

Aerodynamic characterization of rotor sails

An experimental study on the effects of end plate
size

MSc Marine Technology
Sabine A. Dambruin

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size

by

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Preface

Dear reader,

This thesis is the final step before completing my Master's degree in Marine Technology. As a passionate sailor, wind-assisted ship propulsion has sparked my interest as a potential solution for the decarbonization of the maritime industry. I am very grateful that, with this thesis project, I could contribute to the Wind Assisted Ship Propulsion Research Program of TU Delft.

I enjoyed bringing together my maritime knowledge from the master's program and my mechanical engineering background from the bachelor's degree, especially during the design and assembly of the experimental setup. This project also gave me the opportunity to explore the world of aerodynamics, which I greatly appreciated.

I would like to especially thank Alberto, who was my main supervisor and helped me a lot throughout the project. I am grateful for the time and enthusiasm you put into explaining me all about aerodynamics, the late hours spent assembling the setup and the long days working in the wind tunnel. I would also like to thank Martín, Erica, Hendrick and Martino for supporting me during the weeks of testing and for being there to reach out to. My thanks also go to Tom for taking the time to serve as my chair and for providing guidance during this project.

In addition I would like to thank my family for supporting me throughout my entire studies, my roommates and friends for their help and distraction when needed. I would also like to thank my boyfriend, Hidde, for his feedback and encouraging me throughout this project.

I hope you enjoy reading this report!

*Sabine Dambruin
Delft, August 2025*

Summary

The world shipping fleet contributes to 3.1% of global carbon dioxide emissions (Wang et al., 2023). While continuing to grow in size, the International Maritime Organization has set ambitious targets to reduce harmful emissions, aiming to achieve net-zero greenhouse gas emissions from international shipping by or around 2050 (IMO, 2023).

Rotor sails present a promising solution to these challenges, offering the potential for double-digit fuel savings alongside consistent power output and operational simplicity (Tillig & Ringsberg, 2020). Rotor sails, driven by the Magnus effect, were already invented in 1926, but set aside due to growing preference for diesel engines and low oil prices. In recent years they have returned as a good alternative for reducing greenhouse gas emissions, with more than twenty vessels fitted over the past decade.

The performance of a rotor sail is influenced by operating conditions, rotor installation, and geometry. To accurately predict rotor sail performance and develop strategies for improvement, a thorough understanding of the aerodynamics around the rotor is required. From literature the exact performance benefits of the addition of end plates remain unclear. Additionally, there is limited experimental research performed on wake development around (scaled) rotor sails with different end plate configurations. Therefore, the objective of this study is to understand the flow characteristics around a rotor sail, with the focus on the effects of end plate configuration. This is achieved through an experimental campaign on the wake development behind a scaled rotor sail (1:100) with varying end plate configurations in the SLT wind tunnel at TU Delft.

A part of this thesis was to design the experimental setup that could be used for this, and future research on rotor sails. The scaled rotor sail has a diameter of 50 mm and an aspect ratio of $AR = 5.2$, including the end plate thickness. The possible configurations for the end plates include a rotating bottom and top end plates with diameters equal to $De/D = [1.0, 1.6, 3.0]$. The rotor sail was vertically placed, with a static cylinder underneath the rotating one, similar to its arrangement in onboard installations. The experiments were conducted up to a spin ratio of $\alpha = 3$, which is representative of operational conditions for rotor vessels. The Reynolds number was kept constant during the experiments at its maximum of $Re = 1.5 \times 10^4$, given the model motor performance limits.

To minimize vibrations, which is of great importance while operating at high speeds, the rotor sail was equipped with magnetic couplers, shock absorber and the rotor was dynamically balanced. Additionally, the eigenfrequency of the scaled rotor sail was determined by experimental modal analysis with shaker excitation. Three eigenfrequency modes were identified: the first between 1380–1620 RPM and the second and third between 1640–2280 RPM.

Pressure measurements were performed using both a pressure rake and a 7-hole probe, making this the first experimental study to report results on all three velocity components in the wake of a scaled rotor sail. Three main wake characteristics were analyzed: 1) flow direction of the wake, 2) vortical topology of the wake flow and 3) shedding of the flow.

The results confirmed that wake deflection increases with spin ratio, showing an especially large increase in wake deflection for $\alpha = 3$. Interestingly, for this spin ratio, the end plate configuration was found to influence the deflection angle: larger end plates reduced the extent of wake deflection. Furthermore the wake width significantly decreases by increasing the spin ratio from $\alpha = 0$ up to $\alpha = 2$. For $\alpha = 3$ the vortex shedding is suppressed, leading to a steady state flow.

Analysis on the development of tip vortices indicates that the end plate size has a large influence on the vortical topology around the rotor's free ends. The addition of end plates results in the formation of disk vortices. End plates reduce the downwash effect, leading to an increase in effective aspect ratio. The results indicate that an increase in end plate size leads to a delay in the merge of the vortices. For $De/D = 3$, a clear difference in vortex behavior is observed compared with configurations without an end plate or with $De/D = 1.6$.

While the top and bottom end plates have similar aerodynamic roles, the bottom vortices are weaker and spiral in the opposite direction to the top vortices. Another observation is the opposite wake deflection for

the stationary cylinder. It is hypothesized that, without rotation, the flow follows the path of least resistance, moving from high- to low-pressure regions, in the opposite direction to the rotating case.

The spectral analysis of velocity fluctuations confirm results from previous studies (e.g. (Badalamenti, 2010; Ma et al., 2022)): an increase in spin ratio leads to an increase in Strouhal number.

A key limitation of this research is the increase in wake blockage for higher spin ratios affecting the measurements. Due to the complexity and lack of correction methods for wake blockage effects in rotating cylinder experiments, no correction has been applied. Another source of uncertainty lies in the averaging of the pressure measurements, which may influence the representation of unsteady flow features. An uncertainty analysis was carried out for both the pressure rake and the 7-hole probe measurements. For the 7-hole probe, two measurement approaches were compared, weighing the trade-off between spatial resolution and measurement accuracy. Several recommendations are made for future research, the primary is to perform 7-hole probe measurements along the full cylinder height to capture all three velocity components in the lower wake, as the bottom vortices behave differently than those at the top.

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Nomenclature

Abbreviations

Abbreviation	Definition
2D	Two-dimensional
3D	Three-dimensional
CO2	Carbon Dioxide
DNS	Direct Numerical Simulation
FFT	Fast Fourier Transform
FRF	Frequency Response Function
GHG	Green House Gases
IMO	International Maritime Organization
LES	Large Eddy Simulation
PIV	Particle Image Velocimetry
PSD	Power Spectral Density
RPM	Rotation Per Minute
RANS	Reynolds-Averaged Navier-Stokes
SE	Standard Error
URANS	Unsteady Reynolds-Averaged Navier-Stokes
WASP	Wind Assisted Ship Propulsion

Symbols

Symbol	Definition	Unit
a	Speed of sound	[m/s]
AR	Aspect ratio: $\frac{H}{D}$	[-]
C_f	Cylinder skin friction coefficient	[-]
C_l, C_d	Sectional lift and drag coefficient	[-]
C_L or $\overline{C_L}$	Overall lift coefficient	[-]
C_D or $\overline{C_D}$	Overall drag coefficient	[-]
$C_{D,i}$	Induced drag coefficient	[-]
C_{p_s}	Static pressure coefficient: $\frac{p_s - p_{s,\infty}}{\frac{1}{2}\rho U_\infty^2}$	[-]
C_{p_t}	Total pressure coefficient: $\frac{p_t - p_{t,\infty}}{\frac{1}{2}\rho U_\infty^2}$	[-]
C_P	Power coefficient: $\frac{P}{\frac{1}{2}\rho U_\infty^3 D H}$	[-]
C_τ	Torque coefficient: $\frac{\tau}{\frac{1}{2}\rho U_\infty^2 D^2 H}$	[-]
D	Rotor diameter	[m]
D_e	End plate diameter	[m]
E	Energy efficiency	[-]
F_l, F_d	Sectional lift and drag force	[N]
F_L, F_D	Overall lift and drag force	[N]
F_f	Frictional force	[N]
f_s	Natural frequency of oscillatory behavior of flow	[s ⁻¹]
H	Rotor height	[m]
I	Motor electrical current	[A]
Ma	Mach number	[-]
N	Total number of measurements	[-]
N_{eff}	Effective number of independent measurements	[-]
p	Pressure	[Pa]

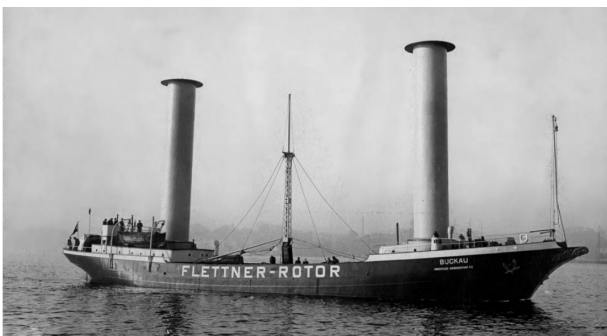
Symbol	Definition	Unit
p_s	Local static pressure	[Pa]
$p_{s,\infty}$	Freestream static pressure	[Pa]
p_t	Local total pressure	[Pa]
$p_{t,\infty}$	Freestream total pressure	[Pa]
P	Power	[W]
Q	Q-criterion for vortex identification: $\frac{1}{2}(\ \Omega\ ^2 - \ S\ ^2)$	[s ⁻²]
q	Dynamic pressure	[Pa]
R	Radius of the cylinder $R = \frac{1}{2}D$	[m]
Re	Reynolds number: $\frac{U_\infty D}{\nu}$	[-]
S	Rate of strain tensor	[s ⁻¹]
St	Strouhal number: $\frac{f_s D}{U_\infty}$	[-]
T	End plate thickness	[m]
t_m	Measurement duration	[s]
u, v and w	Local fluid velocity components in the x -, y -, and z -directions, respectively	[m/s]
U	Local fluid velocity	[m/s]
U_∞	Freestream velocity	[m/s]
u_θ	Tangential velocity of the fluid	[m/s]
U_R	Tangential velocity of the cylinder surface	[m/s]
u_r	Radial velocity of the fluid	[m/s]
V_s	Vessel speed	[kn]
α	Spin ratio: $\frac{U_R}{U_\infty}$	[-]
β	Deflection angle of the wake flow	[deg]
θ	Angle between the incoming free stream flow and a point on the cylinder's surface	[rad]
θ_s	Angle of the stagnation point	[rad]
ν	Kinematic viscosity	[m ² /s]
ρ	Fluid density	[kg/m ³]
ρ_k	Autocorrelation at lag k	[-]
σ	Standard deviation of variable X	[unit of X]
τ	Torque	[Nm]
ψ, θ	Yaw and pitch angle of the flow	[rad]
Ω	Rate of rotation tensor	[s ⁻¹]
ω	Rotor angular velocity	[rad/s]
ω_x	Vorticity in x -direction	[s ⁻¹]

Introduction

The increase in carbon dioxide (CO₂) release into the atmosphere is a major driver in climate change (Mikhaylov et al., 2020). The shipping industry undertakes approximately 90% of the world trade volume and accounts for approximately 3.1% of the global CO₂ emissions (Wang et al., 2023). In July 2023, International Maritime Organization (IMO) Member States adopted the 2023 IMO Strategy on Reduction of Green House Gases (GHG) Emissions from ships, with more ambitious goals to address harmful emissions to reach net-zero GHG emissions from international shipping by or around 2050 (IMO, 2023). From 2016 to 2020, the interest in the market of maritime transport increased from \$4.6 billion to \$8 billion (Park et al., 2022). While the size of the world shipping fleet is increasing, greener methods of ship propulsion are necessary to reduce the fleet's emissions. Greener energy sources in the shipping industry are biofuels, solar photovoltaics and Wind Assisted Ship Propulsion (WASP) (Issa et al., 2022).

According to Argyros (2015), WASP is among the limited technologies that could provide double-digit fuel savings in the present day. Furthermore, wind energy is a renewable resource characterized by its abundant availability, widespread accessibility, and environmentally friendly nature (Issa et al., 2022). Recent investigations (e.g. Lu and Ringsberg (2020), Tillig and Ringsberg (2020), and Traut et al. (2014)) have examined various wind propulsion systems, identifying fixed-wing sails (with or without flaps), rotor sails, suction wings, and kites as the most feasible solutions. According to Chou et al. (2021), the power output of rotor sails tends to be more consistent than kites. This is because rotor sails inherently generate thrust over a broader range of wind directions, making their performance less dependent on geographical location or weather conditions.

In 1926, the first rotor ship, shown in Figure 1.1a, successfully crossed the Atlantic Ocean. Despite its demonstrated potential, the rotor ship was soon set aside due to the growing preference for diesel engines and the low oil prices (Bordogna et al., 2019; Swanson, 1961). More recently, stricter environmental requirements and the volatile fuel prices have renewed interest in rotor ships (Bordogna et al., 2019). Over the past decade more than twenty vessels have been equipped with rotor sails, like vessel E-ship 1 (2010) depicted in Figure 1.1b. An overview of these vessels is provided in Table 1.1, with reported fuel savings of up to 25%. The fuel savings depend on factors such as the ship's route, type, and the number of rotors installed (Tillig & Ringsberg, 2020). Given their significant potential for fuel reduction and operational simplicity, rotor sails are considered one of the most promising solutions for reducing emissions in the shipping industry (Tillig & Ringsberg, 2020).



(a) The Buckau in the 1920s. This was the first vessel to be equipped with rotor sails. Reproduced from Kolodziejewski and Sosnowski (2025).



(b) E-ship 1 (2010) equipped with four rotor sails and top end plates ($H=27.0$ [m], $D=4.0$ [m]). Reproduced from Wikimedia Commons Contributors (2025).

Figure 1.1: Two examples of rotor vessels: (a) a historical vessel and (b) a modern example.

Table 1.1: Specifications of ships equipped with rotor sails, with D the diameter of the rotor, H the height of the rotor, and AR the aspect ratio of the rotor.

