different cases (Lian et al., 2022; Skrzypiński et al., 2014). This may not always be a good choice, because the initial transients may be different for different simulations. It might also give an erroneous perception to the reader of the actual amplitude that would be reached, if it is not very clearly stated that the simulations have not converged to a stationary amplitude value. As an alternative, in this work, it is proposed to employ the dimensionless growth rate of the displacement amplitude as a characteristic of the VIV response. This metric can clearly indicate the VIV lock-in range extent, by differentiating cases with almost no amplitude growth (out of lock-in) from cases with relatively large growth (lock-in). This metric is more consistent between different sets of simulations, and does not require of long time series to be

obtained.

To obtain the dimensionless growth rate of the displacement amplitude from the simulation results, this growth rate is defined as the slope of the straight line which fits a convenient time interval of the magnitude of the Hilbert transform of the dimensionless displacement time series. Thus, to calculate the dimensionless growth rate of the displacement amplitude, we firstly calculate the magnitude of the Hilbert transform of the dimensionless displacement time series $\tilde{x} = x/c$. The resulting time series is denoted $h_{\tilde{x}}(t)$. This time series can be employed as an approximation of the dimensionless displacement envelope or amplitude (F. Zou et al., 2015). Then, a time interval of $h_{\tilde{x}}(t)$, corresponding to the initial displacement growth, is chosen. Finally, a straight line is fitted to this interval of the envelope, and the slope of the line is obtained. This slope, which is the approximation to the dimensionless amplitude growth rate, is denoted $r_{\tilde{x}}$.

In this work, the dimensionless time interval used for the growth rate estimation has been selected after visual inspection of the results. For the 2D and 3D-1c cases, the interval is $150 \leq \tilde{t} \leq 300$. For the 3D-21c case, the interval is $50 \leq \tilde{t} \leq 200$; this earlier start is due to the shorter transient caused by the initial displacement condition. The same time interval has been used for all inflow velocities within each case. Each of these 150 dimensionless time unit intervals corresponds to approximately 15 to 30 vortex shedding periods.

The growth rate calculation process is illustrated in Figure 4.15 for the 2D case. In its top panel, Figure 4.15(a), a time series of the dimensionless displacement $\tilde{x}$ is shown together with the positive part of its Hilbert transform magnitude, $h_{\tilde{x}}(t)$, in a thicker line. Additionally, the selected time interval ($150 \leq \tilde{t} \leq 300$, as mentioned before) is marked with vertical dashed lines. In the bottom panel, Figure 4.15(b), the same time series and envelope as that in the top panel figure is shown, but cropped to only show the selected time interval, together with the straight line fitted to this portion of the envelope using a fine continuous line. The slope of this straight line is the dimensionless amplitude growth rate, $r_{\tilde{x}}$, which is the metric proposed to determine the lock-in range and to compare the different modelling approaches studied. Figures 4.16 and 4.17 show the same information as Figure 4.15, but for the 3D-1c and 3D-21c cases, respectively. The amplitude growth in all the simulations presented in the present work are reasonably well approximated by straight lines. VIV displacement responses with an initial growth rate which is largely non-linear have been reported, e.g. in Derksen (2019) for a 2D cylinder with low mass ratio values. In those cases, an exponential or some other curve could also be fitted to the amplitude time series, achieving the same goal.

This procedure is followed for every simulation, i.e., for every frequency ratio $f_{St,i}/f_s$. The resulting dimensionless displacement amplitude growth rates are presented in Figure 4.18 for the three cases studied. It is observed that all cases show a similar behaviour, with the growth rate increasing as the frequency ratio approaches one, and then decreasing for high frequency ratios. It can be noticed that there is a very good agreement between the 2D and 3D-1c cases. The agreement with the 3D-21c case is also noteworthy, although, in this case, for low frequency ratios the growth rate does not drop to zero, suggesting that the lock-in range is extended in the low velocity side. This can be a consequence of the initial displacement condition, as can be observed from the shape of previously reported forced-vibration aero-damping maps (Lian et al., 2023b). Similarly for high frequency ratios,

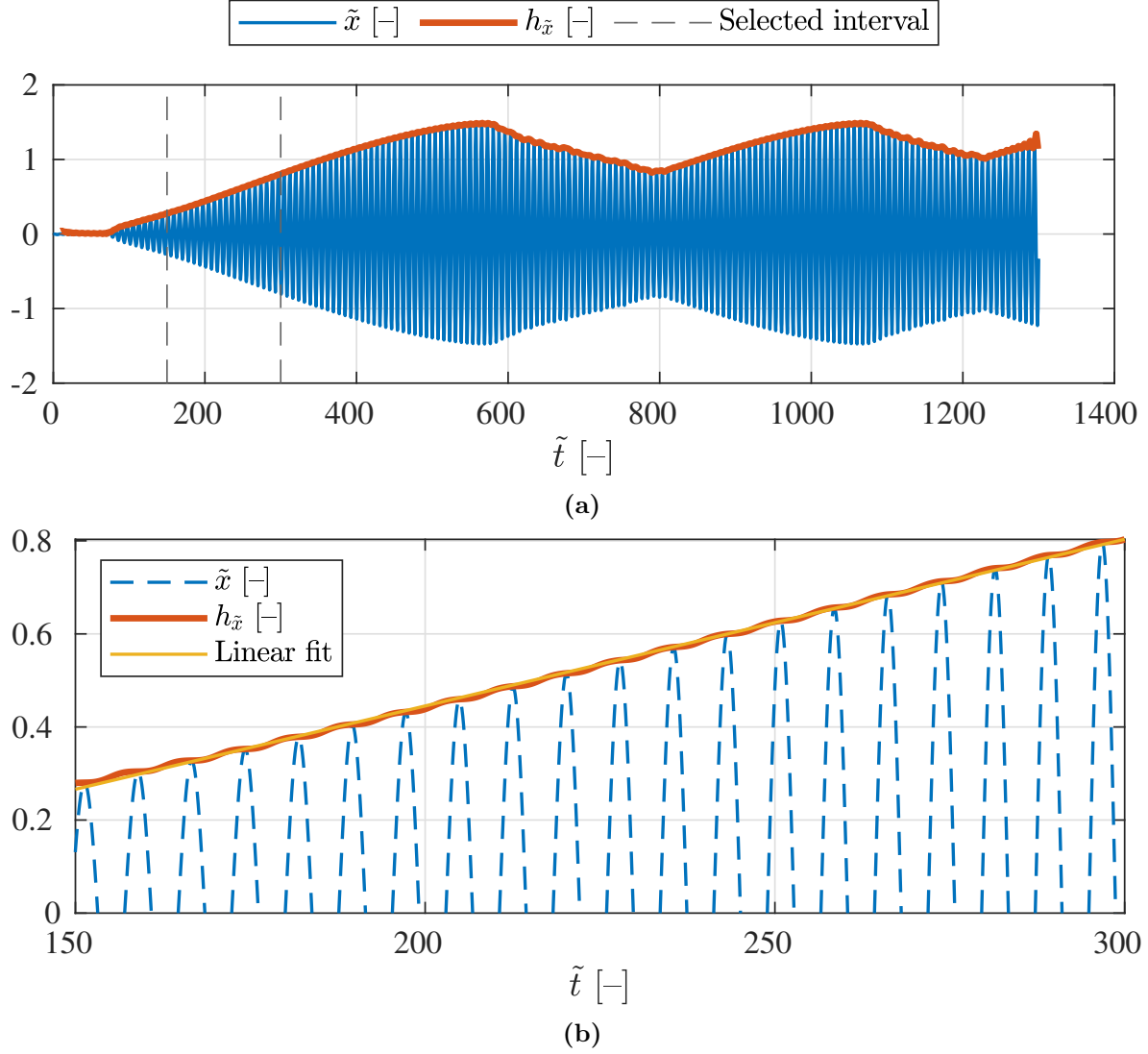


Figure 4.15: (a) Time series of the dimensionless edgewise displacement, $\tilde{x}$, along with, in a thicker line, the envelope of the time series calculated with the Hilbert transform, $h_{\tilde{x}}$. Additionally, the limits of the time interval selected for the linear fit to $h_{\tilde{x}}$ are marked with vertical dashed lines. (b) Time series of the dimensionless edgewise displacement $\tilde{x}$ within the time interval selected for the linear fit, along with the estimated amplitude $h_{\tilde{x}}$, and the straight line fitted to $h_{\tilde{x}}$ in this portion of envelope. All results are for a 2D FSI simulation at $f_{\text{St},2\text{D}}/f_s = 0.98$.

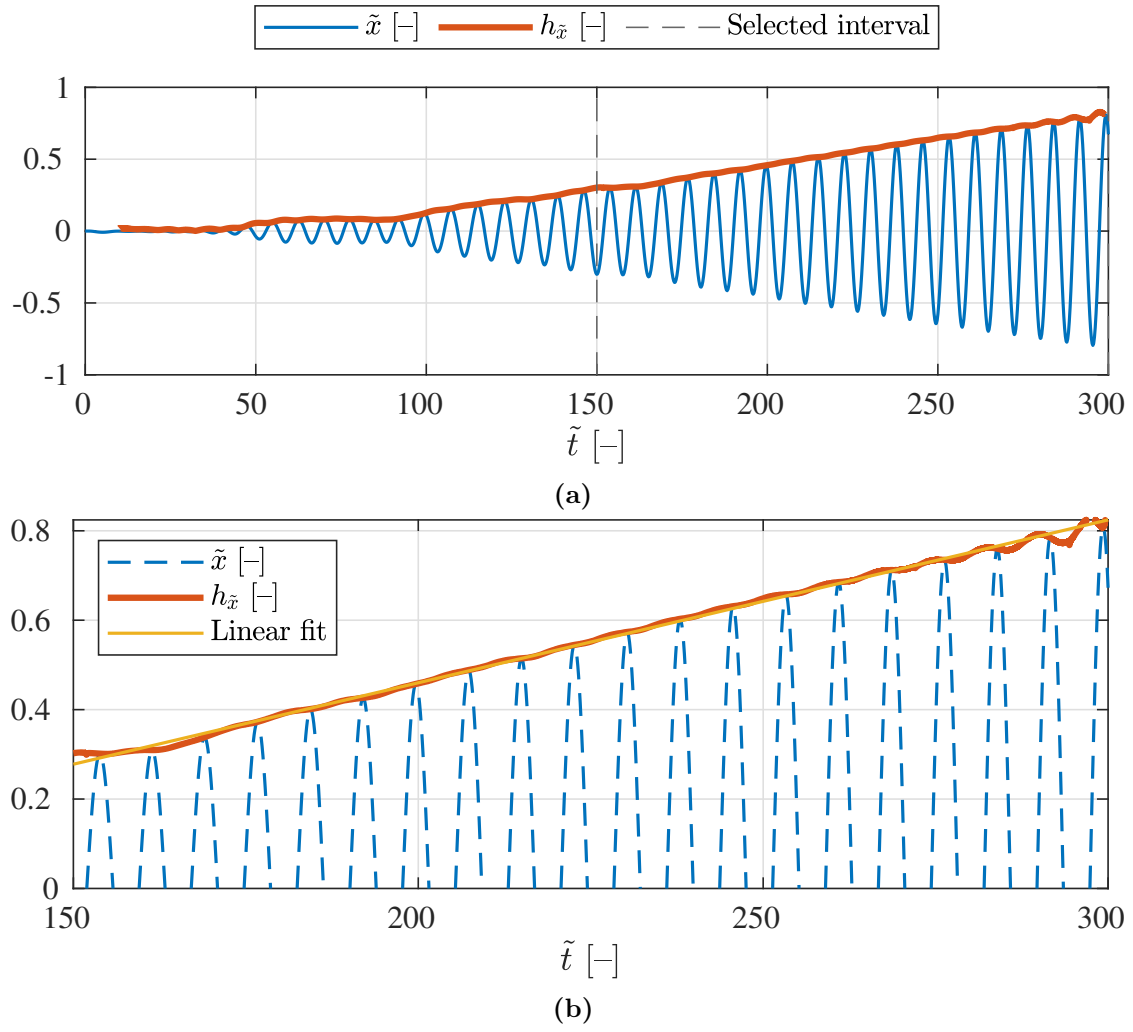


Figure 4.16: Same as Figure 4.15 but for a 3D-1c FSI simulation at $f_{\text{St},3\text{D-1c}}/f_s = 1.02$.

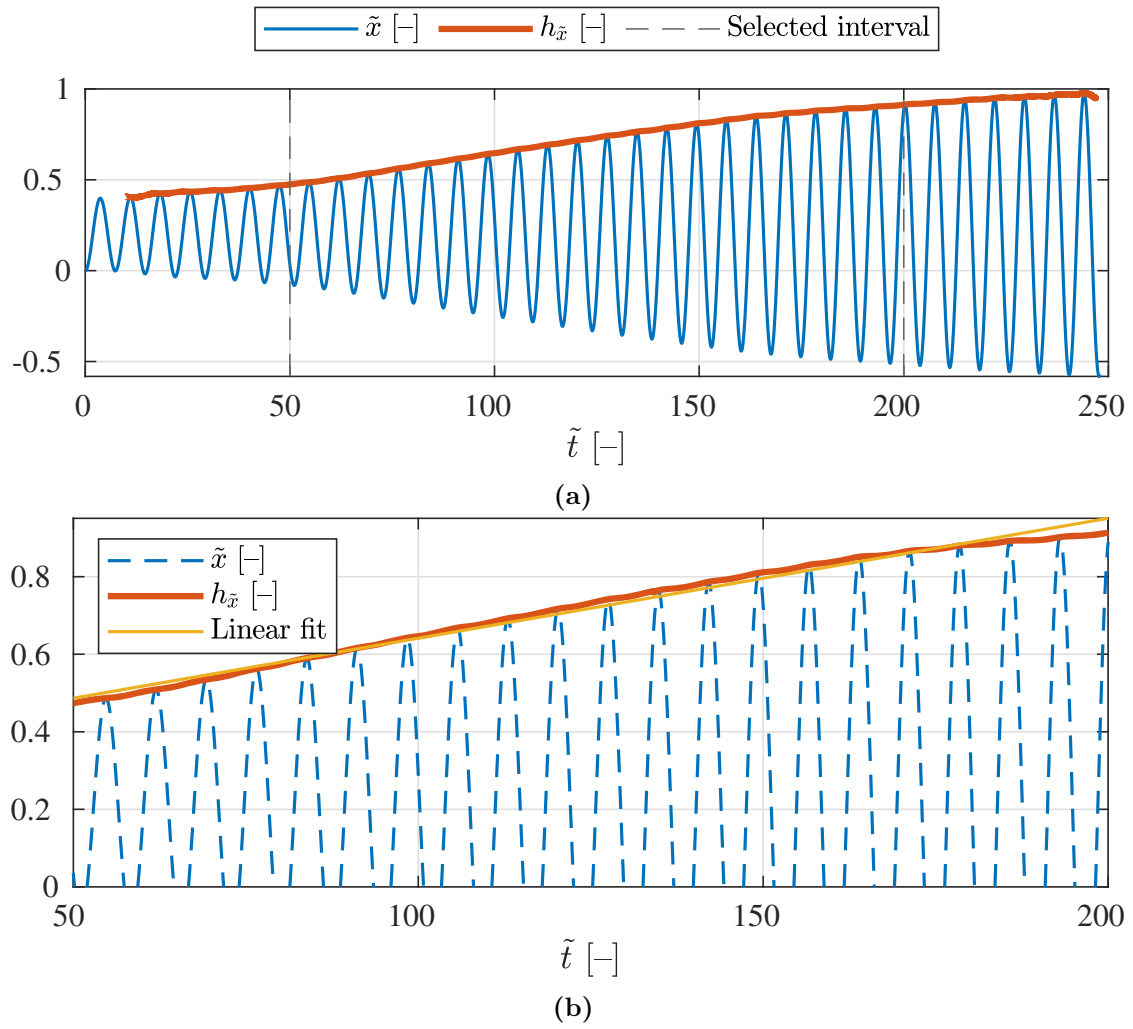


Figure 4.17: Same as Figure 4.15 but for a 3D-21c FSI simulation at $f_{\text{St},3\text{D-21c}}/f_s = 1.05$.