Ship (year)	Installation	Type	Fuel savings	n° rotors	H [m]	D [m]	AR	Material	Max RPM	Company	Source
E-Ship 1 (2010)	Newbuild	RoRo	15.0%	4	27.0	4.0	6.8	NA	NA	Enercon	(Marco et al., 2016)
Estraden (2014)	Retrofit	RoRo	6.1%	2	19.0	3.0	6.3	Composite	250	Norsepower	(Marco et al., 2016)
Axios (2017)	Retrofit	Bulk Carrier	12.5%	4	25.4	3.5	7.3	Composite	NA	Anemol	(Anemol Marine, 2024)
Fehn Pollux (2018)	Retrofit	General Cargo	15.0%	1	18.0	3.0	6.0	Composite	280	Eco Flettner	(Eco Flettner, 2025)
M/V Viking Grace (2018)	Retrofit	Ferry	315 tonnes/year	1	24.0	4.0	6.0	Composite	225	Norsepower	(Norsepower, 2024)
M/V Afros (2018)	Retrofit	Bulk Carrier	12.5%	4	16.0	2.0	8.0	Composite	NA	Anemol	(IRENA, 2023)
Epanastasea (2019)	Retrofit	Tanker	8.2%	2	30.0	5.0	6.0	Composite	180	Norsepower	(Norsepower, 2024)
M/V Copenhagen (2020)	Retrofit	Passenger Ship	7.0%	1	30.0	5.0	6.0	Composite	180	Norsepower	(Norsepower, 2024)
Annika Braren (2021)	Retrofit	General Cargo	15.0%	1	18.0	3.0	6.0	Composite	280	Eco Flettner	(Eco Flettner, 2025)
VLOC Sea Zhoushan (2021)	Newbuild	Bulk Carrier	8.0%	5	24.0	4.0	6.0	Composite	225	Norsepower	(Norsepower, 2024)
SC Connector (2021)	Retrofit	RoRo	25.0%	2	35.0	5.0	7.0	Composite	180	Norsepower	(Norsepower, 2024)
M/S Annika Braren (2021)	Retrofit	General Cargo	15.0%	1	18.0	3.0	6.0	Steel	280	MariGreen	(IRENA, 2023)
Alcyone (2023)	Retrofit	Tanker	8.0%	2	35.0	5.0	7.0	Composite	180	Norsepower	(Norsepower, 2024)
Berbe Neblina (2024)	Retrofit	Bulk Carrier	8.0%	4	35.0	5.0	7.0	Composite	NA	Anemol	(Anemol Marine, 2024)
Oceanus Aurora (2024)	Newbuild	Gas Carrier	4.0%	2	20.0	4.0	5.0	Composite	NA	Norsepower	(Norsepower, 2024)
Delphine (2024)	Retrofit	RoRo	10.0%	2	35.0	5.0	7.0	Composite	180	Norsepower	(Norsepower, 2024)

Accurately predicting savings from wind-assisted propulsion requires estimating yaw and heel moments, thrust, and drift. Unlike conventional propellers, rotor sails generate a side force that affects the ship's drift, rudder angle, and heel, which in turn influences yaw moments and sail efficiency (Tillig & Ringsberg, 2020). To predict these forces and optimize the performance of rotor sails, a thorough understanding of the aerodynamics around a rotor sail is required.

In addition to operating conditions, rotor performance is influenced by its geometry and installation. Chapter 2 reviews the aerodynamic theory of rotor sails, based on existing literature. This chapter is partially based on the literature review conducted earlier as part of this thesis project. It examines the influence of end plate configuration on the lift and drag of the rotor and reveals the unknowns of the exact performance benefits. From this knowledge gap, the research objective and questions are formulated. These questions are addressed through an experimental campaign on wake development around a scaled rotor sail in the SLT wind tunnel at the TU Delft. The results contribute to the fundamental understanding of the flow characteristics around the rotor, particularly in relation to the end plate configuration.

Chapter 3 provides an overview of the experimental setup, which was developed specifically for this study and can also be used in future research on rotor sails. Furthermore, it describes the measurement methods and data processing procedures used during the experimental campaign and examines the limitations on the collected data. Chapter 4 presents the results on the aerodynamic characterization of the scaled rotor sail. Chapter 5 summarizes the conclusions and answers the research questions. Chapter 6 gives an overview of recommendations for future research.

State of the art in rotor sail aerodynamics

In this chapter the fundamental theory related to the aerodynamics of rotor sails is discussed, and the knowledge gap is identified, from which the research objective and questions are formulated. First, in section 2.1 the key performance metrics and their notation are introduced, which are used throughout the report. Secondly, an introduction to the Magnus effect is provided in section 2.2. Then, in section 2.3 the flow topology around a rotating cylinder is examined. In section 2.4, the influence of end plate configuration on aerodynamic performance according to existing literature is discussed. Lastly, in section 2.5 the knowledge gap is identified and the research questions are defined.

2.1. Introduction to definitions and notation

This section defines the key performance metrics and introduces the notation used throughout the report to describe the behavior of a rotor sail in a fluid flow. These definitions are essential for understanding the aerodynamic characteristics of the rotor, including its lift, drag, and rotational effects.

In Figure 2.1a the dimensions of a rotor sail are presented. Where H , in [m], is the rotor height and D the rotor diameter in [m]. On the rotor end plates, round disks, are attached to the end of the rotor to improve aerodynamic performance. D_e is the end plate diameter in [m] and T the end plate thickness in [m]. The freestream wind velocity is indicated with U_∞ in [m/s] and the angular velocity of the rotor with ω in [rad/s]. In Figure 2.1b a section view of a rotor is presented. Here U_R is the tangent velocity in [m/s] on the cylinder surface. The angle θ , in [rad], is the angle between the incoming free stream flow and a point on the rotor surface. Using these notations the following non-dimensional parameters can be defined.

Firstly, the sectional lift and drag coefficient are defined by non-dimensionalizing the sectional lift and drag force, F_l and F_d , defined in Figure 2.1b, given in force per unit span ([N/m]). They are defined as:

$$C_l = \frac{F_l}{\frac{1}{2}\rho U_\infty^2 H^2 D} \quad (2.1)$$

$$C_d = \frac{F_d}{\frac{1}{2}\rho U_\infty^2 H^2 D} \quad (2.2)$$

where ρ is the fluid density.

The overall lift and drag coefficient are defined by non-dimensionalizing the overall lift and drag force, F_L and F_D given in [N]. They are defined as:

$$C_L = \frac{F_L}{\frac{1}{2}\rho U_\infty^2 H D} \quad (2.3)$$

$$C_D = \frac{F_D}{\frac{1}{2}\rho U_\infty^2 H D} \quad (2.4)$$

where ρ is the fluid density.

The pressure coefficient is a dimensionless number and describes the relative pressure variation. To identify pressure variation along the surface of a rotor sail, it is defined by the difference between the static pressure, p_s and the freestream static pressure, p_∞ , given in [Pa].

$$C_{p_s} = \frac{p_s - p_{s,\infty}}{\frac{1}{2}\rho U_\infty^2} \quad (2.5)$$

The pressure variation in the wake of a rotor sail can be studied using the pressure coefficient defined by the difference between the total pressure, p_t and the freestream total pressure, p_∞ , given in [Pa].

$$C_{p_t} = \frac{p_t - p_{t,\infty}}{\frac{1}{2}\rho U_\infty^2} \quad (2.6)$$

The spin ratio, α , is a dimensionless parameter that characterizes the ratio of the tangential velocity of the rotor's surface to the freestream velocity of the fluid. It is defined as:

$$\alpha = \frac{U_R}{U_\infty} = \frac{\omega \cdot \frac{1}{2}D}{U_\infty} \quad (2.7)$$

The aspect ratio, AR , describes the ratio of the height, H , of the cylinder to its diameter, D . It is given by:

$$AR = \frac{H}{D} \quad (2.8)$$

The Reynolds number, Re , is a dimensionless number that characterizes the flow regime around the rotor. For a rotor, the diameter is typically used as the characteristic length to define the Reynolds number. It is defined as:

$$Re = \frac{U_\infty D}{\nu} \quad (2.9)$$

where ν is the kinematic viscosity of the fluid in $[m^2/s]$.

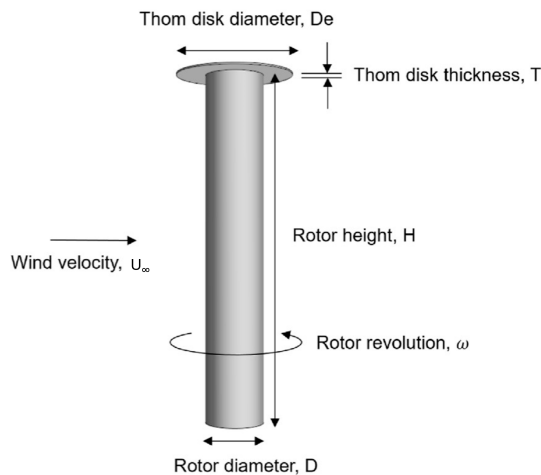
The Strouhal number, St , is a dimensionless parameter that characterizes the oscillatory behavior of the flow using the frequency, f_s in $[s^{-1}]$. For flows around a rotor the Strouhal number is typically used to describe the periodic vortex shedding phenomena. It is given by:

$$St = \frac{f_s D}{U_\infty} \quad (2.10)$$

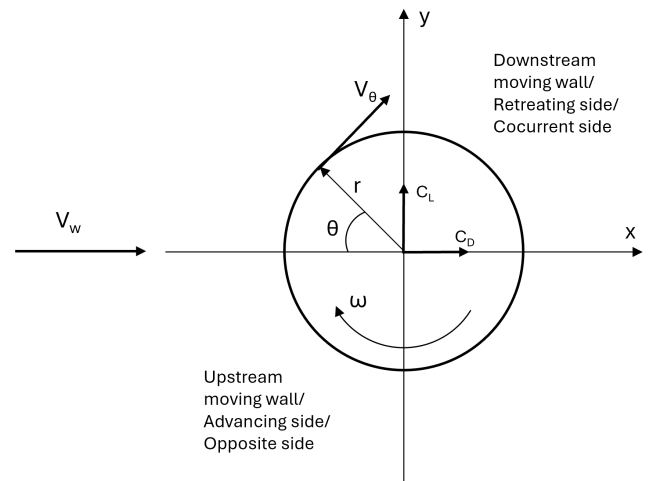
The Mach number, Ma , is another dimensionless number defined as the ratio between the local flow velocity and the speed of sound. The Mach number defines the compressibility of a fluid flow and indicates whether the flow is subsonic, $Ma < 1$, sonic, $Ma = 1$, or supersonic, $Ma > 1$ (Anderson, 2010). When $Ma < 0.3$, the density changes in the fluid are negligible, and the flow can be treated as incompressible (Anderson, 2010). For $Ma \geq 0.3$, compressibility effects become significant, and the changes in density must be accounted for in the analysis. The Ma is defined as:

$$Ma = \frac{U}{a} \quad (2.11)$$

where U is the local fluid velocity in $[m/s]$ and a the speed of sound in $[m/s]$. The speed of sound is defined by rate of change of pressure, δp , with respect to density, $\delta \rho$, at constant entropy s : $a = \sqrt{\left(\frac{\delta p}{\delta \rho}\right)_s}$.



(a) Dimensions of the rotor sail. Reproduced from (Kwon et al., 2022).



(b) Schematic overview of axes and notation for the problem setup of a rotor sail.

Figure 2.1: Schematic overview of the dimensions and notations used in the analysis of the rotor sail.

2.2. Magnus effect

The lift for a rotor sail is caused by the Magnus effect, discovered by Gustav Magnus in 1852 (Magnus, 1852). For a stationary (i.e. non-rotating) cylinder, the flow around the cylinder separates, leading to the formation of a von Kármán vortex street. In contrast, for a rotating cylinder, the streamlines are deflected toward the side of the cylinder rotating in the direction of the flow, creating a pressure differential and altering the wake structure, see Figure 2.2a.

For a certain range of Re and at low α ratio, a lift force can be found in opposite direction of the regular Magnus force and was first observed by Lafay (1910). This is the so called *negative* or *inverse Magnus effect* (Seifert, 2012; Swanson, 1961). The pattern diagram of the negative Magnus force can be seen in Figure 2.2b. Swanson (1961) explained that the inverse Magnus effect arises from variations in boundary layer development and separation between the advancing and retreating surfaces of a spinning sphere. At a critical spin ratio, the shear layer on the advancing side reattaches to the rear of the sphere due to a transition from laminar flow to turbulence. Meanwhile, the boundary layer on the retreating side remains laminar, leading to earlier separation. The boundary layer development and separation are discussed in more detail in section 2.3.

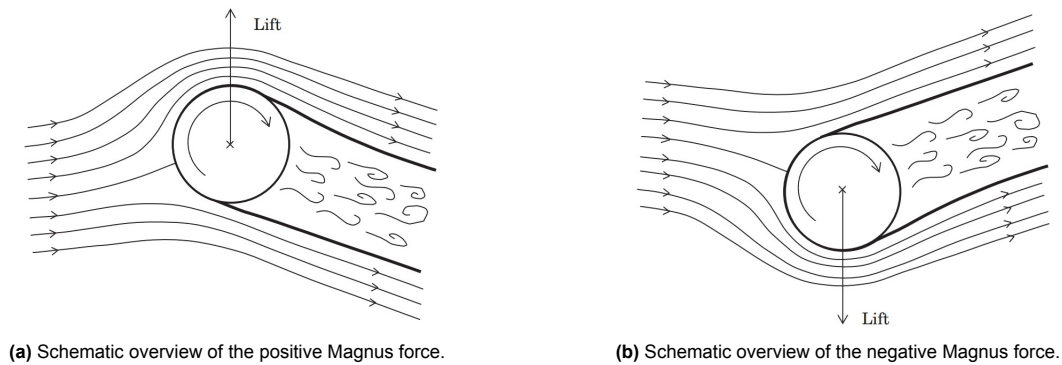


Figure 2.2: Diagrams showing positive and negative Magnus forces. *Reproduced from Maruyama (2010).*

2.3. Flow topology

In this section the flow topology around a spinning cylinder is examined by first discussing the boundary layer flow and the change of separation and stagnation points. Then, the wake characteristics around an infinite span cylinder will be discussed. Lastly, the influence of the end plate configuration on the aerodynamic performance of a rotor sail will be examined.

2.3.1. Boundary layer and wake development across flow regimes and spin ratios of an infinite span rotating cylinder

Boundary layer separation plays a critical role in determining the lift and drag characteristics across various flow regimes. The development of the boundary layer and points of flow separation are affected by Reynolds number and spin ratio. Meaning that the Magnus effect itself depends as well as on the spin ratio as on the Reynolds number.

For a finite span stationary cylinder ($\alpha = 0$), the flow can be characterized into four different regimes using the Reynolds number; the subcritical flow regime, the critical flow, the supercritical flow regime and the transcritical regime (Chen et al., 2023). Figure 2.3 shows these flow regimes. In the subcritical flow regime, the flow is fully laminar along the cylinder's surface. The subcritical regime is found to be until $Re \approx 2.0 \times 10^5$ (Chen et al., 2023). The critical flow regime, is the transitional flow regime from laminar to turbulent flow where laminar separation bubbles and turbulent reattachment form. In the supercritical flow regime the flow is fully turbulent and characterized by less regular vortex shedding behavior and extends to a very high Re . In the transcritical regime, laminar bubbles are absent and the boundary layers on the cylinder transition to turbulence before separation occurs for $Re \geq 5 \times 10^6$ (Chen et al., 2023).

For a rotating cylinder, the spin ratio has a large influence on the flow characteristics and determines its flow topology. The downstream-moving wall experiences delayed separation due to the cylinder surface transferring kinetic energy to the local boundary layer (Badalamenti, 2010). On the upstream moving wall, the cylinder

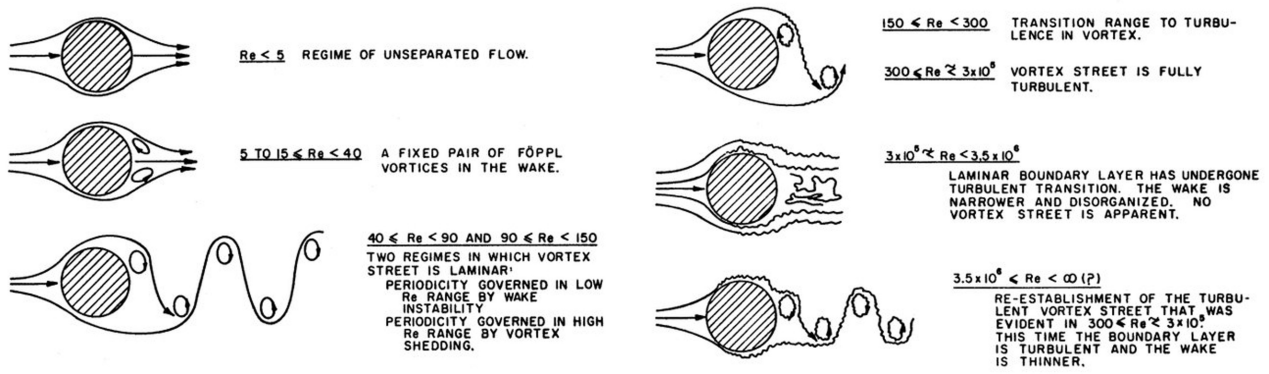


Figure 2.3: Various flow regimes over a finite span stationary cylinder. Reproduced from Lienhard et al. (1966).

rotates against the free stream. This lowers the kinetic energy of the boundary layer, leading to early separation (Badalamenti, 2010). As a result, the wake and boundary layer separation points are moved in the cylinder's rotational direction (Badalamenti, 2010). However, for higher spin ratios the effects of the rotation suppresses the separation completely on both sides, which lead to the steady-state flow (Badalamenti, 2010).

Using the spin ratio, the flow around a rotating cylinder can be categorized into four different regimes. Thouault et al. (2012) did a numerical flow visualization around a rotating cylinder at $Re = 7.2 \times 10^4$. Seifert (2012) made an overview of their results, which are shown in Figure 2.4.

The first regime is found for $0 \leq \alpha < \alpha_{cI}$, where flow separation around the surface of the cylinder appears, separation points can be found and a von Kármán street is seen in the wake behind the cylinder (Chew et al., 1995; Mittal & Kumar, 2003; Padrino & Joseph, 2006; Stojković et al., 2002). In Figure 2.5 the critical velocity ratio, α_{cI} , is plotted against different Reynolds numbers. Both experimental and numerical results indicate that the critical velocity ratio follows a logarithmic relationship with respect to Re and the value of α_{cI} increases with Re .

The second regime is found for $\alpha_{cI} \leq \alpha < \alpha_{cII}$, where vortex shedding is suppressed by the formation of closed streamlines and the flow transforms to a steady-state flow. Mittal and Kumar (2003) and Stojković et al. (2002) found in numerical studies that a second shedding mode is observed for $\alpha_{cII} \leq \alpha \leq \alpha_{cIII}$. This is later confirmed in an experimental study by Kumar et al. (2011). Thouault et al. (2012) observed this shedding regime numerically for $Re = 7.2 \times 10^4$ and $\alpha_{cII} = 3.5$. In the second shedding region, in contrast to the first, the vortices are shed in counter-clockwise sense. The Strouhal number is much lower in the second shedding region and shows a strong dependence on α and relatively less on Re . The fourth and last regime is found when $\alpha \geq \alpha_{cIII}$ where the flow becomes stable again.

$\alpha = 0.0$ $St = 0.20$		First shedding mode. Alternate vortex shedding of Von Kármán type
$\alpha = 1.0$ $St = 0.26$		
$\alpha = 2.0$ $St = 0.46$		
$\alpha = 3.0$		Quasi-steady flow. Closed streamlines around cylinder. Two stationary vortices located on the cylinder lower side.
$\alpha = 3.5$ $St = 0.02$		Second shedding mode. Only one vortex is shed
$\alpha = 4.0$		Quasi steady flow. Flow topology is comparable to the potential theory solution
$\alpha = 6.0$		

Figure 2.4: Flow patterns at $Re = 7.2 \times 10^4$ for various velocity ratios α and Strouhal numbers St (flow from left to right, clockwise rotation) (Thouault et al., 2012). Reproduced from Seifert (2012).

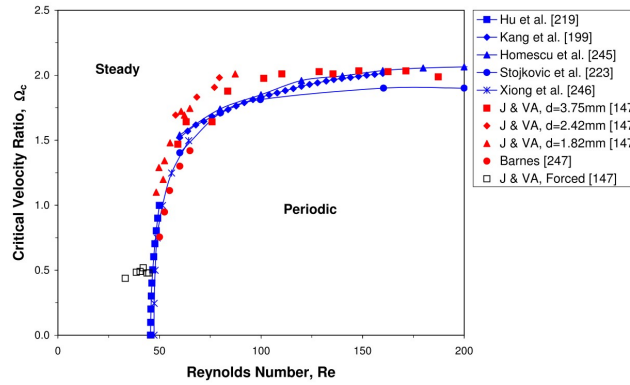


Figure 2.5: Reynolds dependent transition between steady and periodic flows for the first shedding regime. *Reproduced from Badalamenti (2010).*

2.3.2. Wake topology around a finite span cylinder

The wake flow behind a rotating cylinder with a finite span, such as a rotor sail, can exhibit more complex behavior compared to a infinite span cylinder. Chew et al. (1995), Stojković et al. (2002) and Mittal and Kumar (2003) have extensively analyzed the aerodynamic flow around an infinite span rotating cylinder, while Badalamenti and Prince (2008), Thouault et al. (2012), Bordogna et al. (2020), Ma et al., 2022 and Liu, Ma, Zheng, et al. (2024) have investigated the flow characteristics of a finite span cylinder. The wake of a rotating cylinder can be characterized by; the Strouhal number (St), the flow pitch angle (θ), the ratio between the free stream velocity and the wake velocity and lastly the turbulence intensity in the wake.

A finite span cylinder has to deal with flow structures around the free ends, which are completely different from that around its main body (Adaramola et al., 2006; Afgan et al., 2007; Wang et al., 2012b). The flow around the near end consists of tip vortices and base vortices near the wall that create downwash and upwash flows and make the near wake highly three-dimensional (Sumner et al., 2004; Wang et al., 2012a). Figure 2.6 shows a contourplot of the streamwise mean velocity field for a static circular cylinder of $AR=5$, where the strong downwash from the tip vortex structures interfering with the Von Kármán vortex street is observed. Mittal (2004) found that even if the two-dimensional flow is stable, centrifugal instabilities exist along the entire span in a three-dimensional setup. Also, the inflow around the cylinders ends into the wake leads to a higher wake pressure (Badalamenti & Prince, 2008). These combined effects cause a decrease in lift and an increase in drag in comparison to a two-dimensional flow (Mittal, 2004).

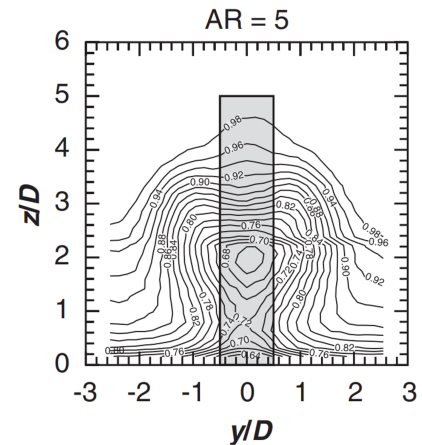


Figure 2.6: Streamwise mean velocity field in the wake of a static cylinder, $X/D = 6$, $AR = 5$ and $Re = 6 \times 10^4$. Contours of U/U_∞ with increments of 0.02. *Reproduced from Adaramola et al. (2006).*

Only limited experimental research has been conducted on the wake of a finite span rotating cylinder. Ma et al. (2022) and Bordogna et al. (2020) performed wake flow measurements using Cobra Probes, allowing for the measurement of both wind velocity and flow angle. A schematic overview of the experimental setup from Ma et al. (2022) is shown in Figure 2.8. Additionally, Badalamenti and Prince (2008) conducted pressure measurements in the wake region, while Liu, Ma, Zheng, et al. (2024) carried out a numerical study that included wake pressure measurements.

Liu, Ma, Zheng, et al. (2024) studied the formation of tip vortices for different end plate configurations. Figure 2.7 shows the size of the tip vortices of a cylinder with rotating end plates for varying spin ratios. Badalamenti and Prince (2008) concluded from their experimental wake measurements that the formation of tip vortices starts from $\alpha \geq 1$. Lift and drag coefficients are affected by variations in the intensity and spanwise location of vortices as α increases.

Ma et al. (2022) analyzed the variation of the wake pitch angle for a rotating cylinder ($AR = 10$, $De/D = 2$) for

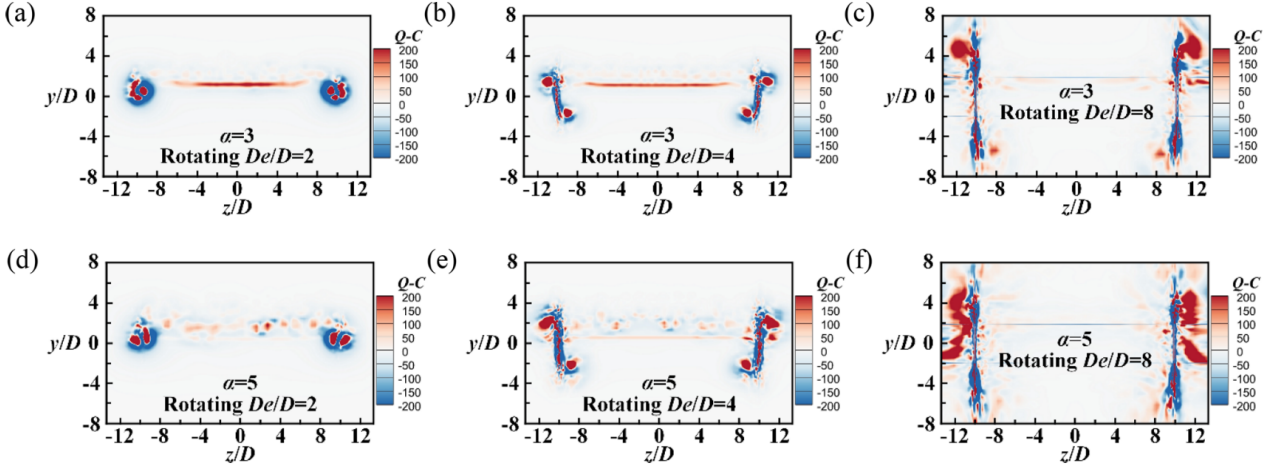


Figure 2.7: Vortex magnitude of the rotor with rotating end plates, $AR = 20$ and $Re = 4.6 \times 10^4$ at $X/D = 3$ for: (a) $\alpha = 3$, $D_e/D = 2$, (b) $\alpha = 3$, $D_e/D = 4$, (c) $\alpha = 3$, $D_e/D = 8$, (d) $\alpha = 5$, $D_e/D = 2$, (e) $\alpha = 5$, $D_e/D = 4$, and (f) $\alpha = 5$, $D_e/D = 8$. Reproduced from Liu, Ma, Zheng, et al. (2024).

different Reynolds numbers in the subcritical and critical flow regime. They found that the variation in wake pitch angle, illustrated in Figure 2.9, can be divided into the effect of the separation bubble and rotating speed ratio: for speed ratio $\alpha < 0.7$, the direction of the wake flow is mainly affected by the separation bubble, while the variation in the wake angle at different measuring points is the same in the range $\alpha > 0.7$, and the wake flow pitch angle increases with the increase in the rotating speed ratio.

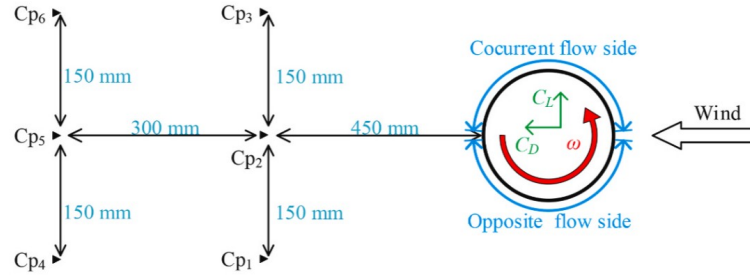


Figure 2.8: Schematic overview of the experimental setup from Ma et al. (2022) for wake measurements using Cobra probes. Reproduced from Ma et al. (2022).

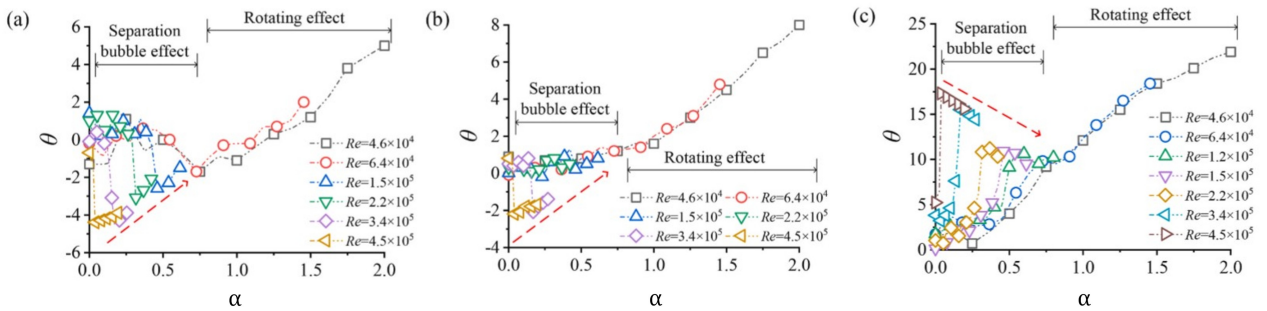


Figure 2.9: Influence of the rotating speed ratio on the wake flow pitch angle at various Reynolds numbers: (a) C_{p4} , (b) C_{p5} (c) C_{p6} ($AR = 10$, $D_e/D = 2$). Reproduced from Ma et al. (2022).

Furthermore, Ma et al. (2022) and Thouault et al. (2012) studied the Strouhal numbers for $10^4 \leq Re \leq 4.1 \times 10^5$. Ma et al. (2022) found that, for the same Reynolds number range, the relationship between the Strouhal number and spin ratio remains consistent across different Cobra probes. This indicates that the vortex shedding frequency of the wake flow is unaffected by the wake deflection. Figure 2.10 and Figure 2.11 show the result of the Strouhal number for various spinning ratios from the studies of Ma et al. (2022) and

Thouault et al. (2012), respectively. The Strouhal number increases for increasing spin ratios. At higher Reynolds numbers, the increase in the Strouhal number happens discontinuously as the flow transforms from the subcritical regime to the one-bubble regime. In the one-bubble regime, the Strouhal number slowly decreases. The results of Ma et al. (2022) correspond for the subcritical flow regime, $Re = 4.6 \times 10^4$, to the results of Thouault et al. (2012) for $Re \geq 10^4$.

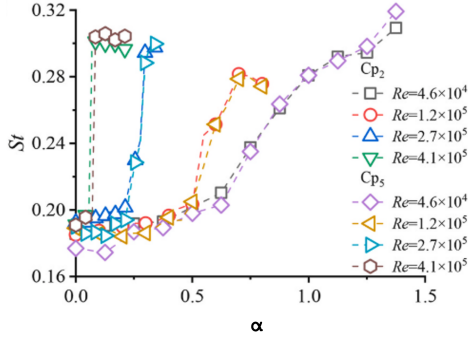


Figure 2.10: Variation in Strouhal number, St , for various spinning ratios and Reynolds number at Cobra probe 2 and 5. Reproduced from Ma et al. (2022).