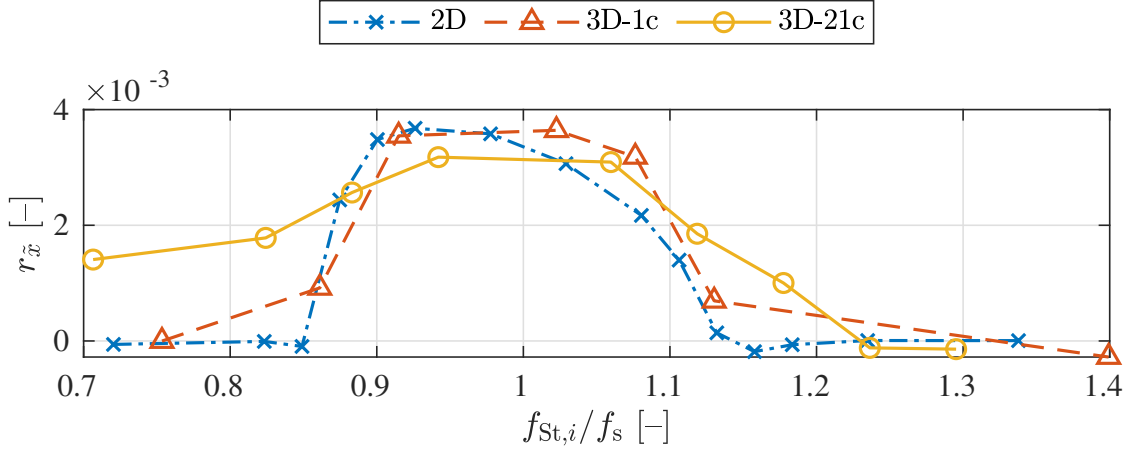


Figure 4.18: Dimensionless displacement amplitude growth rate, $r_{\tilde{x}}$ as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches.

the lock-in range is also slightly extended for the 3D-21c case. Nevertheless, for frequency ratios over 1.2, the 3D-21c case growth rate takes slightly negative values, which implies that the initial condition is being damped out, although the small magnitude of these growth rate values suggests that this damping process is much slower than the build-up of vibrations within the lock-in range. The results for the 3D-21c might be affected by the coarser spanwise mesh density, which may prevent small-scale vortical structures from appearing and breaking the spanwise correlation, thus resulting in an increased lock-in range originating from the initial condition. The lock-in range is here thus defined as the range of frequency ratios where the growth rate is appreciably larger than zero. Considering the results presented in Figure 4.18, the lock-in range is established to be about $f_{St,i}/f_s \in [0.85, 1.15]$ for the 2D and 3D-1c approaches.

The results demonstrate that there is a very good agreement in the transient structural response of the 2D and 3D-1c cases in the lock-in range extent, as well as a good agreement with the 3D-21c case. In the following section, the VIV process will be further characterized through the analysis of the aerodynamic forces experienced by the airfoil.

Aerodynamic coefficients characterization

Some important metrics for characterizing the VIV response, aside from the structural displacement, are the lift coefficient standard deviation σ_{c_l} and the mean drag coefficient $\bar{c}_d$. The lift coefficient standard deviation quantifies the intensity of the lift force fluctuations and is often used in engineering models to predict VIV response. The interaction between structural displacement and lift force amplitude is crucial for accurate VIV predictions. Although the mean drag coefficient is not typically included in VIV engineering models for 1 degree of freedom systems, it is relevant due to the significant increases that can occur during VIV (Blevins, 2001). While the standard deviation of the drag coefficient may be important for fatigue calculations and for predicting systems free to vibrate in-line, its analysis is beyond the scope of this thesis.

The lift coefficient standard deviation can be related to the lift force amplitude, assuming that the mean lift coefficient is approximately zero, and thus the growth rate of the lift coefficient amplitude can be used as an alternative metric to characterize the variation in time of σ_{c_l} . In a manner analogous to the displacement amplitude growth rate, the lift coefficient amplitude growth rate, r_{c_l} , is calculated as the slope of the linear fit to the magnitude of the Hilbert transform of the lift coefficient time series. In Figure 4.19, the lift coefficient amplitude growth rate calculation procedure is illustrated for the 2D case. The time series of the lift coefficient for a case inside the lock-in range is shown in Figure 4.19(a), together with the envelope of the time series calculated with the Hilbert transform, $h_{c_l}(t)$. The selected time interval for the linear fit to $h_{c_l}(t)$ is marked with vertical dashed lines. The same time interval as for the displacement amplitude growth rate is used here. In Figure 4.19(b), the same time series and envelope as that in the top panel is presented, but cropped to only show the selected time interval, together with the straight line fitted to this portion of the envelope using a thin line. The slope of this straight line is the lift coefficient amplitude growth rate, r_{c_l} . The amplitude of c_l is seen to increase substantially during the VIV transient, making its growth rate a useful metric to characterize the VIV response. The magnitude of the Hilbert transform is seen to be more noisy than for the displacement amplitude, but the linear fit still provides a good approximation to the growth rate.

The lift coefficient amplitude growth rate is presented in Figure 4.20 as a function of the frequency ratio for all cases, analogous to the displacement amplitude growth rate shown in Figure 4.18. A good agreement is found between the 2D and 3D-1c case results inside the lock-in range, except for one outlier in the 3D-1c case at $f_{St,3D-1c}/f_s = 1.02$, which deviates significantly from the 2D results. This deviation is attributed to the intrinsic variability of the lift force in the 3D-1c case, as observed in the static case results, particularly at the beginning of the growth transient. The 3D-21c case lift coefficient growth rate shows a similar tendency to the displacement growth rate shown in Figure 4.18, although the lock-in range extent is less well-defined in this case. Due to the less well-defined fluctuations of the lift coefficient compared to the structural displacement, the lift coefficient amplitude growth rate is noisier, and the linear fit is less accurate. Nonetheless, the similarity between the 2D and 3D-1c cases is still notable.

For the mean drag coefficient, the procedure to obtain the growth rate is different. In this case, instead of employing the magnitude of the Hilbert transform, which would give the time-varying amplitude or envelope of the drag coefficient, the time-varying mean drag coefficient value is tracked using a moving average. The mean drag coefficient growth rate, $r_{\bar{c}_d}$, is then estimated as the slope of the linear fit to said moving average. The moving average is calculated using a window of $T_{\text{movmean},i} = 3/St_i$, which corresponds to approximately three vortex shedding periods. An example of the mean drag coefficient growth rate calculation procedure is shown in Figure 4.21 for the 2D case. In the top panel, Figure 4.21(a), the time series of the mean drag coefficient is shown, together with the moving average of the time series. The selected time interval for the linear fit to the moving average is marked with vertical dashed lines, which is the same interval as for the displacement and lift coefficient amplitude growth rates. In the bottom panel, Figure 4.21(b), the same time series and moving average as that in the top panel figure are shown, but cropped to only show the selected time interval, together with the straight line fitted to this portion of the moving average using a

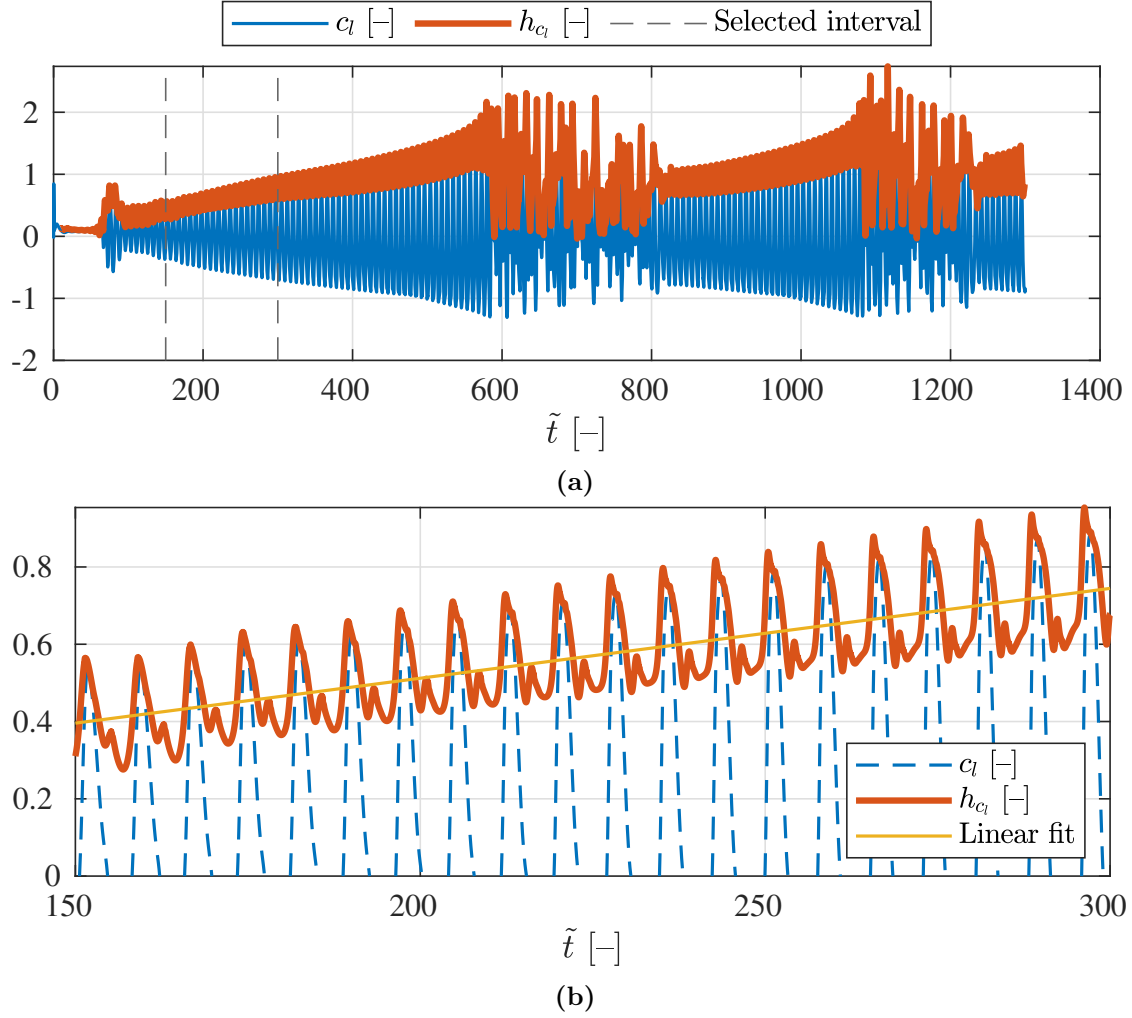


Figure 4.19: Illustration of the lift coefficient growth rate calculation procedure. See Figure 4.15 for details. All results are for a 2D FSI simulation at $f_{St,2D}/f_s = 0.98$.

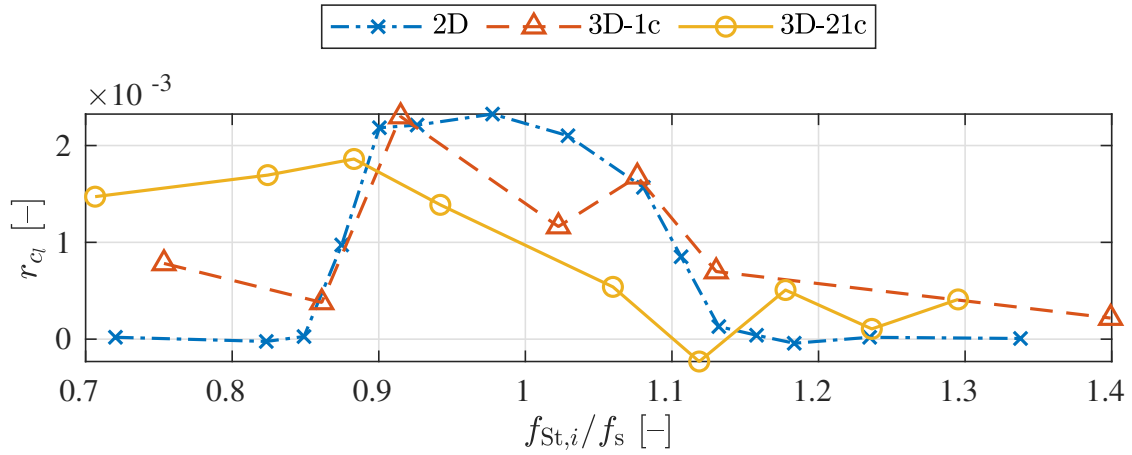


Figure 4.20: Lift coefficient amplitude growth rate, r_{c_l} as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches.

thin line. The slope of this straight line is the mean drag coefficient growth rate, $r_{\overline{c_d}}$. The mean drag coefficient growth rate is a useful metric to characterize the VIV response, as it quantifies the increase in drag force during VIV. The moving average is seen to provide a good approximation to the mean drag coefficient time series, and the linear fit is accurate for the selected time interval.

The mean drag coefficient growth rate is presented in Figure 4.22 as a function of the frequency ratio for all cases, analogous to the previous plots for the displacement amplitude growth rate (Figure 4.18) and the lift coefficient growth rate (Figure 4.20). In this figure, we can observe that the 2D, 3D-1c and 3D-21c cases show a similar behaviour, with the drag growth rate increasing notably inside the lock-in range. The mean drag coefficient growth rate is seen to provide a good approximation to the lock-in range extent.

The growth rates of the displacement amplitude, lift coefficient amplitude, and mean drag coefficient are of particular interest in characterizing the transient behaviour of vortex-induced vibrations (VIV). These growth rates provide insight into the initial build-up of vibrations and the aerodynamic forces acting on the airfoil. By analysing these growth rates, we can also compare different modelling approaches and assess their accuracy in predicting the VIV behaviour of the airfoil.

Due to the limited length of the 3D simulations, it is not possible to obtain the values of the lift coefficient standard deviation and mean drag coefficient for its converged limit-cycle behaviour. Nevertheless, it is interesting to compare the differences in the values of these statistics during the VIV growth transient, between the different configurations studied, and inside and outside the lock-in range. Aiming to make a fair comparison, the statistics for the 2D and 3D-1c simulations are calculated in all cases using the data from the dimensionless time interval $\tilde{t} \in [150, 300]$, and in the interval $\tilde{t} \in [50, 200]$ for the 3D-21c case, as was done for the growth rate calculations, to account for the different transient duration. These two parameters are presented in Figure 4.23, for the three cases studied, as a function of the frequency ratio. It is observed that the lift coefficient standard deviation and mean drag coefficient values increase notably inside the lock-in range, for all configurations, as predicted by the growth rates shown in Figures 4.20 and 4.22, and exemplified for the 2D case in Figures 4.19 and 4.21. The results from the 2D and 3D-1c cases match to a high degree in the vicinity of the resonant frequency, i.e., inside the lock-in range, whereas large discrepancies appear for very small and very large frequency ratios. The 3D-21c case results inside the lock-in range are similar to those of the 2D and 3D-1c cases, its values being a bit lower, and outside the lock-in range these statistics drop even lower. All of these differences can be explained quite satisfactorily as a consequence of spanwise load correlation and vortex shedding intensity and spatial coherence. The 2D case can be considered analogous to a fully correlated case, and so yields the highest loads, which further increase inside the lock-in range due to the increased intensity and spatial coherence of the vortex-shedding process. This increase in loads is a well-known phenomenon of 2D CFD simulations (Mittal & Balachandar, 1995). The 3D-1c case, with its short spanwise direction and periodic boundary condition, may become highly correlated when the vortex-shedding is dictated by the large-amplitude motion of the airfoil, i.e., inside the lock-in range, thus resulting in very similar values to the 2D case. Outside the lock-in range, the three-dimensional characteristics of the wake reduce

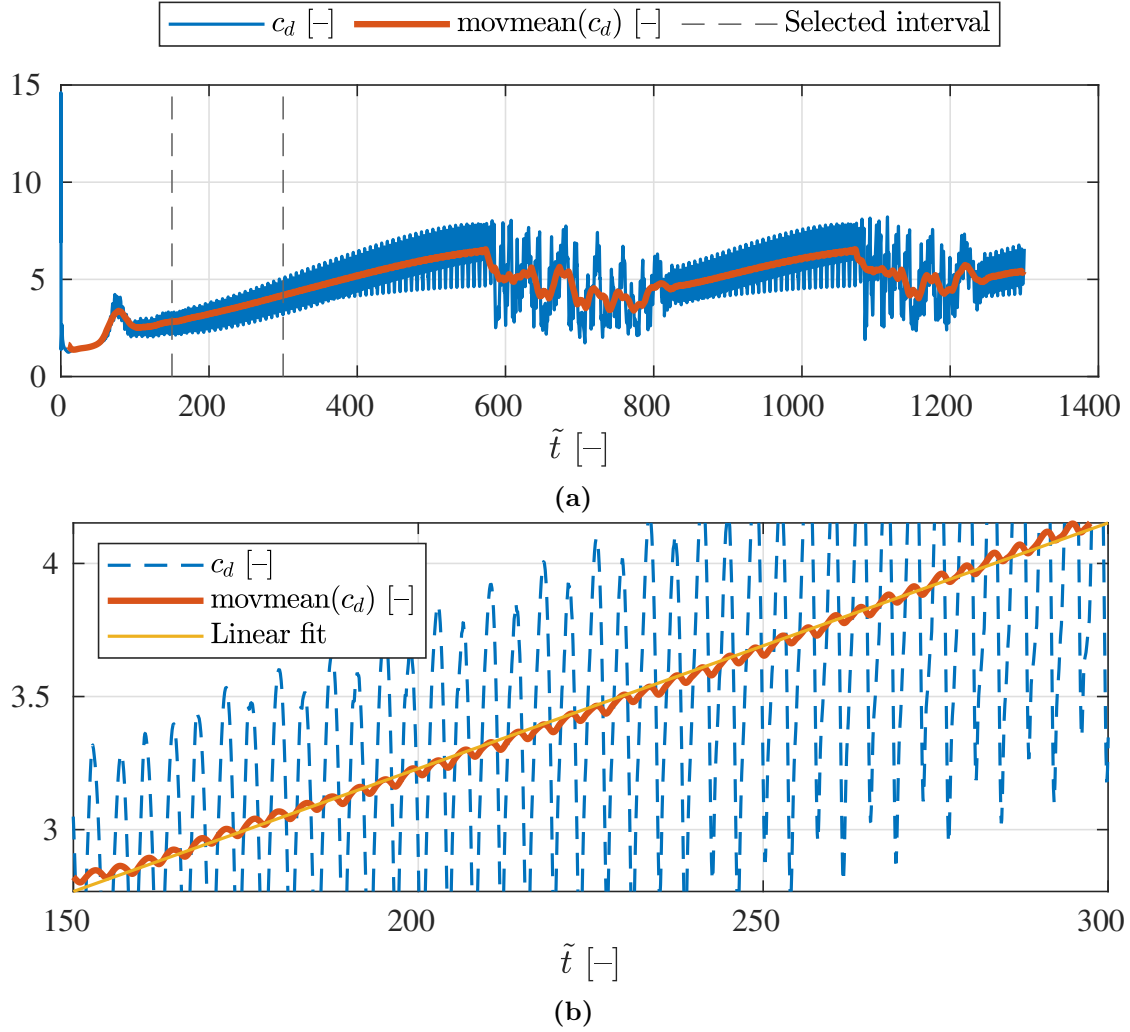


Figure 4.21: Illustration of the mean drag coefficient growth rate calculation procedure. See Figure 4.15 for details. In this case, the linear fit is made to a time-moving average of c_d . The window used for the moving average is $T_{\text{movmean}} = 3/\text{St}_{2\text{D}}$, which is approximately three vortex shedding periods. All results are for a 2D FSI simulation at $f_{\text{St},2\text{D}}/f_s = 0.98$.