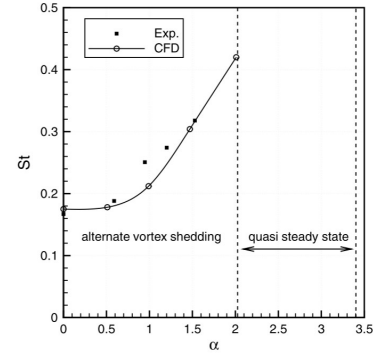


Figure 2.11: Variation of the Strouhal number, St , for various spinning ratios, α , for a finite cylinder ($D_e/D = 2$, $A = 5.1$ and $Re \geq 10^4$). Reproduced from Thouault et al. (2012).

Bordogna et al. (2020) did wake flow measurements for a single rotor sail with dimensions $AR = 5$, $D_e/D = 2$, $7.0 \times 10^4 \leq Re \leq 2.4 \times 10^5$ and found that the cylinder wake is reduced with the increase of the velocity ratio. In Figure 2.12 it can be seen that at $\alpha = 2$, the vortex shedding is fully suppressed and the flow is heavily deviated, up to an angle of about 40° at a distance $x/D = 1.5$. Increasing the velocity ratio strengthens the circulation, leading to a more pronounced downwash effect directly behind the rotor. In the wake of the advancing side ($1 < Y/D < 4$) the circulation causes the flow speed to increase compared to the freestream velocity. On the other hand, the circulation does not seem to cause any decrease in flow speed on the wake of the retreating side of the cylinder ($-1 < Y/D < -4$), where the flow appears to have a speed comparable to the freestream velocity. This result is confirmed by Ma et al. (2022) who found an increase in wake velocity on the direct downstream separated flow region of the cylinder wake for Reynolds numbers in the subcritical regime, $Re = 4.6 \times 10^4$.

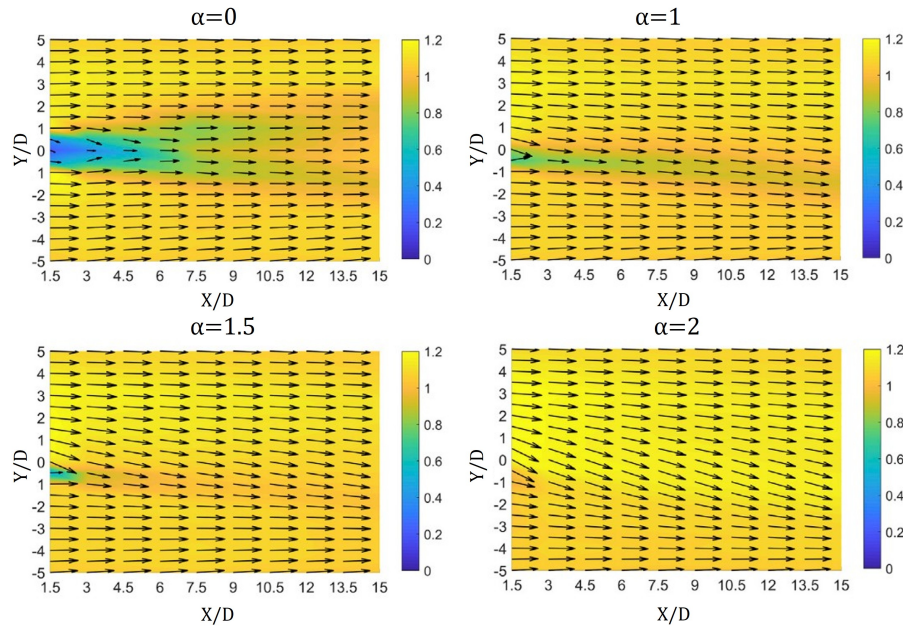


Figure 2.12: Velocity field magnitude, normalized with the freestream velocity, past the single rotor sail with dimensions $AR = 5$, $D_e/D = 2$ and $7.0 \times 10^4 \leq Re \leq 2.4 \times 10^5$ at different velocity ratios. The rotor rotates clockwise and is located at the center of the reference system ($X/D = 0$, $Y/D = 0$). Reproduced from Bordogna et al. (2020).

Wake fluctuation intensity is a measure of the turbulence within the wake, which arises from factors such as vortex shedding and flow separation. Ma et al. (2022) found that the flow state has a significant influence on fluctuation intensity. The subcritical flow state shows a significant reduction in fluctuation intensity on the retreating side as the rotating speed ratio increases, with minimal impact on the advancing side. In the one-bubble regime, only the fluctuation intensity on the advancing side differs to that of the subcritical regime.

2.3.3. Comparison of aerodynamic force coefficients from literature

An overview of selected studies on rotating cylinders, is presented in Table 2.1. The studies are selected on $Re_{max} \geq 1.0 \times 10^4$ and spin ratios $\alpha_{max} \geq 3$ to make the test parameters, and thus the results, more comparable to real ship applications and to each other. In Figure 2.13 an overview is given for the test domain of the experimental studies shown in Table 2.1.

In Figure 2.14 an overview of the lift and drag coefficients, C_L and C_D for these studies is given. It can be seen that for increasing α the lift and drag coefficient generally increases until the lift plafond is reached for a certain spin ratio. An analogy between the velocity ratio of a rotating cylinder and the angle of attack for a wing can be made: as the aerodynamic lift correlates directly with both parameters (Badalamenti, 2010). The differences between the results of the studies are mainly explained by the different AR , Re and end plate configurations.

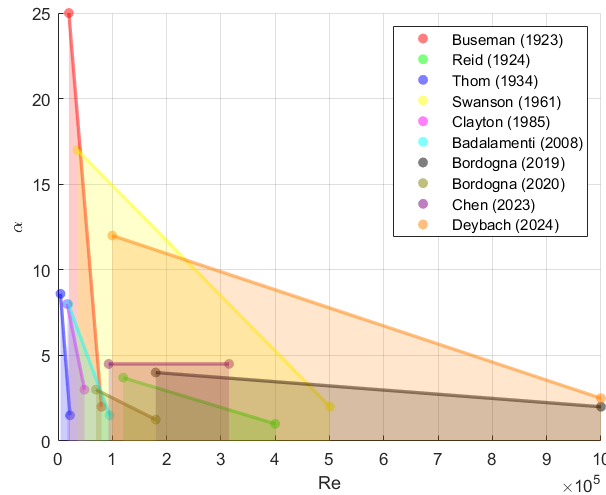


Figure 2.13: Domains of tested spin ratios and Reynolds number across experimental studies presented in Table 2.1.

Table 2.1: Overview of studies on the aerodynamic performance of rotating cylinders. The table includes both experimental (E) and numerical (N) studies, showing a range of parameters including diameter of the cylinder (D), aspect ratio (AR), diameter ratio of the end plate (D_e/D), Reynolds number (Re), spinning ratio (α), and the facility or method used with Reynolds-Averaged Navier-Stokes (RANS) and Large Eddy Simulation (LES).

Researchers	E/N	D [mm]	AR [-]	D_e/D [-]	Re [-]	α [-]	Facility/method
Reid (1924)	E	114.3	2D	None	$\approx 4.0 \times 10^4 - 1.2 \times 10^5$	0–3.7	Wind tunnel
Busemann (1932)	E	60	1.68–12	1.0–3.0	$\approx 2 \times 10^4 - 8 \times 10^4$	0–10	Wind tunnel
Thom (1934)	E	25.4–50.8	12.5–26	3	$4.5 \times 10^3 - 4.73 \times 10^4$	0–7.3	Wind tunnel
Swanson (1961)	E	152.2	2D	None	$3.58 \times 10^4 - 5.01 \times 10^5$	0–17	Wind tunnel
Clayton (1985)	E	48	10.4	1.0–3.0	$1.7 \times 10^4 - 4.89 \times 10^4$	0–8	Wind tunnel
Badalamenti and Prince (2008)	E	88.9	5.1	1.0–3.0	$1.6 \times 10^4 - 9.5 \times 10^4$	0–8	Wind tunnel
Thouault et al. (2012)	E/N	88.9	5.1	1.1–3.0	$1.8 \times 10^4 - 9.6 \times 10^4$	1–3.4	Wind tunnel/RANS
Karabelas et al. (2012)	N	-	2D	None	$200 - 5 \times 10^6$	0–8.0	RANS
Bordogna et al. (2019)	E	1000	3.73	None	$1.8 \times 10^5 - 10 \times 10^5$	0–5	Wind tunnel
Bordogna et al. (2020)	E	300	5	2.0	$7.0 \times 10^4 - 2.5 \times 10^5$	0–3	Wind tunnel
Kwon et al. (2022)	N	158	4.0–8.0	1.5–8	$2.24 \times 10^4 - 7.66 \times 10^5$	1.0–4.0	RANS
Chen et al. (2023)	E	350	3.5–6.0	1.0–2.0	$9.33 \times 10^4 - 3.15 \times 10^5$	0–4.5	Wind tunnel
Deybach et al. (2024)	E	400	6	1.0–2.0	$1 \times 10^5 - 1 \times 10^6$	0–12	Wind tunnel
Liu, Ma, Jin, and Liu (2024)	N	150	20	None	4.6×10^4	0–3	LES
Liu, Ma, Zheng, et al. (2024)	N	150	20	2.0–8.0	4.6×10^4	0–5	LES

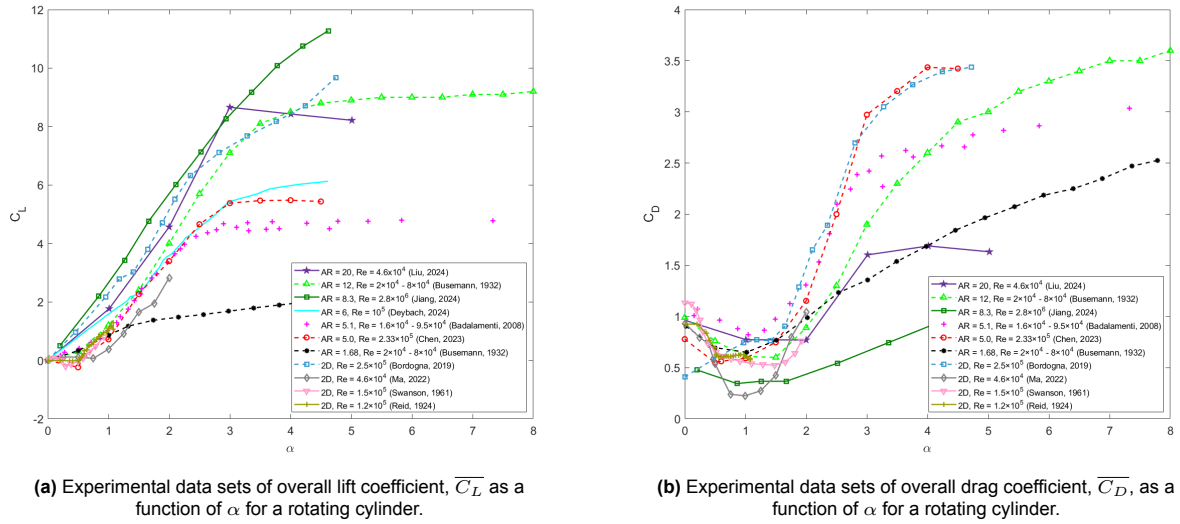


Figure 2.14: Comparison of experimental data for lift and drag coefficients as a function of α for a rotating cylinder without end plates.

2.4. Influence of end plate configuration on aerodynamic performance of rotor sails

End plates have been explored as a means to improve aerodynamic performance of rotors. The idea of adding end plates, to increase the lift-to-drag ratio, was first proposed by Prandtl (Prandtl, 1925). Ten years later, Thom (1934) conducted the first research on large end plates with a size ratio of $D_e/D = 3.0$. This research suggested that end plates could double the lift at high velocity ratios ($\alpha \geq 2$), which is why end plates are sometimes referred to as Thom disks. More recently Badalamenti (2010), Chen et al. (2023), Deybach et al. (2024), Liu, Ma, Zheng, et al. (2024), and Thouault et al. (2012) did research on the influence of end plates on the flow topology around a rotating cylinder.

Figure 2.15 shows the percentage change in lift and drag coefficients for a cylinder with rotating end plate, varying in size, ranging $1.1 \leq D_e/D \leq 3$, in comparison to no end plate case obtained from experimental research from Badalamenti and Prince (2008). From these results it can be concluded that an increase in end plate size or aspect ratio is similar: an increasing trend in lift, along with the postponement of the lift plateau for higher spinning ratios. Badalamenti and Prince (2008) found that the relationship between plate size and the increase in the maximum lift coefficient was proportional: the percentage increase in the maximum lift (based on $\alpha > 4$) relative to the case without end plates was approximately the same as the percentage increase in the plate size ratio in comparison to $D_e/D = 1$. Furthermore, they concluded that the optimum end plate size depends on the spinning ratio. The lift-to-drag optimum occurs close to $\alpha = 2$. For higher spin ratios, the lift-to-drag ratio was substantially increased relatively to no end plates.

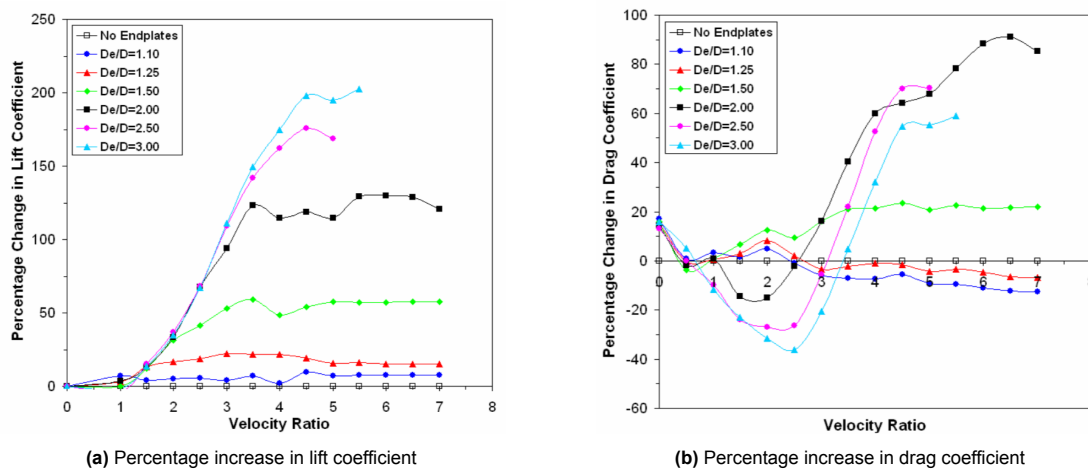


Figure 2.15: Effect of end plate size on the cylinder's force coefficients in comparison to no end plate. Results for end plate sizes $1.1 \leq D_e/D \leq 3$ on both ends of the cylinder, $AR = 5.1$, $Re = 1.6 \times 10^4 - 9.5 \times 10^4$. Reproduced from Badalamenti and Prince (2008).

The increase in lift-to-drag ratio can be explained by the change in formation of vortices around the rotors end. The pressure gradients around the rotors end cause cylinder axial flow and result in the formation of tip vortices. Liu, Ma, Zheng, et al. (2024) performed a numerical investigation on the effect of end plates using LES and made a schematic overview on the formation of vortices at the rotor free end, shown in Figure 2.16. Static end plates can reduce the intensity of the rotor tip vortex to a certain degree by restricting the axial flow, while two weaker disk vortices will be generated at the edge of the end plate, having a triple vortex pattern. The rotating end plate increases the strength of the disk vortices and preventing the formulation of big tip vortices. This results in a reduction of the induced drag. In the numerical research of Thouault et al. (2012) on end plates, using Unsteady Reynolds Averaged Navier-Stokes (URANS) simulations, an overview of this flow topology in the first shedding mode, $\alpha = 2$, is shown, see Figure 2.17. Vortex shedding is observed between the end plates. Along the end plates, the flow spirals toward the edges, eventually separating and forming two tip vortices. Additionally, Figure 2.18 clearly illustrates, for a cylinder with rotating end plates, the formation of small tip vortices and the formation of the two disk vortices, that merge further downstream rotating end plate.

Additionally, Liu, Ma, Zheng, et al. (2024) studied the pressure field in the wake at a spin ratio of $\alpha = 5$ for different end configurations, shown in Figure 2.19. The different end plate configurations, causes variations in the pressure distribution along the cylinder axis. For the rotor without end plates the extent of negative pressure on the cocurrent side slowly decreases, but the effect of the opposite side is more significant. For a static end plate the negative pressure area at the opposite is weakened, but still exists. The rotating end plate eliminates the negative pressure area at opposite side and extends the negative pressure at the cocurrent side leading to an increase in lift and demonstrating a better Magnus effect at the cylinders end.

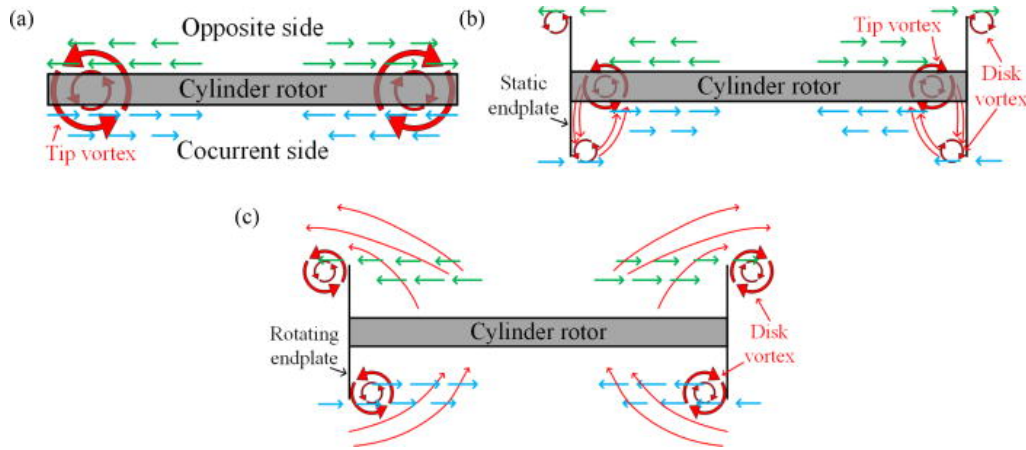


Figure 2.16: Schematic diagram of vortices at the rotor free end: (a) cylinder rotor, (b) static end plates, and (c) rotating end plates. Reproduced from Liu, Ma, Zheng, et al. (2024).

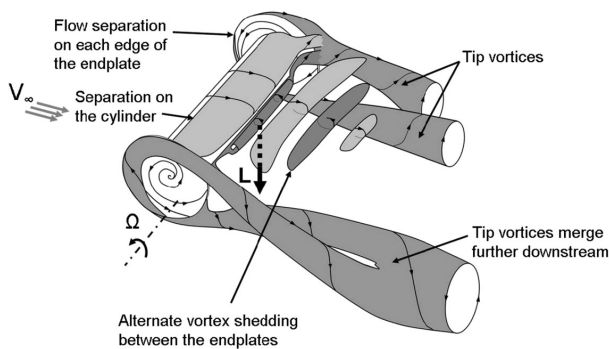


Figure 2.17: Flow topology at $\alpha = 2$ ($Re = 7.1 \times 10^4$, $D_e/D = 2$, $AR = 5.1$). Reproduced from Thouault et al. (2012).

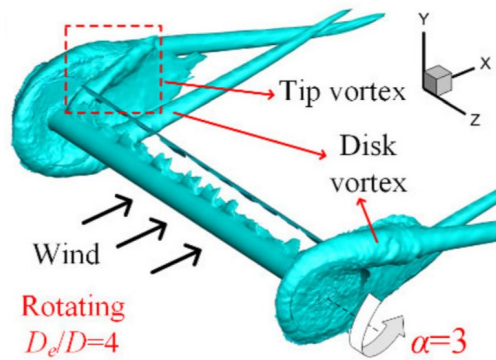


Figure 2.18: Vortical structures with rotating end plates of size $D_e/D = 4$ for $\alpha = 3$, $Re = 4.6 \times 10^4$. Reproduced from Liu, Ma, Zheng, et al. (2024).

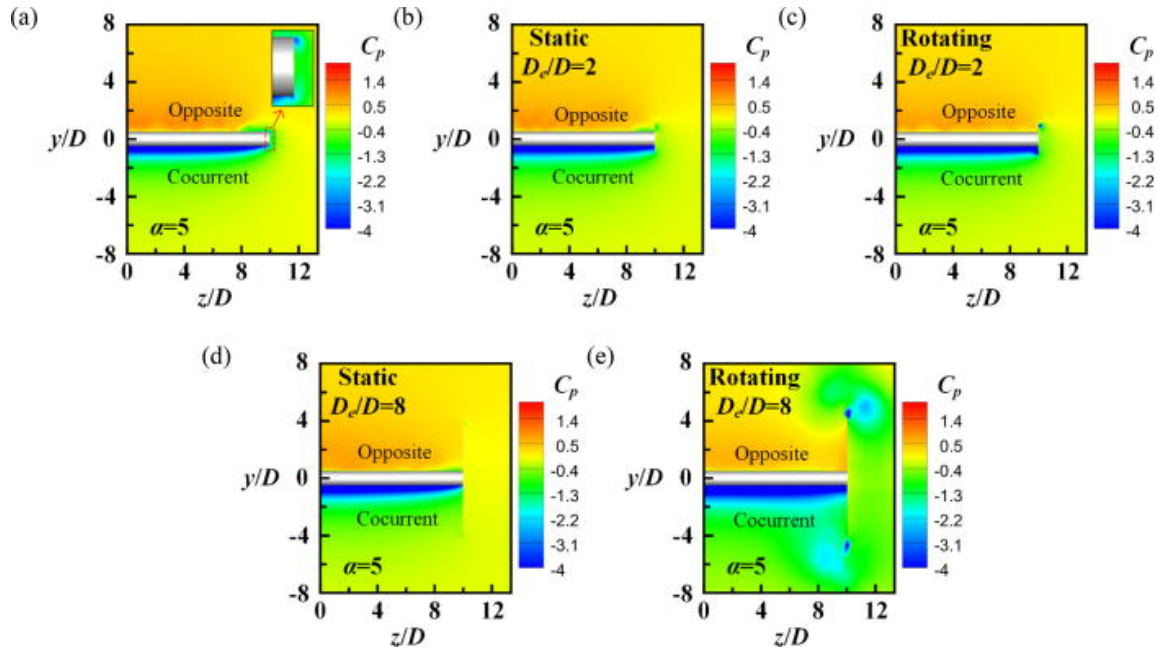


Figure 2.19: Pressure coefficient in the flow field at $\alpha = 5$: (a) cylinder rotor, (b) static end plate with $D_e/D = 2$, (c) rotating end plate with $D_e/D = 2$, (d) static end plate with $D_e/D = 8$, and (e) rotating end plate with $D_e/D = 8$. *Reproduced from Liu, Ma, Zheng, et al. (2024).*

In addition to force measurements, Badalamenti and Prince (2008) conducted wake measurements for varying end plate sizes and spin ratios. As shown in Figure 2.20, the results indicate that increasing end plate sizes strengthens the vortices, but postpone their formation to higher spin ratios and limit their inwards movement. Based on the combined analysis of force coefficients and wake pressure data, Badalamenti and Prince (2008), concludes that the addition of end plates lead to: (1) an improved pressure distribution around the cylinders end, and (2) a reduction of induced drag due to limitation in formation of tip vortices. For low spinning ratios ($\alpha \leq 1$), the first phenomenon is more dominant and larger plates tend to give a slightly larger drag. While at higher spinning ratios, the second becomes more influential. Larger plates leads to stronger vortices and delay their occurrence until much higher velocity ratios. This allows both lift and drag to increase for much higher spin ratios. Therefore, for moderately high velocity ratios ($0.7 \leq \alpha \leq 3.5$) a larger end plate is more desirable until the spin ratio becomes approximately $\alpha \approx 3.5$ where the drag behavior of the smaller plates sizes is drastically lower.

Bottom end plates are less applied in practice than top end plates. Understanding the impact of adding a bottom end plate is therefore essential. Deybach et al. (2024) conducted experimental research on end plates, testing the bottom plate and the top end plate independently. Their findings, shown in Figure 2.22, indicate that a rotating bottom plate has a positive effect on the force coefficients, while a stationary bottom plate has no effect relative to having no bottom end plate.

Badalamenti (2010) and Clayton (1985) did power measurements for different disk configurations. They concluded that power consumption increases with the size and number of disks. Badalamenti and Prince (2008) found especially a huge increase in power consumption using two end plates of size $D_e/D = 3$ in comparison to $D_e/D = 2$. The study concluded that the combination of very large end plate size and high aspect ratio could be used to produce exceptionally high lift, but assessments on the power suggests that this would be highly inefficient. The results are shown in Figure 2.21.

Considering that the formation of the vortices explains the working principle of the increase in lift-to-drag ratio due to the end plates, it is interesting to study the formation of these vortices and development in the wake. Unfortunately, experimental wake measurements are scarce to confirm the numerical results. Badalamenti performed wake measurements, but as his cylinder was mounted in the test section using steel struts, very close to the end plate, the influence of the struts on the topology is unknown. Thereby, as the wake deflected for higher spin ratios, they did not manage to accurately measure the wake pressure with the pitot tubes, which perform badly with inclined flows.

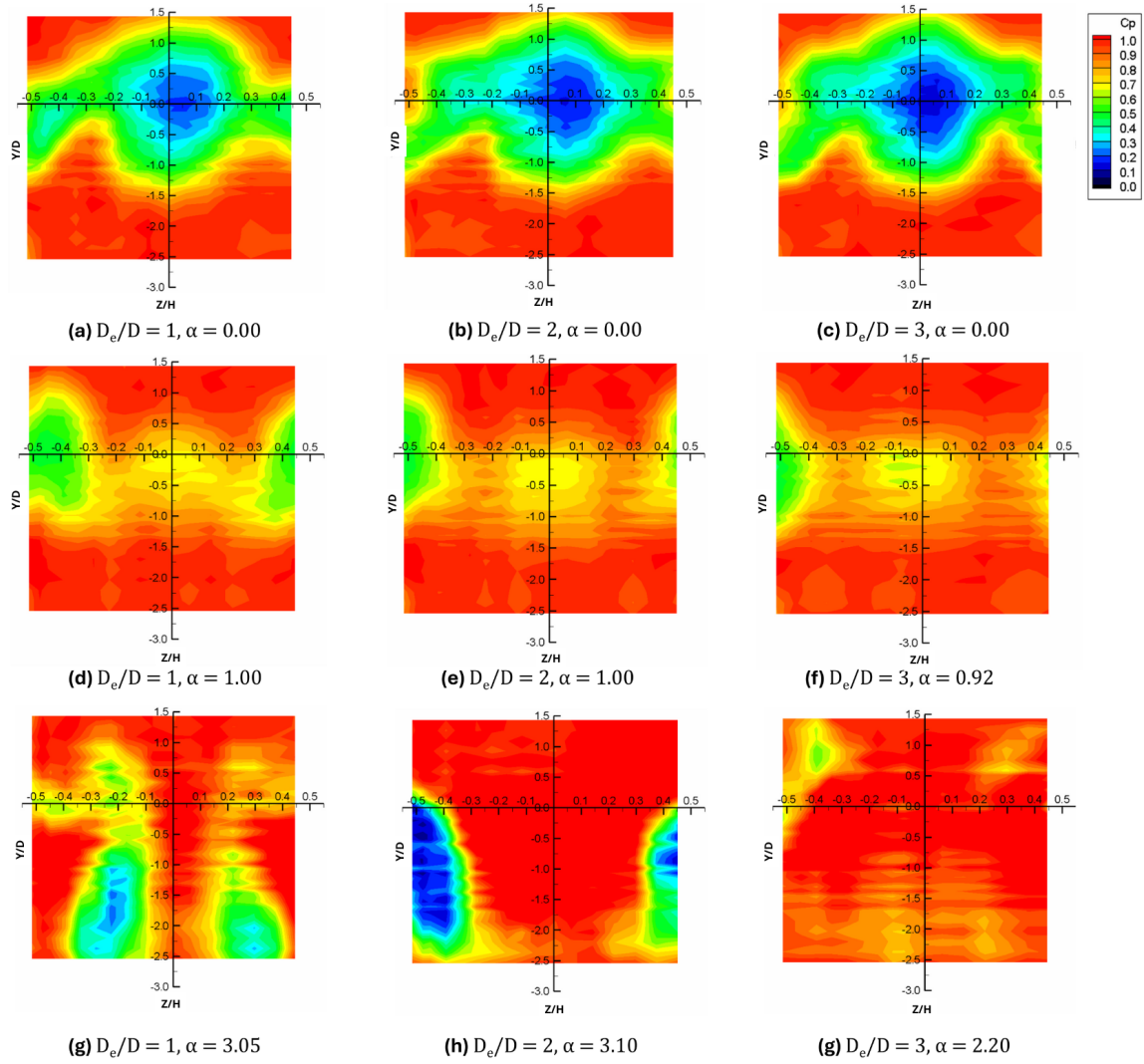


Figure 2.20: Spanwise total pressure variation using $C_p = \frac{p_t - p_{s,\infty}}{0.5\rho U_\infty^2}$, in the wake at $X/D = 5$, $Re = 7 \times 10^4$ and $AR = 5.1$. The cylinder is mounted in the horizontal direction, with both end plates situated at $Z/H = \pm 0.5$ and the wake deflection in negative y -direction. *Readapted from Badalamenti and Prince (2008)*

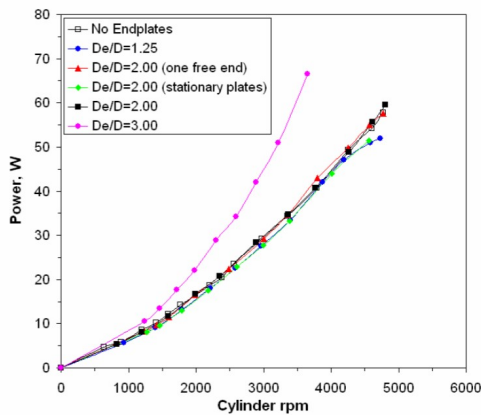


Figure 2.21: Power usage for a spinning cylinder with different end configuration for $Re = 4 \times 10^4$, $AR = 5.1$. *Reproduced from Badalamenti and Prince (2008)*.