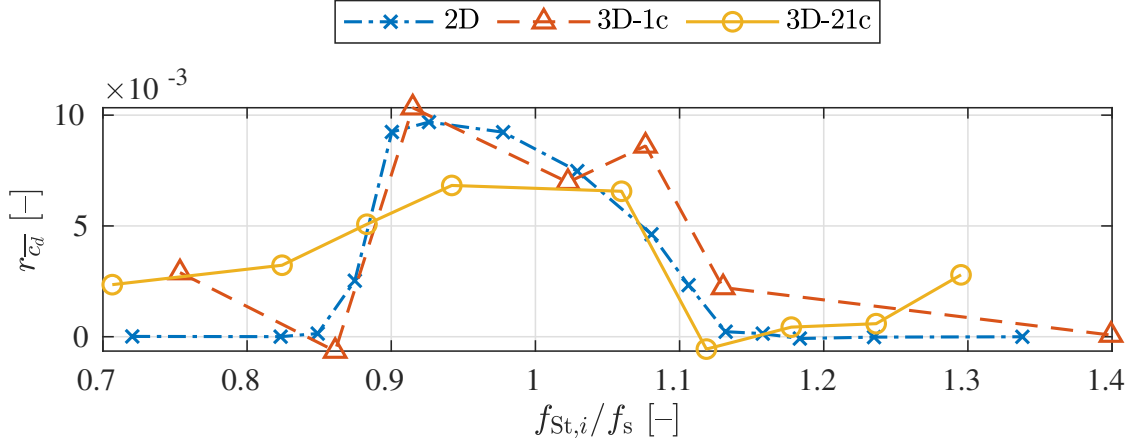


Figure 4.22: Mean drag coefficient growth rate, $r_{\overline{c_d}}$ as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches.

the spanwise correlation of the loads, as well as their intensity through the reduced spatial coherence, and therefore the values of the lift coefficient standard deviation and mean drag coefficient drop notably below those of the 2D case. In the 3D-21c case, considering the initial displacement condition, the spanwise averaged loads are expected to be lower than in the 3D-1c case, even if there is also an increase in spanwise correlation. This is reflected in the lower values of σ_{c_l} and $\overline{c_d}$ compared to the 3D-1c case, even inside the lock-in range. Note that for the low frequency ratio cases, the 3D-21c case is still experiencing lock-in behaviour, as was observed from the growth rate values in Figure 4.18. In summary, it is observed that the higher the aspect ratio, the lower the lift coefficient standard deviation and mean drag coefficient values become, although these differences are reduced by the increased correlation of the vortex shedding in lock-in conditions. The evaluation of this hypothetical explanation will be further substantiated by the analysis of flow field visualizations in Section 4.2.5. The actual values obtained are affected by the lateral boundary condition effect for the 3D-1c case, and the spanwise mesh density for the 3D-21c case, especially for the cases outside lock-in. Nevertheless, it is worth noting that the mean drag coefficient predicted by the 3D-21c case outside lock-in (which is the same as in the static case) is very similar to those reported from wind tunnel tests of static airfoil sections (Timmer & van Rooij, 2001; G. Xu et al., 2025).

To further illustrate the similarities of the VIV airfoil response predicted by the 2D and 3D-1c approaches inside the lock-in range, one example of the time series of the dimensionless displacement and the aerodynamic coefficients c_l and c_d are shown in Figure 4.24, cropped in time for better visualization. Note that the frequency ratio $f_{St,i}/f_s$ is not exactly the same for both models, but similar enough to be comparable, and well within the lock-in range. In Figure 4.24, the x -axis is the time t , and not the dimensionless time $\tilde{t}$ (as in previous figures), because inside the lock-in range the vortex-shedding frequency is locked-in to the natural frequency of the structure f_s , which is the same for both configurations. Therefore, the oscillation period is the same for both approaches and equal to $T = 1/f_s$, but the dimensionless oscillation period, $\tilde{T} = U_\infty/(f_s c)$, will be different for any given frequency ratio $f_{St,i}/f_s$, because the velocity at which this ratio is achieved, $U_\infty = f_{St,i} c / St_i$, depends on the

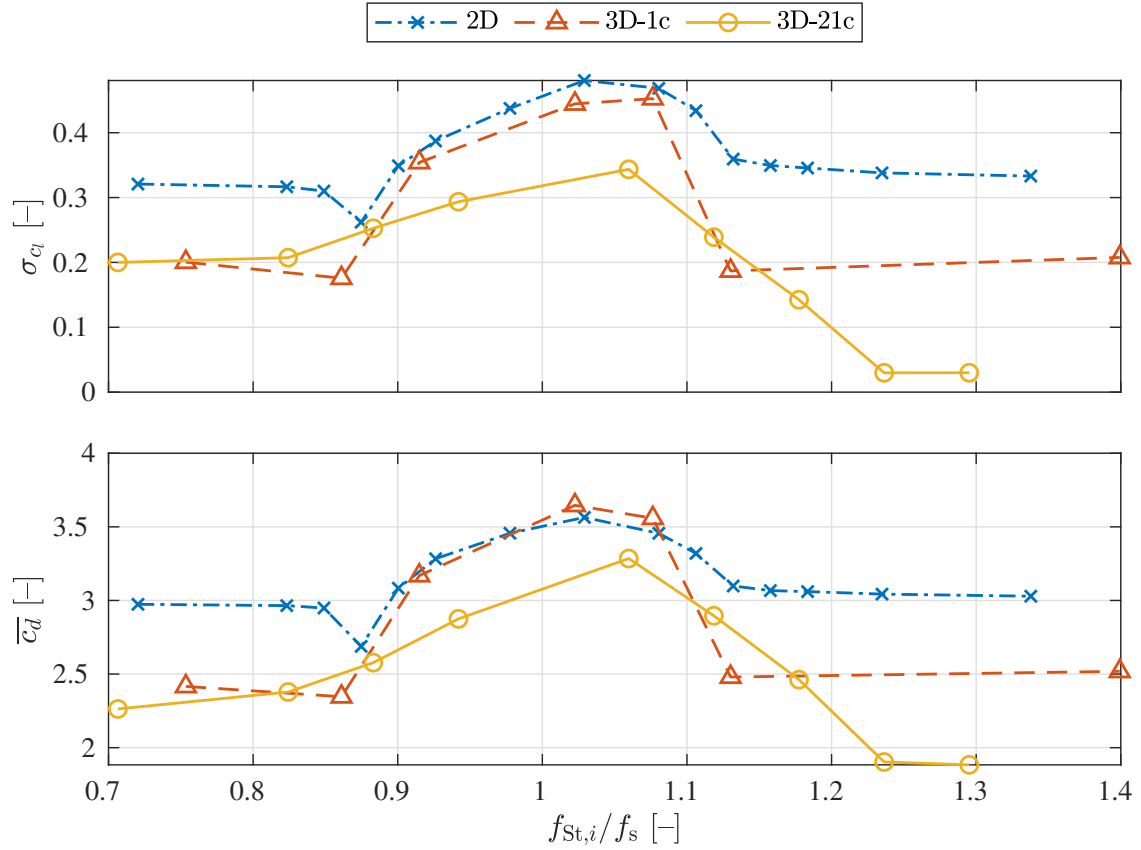


Figure 4.23: Lift coefficient standard deviation (top) and mean drag coefficient (bottom) as a function of the ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency, for the three modelling approaches. In the 2D and 3D-1c cases the time series used to calculate these statistical parameters are limited to the dimensionless time interval $\tilde{t} \in [150, 300]$, and in the 3D-21c case it is limited to $\tilde{t} \in [50, 200]$.

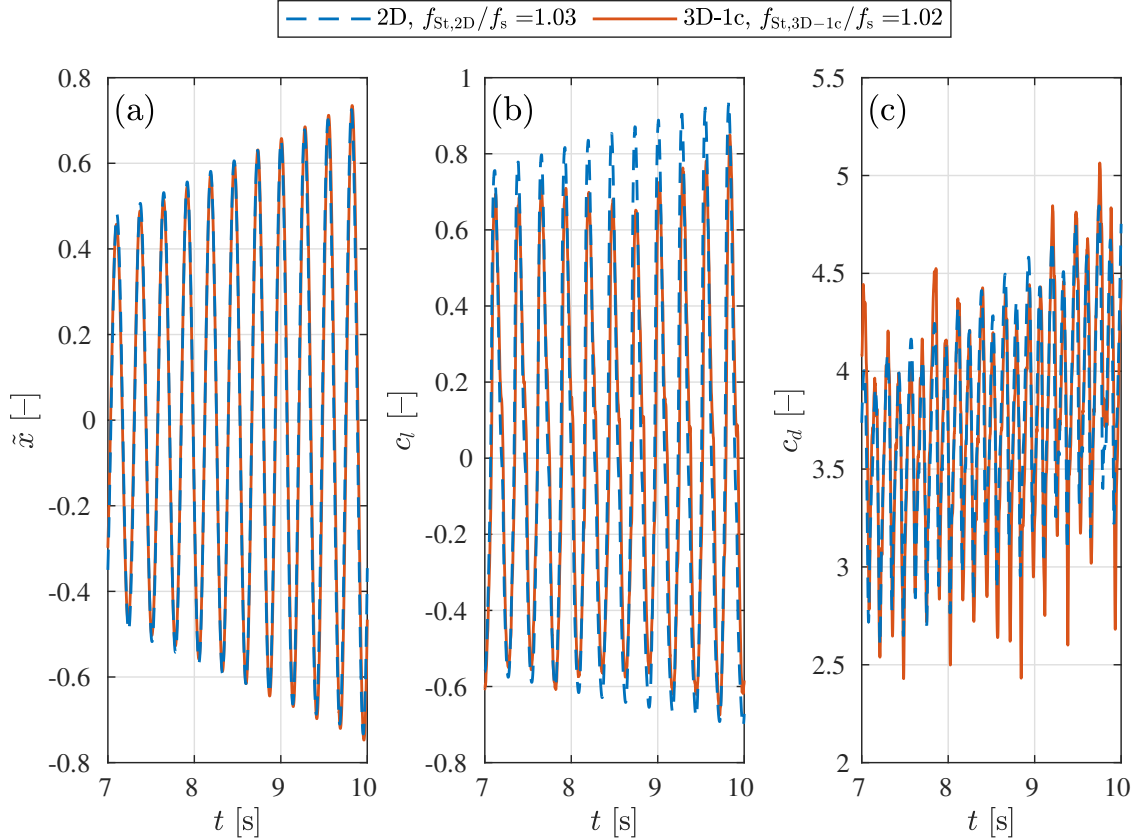


Figure 4.24: Time series segment of the airfoil (a) dimensionless displacement, (b) lift coefficient and (c) drag coefficient, for the 2D and 3D-1c approaches, at a ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency close to one (inside the lock-in range).

Strouhal number chosen for each approach, which are different in the present work. Note that in the comparison shown in Figure 4.24, the time series predicted by both approaches happen to be in phase, but due to differences in the initial transients this does not generally occur. It is observed that both 2D and 3D-1c approaches predict similar time series for $\tilde{z}$, c_l and c_d . The similarity in the oscillation frequency was expected, as explained above. It was also expected, following the results in Figures 4.18, 4.20 and 4.22, that the displacement, lift coefficient amplitude and mean drag coefficient growth rates would be similar. It can be seen that the 3D-1c results show a slightly less periodic behaviour of both aerodynamic coefficients, with much more pronounced variations from one period to another than in the 2D case. On the other hand, the displacement time series is notably smooth in both cases. The similarities of the aerodynamic coefficient time series inside the lock-in range are in contrast to the results previously shown in Figure 4.7 for the static airfoil case, where the differences were noticeably larger. This may be attributed to the locking of the wake to a single vortex-shedding regime –propitiated by the large airfoil motion and increased spanwise correlation– which happens to be almost identical for the 2D and 3D-1c simulations.

To represent the relationship in time between the airfoil dynamics and the lift force, a phase plane is shown in Figure 4.25, for both the 2D and 3D-1c cases. The frequency

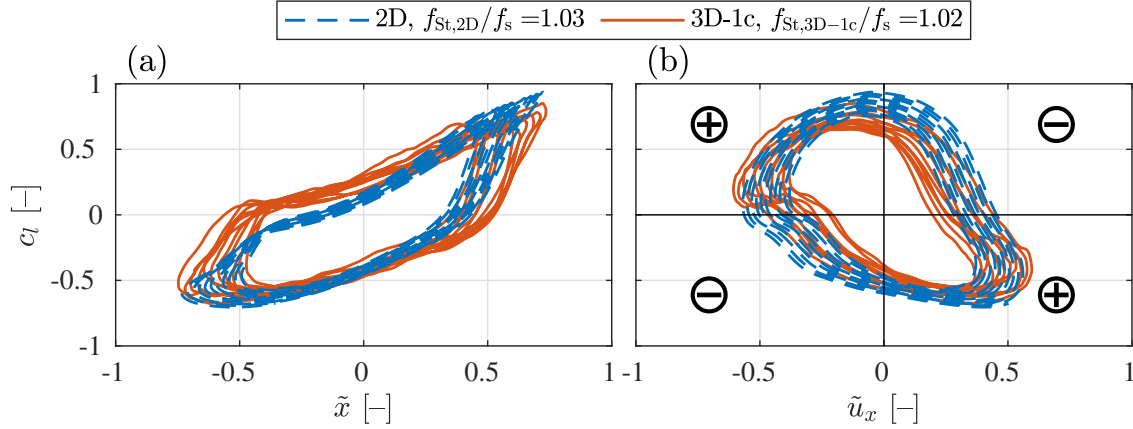


Figure 4.25: Phase plane of lift coefficient versus (a) dimensionless displacement and (b) dimensionless velocity, for simulation time between $t = 7$ and $t = 10$ s, for the 2D and 3D-1c approaches, at a similar ratio between the Strouhal-defined vortex-shedding frequency and the structural natural frequency inside the lock-in range. The quadrants with the + sign correspond to segments of the time series where the lift coefficient is adding energy to the motion, and the quadrants with the – sign correspond to those where energy is extracted from the motion.

ratios chosen (corresponding to the lock-in range), and the time series segment length shown ($7 < t < 10$ s), are the same as those of Figure 4.24. The left panel contains the phase plane of the dimensionless displacement and the lift coefficient, where the time series are shown as lines evolving anticlockwise. Note that the lift force is defined as positive when pointing in the negative x direction. The right panel contains the phase plane of the lift coefficient and the dimensionless airfoil velocity $\tilde{u}_x = d\tilde{x}/d\tilde{t}$, where the time series are also shown as lines evolving anticlockwise. The aerodynamic power added from the flow to the airfoil motion can be defined as the product of the airfoil velocity times the aerodynamic force projected in the direction of motion (Skrzypiński et al., 2014). With the axis system used in the present work, energy is added to the airfoil when the lift coefficient has the opposite sign as the airfoil velocity. In Figure 4.25 (b), the quadrants with the circled + sign correspond to time series segments where the fluid is adding energy to the airfoil motion and those with the circled – sign are time segments where the airfoil is losing energy. The shape of the phase plane can provide information about the phase difference between the two variables represented. In this case, the deviation from the purely elliptical shape means that the phase is changing within the duration of each oscillation period. This deviation is due to the not perfectly sinusoidal nature of the lift coefficient, which is often better modelled as a van der Pol type oscillator (Hartlen & Currie, 1970; Skop & Balasubramanian, 1997), resulting in the aforementioned time-dependent phase difference between lift and displacement or velocity (as displacement and velocity are almost perfectly sinusoidal). In both panels of Figure 4.25, an excellent agreement is observed between the 2D and 3D-1c cases at the magnitudes of the lift coefficient, dimensionless displacement and dimensionless velocity, as well as their relative phases.

In conclusion, the free to move airfoil simulations have demonstrated that both the 2D and 3D-1c approaches predict similar VIV responses in terms of airfoil motion and aerodynamic forces within the lock-in range. The 3D-21c case, with an initial displacement condition, also

shows comparable results, although with slightly lower aerodynamic loading, possibly due to spanwise averaging. These findings highlight the importance of considering three-dimensional effects and initial conditions in VIV simulations to accurately capture the airfoil's dynamic response. The next section will delve into the three-dimensional flow field analysis to further investigate the underlying mechanisms driving these observations.

4.2.5 Flow field analyses

As suggested in Section 4.2.3, the amplitude of the loads on the airfoil is expected to be related to the coherence of the vortical structures. For instance, the vortex shedding predicted by the 3D approaches should be more coherent within the lock-in range, where the lift coefficient amplitude is greater, compared to outside the lock-in range or for the static case (Lian et al., 2023a), at least during the initial build-up of vibration amplitude. This enhanced coherence might explain why the 2D and 3D predictions of the airfoil displacement and aerodynamic coefficients closely match in the lock-in region. Note that an increased coherence of the vortical structures also results in an increase in their spanwise correlation.

To qualitatively observe the coherence in the shedding of vortices for the 3D cases, flow visualizations of four cases with different expected levels of coherence are shown in Figure 4.26. The four cases shown correspond to, from top to bottom, (i) the 3D-21c static simulation, (ii) the 3D-1c static simulation at time instant $\tilde{t} = 300$ (i.e. within the L regime), (iii) the 3D-1c static simulation at time instant $\tilde{t} = 500$ (i.e. within the H regime), and (iv) the 3D-1c free to move simulation for the frequency ratio $f_{\text{St},3\text{D-1c}}/f_s = 1.08$ (i.e. within the lock-in range). In Figure 4.26(a), snapshots of the pressure coefficient field c_p are presented in a slice in spanwise position $z = 0.5c$ of the 3D airfoil. This slice corresponds to the midspan section for the 3D-1c case. The pressure coefficient is defined as $c_p = 2(p - p_\infty) / (\rho_\infty U_\infty^2)$, where p is the pressure field and p_∞ is the pressure at the inlet. In Figure 4.26(b), snapshots of the Q -criterion isosurfaces, for a value of the dimensionless second invariant of the velocity gradient tensor $\tilde{Q} = Qc^2/U_\infty^2 = 1$, are superimposed on snapshots of the dimensionless spanwise velocity field, $\tilde{U}_z = U_z/U_\infty$, at the $z = 0.5c$ slice. The Q -criterion isosurfaces are coloured by the values of the spanwise velocity field. The $\tilde{U}_z$ colour range is limited to ± 0.2 for improved visualization. The time instant $\tilde{t} = 300$ of the 3D-1c static simulation, shown in panel (ii), is considered to belong to the L regime due to the irregular lift coefficient behaviour and the high Strouhal number around this instant observed in Figure 4.9. On the other hand, the time instant $\tilde{t} = 500$, shown in panel (iii), is seen to belong to the H regime, where the lift coefficient exhibits high amplitude and regular oscillations, while the associated Strouhal number remains low. The presence of vortices can be recognized in the wake pressure field, as vortices correspond to low-pressure areas. It is observed in Figure 4.26(a) that the intensity of the shed vortices, as indicated by the values of the pressure coefficient, is lower in the L regime case than in the H regime, and it is highest for the free to move case. The 3D-21c case, panel (i), shows a pressure field similar to the L regime case. Similarly, regarding the Q -criterion isosurfaces in Figure 4.26(b), it is noticeable that the L regime has less spatially coherent vortical structures than the H regime, being most coherent in the free to move case. In these three cases there are clear streamwise vortical structures (ribs), which induce spanwise velocities in the wake in the order of 10% of the inflow velocity, i.e. about $\tilde{U}_z = \pm 0.1$, as

shown by the dimensionless spanwise velocity field presented. Regarding the 3D-21c case, it is observed that the Q-criterion isosurfaces are barely visible. There are no appreciable vortical ribs, but this is because the isosurfaces are shown between $z = 0$ and $z = 0.5c$, which is a region smaller than the size of the actual ribs present. Note that the spanwise velocity is much higher than in the other cases, being about $\tilde{U}_z = \pm 0.5$, which indicates the presence of intense spanwise structures. The vortical structures of the 3D-1c static cases agree well qualitatively with the static case presented by Lian et al. (2023a), although in said work there is no distinction between the L and H regimes. The 3D-1c lock-in case also agrees well with that presented by Lian et al. (2023a) for a low-damping case well inside the lock-in range. The results presented in this section agree well with the interpretation previously given in Section 4.2.3, showing that the L regime is characterized by lower coherence of its vortical structures, as well as less intense and irregular vortex shedding, compared with the H regime. Moreover, it is shown that large and growing edgewise motions further increase this coherence and the vortex-shedding intensity.

An additional comparison is made in Figure 4.27 between near-wake dimensionless velocity flow fields, for a slightly different set of cases, namely, from top to bottom, (i) the 3D-21c static simulation, (ii) the 3D-1c static simulation at time instant $\tilde{t} = 300$ (i.e. within the L regime), (iii) the 3D-1c static simulation at time instant $\tilde{t} = 500$ (i.e. within the H regime), and (iv) the 2D static simulation. In Figure 4.27(a), the instantaneous dimensionless velocity magnitude $||\tilde{U}||$ is shown, and in Figure 4.27(b) and Figure 4.27(c) the instantaneous dimensionless lateral velocity $\tilde{U}_x$ and longitudinal velocity $\tilde{U}_y$ are shown, respectively. The results of the 3D cases are shown at a slice in spanwise position $z = 0.5c$. As expected, the 3D DDES results show resolved vortical structures smaller than those of the 2D URANS case. But it is observed that the 2D flow field is quite similar to the 3D-1c H regime case on average, for all three presented velocity components. On the other hand, in the 3D-21c and the 3D-1c L regime cases, the flow field is notably different, having less defined large-scale structures. The similarities in the flow fields between the 2D and H regime cases, and between the 3D-21c and L regime cases, are in agreement with the time series presented earlier in Figure 4.7, where both the lift and drag coefficients of the 3D-1c case were seen to match closely either the 2D case or the 3D-21c case, in an intermittent manner.

In summary, the flow field analyses have provided insights into the vortex-shedding behaviour and spatial coherence of the vortical structures for the different configurations studied. The results have shown that the 3D-1c case exhibits intermittent shedding regimes, with the H regime being characterized by higher coherence and more intense vortex shedding compared to the L regime. The 3D-21c case, on the other hand, shows less coherent and less intense vortex shedding, similar to the L regime of the 3D-1c case. The 2D case, while not capturing the three-dimensional effects, provides a good approximation to the H regime of the 3D-1c case. These findings highlight the importance of considering three-dimensional effects and the potential limitations of 2D simulations in accurately capturing the complex vortex-shedding dynamics of an airfoil, but also the opportunity to use 2D simulations to approximate the aerodynamic loading and structural response of a 3D case in the lock-in regime.

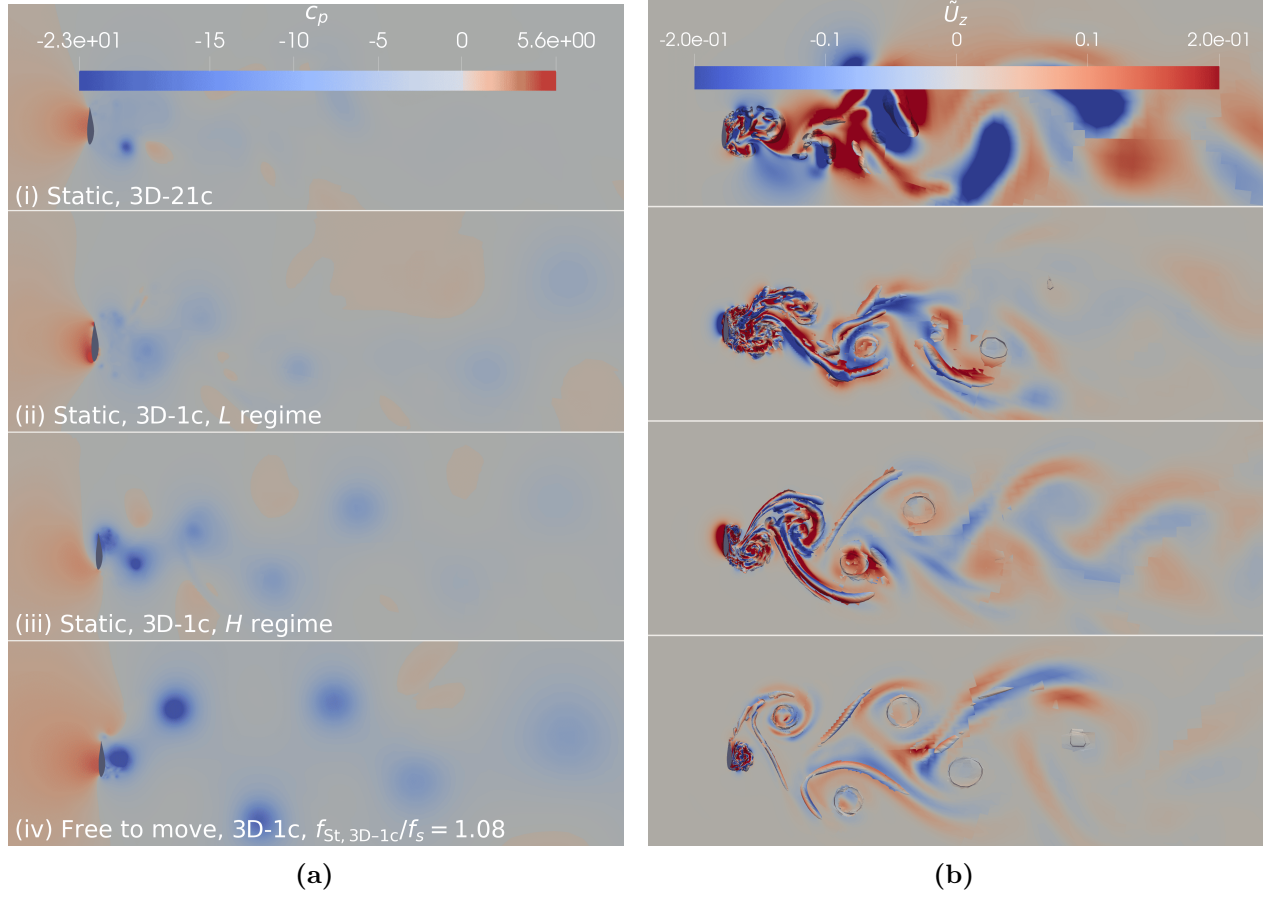


Figure 4.26: (a) Instantaneous pressure coefficient field, $c_p = (p - p_\infty) / (0.5\rho_\infty U_\infty^2)$, at a slice in spanwise position $z = 0.5c$, and (b) instantaneous Q-criterion isosurfaces, for $\tilde{Q} = Qc^2/U_\infty^2 = 1$, coloured by the dimensionless spanwise velocity field, $\tilde{U}_z = U_z/U_\infty$, and superimposed to the instantaneous dimensionless spanwise velocity field at the $z = 0.5c$ slice. The $\tilde{U}_z$ colour range is limited to ± 0.2 for improved visualization. In both panels four different cases are shown, from top to bottom: (i) 3D-21c static simulation, (ii) a time instant in the L regime of the 3D-1c static simulation, (iii) a time instant in the H regime of the 3D-1c static simulation, and (iv) 3D-1c free to move simulation with frequency ratio $f_{St, 3D-1c}/f_s = 1.08$.

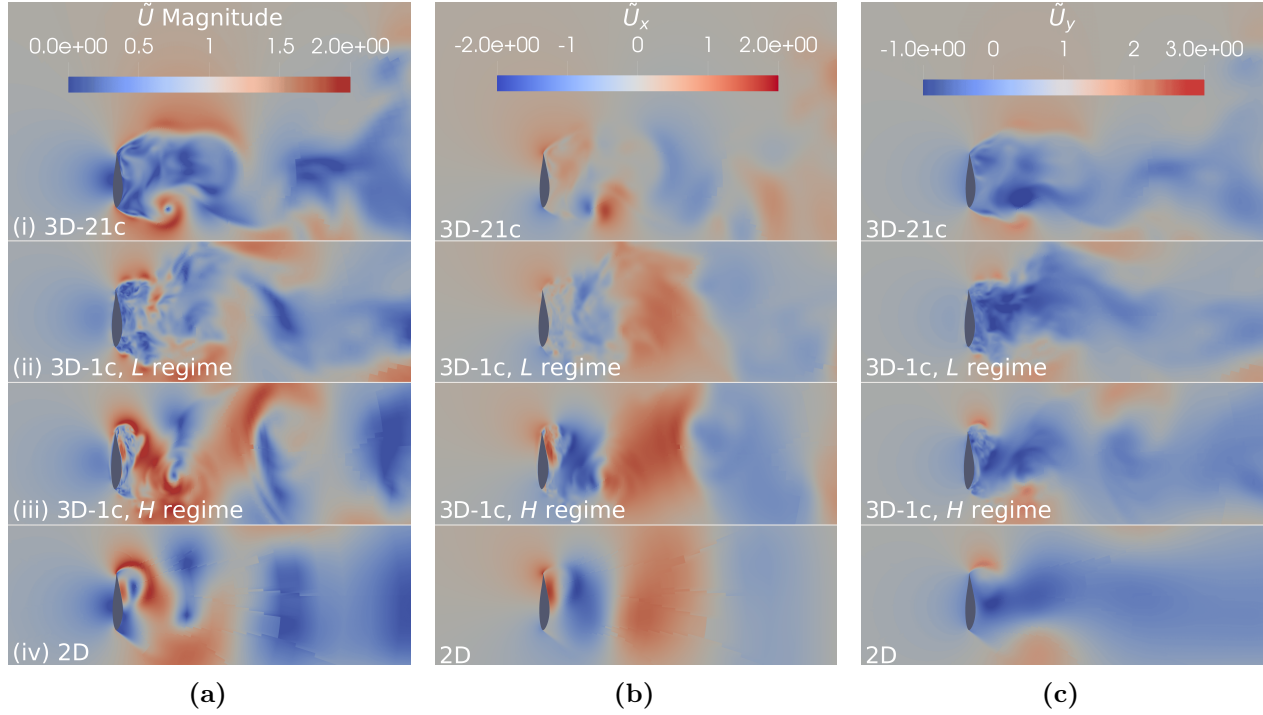


Figure 4.27: (a) Instantaneous dimensionless velocity magnitude $||\tilde{U}||$, (b) instantaneous dimensionless lateral velocity $\tilde{U}_x$ and (c) instantaneous dimensionless longitudinal velocity $\tilde{U}_y$. In the three panels, four different cases are shown, from top to bottom: (i) 3D-21c simulation, (ii) a time instant in the L regime of the 3D-1c simulation, (iii) a time instant in the H regime of the 3D-1c simulation, and (iv) 2D simulation. All panels correspond to the static simulations. For the 3D cases, the results are shown at a slice in spanwise position $z = 0.5c$.

4.3 Conclusions and future work

In this chapter, the vortex-induced vibration (VIV) response of an airfoil designed for wind turbines has been obtained using high-fidelity FSI simulations. Different aspect ratio configurations, with differing levels of fidelity and computational cost, have been simulated and compared, namely a 2D airfoil with the URANS turbulence-modelling approach (case “2D”), and two cases of a 3D airfoil with the DDES turbulence-modelling approach; one with an aspect ratio of one and the other with an aspect ratio of twenty-one (cases “3D-1c” and “3D-21c”).

To characterize the Strouhal number of the vortex shedding process, simulations of the static airfoil have been performed. It is shown that only in the 3D-1c configuration is there intermittent switching of the flow regime, resulting in a time-varying Strouhal number without a clear dominant value. The Strouhal numbers predicted with this approach span a considerable range, of $St \in [0.11, 0.16]$. It is observed that long simulations are required to characterize the switching of flow regimes, and it is noted that short simulations may produce the misconception that there is a single dominant vortex-shedding frequency. Further exploration of the mechanism responsible for this intermittent switching of the flow regimes could be performed in the future, for example by means of multivariate–multidimensional empirical mode decomposition, as performed by de Souza et al. (2024) for the cases of airfoil dynamic stall conditions and for transitional Reynolds number conditions.