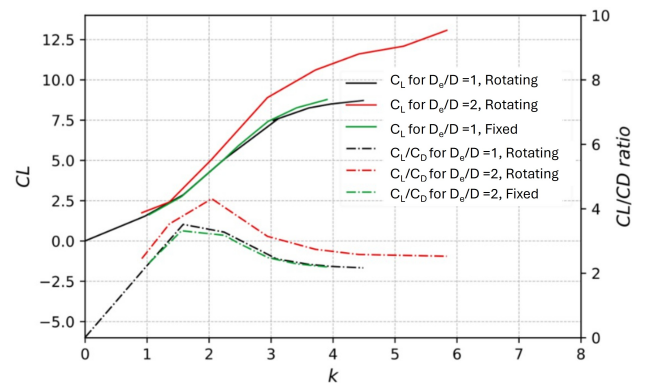


Figure 2.22: Influence of bottom end plate for rotating and fixed configuration. The model was equipped with top end plate ($D_e/D = 2$) and $Re = 2.5 \times 10^5$. *Reproduced from Deybach et al. (2024)*.

2.5. Research objective and research questions

Existing literature and experimental studies indicate that end plates can significantly enhance lift and reduce induced drag by influencing the formation of tip vortices. However, their use also increases the rotor's power consumption, and the optimal end plate size appears to depend on the spin ratio. The exact performance benefits of both top and bottom end plates remain unclear. Additionally, there is limited experimental research available on wake development around finite rotating cylinders with different end plate configurations. Experimental wake measurements around the rotor would also contribute to the fundamental understanding of flow development and the role of end plates. Therefore, detailed wake measurements are needed to verify and extend the findings of Badalamenti and Prince (2008). In conclusion, the reviewed literature reveals a clear knowledge gap regarding the exact performance benefits and fundamental understanding of flow development regarding end plate configuration on rotor sails.

Therefore, the research objective of this thesis is to understand the impact of end plate configuration on wake flow topology of a rotor sail by performing wind tunnel experiments on a scaled rotor sail model. Another part of this research is therefore to design and perform a mechanical characterization on an experimental set up that can be used for the campaign and future research on rotor sails.

Based on the research objective, the following research questions are defined:

How does the end plate configuration influence the wake flow characteristics of a finite rotating circular cylinder?

Based on this objective the following sub-questions are formulated:

1. *What is the influence of the end plate configuration on the wake deflection?*
2. *What is the influence of the end plate configuration on the vortical structures in the wake flow?*
3. *What is the influence of the end plate configuration on the vortex shedding of the wake flow?*

Chapter 3 addresses the design and operational lay out of the experimental setup and provides an overview on the measurement techniques. Furthermore, it gives an overview on the used coordinate system, describes how the data was processed and examines the limitations of the collected data. Chapter 4 presents the results for the aerodynamic characterization of the scaled rotor sail and includes a discussion with findings from existing literature. Chapter 5 summarizes the conclusions and answers the research questions. Chapter 6 gives an overview of recommendations for future research.

Methodology

In this chapter the methodology of the research is described. Firstly, the design of the experimental setup is described in section 3.1. Secondly, an explanation is given on the mechanical characterization of the scaled rotor sail in section 3.2. Following this, the coordinate system is defined and the measurement matrices are presented in section 3.3. Then, the measurement methodologies are given for the pressure rake wake surveys and the 7-hole probe wake velocity measurements in section 3.4 and section 3.5, respectively. Lastly, a characterization of the measurement errors is given in section 3.6.

3.1. Experimental setup

This section describes the setup used for the wind tunnel experiments. Subsection 3.1.1 provides the characteristics of the wind tunnel. Subsection 3.1.2 gives a detailed explanation on the design of the scaled rotor sail.

3.1.1. SLT wind tunnel TU Delft

To address the research questions, experiments were conducted on a scaled rotor sail (1:100). The experiments consisted of a two week campaign in the SLT wind tunnel at TU Delft. Figure 3.1 shows a schematic overview of the wind tunnel in open return configuration. The nozzle exit has a rectangular 900 mm to 600 mm cross-section. The wind tunnel has a maximum freestream velocity of 50 m/s. To make the results best comparable to real-life implementation, the experiments were conducted at the highest Reynolds numbers achievable considering the scaled rotor sail and SLT wind tunnel parameters.

The scaled rotor sail has a diameter of 50 mm and an aspect ratio of $AR = 5.2$, including the end plate thickness. Badalamenti and Prince (2008), Bordogna et al. (2020), Chen et al. (2023), and Thouault et al. (2012) used models with equal aspect ratios. This choice ensures that the experimental results are most comparable to literature. The possible configurations for the end plates include a rotating bottom and top end plates with diameters equal to $D_e/D = [1.0, 1.6, 3.0]$.

The maximum Reynolds number achievable in the experiments were determined by the model dimensions, model motor performance limits and compressibility effects. Based on the model diameter and a maximum rotational speed of 6000 RPM, the upper Reynolds number limit was estimated to be $Re_{\max} = 3.0 \times 10^4$, assuming a kinematic viscosity of $\nu = 1.48 \times 10^{-5} \text{ m}^2/\text{s}$ for air at 15°C. At this Reynolds number, the highest attainable spin ratio would be $\alpha = 1.5$ with a freestream velocity of 8.9 m/s. The minimum Reynolds number that could be tested was $Re = 1.0 \times 10^4$, which corresponds to a freestream velocity of 3.0 m/s and would allow for a maximum spin ratio of $\alpha = 5$. Table 3.2 compares the possible test range of this study to the experimental data summarized in Table 2.1. Since the test range of Reynolds numbers for the current setup falls completely within the subcritical regime (see subsection 2.3.1), the decision was made to fix the Reynolds number and focus on the end plate configurations, as was the focus of the research objective. The experiments were conducted at $Re = 1.5 \times 10^4$, enabling the testing of spin ratios up to $\alpha = 3$, which is representative of operational conditions for rotor vessels. The parameters of the scaled rotor sail and the wind tunnel used during the experiments are summarized in Figure 3.2.

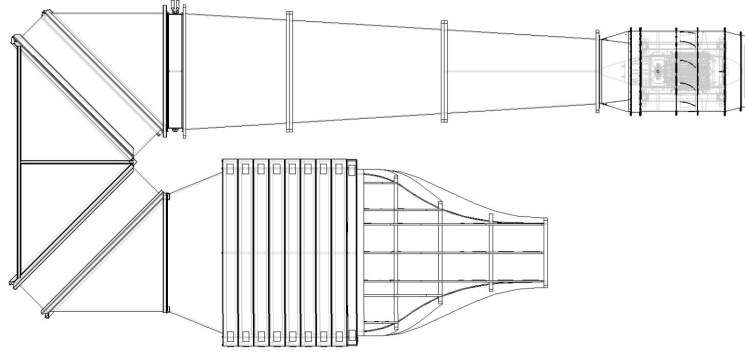


Figure 3.1: Schematic overview of SLT wind tunnel at TU Delft in open return configuration.

Table 3.1: Wind tunnel and model rotor parameters used during wind tunnel experiments in the SLT wind tunnel at TU Delft

Parameter	Value	Unit
U_∞	4.6	[m/s]
$RPM_{\max}$	5215	[RPM]
$\omega_{\max}$	546	[rad/s]
D	50	[mm]
AR	5.2	[-]
D_e/D	1.0, 1.6, 3.0	[-]
Re	1.5×10^4	[-]
α	0 – 3	[-]
Blockage ratio	2	[%]
$Ma_{\max}$	0.11	[-]

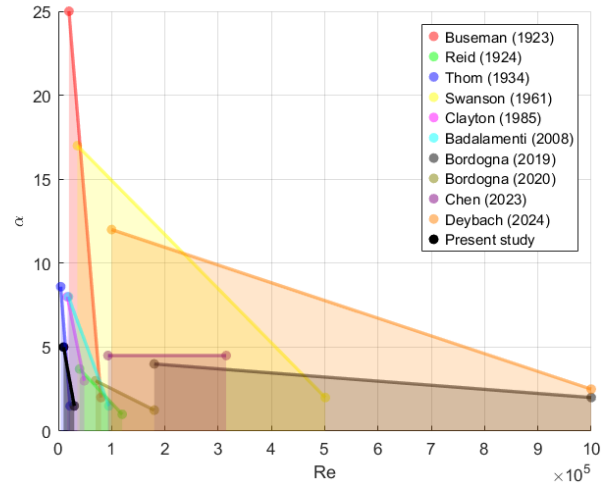


Figure 3.2: Domains of tested spin ratios and Reynolds number across experimental studies presented in Table 2.1 in comparison to the present study.

3.1.2. Rotor sail model

The scaled rotor sail was designed within a test section, which is specifically configured for use in the SLT wind tunnel. Figure 3.3a provides a CAD overview of the scaled rotor sail installed in the test section. A detailed view of the scaled rotor sail assembly is shown in Figure 3.3b, where the base plates are labeled with letters. A schematic layout of the shaft and bearing arrangement, with numbered components, is presented in Figure 3.5. The setup is divided into three main parts, which are described below with reference to the labels and numbers in Figure 3.3b and Figure 3.5.

1. **Rotor sail model:** The rotor sail model was originally designed in a setup for high-pressure measurements (Rius-Vidales et al., 2025). From this previous setup, the rotating cylinder (3), the upper end plate (1), the bottom end plate (5) and cylinder base block (A) have been reused. Additionally, the design of the bearing housing remains unchanged. The cylinder has a width of 50 mm and a height of 260 mm (including the end plates). The available end plates have a diameter of 50 mm, 80 mm and 150 mm. Figure 3.4 shows the three end plate configurations used during the experimental campaign. The end plates are 5 mm thick and their edges are defined using a modified superellipse profile. This makes the end plates more aerodynamic, avoiding flow separation on the edges. The end plates are mounted to the cylinder using a cap and two pins and an M8 high grade (12.9) steel bolt. The rotating cylinder is clamped to the inner rotating shaft (4). Underneath the lower end plate the shaft is covered by a static cylinder (6). The static cylinder is mounted to the cylinder base block (A) of the cylinder. A balancing ring (2) was added on top of the cylinder. In this balancing ring socket set screws can be added for correcting any imbalance. Figure 3.3c shows the design of the balancing ring.
2. **Bearing block:** The inner rotating shaft (4) is mounted to the stage shaft (7), which is supported by the bearing block (B). The bearing block consists of the lower bearing (10) (SKF 6205-2Z) and upper

bearing (8) (SKF 6206-2Z), a stage shaft sleeve (9), a lock nut (SKF KM 5) and washer (SKF MB 5) (11). The bearing block is supported by a supporting plate (D) which is attached to the test section using a base plate (C).

3. **Motor and coupler:** The Festo servo motor (13), type EMMT-AS-80-HKR-HS-RMB, drives the Mobac magnetic disk coupler (12). The coupler is connected to the motor and shaft by clamping and a feather key. A magnetic disk coupler was selected to prevent the transmission of mechanical vibrations from the motor, thereby reducing the impact of any misalignment. Technical specifications of the motor and coupler are provided in Appendix A.1 and Appendix A.2. The motor is mounted to the setup using the base plates (E, F). Plate E is connected to plate F using shock absorbers (Paulstra Mount 22000 - 530903 12) to minimize the transmission of vibrations of the motor.

The total weight of the entire assembly is estimated at 40 kg, excluding the test section. In section A.3 an overview is given of all the components and their mass. The design includes space for adding 6DOF loadcell for future research.

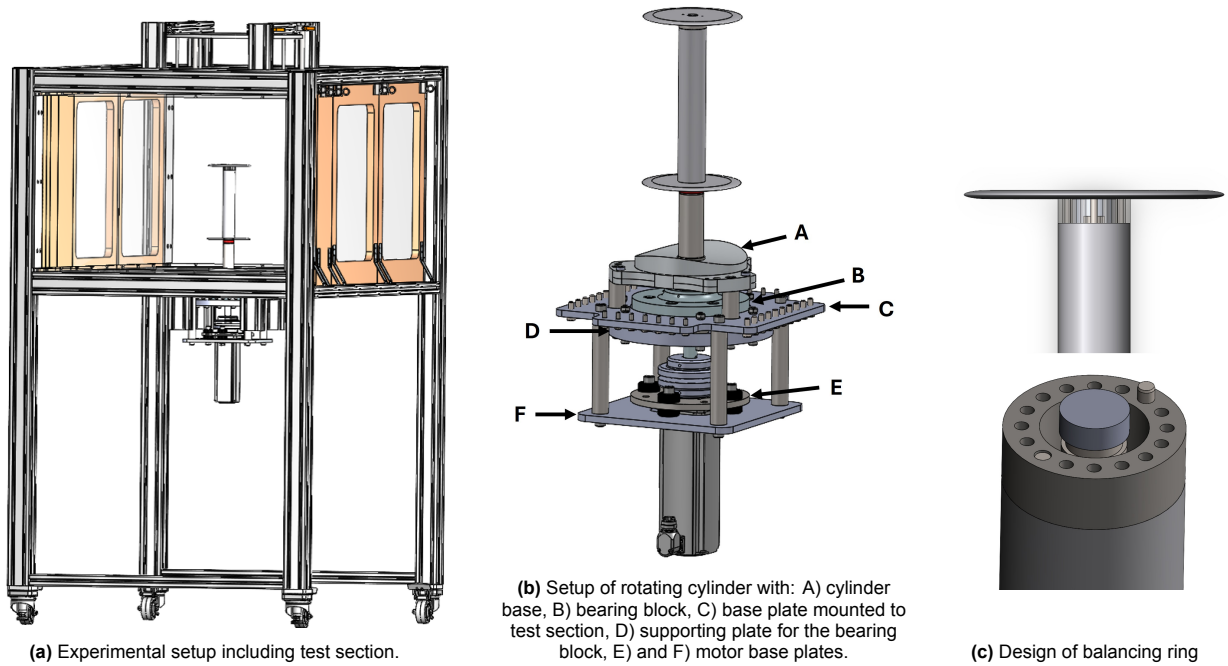


Figure 3.3: CAD design of experimental setup.