To compare the VIV simulation results between different approaches, the results are typically plotted in terms of the ratio between the Strouhal-defined frequency and the structural natural frequency, where the Strouhal number employed is observed from a static case. For the 3D-1c approach, as there is not a single Strouhal number in the static case, the Strouhal number is chosen based on a Gaussian fit to the underlying spectral content of the vortex-shedding process. The Strouhal number thus defined has a value of 0.132, and is seen to correspond to one of the multiple dominant frequencies in the static case. Comparing the 3D-1c results to the 2D results employing this value for the Strouhal number, an almost perfect match of the lock-in range behaviour is obtained, both in terms of structural motion and aerodynamic loads. Such a result is supported by the well-known fact that the coherence of the wake flow is increased as the body undergoes large and growing vibrations, thus implying that the VIV response of the 3D airfoil under such conditions may be similar to that of the 2D simulations. This is further confirmed by a good match also of the long aspect ratio case (3D-21c) with the 2D results in the lock-in range, although some differences in the lock-in range extent are found, which may be attributed to the coarser spanwise mesh of this case.

Comparing the VIV response between different configurations using the dimensionless displacement growth rate has proven to be a robust metric, requiring only short-time simulations. This metric is used to define the extent of the lock-in range. A very good level of agreement is found between the 2D and 3D approaches for the growth rate and lock-in range. Additionally, during the VIV initial growth period, the obtained lift coefficient growth rate, mean drag growth rate, lift coefficient standard deviation, and mean drag values are also found to be in good agreement between all approaches. The strong agreement between 2D and 3D predictions of the airfoil lock-in range extent, vibration growth rate, and aerodynamic forces

is a significant and novel finding.

It is observed that outside the lock-in range, the different models studied produce significantly different results in terms of lift and drag forces. The prediction of these forces may be significantly affected by the periodic boundary condition for the short aspect ratio case, and the spanwise mesh density for the long aspect ratio case. Therefore, caution is advised when using simulation results to predict full-blade responses in cases where a significant portion of the blade might be outside the lock-in condition. Especially in the case of 2D simulations, where the forces might be severely overpredicted.

It is thus concluded that relatively cheap URANS 2D simulations may be as good as DDES 3D simulations for obtaining the VIV initial response of an airfoil section inside the lock-in range. Due to their computational cost, the computed 3D time series have not been run for long enough for the maximum displacement amplitudes to be reached, and therefore it has not been established whether they would compare well to those from the 2D approach in this sense.

Further relatively inexpensive 2D simulations can be used to characterize the VIV response under various conditions, such as different Reynolds numbers, mass ratios, structural damping values, airfoil shapes, angles of attack, initial displacement values, and concurrent forces or displacements. These results can then be utilized to inform 2D or lumped-element engineering models of the airfoil VIV response (Rigo et al., [2022b](#)), which are significantly more cost-effective. However, wind tunnel experiments or higher-fidelity simulations may still be necessary to accurately capture 3D phenomena related to spanwise geometry, such as full-blade mode shapes, twist, and chord taper, as well as non-homogeneous inflow conditions, including shear, atmospheric boundary layer turbulence, or flow inclination.

Chapter 5

Discussion

This chapter synthesizes the findings from wind turbine tower vortex-induced vibrations (VIV) analysis using engineering models (Chapter 3) and wind turbine blade VIV analysis using high-fidelity simulations (Chapter 4), and explores the applicability of existing VIV models to the different wind turbine components, discussing their limitations.

The VIV problem has been extensively studied in various engineering fields, particularly for slender structures such as chimneys, where well-established engineering models have been developed to predict their response. Despite ongoing refinements and persistent uncertainties—such as the influence of turbulence, Reynolds number effects, and the formulation of the nonlinearity in the response—these models are applied in structural design to predict vibration amplitudes and fatigue damage. In contrast to wind turbine towers, for which the knowledge from chimneys can be easily transferred, wind turbine blades represent a relatively unexplored case of VIV, with significantly fewer experimental and field data available. The complexity of the blade geometry (airfoil-shaped cross-section and spanwise variations of chord length and twist angle), the range of possible operating conditions (blade inclination and pitch angle), and the richness in the wind turbine dynamics (e.g. due to mode coupling), introduce additional challenges not typically encountered in classical VIV problems. As a result, it remains unclear whether existing VIV models can be effectively adapted to predict blade response, or whether new modelling approaches are required.

The research objectives of this thesis were structured around three main questions: (1) assessing the predictive capabilities and differences between well-established VIV engineering models when applied to wind turbine towers, (2) comparing response predictions from high-fidelity simulations of wind turbine airfoil sections, and (3) evaluating the suitability of existing engineering models for blade VIV based on insights from high-fidelity simulations. The application of engineering models to the wind turbine case in Chapter 3 has highlighted the challenges associated with the semi-empirical nature of these models. Reliance on curve-fitting and limited experimental data can lead to uncertainties in the VIV response and a lack of generalization capability to new geometries or flow conditions. The high-fidelity simulations of the blade VIV response in Chapter 4 have provided detailed insights into the complex fluid-structure interactions that govern the VIV response of airfoils. These simulations demonstrate the potential of using less resource-intensive methodologies, such as

2D simulations, while also highlighting the necessity of considering the full three-dimensional flow field to accurately characterize the blade response. The question that remains is the extent to which existing engineering models can be applied to the wind turbine blade case, and how high-fidelity simulations can inform the development of such models. This chapter aims to bridge the gap between the two approaches and identify research gaps that need to be addressed to enhance the predictive capabilities of VIV models for wind turbine components. To do so, the three engineering models presented in Chapter 3 are discussed next in the context of the wind turbine blade case, and recommendations are made for their application to this geometry, including the evaluation of the parameters involved.

5.1 Type A

The VIV Type A –or forced system– engineering model studied in this work is based on the assumption that the fluid forces are not dependent upon the structural dynamics. This model is well-suited to analyse the forces on a static structure, and it can be used to approximate the response of highly damped structures. One advantage of this model is that it can be informed from experimental or numerical data of the static structure, which greatly simplifies the procedure and reduces the parameter space and costs, compared to coupled fluid-structure interaction cases. The present implementation of the model, described in Chapter 3, is based on limited full scale and experimental data on chimneys, with which a semi-empirical approximation to the power spectral density (PSD) of the lift force was distilled by B. J. Vickery and Clark (1972). This approximation relies on three main parameters, the bandwidth of the PSD, B , the standard deviation of the lift coefficient, σ_{c_l} , and the Strouhal number of the shedding for the static case, St .

Bandwidth parameter, B_0 The bandwidth in the Type A model is a function of turbulence intensity, and its value for zero turbulence intensity is parametrized by the constant B_0 , which is taken to be 0.1 in the present work. Several values for B_0 have been proposed over the years, with values ranging from 0.05 (B. Vickery, 1978), up to the value of 0.1 employed here (B. J. Vickery, 1998). In the numerical simulations presented in Chapter 4, the PSD of the lift coefficient was calculated from the time series of the lift force on the static blade. The fitting of the PSDs obtained from the high-fidelity simulations to the Gaussian distribution proposed in B. J. Vickery and Clark (1972) yields values of $B_0 = [0.0113, 0.164, 0.0393]$ for the 2D, 3D-1c and 3D-21c cases respectively. The low value for the 2D case may actually be an artefact of the PSD calculation procedure due to the limited length of the time series, and longer time series may result in even lower bandwidth values for this case. In the 3D-1c case, the vortex shedding process presented an intermittent behaviour, which was seen to greatly increase the bandwidth of the PSD. For the 3D-21c case there was also significant spectral content present at separate frequencies, which made the Gaussian fit not so good and somewhat increased its bandwidth. In the absence of turbulence, the origin of the bandwidth may be attributed to the turbulent nature of the wake. It would be interesting to analyse if the bandwidth values reported in the literature originate from an intermittent behaviour of the wake or not. This is specially relevant for the application of the Type A model in the time-domain, as the synthetic generation of lift time series may not necessarily reproduce

this intermittent behaviour. To the best of the author’s knowledge, there are no published experimental results on the bandwidth of the lift coefficient of airfoils or blades at 90° angle of attack. The Type C model does not have an equivalent parameter to the bandwidth parameter B_0 used in the Type A model. This is because the lift coefficient in the Type C model does not account for the stochastic nature of the wake or the influence of turbulence on the lift force, being the solution of the wake oscillator equation a quasi-harmonic lift force. Efforts to include wake turbulence effects in the Type C model have been recently made by Rigo et al. (2022a), who proposed modifications to the wake oscillator model to account for the stochastic nature of the wake.

Lift coefficient standard deviation, σ_{c_l} Another parameter present in the Type A model is the standard deviation of the lift coefficient for the static tower, σ_{c_l} , which has also been modelled as a function of the turbulence intensity. The value of σ_{c_l} for zero turbulence intensity is taken to be 0.06 in the present work, by application of the expression in B. J. Vickery (1995). For the Type C model, the value used for the amplitude of the lift coefficient is $c_{l,0} = 0.3$, which, for the present quasi-harmonic implementation gives an approximate value of its standard deviation of $\sigma_{c_l} = c_{l,0}/k_p = 0.212$, where $k_p = 1.41$ is the peak factor according to EN 1991-1-4:2005+A1 (2010). On the other hand, for a stochastic process the peak factor tends to $k_p = 4.08$, and the same amplitude of 0.3 can yield a standard deviation as low as $\sigma_{c_l} = 0.07$, well in line with the value used in the Type A model. The high-fidelity simulations presented in this thesis have resulted in values of $\sigma_{c_l} = [0.33, 0.264, 0.0112]$ for the 2D, 3D-1c and 3D-21c static cases, respectively. The values of the 3D cases correspond to spanwise integral values, where spanwise differences in the lift force get averaged out, reducing its standard deviation, and therefore are not comparable with the sectional results that should be used in engineering models. Values of the lift coefficient standard deviation for airfoils at 90° angle of attack have been reported from experimental results to be between $0.022 \leq \sigma_{c_l} \leq 0.1$; a value of about 0.022 was reported by Fallahpour et al. (2023), values between 0.025 and 0.06 –depending on the Reynolds number– have been reported by G. Xu et al. (2025), and values of 0.025 and 0.1 were reported by Lind and Jones (2016) for two airfoils with different thicknesses. These results are notably lower than the values obtained in the high-fidelity simulations, which may be due, among others, to the lower Reynolds numbers and larger surface roughness in experimental conditions, resulting in smaller lift coefficient amplitudes or a more stochastic behaviour of the lift force in the experiments.

Strouhal number, St Lastly, the Strouhal number appears as a parameter in the Type A model, as well as in the Type B and C models. As shown in Chapter 3, the Strouhal number is a function of Reynolds number and surface roughness, and there are some uncertainties in its value for the very high Reynolds numbers relevant to wind turbine tower VIV. In the present work, the tower Strouhal was defined as a function of Reynolds number based on empirical results, and was seen to range between 0.23 and 0.25 when applied to the NREL 5-MW tower. For the blade case, in the high-fidelity simulations, the Strouhal number was found to be between 0.11 and 0.16, with significant uncertainties due to the different vortex shedding regimes observed in the simulations. Experimental results for different airfoils at 90° angle of attack, have shown Strouhal numbers in the range $0.12 < St < 0.2$; Strouhal numbers about $St = 0.12$ were reported in Huang and Lin (1995) and H.-W. Lee and Huang (1998),

$0.14 < St < 0.18$ in Besem et al. (2015), $St = 0.16$ in Iyer (2023) and G. Xu et al. (2024, 2025), $St = 0.17$ in Lind and Jones (2016), and $0.18 < St < 0.2$ in Fallahpour et al. (2023). The scatter in the reported Strouhal numbers could be due to blockage effects (J. M. Chen & Fang, 1996). The Strouhal number has been shown to be quite insensitive to variations of the Reynolds number in geometries where the separation point is fixed, such as rectangular prisms (Nakagawa, 1987; Okajima, 1982) and flat plates (J. M. Chen & Fang, 1996), for Reynolds numbers above the critical regime. In the airfoil case, the trailing edge separation point is fixed, while the leading edge separation may be slightly affected by the boundary layer development and thus the Reynolds number. More data at the transcritical Reynolds number regime and on the effect of surface roughness would be needed to accurately predict the Strouhal number of wind turbine blades.

Application to the blade case The application of the Type A model to a wind turbine blade would require a reassessment of the lift coefficient PSD fit, with special attention to the intermittent behaviour of the wake and its dependency with inflow turbulence intensity and length scales, as well as surface roughness and Reynolds number. As a first approximation, the present author suggests employing the same model as for the tower, while adjusting its parameters. The bandwidth parameter should be derived from full-scale data under different turbulence conditions, but the author suggests using the value of $B_0 = 0.1$ as a first approximation, without modifying the dependency on the turbulence intensity. It may be pertinent to increase the standard deviation value, σ_{c_l} , by a factor of 1.67 to account for the higher lift coefficient amplitudes observed in the airfoil simulations, of about $c_{l,0} = 0.5$, compared to typical lift coefficient amplitudes reported for circular cylinders, of about $c_{l,0} = 0.3$. Due to lack of additional data, the same expression as for the tower case may be used to account for the effect of inflow turbulence characteristics on σ_{c_l} . Regarding the Strouhal number, the limited available data and its dispersion does not allow for a clear recommendation on the values to be used for the application of the VIV models to wind turbine blades, but a tentative value of $St = 0.16$ may be used. No variation of the Strouhal value with the Reynolds number is recommended for the Reynolds numbers of interest for wind turbine blades.

5.2 Type B

The Type B –or fluidelastic– model, is based on the assumption that the fluid forces have a dependency on the structural dynamics, acting as a negative damping which becomes nonlinear for large vibration amplitudes, making the response self-excited and self-limited. Unlike the Type A model, this model may be appropriate for structures with medium or low damping. Modelling the fluid forces as a function of the structural dynamics requires data from experiments with either forced vibration or free-to-move setups. The implementation of the Type B model presented in Chapter 3 of this thesis is based on the work of B. J. Vickery and Basu (1983), applied to a modal representation of the wind turbine tower. Two modal quantities are needed to model the aeroelastic forces; a linear negative damping parameter $k_{a,j}$, and a nonlinear damping parameter $G_{a,j}$. In this work, both quantities are functions of the Reynolds number, the turbulence intensity and the ratio between the vortex shedding

frequency and the natural frequency of the corresponding mode. This formulation is based on physical arguments made by Marris (1964), which rely on the acceleration of the boundary layer separation points. It is unclear whether this formulation is applicable to the wind turbine blade case, where the separation points may exhibit significantly less movement compared to circular sections.

Linear negative damping, $k_{a,j}$ If the Type B model were to be applied to the wind turbine blade case, the linear negative damping parameter $k_{a,j}$ would need to be derived from experimental data of the vibrating blade. The high-fidelity simulations of the airfoil show that the lock-in region is centred around a frequency ratio of one, which is not the observed behaviour for circular cylinders, where the lock-in range typically starts at about the frequency ratio of one. Thus, the formulation of $K_{U_{\text{ratio}},j}$ employed in this work may have to be fundamentally modified. Furthermore, the lock-in range observed in the high-fidelity simulations is quite narrow, approximately between $0.85 \leq U_{\text{ratio}} \leq 1.15$. This is in contrast to the broader lock-in range defined in this work and in Hansen (1998) for circular cylinders, which ranges between $1 \leq U_{\text{ratio}} \leq 2$, as well as that reported in standards such as DNV-RP-C205 (2010), which suggests that VIV may occur when $0.8 \leq U_{\text{ratio}} \leq 1.6$. Experimental results by Benner et al. (2019) for an airfoil at 90° angle of attack, on the other hand, show a much larger lock-in range, of about $0.3 \leq U_{\text{ratio}} \leq 2$, also centred around one. Nevertheless, the results in Benner et al. (2019) are for subcritical Reynolds numbers, and for an airfoil with a quite small mass ratio of $\tilde{m} = m/(\rho c^2) = 4.82$, where m is the mass per unit length and c is the chord, which is closer to hydrodynamical applications. The mass ratio typically ranges from order 1 or 10 for underwater flexible bodies, e.g. $\tilde{m} \approx 2$ for an offshore riser Mukundan et al. (2009), to order 100 for structures in air, e.g. $\tilde{m} \approx 300$ for the NREL-5MW wind turbine tower Jonkman et al. (2009), as outlined by Jauvtis and Williamson (2003). For wind turbine blades, the present author estimates the mass ratio for the NREL-5MW (Jonkman et al., 2009) and DTU-10MW (Bak et al., 2013) blades to be about $\tilde{m} \approx 15$, for the NREL-UAE blade (Hand et al., 2001) it is $\tilde{m} \approx 30$, and for the IEA-22-280-RWT blade (Zahle et al., 2024) it is in the range $\tilde{m} \in [12, 23]$. A value of $\tilde{m} = 32.7$ is used in the high-fidelity simulations presented in Chapter 4, which is closer to that of the shorter NREL-UAE blade, but still similar to those of the other aforementioned full scale blades. The fact that the mass ratio for blades is about one order of magnitude smaller than for the NREL 5-MW tower suggests that there may be fundamental differences in the nature of the forces induced by the structural motion. To completely characterize the Type B aerodynamic damping, additional data would be needed on its sensitivity to the Reynolds number and turbulence intensity, parametrized in the K_{Re} and K_{I_u} terms of the model, respectively.

Nonlinear positive damping, $G_{a,j}$ The nonlinear modal damping parameter $G_{a,j}$ is here modelled as a function of the negative linear damping and a limiting amplitude factor a_L . The high-fidelity simulations of the airfoil show a self-limitation of the displacement amplitude at around 1.5 chords, which is significantly higher than the value of 0.4 diameters employed here for the tower. Nevertheless, the value of $a_L = 0.4$, as proposed in EN 1991-1-4:2005+A1 (2010), is based on limited empirical data (Basu, 1983), and higher displacement amplitudes have been observed for cylinders in wind tunnel experiments (Blevins, 2001). It is unclear

how an equivalent value to that of the Eurocode EN 1991-1-4:2005+A1 (2010) should be obtained for the blade case.

Application to the blade case To accurately accommodate a Type B VIV model to the wind turbine blade case, extensive experimental data would be needed to derive the linear and nonlinear damping parameters. High-fidelity simulations of the 2D airfoil have been shown to provide accurate results in the lock-in range when compared to 3D models. Therefore, 2D simulations may be a good starting point to calibrate the Type B model, given that it will probably yield conservative results when compared with real scenarios involving turbulence and reduced coherence due to 3D effects. Additionally, as a conservative first approximation, the author suggests using a limiting amplitude factor of $a_L = 1.5$ for the blade case. It is considered that, as such large deflections should be avoided altogether, it is desirable to be conservative in the estimation of the nonlinear limiting damping parameter. The high-fidelity results in the present thesis show an almost linear increase of the displacement amplitude with time, suggesting that the aerodynamic damping diminishes with amplitude with a hyperbolic form, as suggested by Scruton and Flint (1964). This evolution of the aerodynamic damping is also in line with recent results by Lupi et al. (2017b, 2018), who argue against the functional form of the nonlinear damping proposed by B. J. Vickery and Basu (1983) and employed in the present work. Furthermore, in the present Type B VIV model, the negative aerodynamic damping includes the assumption that its modal counterpart, $k_{a,j}$, is a function of the velocity ratio for each mode j independently, $U_{\text{ratio},j} = \frac{f_{\text{St}}}{f_{s,j}}$. This assumption implies that negative damping appears only when the motion occurs close to the structural natural frequencies, neglecting contributions from vibrations at other frequencies. While this simplification may be reasonable for the tower case, it could be problematic for the blade case. This is because the first tower mode can induce significant motion of the blades, meaning that if there is tower motion and the inflow wind speed equals the critical wind speed of the blade, the blades may experience a negative damping force at the tower motion frequency rather than at its own natural frequency, as demonstrated with high-fidelity simulations by Vicente-Ludlam et al. (2024). To address this issue, a higher-fidelity representation of the structural dynamics, such as a finite-element beam model, could be considered instead of the modal approximation, and the aerodynamic damping formulation would need to be adapted to make it a function of the instantaneous vibration frequency. In contrast, the Type C model inherently captures this effect, since its equations do not explicitly include parametric variations within the lock-in range. Thus, any vibration near the Strouhal frequency, f_{St} —whether that is close to a natural frequency or not—, can excite the Type C wake oscillator.

5.3 Type C

The Type C—or coupled system— model, is based on the assumption that the fluid force has its own dynamics, which is dependent on the structural dynamics, and vice versa. This model is well-suited for lowly damped structures. Typical formulations of the Type C model, like the one employed in this work, do not account for the influence of turbulence on the wake, or the inherently stochastic behaviour of the wake at some Reynolds number regimes, which are

crucial to accurately represent the VIV response of highly damped structures. The Type C model presented in Chapter 3 is based on the work by Skop and Balasubramanian (1997) and K. Xu et al. (2015). The structural dynamics depend on a stall parameter, H_1 , a static lift coefficient parameter, $c_{l,0}$, and on the dynamic wake variable, q . The wake dynamic equation is dependent upon the wake oscillator damping parameter, ε , and the parameter quantifying the influence of the structural velocity on the wake, β . It is also dependent on the Strouhal number, which was discussed before for the Type A model.

Stall parameter, H_1 The stall damping parameter values, H_1 , employed in Chapter 3 were derived from curve-fitting the response of the Type C model to that obtained with the Type B model, which resulted in values of $H_1 = -0.555$ for the low damping case and $H_1 = -33.3$ for the high damping case. The stall parameter may be related to the drag coefficient by physical argumentations, such as the one presented in Tamura and Matsui (1980), where the stall force presented in Skop and Balasubramanian (1997) is derived as the projection of the drag force in the lateral direction, due to changes in the effective angle of attack as a result of the bluff body oscillation. Using this approach, it results that $H_1 = -c_d$. The mean drag coefficient of a static cylinder is about $0.2 \leq c_d \leq 0.5$ (Schewe, 1983) for a clean configuration, being quite sensitive to the Reynolds number and surface roughness (Achenbach & Heinecke, 1981). The significant difference between the values of H_1 for the high damping case and reported c_d values suggests that the Type C model fitting procedure employed might not be a good approach in this case. In the static airfoil high-fidelity simulations presented in Chapter 4, the mean drag coefficient values obtained were $c_d = [3, 2.39, 1.76]$ for the 2D, 3D-1c and 3D-21c cases, respectively. The 3D-21c value is quite close to the value of 1.914 reported for this airfoil from wind tunnel experiments by Timmer and van Rooij (2001), and also to the values of $1.8 \leq c_d \leq 1.9$ for a similar airfoil reported by G. Xu et al. (2025), were both experiments were performed at supercritical Reynolds numbers. The 3D-1c mean drag was shown in Chapter 4 to fluctuate significantly over time, thus defining two vortex-shedding regimes; a high-drag regime, for which the mean drag coefficient is about 3 (as in the 2D case), and a low-drag regime for which it is about 1.6 (more similar to the 3D-21c case). It may be of interest to discern whether reported experimental data also suffer from intermittent regimes and therefore considerable mean drag fluctuations. It is known that 2D CFD simulations overestimate the drag coefficient (Mittal & Balachandar, 1995), and the value obtained here for the 2D case compares well with the value of $c_d = 3.1$ reported by Skrzypiński et al. (2014) also obtained from 2D high-fidelity simulations.

Additionally, the mean drag coefficient is known to increase inside the lock-in range (Blevins, 2001), as seen for instance in Figure 4.23 of Chapter 4. Furthermore, the physical argumentation for the relationship between H_1 and c_d refers to the instantaneous drag force, which can experience significant oscillations around its mean value, as seen for example in the simulation results in Figures 4.7 and 4.21 of Chapter 4, although in Tamura and Matsui (1980) a constant value is employed for the drag coefficient. This may be especially relevant for non-symmetrical bodies such as airfoils at angles of attack different from 90° . Therefore, it might be better to consider H_1 as a time-dependent drag-related variable than a parameter, but this requires verification.

Lift coefficient amplitude, $c_{l,0}$ The static case lift coefficient parameter, $c_{l,0}$ may be related to the σ_{c_l} parameter discussed for the Type A model by the peak factor. The value chosen here of $c_{l,0} = 0.3$ is close to the value $c_{l,0} = 0.245$ obtained for the tower case, from $\sigma_{c_l} = 0.06$ and a peak factor of $k_p = 4.08$, corresponding to stochastic loading (EN 1991-1-4:2005+A1, 2010). This value is also in line with the values of $0.2 \leq c_{l,0} \leq 0.3$ typically reported for circular cylinders (Blevins, 2001; EN 1991-1-4:2005+A1, 2010), for Reynolds numbers above the critical. Nonetheless, lower values of $0.033 \leq c_{l,0} \leq 0.163$ have been reported from field measurements by Hansen (1981). A value of $c_{l,0} = 0.04$ has been applied for an oscillator model calculation at transcritical Reynolds numbers by Tamura (2020). The maximum lift coefficient amplitude obtained with the high-fidelity simulations of the airfoil presented in Chapter 4 are $c_{l,0} = [0.488, 0.62, 0.0647]$, for the 2D, 3D-1c and 3D-21c cases, respectively. Once again, the 3D cases may not be representative of sectional values, as they correspond to the spanwise integral lift force. The 3D-1c value is notably high, corresponding to the H regime, where the vortex shedding is highly spanwise coherent. To the best of the author's knowledge, there are no published experimental results on the lift coefficient amplitude for static airfoils at 90° angle of attack.

Nonlinear damping parameter, ε In the present implementation of the Type C model, the values of the wake oscillator damping parameter are $\varepsilon = [0.075, 0.3]$ for the low structural damping and high structural damping cases respectively, as obtained from the fit to the Type B response. A value of $\varepsilon = 0.3$ is proposed by Facchinetti et al. (2004). Values of $\varepsilon = 2\zeta = [0.076, 0.09]$ have been proposed for subcritical and transcritical Reynolds numbers cases, respectively, where ζ is the equivalent damping parameter, as presented in Tamura (2020). A curve-fitting approach to wind tunnel data is followed in K. Xu et al. (2015) to obtain a value of $\varepsilon = 0.0219$ for a bridge section. As an alternative to curve-fitting procedures, a different approach is presented in Tamura and Matsui (1980) and Tamura (2020), where the damping parameter of a circular cylinder is related to its fluctuating near-wake length, which may be obtained from wake visualizations, and a Magnus effect factor, which can be obtained from experimental results (Funakawa (1969), as mentioned in Tamura (2020)). It remains to be seen if such a formulation would hold for the airfoil case.

Velocity coupling parameter, β Lastly, the parameter quantifying the influence of the structural velocity on the wake, β , was also derived from curve-fitting the response, resulting in $\beta = [99.5, 298] \text{ s}^{-1}$ for the low damping and high damping cases, respectively. This is possibly the most difficult parameter to characterize, as different implementations of the wake oscillator model in the literature use a variety of definitions for the structural velocity coupling (Facchinetti et al., 2004; Skop & Balasubramanian, 1997; Tamura, 2020; K. Xu et al., 2015). According to the formulation by Tamura and Matsui (1980), the structural velocity changes the relative angle of attack, affecting only the stiffness term of the wake oscillator, and being the stiffness related to the dimensions of the oscillating near-wake, making it directly obtainable without the need for curve-fitting.

Application to the blade case The discussion just presented on the Type C model parameters highlights some of the multiple complexities and uncertainties of the model. Any

application of such a model for practical engineering purposes will pivot between two main strategies; one is to choose and tune semi-empirical parameters derived from curve-fitting the model to some measurement of the response of the structure, and the other is to derive a model based on meaningful physical properties of the system which can be measured directly. Despite significant efforts by previous researchers, as highlighted in Tamura (2020), the task of finding suitable physical models is far from complete. Regarding the application of such a physical model, like the one by Tamura and Matsui (1980), to a wind turbine blade, a first step would be to adapt the formulation to the airfoil geometry case. Similar efforts have been done for square and rectangular cylinders by C. Chen et al. (2022), Mannini (2020), and Tamura and Shimada (1987), among others. Such a derivation is out of scope of the present thesis. A procedure to derive the parameters of the model by Tamura and Matsui (1980) from experimental data has been proposed by Rigo et al. (2022b). It involves reducing the differential equations to an algebraic system, which is dependent on some parameters to be obtained by curve-fitting. The same procedure could be applied using high-fidelity simulation results. Both the curve-fitting approach and the physical parameters derivation and measurement require high-quality data and careful consideration. Some recommendations are now given on the parametrization of the Type C model applied to the blade VIV, based on the circular cylinder formulation used throughout the thesis, which may be used for an approximate assessment of the blade VIV response. As a first approximation, the parameters of the Type C model can be assumed from data available in the literature and the high-fidelity simulations results presented here, as discussed earlier. For the stall parameter, H_1 , the author suggests using the relationship $H_1 = -c_d$, where c_d is the mean drag coefficient value as obtained from static experiments or high-fidelity 3D simulations of the airfoil. The lift coefficient amplitude parameter should be obtained from experiments or high-fidelity simulations. As this data may not be readily available, the author suggests using the value of $c_{l,0} = 0.5$ as a first approximation, based on the present high-fidelity simulation results. For the wake oscillator damping parameter, ε , the author suggests using a value of $\varepsilon = 0.075$ as a first approximation, based on the results obtained in the present curve-fitting approach, although a great dispersion of values is reported in the literature for this parameter. Lastly, as mentioned earlier, the structural velocity coupling parameter, β , may be difficult to characterize and compare with values reported in the literature. The author suggests using the value of $\beta = 99.5 \text{ s}^{-1}$ as a first approximation, based on the results obtained in the present curve-fitting approach. Nonetheless, the high-fidelity simulation results for the airfoil show that the lock-in range is centred around a velocity ratio of one. This behaviour can be reproduced with the wake oscillator engineering model by including an acceleration coupling, instead of or additionally to the velocity coupling considered for the tower case (Facchinetti et al., 2004; Rigo et al., 2022b). This suggests that the added mass effect may not be negligible for the blade case.

5.4 Preliminary application of engineering models to the NREL 5-MW blade

Following the recommendations given in the previous sections, the three VIV engineering models are applied to the NREL 5-MW wind turbine blade (Jonkman et al., 2009), to obtain

a first approximation of the response of the blade to VIV. As for the tower case, the blade is modelled as a cantilever beam, and the response is approximated using modal superposition. The blade is modelled in a pitched to feather position, no flow inclination is considered, and only the first edgewise mode is retained, being the dominant mode for blade VIV (Manolas et al., 2022). This blade position is in line with the high-fidelity simulations performed in Chapter 4 and may be considered the canonical situation for blade VIV, although recent investigations have shown that other combinations of flow inclination and pitch angle may be more prone to VIV (Horcas, Sørensen, et al., 2022). The modelling of VIV for different flow inclination angles is out of scope of the present thesis, as it presents unique flow configurations and other challenges (Horcas, Sørensen, et al., 2022).

A preliminary estimate of the sensitivity of the blade to VIV can be achieved by comparing the blade structural damping value to the negative aerodynamic damping ratio values, $\zeta_{a,1}$ predicted with the Type B model. Two different values for the structural damping ratio, $\zeta_{s,1}$, are considered here; one is $\zeta_{s,1} = 0.477\%$, as originally defined in Jonkman et al. (2009), and the other is $\zeta_{s,1} = 0.25\%$, as suggested in Manolas et al. (2022) to be more consistent with state-of-the art wind turbine blade damping values. To obtain this aerodynamic damping ratio, it is only necessary to define the geometry of the blade (chord distribution)¹, the modal structural properties of the blade, $f_{s,1} = 1.12$ Hz and $m_1 = 1.39 \times 10^3$ kg, in this case corresponding to the first edgewise blade mode, and the aerodynamic damping factor, K_a . This aerodynamic damping factor was defined for the tower in Equation (3.19), and it controls the extent of the lock-in range. As discussed in Section 5.2, for the airfoil case, the lock-in range may extend to values below $U_{\text{ratio},j} = 1$. For this preliminary application to the blade case, the aerodynamic damping factor definition used for the tower is modified based on the airfoil experiments performed by Benner et al. (2019). Specifically, the aerodynamic damping parameter $K_{U_{\text{ratio},j}}$ is defined as increasing linearly from zero to one in the range $0.3 < U_{\text{ratio},j} < 1$, and then, as for the tower model, it is set to decrease linearly from one to zero in the range $1 < U_{\text{ratio},j} < 2$. The resulting expression is shown in Equation (5.1),

$$K_{U_{\text{ratio},j}}(z) = \begin{cases} 0, & \text{if } U_{\text{ratio},j}(z) \leq 0.3 \\ \frac{1}{0.7}U_{\text{ratio},j}(z) - \frac{0.3}{0.7}, & \text{if } 0.3 < U_{\text{ratio},j}(z) \leq 1 \\ 2 - U_{\text{ratio},j}(z), & \text{if } 1 < U_{\text{ratio},j}(z) < 2 \\ 0, & \text{if } U_{\text{ratio},j}(z) \geq 2. \end{cases} \quad (5.1)$$

This formulation extends the lock-in range towards velocities below the critical wind speed, which is also in line with the high-fidelity simulations presented in Chapter 4.

Having defined the necessary parameters, the aerodynamic damping ratio for the first edgewise mode of the blade, $\zeta_{a,1}$, is obtained. This negative aerodynamic damping ratio is shown in Figure 5.1, as a function of the velocity ratio at the reference blade spanwise position, $U_{\text{ratio,ref},j}$. The aerodynamic damping ratio is notably higher than the structural damping for most velocities within the predefined lock-in range, $0.3 < U_{\text{ratio},1} < 2$, which suggests that

¹Note that the reference length at each blade station is defined to be the local chord length, as if all sections were at an angle of attack of $\alpha = 90^\circ$, without accounting for the twist distribution along the blade. The twist spans less than 13.5° , which gives a projection factor error under 3%.