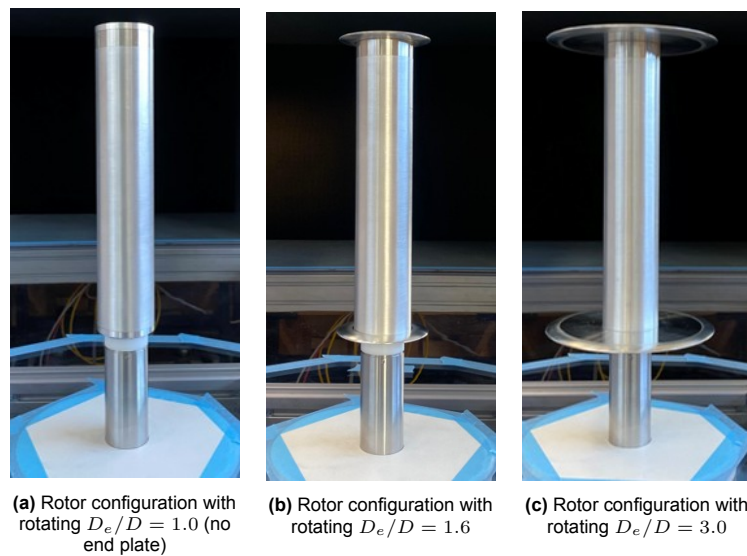
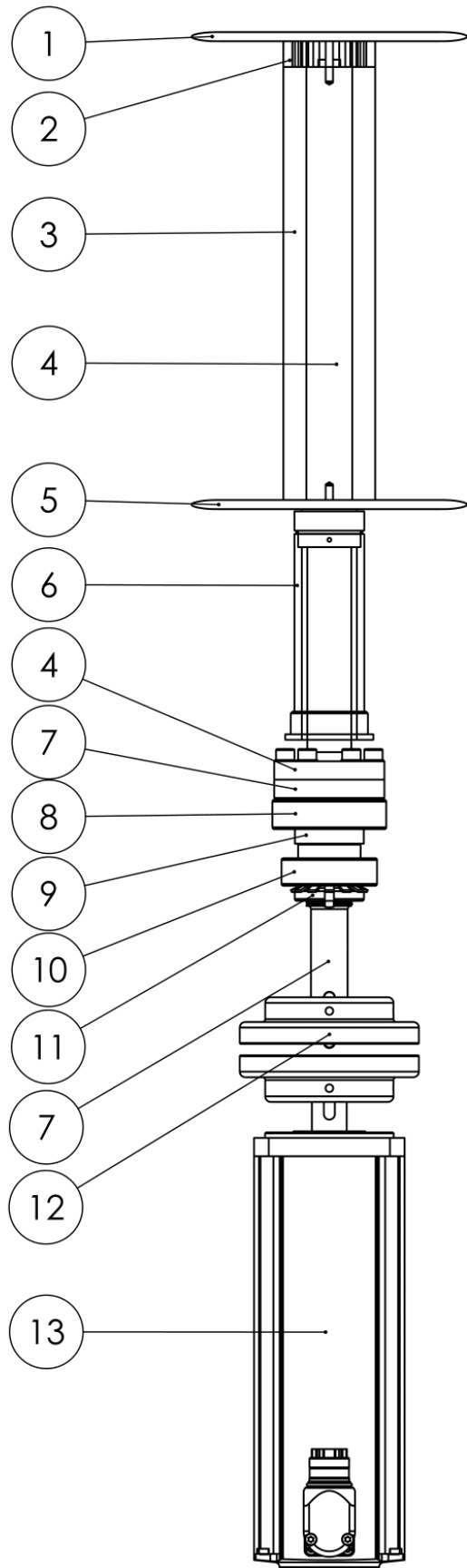


Figure 3.4: Rotor with three end plate configurations.

**Table 3.2:** Overview of parts

Number	Name
1	Upper end plate
2	Balancing ring
3	Outer rotating cylinder
4	Inner rotating shaft
5	Lower end plate
6	Outer static cylinder
7	Stage shaft
8	Upper bearing, SKF 6206-2Z
9	Stage shaft sleeve
10	Lower bearing, SKF 6205-2Z
11	Lock nut, SKF KM 5, and washer, SKF MB 5
12	Mobac magnetic coupler, MTD-5
13	FESTO servo motor, EMMT-AS-809-HKR-HS-RMB

Figure 3.5: Shaft and bearing layout with end plate size $D_e/D = 3.0$.

As the rotor will operate at a high rotational speed, it is of great importance to minimize vibrations. Next to the use of a magnetic disk coupler and shock absorbers to minimize vibrations, the rotor was dynamically balanced at an external company named Schenck. At Schenck, the unbalance can be determined while rotating your part in their machine as shown in Figure 3.6. This machine measures the forces and gives an imbalance rapport. The shaft was tested independently to determine whether any imbalance originated from the shaft itself or from the addition of the outer cylinder and end plates. The imbalance of the bare shaft was found to be minimal, requiring no further corrective measures. Subsequently, the cylinder was tested with each set of end plates ($D_e/D = [1.0, 1.6, 3.0]$). In all cases, the imbalance consistently occurred at the same angular position relative to the outer cylinder, indicating that the largest source of imbalance was the cylinder itself rather than the end plates. Accordingly, material was removed from one side of the balancing ring, and set screws were added on the opposite side. This reduced the imbalance sufficiently for safe use of the rotor at high rotational speed.

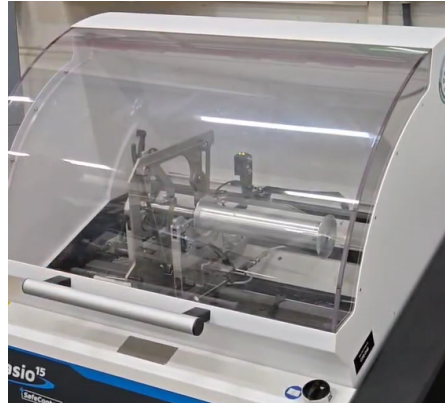


Figure 3.6: Measurement of the rotor's unbalance using Schenck balancing machine.

Additionally, it is essential to avoid operating the rotor at its eigenfrequency. To identify the system's eigenfrequency tests were conducted. The method and results of this characterization are discussed in section 3.2 as well as an explanation on the monitoring of vibrations during the experiments.

Figure 3.7 and Figure 3.8 show, respectively, the schematic layout and a photograph of the wind tunnel during operation. The measurement instruments were moved using an automated traversing system capable of moving the measurement probe in the x -, y - and z -direction with a position accuracy of $\pm 6.2 \mu\text{m}$, as shown in Figure 3.8. This allowed for accurate and automatic placement of the measurement instruments. The system had a reach of 600 mm in the spanwise and vertical directions, and 300 mm in the downstream direction.

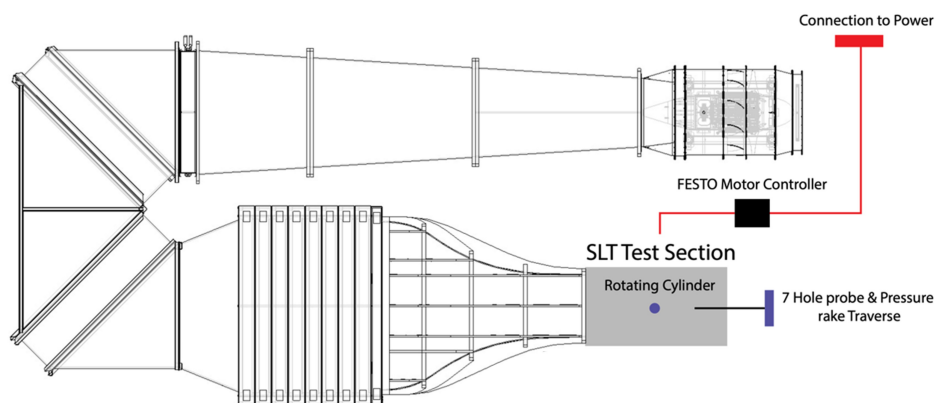


Figure 3.7: Operational layout of SLT wind tunnel.



Figure 3.8: Overview of wind tunnel with test section and traverse system, barriers and operators location.

3.2. Mechanical characterization of the scaled rotor sail

This section provides an overview on the mechanical characterization of the rotor sail model. Firstly, the measurement method is explained in subsection 3.2.1. Then, the results are presented in subsection 3.2.2.

3.2.1. Measurement method

A major challenge when doing experiments with rotating cylinders are the vibrations caused by the rotation, especially when using end plates. To omit resonance frequencies causing the heaviest vibrations, an analysis is performed on the eigen frequencies of the setup. The experiments are performed for the static rotor sail¹. During the measurements a shaker excited a sinusoidal excitation force with a gradually increasing frequency. This excitation force was measured by the force sensor, while the response of the system was captured using five accelerometers (PCB Piezotronics model 352C22), to compare the response at different points on the structure making it possible to detect mode shapes. The sensor and accelerometers were connected to the Dewesoft module, which enabled real-time data acquisition and signal processing through the analysis software. In Figure 3.9a the setup is shown, including the shaker on the right side, a coupling part between the sting and the cylinder to improve transmission of the excitation force and the accelerometers on the cylinder. Measurements were conducted for all three end plate configurations. Two accelerometers are placed on the top end plate, one on the bottom end plate and three on the rotors surface across the shaker, where the highest accelerations were expected. A schematic overview of the placements of the accelerometers is given in Figure 3.9b. One accelerometer was placed on the bearing block to evaluate whether vibrations observed on the cylinder could also be detected at the bearing location, as this was also one of the vibration monitoring points during the actual experiments.

A linear sinusoidal sweep was applied from 1 to 200 Hz over a duration of 120 seconds, with an output amplitude of 1.5 V. Using the analysis software the frequency response function (FRF) was computed. The FRF relates the output response (accelerometers) to the input excitation (force sensor) as a function of frequency (Rao & Yap, 1995). Mathematically, the FRF is defined as

$$H(f) = \frac{Y(f)}{X(f)} \quad (3.1)$$

where $H(f)$ is the FRF, $Y(f)$ is the Fourier transform of the system's output signal, $X(f)$ is the Fourier transform of the input excitation signal and f is the frequency in Hz.

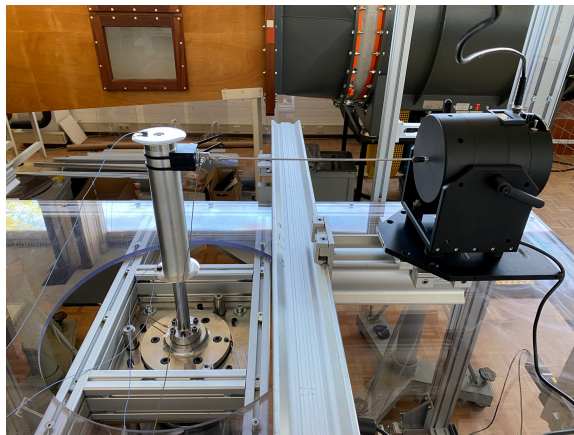
The FRF is represented in terms of amplitude and phase across the frequency spectrum. By locating peaks in the amplitude plot resonance frequencies can be identified, as the system responds most strongly to the input excitation, indicating a natural mode of vibration. The phase response also typically exhibits a rapid shift near resonance points, further confirming their location. The FRF was computed using a frequency resolution

¹Data acquisition was carried out using Dewesoft equipment.

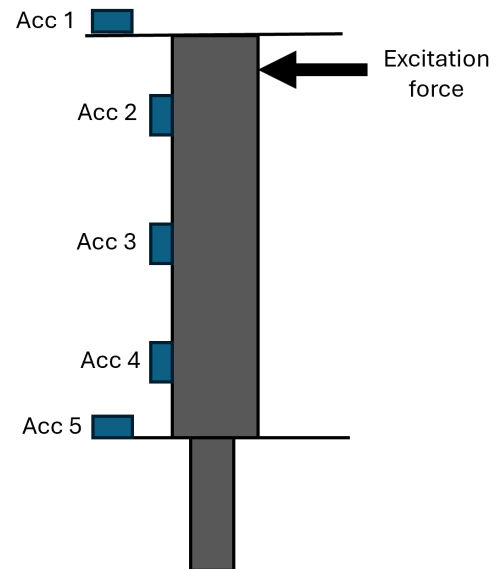
of 0.5 Hz, based on 20000 lines and a block duration of 2 seconds. A linear averaging method was used, with a Hanning window and 66.7% overlap between segments.

To evaluate the reliability of the measured FRF, the coherence function is used. Coherence values range from 0 to 1 and indicate how well the output signal corresponds to the input at each frequency. A value close to 1 suggests a strong linear relationship and good signal quality, while lower values may indicate noise, non-linear behavior, or insufficient excitation. The coherence function is computed using the auto- and cross-power spectral densities of the input and output signals, providing a frequency-dependent measure of the consistency between the two signals.

To extract resonance frequencies and damping coefficients from the measured FRFs, Dewesoft's modal analysis module uses a stabilization diagram. The diagram plots identified poles against increasing model order. Physically meaningful modes appear as stable poles that repeat across multiple model orders with similar frequency, damping, and mode shape. Unstable poles, which vary significantly with model order, are considered numerical artifacts or noise and are excluded. The remaining stable poles define the system's natural frequencies and damping ratios.



(a) Setup used for eigenfrequency measurements, including a Dewesoft shaker, accelerometers, a force sensor, and a coupling part between the sting and the rotor to improve transmission of the excitation force.



(b) Schematic overview of accelerometer placement: Accelerometer 1 is mounted on the top end plate; Accelerometers 2, 3 and 4 are positioned on the surface of the cylinder; Accelerometer 5 is attached to the bottom end plate; all located opposite to the direction of the excitation force.

Figure 3.9: Experimental setup and accelerometer placement for eigenfrequency measurements.

3.2.2. Results

Figure 3.10 shows the FRF results for accelerometer 2 and 4 for three end plate configurations a) $D_e/D = 1.0$ (no end plate), b) $D_e/D = 1.6$, and c) $D_e/D = 3.0$. Using the stabilization diagram the resonance frequencies are determined and shown in Table 3.3. As no resonance frequencies were found above 38 Hz, the FRF diagrams are shown up to 50 Hz.

In the range from 0 to 5 Hz, the phase difference between the two accelerometers is inconsistent, and the coherence is low. From approximately 5 Hz onward, coherence increases, indicating that the results become more reliable in this frequency range. A noticeable drop in coherence between 8–10 Hz may indicate a structural response of the test section at that frequency.

For all three configurations, three resonance frequencies were identified: one at a lower frequency and two closely spaced at higher frequencies. In the no end plate case, the first mode occurs at 23.76 Hz, which is notably higher than the first modes of the $D_e/D = 1.6$ and $D_e/D = 3.0$ configurations, both appearing around 17 Hz. Additionally, the damping ratios of the first modes for the $D_e/D = 1.6$ and $D_e/D = 3.0$ configurations are significantly higher than that of the $D_e/D = 1.0$ case.

The first resonance mode was identified around 1380–1620 RPM. While operating the rotor, it was found that it was still possible to spin the rotor within this range although light vibrations were observed. The second and third modes fall within a range of 1640–2280 RPM. During operation, it was avoided to spin the rotor within this range, as very strong vibrations were observed when ramping through toward higher rotational frequencies. It was especially important to avoid running the rotor near 2000 RPM and its harmonics. Above 2000 RPM, the setup performed very steadily. From these observations during rotor operation, it can be seen that the results from the eigenfrequency tests match quite well with the vibrations seen during actual operation. Since the eigen frequency tests were done for a static rotor, small differences could be expected during operation in dynamic conditions.

Table 3.3: Resonance frequencies (f_s), damping ratios (ζ), and corresponding RPM values for three configurations across the first three modes.