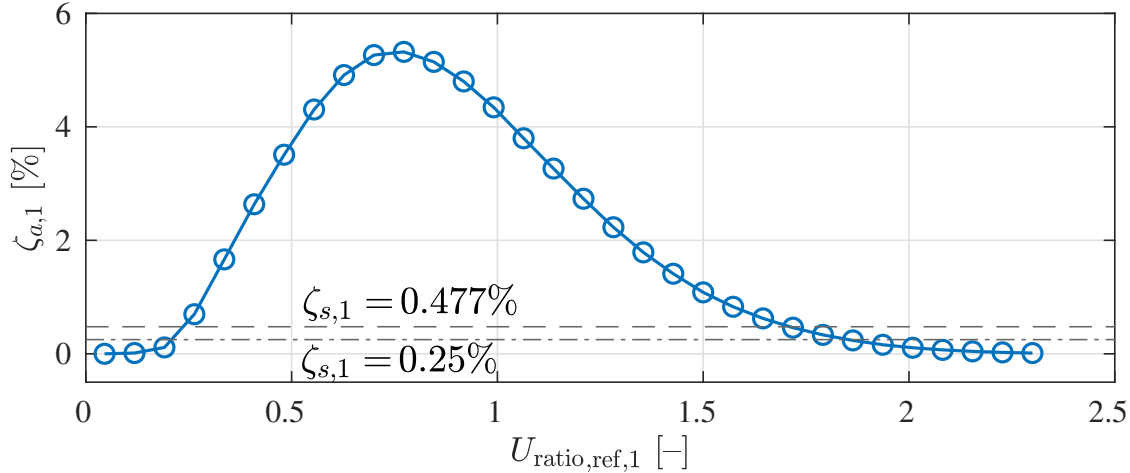


Figure 5.1: Type B modal linear aerodynamic damping ratio for the first NREL 5-MW blade edgewise mode, as a function of the velocity ratio, together with two representative values for the structural damping ratio, $\zeta_{s,1}$.

the blade VIV response will be characterized by its self-limiting or “lock-in” behaviour. The aerodynamic damping ratio for the blade is notably higher than for the tower, which was shown in Figure 3.11 to be lower than 0.6. This difference may be attributed to the significantly smaller equivalent mass of the blade, as the damping ratio is inversely proportional to the equivalent mass. The equivalent mass of the blade is $m_{eq,1} = 1.05 \times 10^2$ kg/m, while the tower equivalent mass is $m_{eq,1} = 3.01 \times 10^3$ kg/m, which is one order of magnitude larger. This result suggests that the blade may be significantly more sensitive to VIV than the tower, but this would have to be confirmed with additional data.

The blade response will be calculated next, for the three VIV engineering models described in Chapter 3. The choice of parameters for the three models is summarized in Table 5.1. The table shows the parameters for the blade and the tower cases, and the choice of each parameter and how they compare between cases is explained below.

Note that for the tower case, two sets of Type C parameters were defined in Section 3.3.2; one for the high damping case and one for the low damping case. In Table 5.1, the low damping case, $\zeta_{s,1} = 0.25\%$, is the one presented, as it was shown in Section 3.3.3 to result in a self-limited or “lock-in” VIV response of the tower, making it more comparable to the blade case, which is also expected to be self-limited as determined by Figure 5.1.

For the Type A model, as discussed in Section 5.1, the bandwidth parameter is kept at $B_0 = 0.1$, the lift coefficient is increased for the blade case to $\sigma_{cl} = 0.1$, and the Strouhal number is set to $St = 0.16$ instead of having the Reynolds-number dependency employed for the tower case.

For the Type B model, as explained earlier, the aerodynamic damping factor $K_{U_{ratio,j}}$ is extended towards velocities below the critical wind speed, thus increasing the lock-in range. The other components of the total aerodynamic damping factor, K_{Re} and K_{I_u} , are kept identical to the tower case. Additionally, the limiting amplitude factor is considerably increased from the tower to the blade case, from $a_L = 0.4$ to $a_L = 1.5$. As the response is

Parameter	Blade	Tower
Type A		
B_0 [-]	0.1	0.1
σ_{c_l} [-]	0.1	0.06
St [-]	0.16	(prEN 1991-1-4:2019, 2020)
Type B		
K_a [-]	Equation (5.1)	Equation (3.19)
a_L [-]	1.5	0.4
Type C		
H_1 [-]	-1.9	-0.555
$c_{l,0}$ [-]	0.5	0.3
ε [-]	0.075	0.075
β [s ⁻¹]	99.5	99.5

Table 5.1: Model parameters of the three engineering models, as applied to the blade, and to the tower cases with a structural damping ratio value of $\zeta_{s,1} = 0.25\%$. Note that all Type A parameters are also employed in the Type B model, and that the Strouhal number, St, is also present in the Type C model.

expected to be in the self-limited regime, one can expect the blade response to be limited to a maximum amplitude of about 1.5 chords.

For the Type C model, the stall parameter, H_1 , is increased in magnitude, to account for the higher drag coefficient of the airfoil. The lift coefficient amplitude, $c_{l,0}$, is also increased based on the 2D CFD data presented in Chapter 4. The wake oscillator damping parameter and the structural velocity coupling parameter are kept as obtained for the tower from the curve-fitting procedure, as discussed earlier in Section 5.3. The increase in H_1 acts as a damping factor, whereas the higher lift coefficient will increase the blade response magnitude.

Regarding the numerical parameters of the simulations, they are all kept the same as for the tower, as presented in Section 3.3.3, except the time step, which is decreased from $\Delta t = 0.1$ to $\Delta t = 0.06$ to account for the higher velocities covered in the blade case. The dimensionless velocity range for the blade case consists of 32 linearly spaced velocities in the range $0.0455 \leq U_{\text{ratio,ref},1} \leq 2.3$, which results in an inflow velocity range between 7.58 and 50.52 m/s. This inflow velocity range is larger than the one used for the tower case, which was between 12.21 and 30.52 m/s. Note that the velocity range extends to higher values due to its higher critical wind speed, which is about 25.26 m/s for the first edgewise mode of the NREL 5-MW blade, compared to about 15.26 m/s for the first mode of the tower, and it also extends to lower values to account for the lock-in range extension towards lower velocities, as discussed earlier. Additionally, for the blade case no wind shear is considered, which corresponds to a situation where the blade is positioned horizontally.

The response of the NREL 5-MW wind turbine blade is shown in Figure 5.2, as a function of the ratio between the inflow and critical wind speeds at the reference height, $U_{\text{ratio,ref},1}$. The

maximum dimensionless blade tip displacement, $\max(\tilde{w}_t)$ is shown for the three VIV models. The Type A model predicts the lowest blade response, being the Type B response about an order of magnitude higher. This is consistent with the expected effect of the aerodynamic damping shown earlier in Figure 5.1. The Type C response lies between the two, its parameters not being adjusted to match the Type B response.

The Type A response is shown in Figure 5.2 to be centred around a velocity ratio of approximately one, and extends from about 0.5 to 1.5. This range is more extensive than for the tower case, where the Type A response extended for velocity ratios from about 0.8 to 1.2. The reason for this extended range may be the larger relative differences in chord length along the blade span than the differences in tower diameter along its height, which leads to a greater range of vortex shedding frequencies along the blade span. Thus, at relatively low wind speeds, the tip of the blade will be shedding vortices at a frequency close to the structural one and therefore inducing significant aerodynamic forces. This effect would be more pronounced in the case of the blade pointing upwards due to the wind shear, and possibly less pronounced for blades pointing downwards. In the Type B case, the lock-in range extends from about 0.3 to 1.8, as predicted by the crossing points of the aerodynamic and structural damping values shown in Figure 5.1. The shape of the response is quite distinct from that of the tower, as in this case the maximum response is reached at the high-velocity end of the lock-in range, rather than at the middle. Again, this might be a consequence of the complex blade chord distribution, which leads to different sections of the blade driving the response at different velocities. The Type C response is also centred around a velocity ratio of about one, quite differently to the tower case, where the response started at about the velocity ratio of one and extended up to 1.5 in the low damping case. The position of the beginning of the Type C lock-in range responds to the modal wake oscillator frequency, which is related to the modal projection of the spanwise local vortex shedding frequency, so, once again, the response begins before the velocity ratio of one due to the reduced chord towards the blade tip, which induces a higher modal frequency than that expected at the reference height according to the Strouhal law. It is worth noting that the extension of the lock-in range towards lower velocities observed in the Type A and Type C models, being a consequence of the spanwise distribution of the chord length, is not related to that similar extension observed in the high-fidelity simulations presented here and in the experimental results from Benner et al. (2019), and applied to the Type B model aerodynamic damping definition, which stem from cases with constant chord, where the vortex shedding frequency is constant along the span.

Regarding the response amplitudes shown in Figure 5.2, the three models predict considerably larger amplitudes than those obtained for the tower case. The Type A model predicts a maximum dimensionless displacement, $\max(\tilde{w}_t)$, of about 0.25, whereas for the tower it was below 0.01. This is possibly a consequence of the smaller blade modal mass, which results in a larger response when subjected to a lift force of similar amplitude. Note that the Type A lift coefficient standard deviation was slightly increased for the blade case, from 0.06 to 0.1, but this alone does not explain the change observed in the response amplitude. Similarly, the Type B model predicts a maximum dimensionless displacement of about 3.6, which is consistent with the high aerodynamic damping ratio discussed earlier, and it is also one order of magnitude larger than that of the tower case, of $\max(\tilde{w}_t) = 0.38$, again possibly due to the

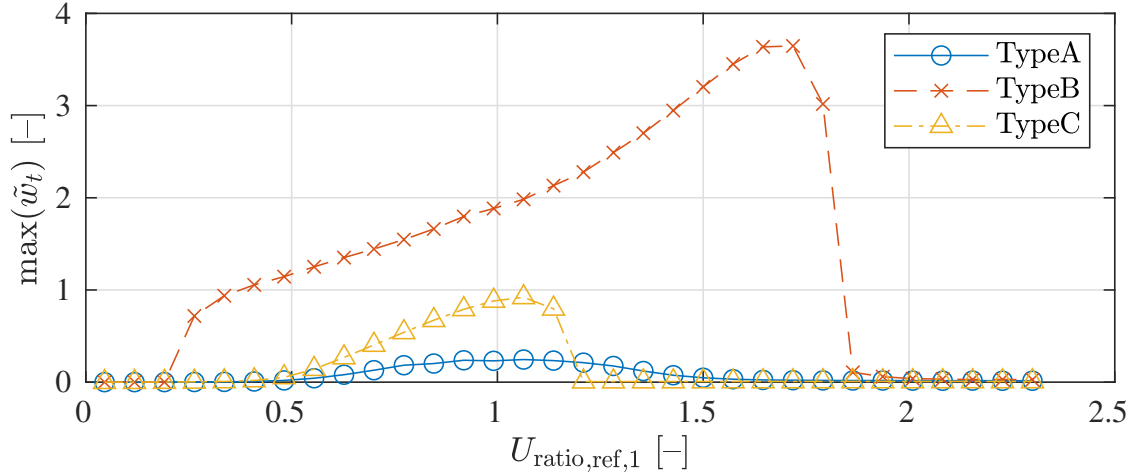


Figure 5.2: Maximum dimensionless blade tip displacement as a function of the ratio between the reference inflow and critical wind speeds, for the three VIV models. For Type A and Type B, the ensemble average of the maximum dimensionless displacement is shown.

smaller equivalent mass. The Type C model predicts a maximum dimensionless displacement of about 1, whereas for the tower it was 0.35. The smaller difference between the Type C results for blade and tower, compared to the Type A and Type B results, is possibly due to the modified value of the stall damping parameter, H_1 . More research is needed to assess whether the large amplitudes obtained for the blade case are realistic or not.

5.5 Closing remarks

For the practical application of VIV engineering models, the designer has a choice on the level of detail they are interested in, which is related to the computational cost and complexity of the models.

The most basic consideration would be to determine at which wind speeds large VIV may occur. For this, knowledge of the Strouhal number and the maximum expected lock-in range extent is necessary. This basic information can be used to define operational procedures to avoid large VIV, such as avoiding installation under unsuitable wind speed conditions, or the necessity of additional damping or mitigation devices. For the tower VIV case, the Strouhal number and lock-in range extent are well documented, and included in some standards (DNV-RP-C205, 2010), albeit with considerable uncertainties for very high Reynolds numbers and in the consideration of surface roughness and turbulence effects. For the blade VIV case, the available data on the Strouhal number and lock-in range is limited, and its dispersion is large, as discussed above.

A more advanced consideration is to estimate the maximum expected response. If this response is within reasonable margins then no action may be required, even within the lock-in range extent. The maximum response is a function of the structural damping, and requires estimating the vortex-induced aerodynamic damping, which depends on the turbulence intensity, among other things. This is the recommended procedure in several standards (DNV-RP-C205, 2010;

EN 1991-1-4:2005+A1, [2010](#); IEC 61400-6 Ed.1, [2020](#)), but these standards are not tailored to the wind turbine blade case.

A third level of consideration would be to estimate the maximum response for each inflow velocity. This requires parametrization of the aerodynamic damping with the velocity for a Type B model (Hansen, [1998](#); B. J. Vickery & Basu, [1983](#)), or the application of a Type C model, which intrinsically accounts for the lock-in behaviour. With this calculation, a conservative estimate of the fatigue damage can be obtained for a given wind speed statistical distribution. This method is not present in design standards.

Finally, the designer may be interested in the time history of the response, for example for a typical load calculations procedure (IEC 61400-1 Ed.3.1, [2010](#)), or the consideration of transient events such as a yaw alignment manoeuvre. By integrating the VIV model in load calculation aeroelastic solvers, a detailed fatigue analysis may be performed, considering the effect of the overall wind turbine dynamics on the vortex-induced loading, and vice versa. For this, a Type B model could be used, if it is extended to account for the possibility of vortex-induced loading due to forced vibrations, as described earlier. A Type C model would also be suitable. The Type A model is not recommended for this purpose, even if the structural damping is large, as any considerable motion, even if not originated by VIV, may result in a considerable vortex-induced loading due to the lock-in phenomenon.

In conclusion, this chapter has provided a discussion on the suitability of existing VIV engineering models to wind turbine blades, a first estimate of the values of the parameters involved, and their preliminary application to predict the blade response over the lock-in range. The analysis has highlighted the complexities and limitations of current models when applied to wind turbine blades, emphasizing the need for further research and data to enhance their predictive capabilities. By bridging the insights from engineering models and high-fidelity simulations, this work contributes to a deeper understanding of VIV in wind energy applications, serving as a foundation for future advancements in the field.

Chapter 6

Conclusions

6.1 Conclusions

This thesis set out to address the following question: How can we improve the time-domain modelling of vortex-induced vibrations (VIV) for wind turbine towers and blades? This question is motivated by the need for accurate and reliable aeroelastic load calculations in wind turbine design and certification, particularly in the context of increasing turbine sizes and the complexity of their operating environments. The primary objectives of the thesis were to evaluate the applicability of existing engineering models to wind turbine towers (Chapter 3), to investigate the role of high-fidelity simulations in understanding blade VIV (Chapter 4), and to explore the integration of these approaches to enhance predictive capabilities (Chapter 5). In doing so, this work provides an integrated perspective on the modelling and prediction of VIV for aeroelastic load calculations in wind turbines. The findings highlight the complexities of VIV phenomena, the applicability and limitations of existing models, and the need for further research to enhance predictive capabilities.

The time-domain semi-empirical engineering models on one hand, and the high-fidelity fluid-structure interaction simulations on the other, represent two complementary approaches to predicting VIV phenomena. While engineering models offer computational efficiency and may be well-suited for design and parametric studies, high-fidelity simulations provide detailed insights into complex fluid-structure interactions, capturing phenomena that are often oversimplified or neglected in engineering models.

This thesis builds upon prior work on the VIV response of cylindrical structures with circular cross-sections, which has lead to the development of practical engineering models for predicting VIV responses, and applied in numerous building codes and standards (DNV-RP-C205, 2010; EN 1991-1-4:2005+A1, 2010; IEC 61400-6 Ed.1, 2020). These models are often condensed into simplified analytic solutions, whereas the prediction of VIV for aeroelastic load calculations calls for time-domain simulations that can capture the complex interactions between the wind turbine components and the turbulent wind field, as well as transient events. Thus, Chapter 3 of this thesis focused on the implementation in the time-domain of existing engineering models, and their application to wind turbine towers, specifically that of the NREL 5-MW reference

wind turbine. Three types of VIV models were considered, in order of complexity: Type A (forced system), Type B (fluidelastic system), and Type C (coupled system), as defined in Section 2.2 and formulated in Section 3.1.

The Type A model, based on a stochastic lift force characterized by its power spectral density, provides a representation of the vortex shedding process without considering the interaction between the structural motion and the fluid forces. Although simple, it does not account for the interaction between the structural motion and the fluid forces, limiting its applicability to highly damped structures. It serves as a baseline case for comparison with the more complex models.

The Type B model introduced an aeroelastic component by incorporating both a negative linear damping term and a nonlinear positive damping term, which allowed it to capture key features of the lock-in phenomenon and the self-limiting behaviour of VIV. Its validation against full-scale data from the Aachen chimney demonstrated reasonable agreement in terms of maximum displacement and the onset of lock-in, despite uncertainties in the Strouhal number and turbulence intensity. By incorporating both the stochastic lift force and the aeroelastic damping terms, the Type B model is regarded as appropriate for both low and high structural damping cases.

Finally, the more complex Type C model, which couples the tower dynamics with a wake oscillator representing the fluid flow, offered a more physically sound description of the system. Although more complex to implement, it captures intermodal and wake-structure interactions that are critical for accurately predicting the dynamic response. The Type C model is effective for low damping cases, where the coupling between structural and fluid dynamics plays the major role in the VIV response. Nonetheless, it fails to characterize the stochastic nature of the response in high damping cases, where the turbulent vortex shedding dominates, thus showcasing the ineffectiveness and risk of performing simple calibration of its parameters to match limited results such as the maximum displacement.

The comparative analysis of these models reveals several important insights. First, the engineering models, though semi-empirical, provide a practical means of estimating the VIV response despite its acute sensitivity to uncertainties in the environmental and structural parameters. In particular, the ability of the Type B model to predict both stochastic loading and negative aerodynamic damping underscores its value in time-domain loads estimation. However, one limitation of the present Type B model is the assumption that negative damping only occurs at the structural natural frequencies, which may not hold true in scenarios where the structural motion is forced at a different frequency (such as blade VIV injecting power at the tower natural frequency). Additionally, the Type B model depends on several semi-empirical parameters, whose calibration relies on very limited data. The Type C model, while more complex, offers a more physically grounded representation of the VIV response, with reduced reliance on empirical parametrizations. However, its accuracy depends on a careful consideration of the Reynolds number effect on its parameters, and the robust incorporation of turbulence effects in the wake dynamics.

Unlike for the tower case, research on wind turbine blades VIV remains scarce, with almost no documented applications of VIV engineering models specifically tailored to blades. As

a first approximation towards developing VIV models for the blades, Chapter 4 focused on the high-fidelity fluid-structure interaction simulation of wind turbine airfoil sections at a 90° angle of attack. Three cases were considered and compared; the 2D airfoil using the URANS turbulence-modelling approach, case “2D”, and two cases of 3D simulations using the DDES turbulence-modelling approach, one for the airfoil section extruded one chord, and the other twenty-one chords, cases “3D-1c” and “3D-21c” respectively. This setup allowed for the consideration of the simplest and cheapest representation of the wind turbine blade as a 2D airfoil, to the accurate examination of three-dimensional flow structures with the 3D-1c case, up to a realistic aspect ratio for a wind turbine blade with the 3D-21c case. These two last cases were performed at a similar computational cost, as limited by time constraints.

The 2D and 3D simulations of the airfoil in a static position provided critical insights into the complex vortex shedding behaviour of wind turbine blades, revealing intermittent vortex-shedding regimes in the 3D-1c case. Two main regimes were identified, the high-drag regime H , and the low-drag regime L , with regime H resembling the 2D results, and the L regime being comparable to the 3D-21c case. The 3D simulations also highlighted the challenge of defining a single Strouhal number due to these intermittent shedding regimes. The central frequency of a Gaussian fit to the lift coefficient power spectral density proved more robust at characterizing the Strouhal number than the typical consideration of the highest peak frequency, thus offering a more reliable measure for engineering models. The importance of running the simulations for a sufficiently long time, so as to accurately capture the PSD of the lift coefficient, was also highlighted.

Having defined the Strouhal number for each case, the VIV response of the airfoil sections was simulated. The response was plotted in terms of the frequency ratio, that is, the ratio between the Strouhal-defined vortex shedding frequency and the structural natural frequency. An almost perfect match of the 2D and 3D-1c results in the lock-in range is obtained, both in terms of displacements and aerodynamic loads. Such a result is supported by the fact that the coherence of the wake flow is increased as the body undergoes large and growing vibrations, which suggests that the VIV response of the 3D airfoil in lock-in may be similar to that of the 2D simulations. This could also explain the similarity between the 3D-1c H regime and the 2D results. A good agreement between the responses predicted with the 2D and 3D-21c cases was also observed.

It was found, for all cases inside the lock-in range, that the growth in time of the response amplitude can be approximated by a linear function. In this manner, a growth rate metric was proposed, and used to effectively characterize the extent of the lock-in range. This metric is useful to identify the lock-in state in short simulations where the response is not fully developed, enabling reproducible comparisons across different studies.

The final main contribution of this thesis is the discussion presented in Chapter 5, on the applicability of well-established time-domain engineering models to the wind turbine blade case, in light of the results from the high-fidelity simulations performed. The analysis has shown that while traditional engineering models (especially the Type B model) perform well for the tower case, modifications are necessary to accurately capture the VIV behaviour in blades. For instance, parameters such as the Strouhal number, lift coefficient spectral bandwidth, and aerodynamic damping may require re-evaluation for the airfoil shape. Recommendations

have been given for the application of the three VIV models presented to the wind turbine blade case, and specific needs and considerations for further research to apply these models have been highlighted. These considerations are not limited to parameter evaluations, but also to potential inadequacies in the formulation of the models, such as the form of the nonlinearity of the Type B aerodynamic damping or the nature of the coupling between the Type C structural and wake equations. Despite these challenges, a preliminary application of the engineering models to the wind turbine blade case has been conducted, providing initial insights into their performance and limitations.

While high-fidelity simulations have the potential to serve as a benchmark to assess these modelling choices, the dependency of the VIV response on the large parameter space of relevant blade geometries, operating conditions, and environmental parameters, together with the large Reynolds numbers involved, make such characterizations highly challenging and costly. The same is true for experimental campaigns and full-scale measurements. Therefore, a combination of approaches with different fidelities remains necessary to develop more accurate and reliable models for wind turbine blades VIV.

6.2 Future work

The next steps in this research should focus on the integration of the findings from both the tower and blade studies, with an emphasis on refining the engineering models based on high-fidelity simulation data, with the final goal of introducing VIV models into the coupled aeroelastic load calculations of wind turbines. This integration is needed to evaluate the effect of VIV on ultimate limit state and fatigue loads.

Regarding the Type B models, three main opportunities for improvement have been identified. One is to make the nonlinear aerodynamic damping value a function of the instantaneous vibration frequency, and not simply of the structural natural frequency, in order to allow forced vibrations to induce negative damping, which may be important for coupled blade-tower vibrations modes. This may require a real-time calculation of the vibration frequency, and a beam-like finite element model of the structure, as opposed to the modal truncation approach currently used. The second improvement would be to adapt the Type B model to the wind turbine blade case, which would require a careful re-evaluation of the multiple parameters used in the model, such as the Strouhal number, lift coefficient power spectral density function, and aerodynamic damping. This adaptation could be partly informed by the high-fidelity simulations conducted in Chapter 4, but may require further experimental data to validate the model parameters. A third and final possibility would be to reassess the form of the nonlinear aerodynamic damping term, as the current formulation may not be suitable for the tower nor the blade case. The results from the high-fidelity simulations suggest that the aerodynamic damping should have a positive curvature for increasing displacements, as also suggested by Cheung and Melbourne (1983a), Lupi et al. (2018), and Scruton and Flint (1964). But also that the damping should not be unbounded for small displacements, as seen in the twenty-one chord case blade simulations, and that it should reach zero for large enough displacements, even for zero structural damping cases, due to the breakdown of vortex shedding (Blevins, 2001) or the establishment of a limit cycle. These two last characteristics are not captured by

the damping models proposed by Cheung and Melbourne (1983a), Lupi et al. (2018), and Scruton and Flint (1964).

The present implementation of the Type C model could also be improved upon. Most notably, the effect of turbulence in the wake and in the inflow should be considered, both in terms of its effect on the model parameters, and as an instantaneous modulation of the vortex shedding frequency. Also, it should be assessed whether considering the wake mode shape to be identical to the structural mode is a valid assumption, especially if more than one structural mode is considered. An alternative approach would be to solve the wake equation without applying the modal decomposition, for example by considering a diffusive equation for the wake oscillator, as proposed by Balasubramanian and Skop (1996). Lastly, the Type C model should be adapted to the wind turbine blade case, in a manner equivalent to the Type B model, or through more physically-based considerations, such as the wake oscillator with variable length formulation (Tamura, 2020).

Regarding the high-fidelity simulations of the airfoil sections, a deeper analysis of the simulated data could be carried out, for example by investigating the coherence of the pressure and aerodynamic loads along the blade span. Future work could focus on extending the current simulations to include a wider range of conditions, such as different angles of attack, Reynolds numbers or structural damping values. For these simulations, the 2D URANS approach could be employed for lock-in situations, as it has been found to give equivalent results to the 3D DDES approach while being computationally much less expensive. On the other hand, the 3D DDES approach could be used to characterize the behaviour outside the lock-in range, and in situations where three-dimensional effects are not negligible, such as for inclined flows or in the case of a full wind turbine blade. All this information would help to characterize the VIV response and its dependence on these parameters for the development of engineering models. Nonetheless, full-scale or wind tunnel measurements are needed to validate the simulations and engineering models, particularly for high Reynolds numbers, and are essential to characterize the effect of atmospheric boundary layer turbulent structures on VIV.

Investigating the interactions between tower and blade vibrations through measurements, high-fidelity simulations, and fully coupled aeroelastic simulations with VIV engineering models, could provide deeper insights into the integrated dynamics of wind turbines, which is crucial for the design of next-generation turbines. Some practical results that could be derived from such an integrated approach are the determination of assembly strategies that minimize aerodynamic loads, the development of control laws to minimize aerodynamic loads under standstill and idling conditions, and the determination of azimuthal positioning of blades to minimize aerodynamic loads under standstill conditions.

6.3 Final remarks

In summary, this thesis bridges the gap between traditional VIV models and the unique challenges posed by wind turbine towers and blades, offering a foundation for future research. The research has highlighted the strengths and limitations of existing models, and has provided insights into the complex fluid-structure interactions that govern VIV phenomena. The recommendations provided aim to guide further investigations, contributing to the

development of more accurate and reliable tools for wind turbine design and certification, safer operations and longer service life, and thereby supporting the global transition towards sustainable energy.

Part of the research presented in this thesis has been published in a peer-reviewed journal indexed in the Journal Citation Reports (Fernandez-Aldama et al., [2025](#)) as well as in four international conference proceedings (Fernandez-Aldama, Lopez-Garcia, Cuerva-Tejero, et al., [2024](#); Fernandez-Aldama, Papadakis, Lopez-Garcia, Avila-Sanchez, & Riziotis, [2024](#); Fernandez-Aldama et al., [2021](#), [2022](#)).

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Appendix A

Tower structural equation of motion derivation

The Euler-Bernoulli beam formulation for the tower displacement, including the tower self weight and a general external stochastic distributed force $F(z, t; \alpha)$, is:

$$m(z)\ddot{w} + t_{sd}\frac{\partial^2}{\partial z^2}\left(\overline{EI}(z)\frac{\partial^2\dot{w}}{\partial z^2}\right) + \frac{\partial^2}{\partial z^2}\left(\overline{EI}(z)\frac{\partial^2 w}{\partial z^2}\right) = F(z, t; \alpha), \quad (\text{A.1})$$

where m is the tower mass per unit length; z is the height from the ground at $z = 0$ to the tower top at $z = H$; the displacement is $w(z, t; \alpha)$, written as w for the sake of clarity, and where α is the realization number of the stochastic process, being $\alpha = 1, \dots, N_\alpha$, where N_α is the total number of processes considered; t_{sd} is a structural damping constant; and $\overline{EI}(z)$ is the sectional bending stiffness defined as $\overline{EI}(z) = \int_A z^2 E dA$.

The boundary conditions for the cantilevered beam are:

$$\begin{aligned} w(0, t) &= 0; & \left(\overline{EI}(z)\frac{\partial^2 w}{\partial z^2} + t_{sd}\overline{EI}(z)\frac{\partial^2 \dot{w}}{\partial z^2}\right)\Big|_{z=H} &= 0; \\ \frac{\partial w}{\partial z}(0, t) &= 0; & \frac{\partial}{\partial z}\left(\overline{EI}(z)\frac{\partial^2 w}{\partial z^2} + t_{sd}\overline{EI}(z)\frac{\partial^2 \dot{w}}{\partial z^2}\right)\Big|_{z=H} &= 0. \end{aligned} \quad (\text{A.2})$$

Note that these boundary conditions represent the case of an isolated tower, without the rotor-nacelle assembly weight.

Modal decomposition will now be applied. Firstly, to calculate the mode shapes, the assumption is made here that the real mode shapes can be approximated by the mode shapes of the conservative free-vibrations problem (Meirovitch, 2001), which is

$$m(z)\ddot{w} + \mathcal{L}[w] = 0, \quad (\text{A.3})$$

where $\mathcal{L}[\square]$ is defined as the following linear operator:

$$\mathcal{L}[\square] = \frac{\partial^2}{\partial z^2}\left(\overline{EI}(z)\frac{\partial^2 \square}{\partial z^2}\right). \quad (\text{A.4})$$

The classical modal decomposition is now applied by defining the displacement $w(z, t; \alpha)$ as the sum over the product of generalized coordinates and mode shapes, such that $w(z, t; \alpha) = \sum_{j=1}^{\infty} p_j(t; \alpha) \Psi_j(z)$. Equation (A.3) is thus rewritten as

$$m(z) \ddot{p}_j(t; \alpha) \Psi_j(z) + p_j(t; \alpha) \mathcal{L} [\Psi_j(z)] = 0, \quad (\text{A.5})$$

which can be reformulated as:

$$\frac{\ddot{p}_j(t; \alpha)}{p_j(t; \alpha)} = -\frac{\mathcal{L} [\Psi_j(z)]}{m(z) \Psi_j(z)} = \text{const} = -\omega_j^2, \quad (\text{A.6})$$

where ω_j is the eigenvalue of the problem, which corresponds to the natural angular frequency of mode j , as per the ordinary differential equation $\ddot{p}_j + \omega_j^2 p_j = 0$.

The natural frequencies and mode shapes are obtained by solving the following ordinary differential equation with the boundary conditions in equation (A.2):

$$\mathcal{L} [\Psi_j(z)] - m(z) \Psi_j(z) \omega_j^2 = 0. \quad (\text{A.7})$$

Once the natural frequencies and mode shapes have been solved for, to obtain the differential equation for the generalized coordinates, $p_j(t; \alpha)$, the original Euler-Bernoulli beam formulation, equation (A.1), is recovered. Applying the modal decomposition:

$$m(z) \Psi_j(z) \ddot{p}_j + \dot{p}_j t_{sd} \frac{\partial^2}{\partial z^2} \left(\overline{EI}(z) \frac{\partial^2 \Psi_j(z)}{\partial z^2} \right) + p_j \frac{\partial^2}{\partial z^2} \left(\overline{EI}(z) \frac{\partial^2 \Psi_j(z)}{\partial z^2} \right) = F(z, t; \alpha). \quad (\text{A.8})$$

Substituting $\frac{\partial^2}{\partial z^2} \left(\overline{EI}(z) \frac{\partial^2 \Psi_j(z)}{\partial z^2} \right) = m(z) \Psi_j(z) \omega_j^2$, obtained from equations (A.7) and (A.4), equation (A.8) is written as

$$m(z) \Psi_j(z) \ddot{p}_j + \dot{p}_j t_{sd} m(z) \Psi_j(z) \omega_j^2 + m(z) \Psi_j(z) \omega_j^2 p_j = F(z, t; \alpha). \quad (\text{A.9})$$

This expression can be further simplified by using the orthogonality property of the mode shapes with respect to the mass, which is:

$$\int_0^H \Psi_i(z) m(z) \Psi_j(z) dz = m_i \delta_{ij}, \quad (\text{A.10})$$

being m_i the modal mass per definition, and δ_{ij} the Kronecker delta. This condition is exploited by multiplying both sides of equation (A.9) by Ψ_i and integrating along the height, such that:

$$\begin{aligned} & \int_0^H \Psi_i(z) m(z) \Psi_j(z) \ddot{p}_j dz + \int_0^H \Psi_i(z) \dot{p}_j t_{sd} m(z) \Psi_j(z) \omega_j^2 dz + \int_0^H \Psi_i(z) m(z) \Psi_j(z) \omega_j^2 p_j dz \\ &= \int_0^H \Psi_i(z) F(z, t; \alpha) dz, \end{aligned} \quad (\text{A.11})$$

which is equivalent, for each mode i , to

$$m_i \ddot{p}_i + t_{sd} m_i \omega_i^2 \dot{p}_i + m_i \omega_i^2 p_i = \int_0^H \Psi_i(z) F(z, t; \alpha) dz, \quad (\text{A.12})$$

and defining the modal force as $F_i(t; \alpha) = \int_0^H \Psi_i(z) F(z, t; \alpha) dz$, it is obtained that

$$m_i \ddot{p}_i + t_{sd} m_i \omega_i^2 \dot{p}_i + m_i \omega_i^2 p_i = F_i(t; \alpha). \quad (\text{A.13})$$

It naturally follows to define a modal structural damping ratio as $\zeta_{s,i} = 1/2t_{sd}\omega_i$ to obtain the canonical modal equation, which is

$$m_i \ddot{p}_i + 2\zeta_{s,i} m_i \omega_i \dot{p}_i + m_i \omega_i^2 p_i = F_i(t; \alpha). \quad (\text{A.14})$$