Configuration	Mode 1			Mode 2			Mode 3		
	f_s [Hz]	ζ [%]	RPM	f_s [Hz]	ζ [%]	RPM	f_s [Hz]	ζ [%]	RPM
$D_e/D = 1.0$	23.76	3.8	1426	36.79	2.2	2207	38.10	3.4	2286
$D_e/D = 1.6$	17.41	10.4	1045	33.17	3.9	1990	36.07	7.5	2164
$D_e/D = 3.0$	17.16	10.5	1030	27.28	6.4	1637	33.40	5.4	2004

Vibration monitoring

During the experiments, the rotor vibrations were monitored to assess the system's response while accelerating through different rotational frequencies and to detect any potential signs of malfunction or failure. Additionally, the measured vibrations of the system can be compared to the shedding frequencies to investigate possible coupling between structural and flow-induced vibrations.

The real-time vibrations were monitored by mounting accelerometers to the bearing block². From the real-time vibration measurements, a spectral analysis is conducted to determine how the vibration energy is distributed across different frequencies. This allows identification of the dominant frequency components in the system. The PSD was computed using a Hanning window with 66.7% overlap and 8192 spectral lines.

Figure 3.11 and Figure 3.12 show the power spectral density (PSD) of the accelerometer mounted on the bearing block during experimental tests at $D_e/D = 1.6$ and $D_e/D = 3.0$, respectively. The peaks in the PSD indicate the dominant frequencies present in the mechanical vibrations of the setup. Where applicable, rotational frequencies are indicated with a dashed line. These measurements correspond to test runs performed for the identification of the Strouhal number, the results of which will be discussed in section 4.3. Monitoring the vibrations in this way enables a direct comparison between the mechanical response of the system and the frequencies associated with vortex shedding. It can be seen that the power spectral density is dominated by the second and third vibration modes.

For $\alpha = 0$, the highest peaks are observed at 32.3 Hz and 32.9 Hz, which are close to the system's eigenfrequencies. For $D_e/D = 1.6$, secondary peaks are observed at 9.1 Hz and 12.8 Hz, while for $D_e/D = 3.0$, a secondary peak appears at 12.8 Hz.

For $\alpha = 0.5$ and $D_e/D = 3.0$, the rotational frequency and its harmonics are clearly visible, while the eigenfrequency peaks are not distinguishable.

For spin ratios $\alpha = 1$ and $\alpha = 3$, the dominant peaks correspond to the rotational frequencies of the cylinder. In the case of $\alpha = 1$, the second and third highest peaks are found at the second and third harmonics. Additional harmonic components are visible around 12 Hz and 16 Hz. For both $\alpha = 1$ and $\alpha = 3$, further peaks appear around 32 Hz and 35 Hz, again closely matching the system's eigenfrequencies.

For $\alpha = 2$ and $D_e/D = 3.0$, the dominant peak corresponds to the rotational frequency of the cylinder. A secondary peak is observed at 29 Hz, which is approximately half the rotational frequency. Peaks near 32 Hz and 35 Hz are also present, consistent with the system's eigenfrequencies.

²Data acquisition was carried out using Dewesoft equipment.

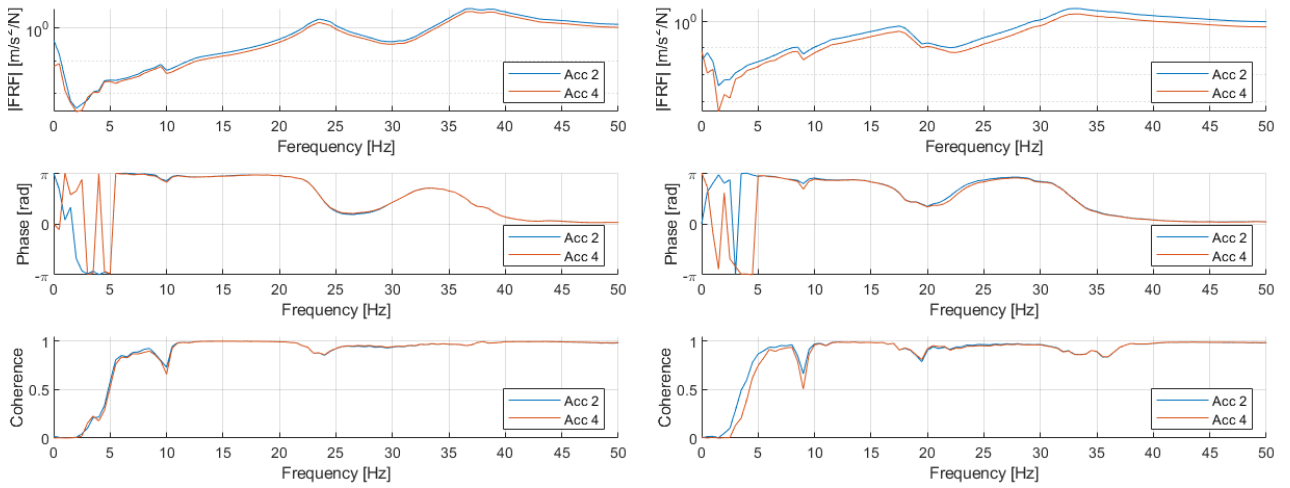
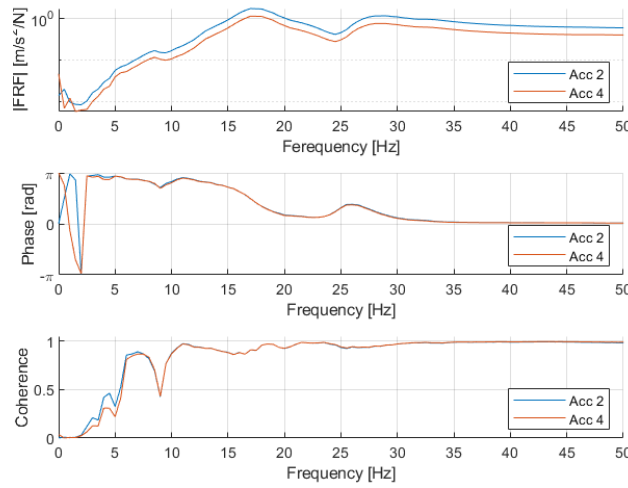
(a) $D_e/D = 1.0$ (no end plate)(b) $D_e/D = 1.6$ (c) $D_e/D = 3.0$

Figure 3.10: Amplitude, phase, and coherence plots of the frequency response function (FRF) for three endplate configurations: a) $D_e/D = 1.0$ (no end plate), b) $D_e/D = 1.6$, and c) $D_e/D = 3.0$, measured using accelerometer 2 and 4.

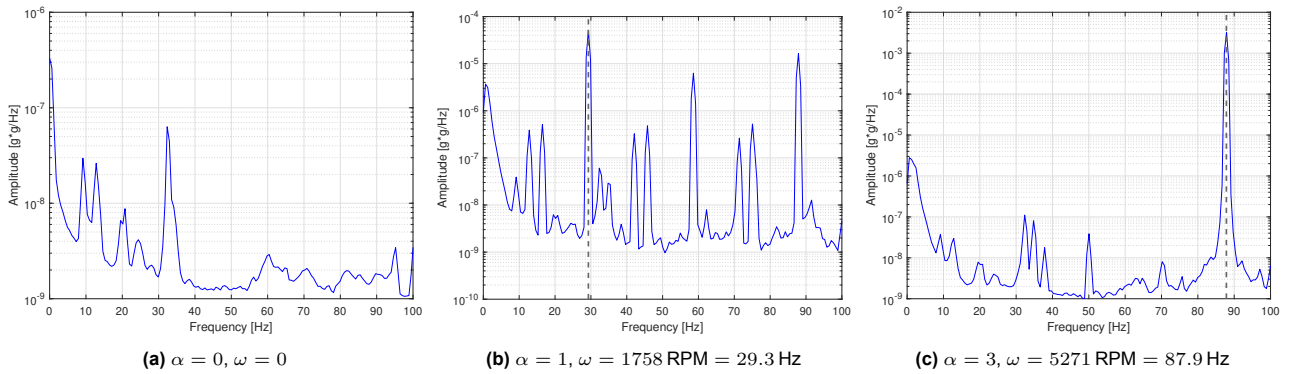


Figure 3.11: Power spectral density of accelerometer mounted to the bearing block during experimental tests for $D_e/D = 1.6$. The rotational frequencies are indicated with a dashed line, where applicable.

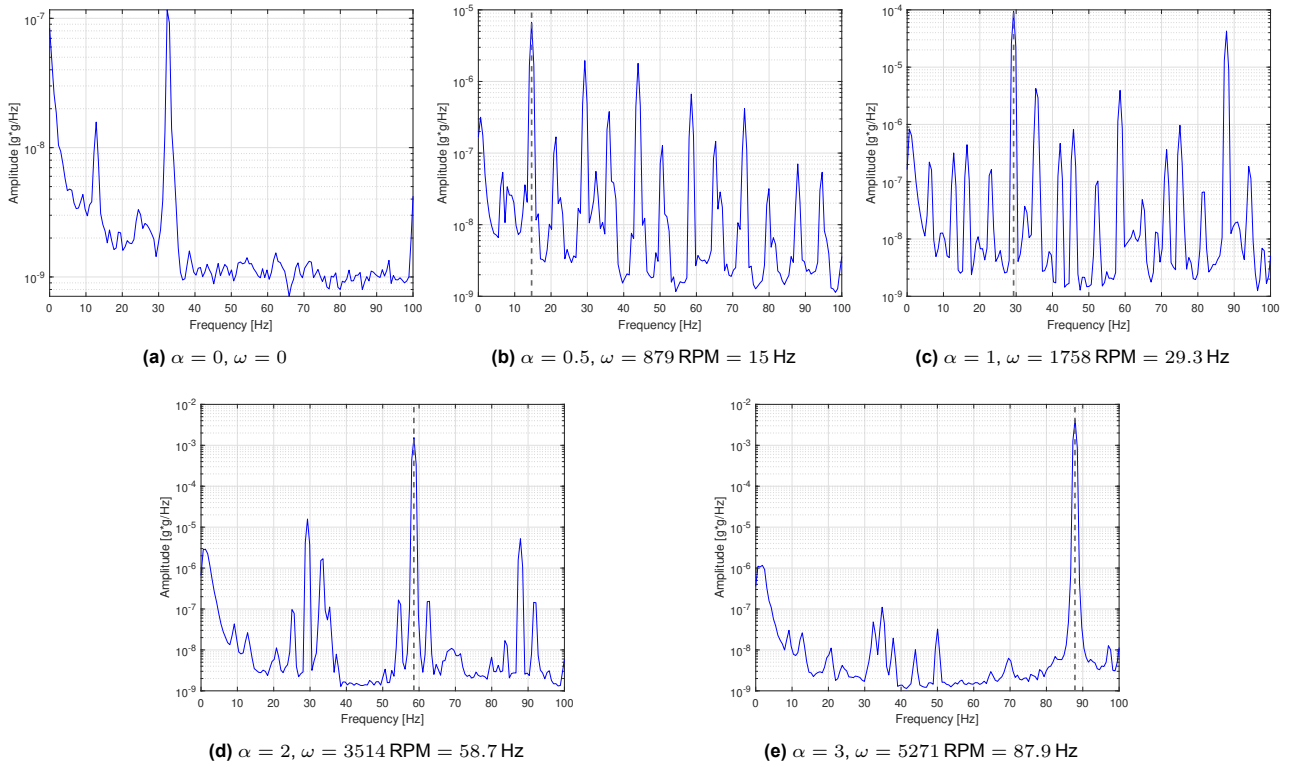


Figure 3.12: Power spectral density of accelerometer mounted to the bearing block during experimental tests for $D_e/D = 3.0$. The rotational frequencies are indicated with a dashed line, where applicable.

3.3. Measurement matrix and coordinate system

This section introduces the coordinate system used for the wind tunnel experiments on the scaled rotor sail. Figure 3.13 shows the rotor with the right-handed coordinate axes and the distances relative to the measurement device, whose center is indicated by a red dot. The x -axis is aligned with the streamwise direction, and the y -axis corresponds to the transverse direction of the measuring section in the wind tunnel. The $z = 0$ plane is defined at the rotor's mid-height, placing the origin $(0, 0, 0)$ at the geometric center of the rotor. A pitot tube measured the free stream pressures next to the rotor, as indicated by in Figure 3.13a. The pitot tube was placed at location $X/D = 1.5$, $Y/D = 7.1$, $Z/H = 0.37$.

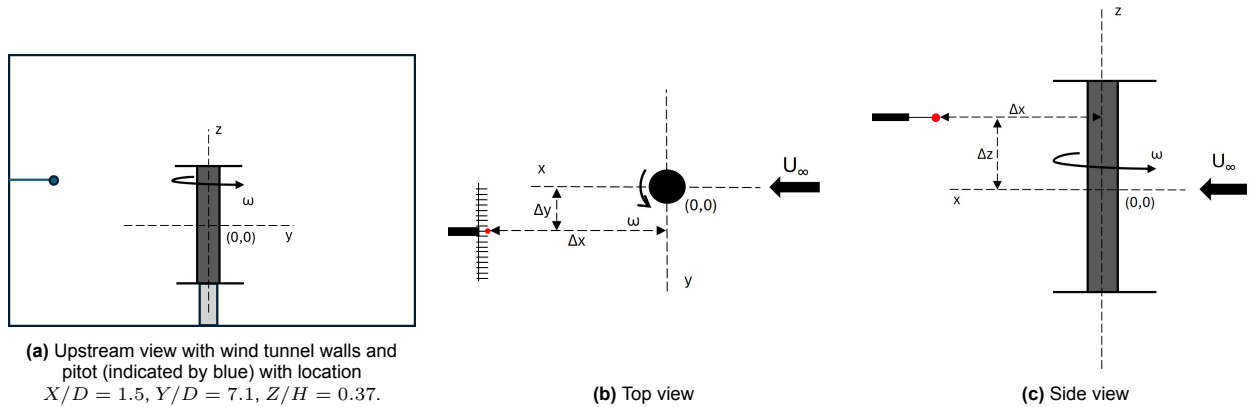


Figure 3.13: Overview of the coordinate system used during the wind tunnel experiments on the scaled rotor sail. The center of the measurement device is represented by a red dot.

Measurements were conducted using a pressure rake and a 7-hole probe. Table 3.4 and Table 3.5 show the experimental matrices of the pressure rake for each end plate configuration and spin ratio and the X/D and Z/H coordinates, respectively. Similarly, Table 3.6 and Table 3.7 show the experimental matrix of the 7-hole probe for each end plate configuration and spin ratio with the X/D coordinates. Figure 3.14 show a

three dimensional visualization of the measurement planes. An explanation on the purpose of the different measurement planes is provided in section 3.4 and section 3.5.

D_e/D	$\alpha = 0$		$\alpha = 1$		$\alpha = 3$	
0	3.5	5	3.5	5	3.5	5
1.6	3.5	5	3.5	5	3.5	5 9.5
3	3.5	5	3.5	5	3.5	5

Table 3.4: Measurement positions X/D of the pressure rake in the y, z -plane for each D_e/D and spin ratio α , with $Re = 1.5 \times 10^4$, acquisition frequency 2kHz and measurement time 5 seconds.

D_e/D	$\alpha = 0$	$\alpha = 1$	$\alpha = 3$
0			
1.6			2.2
3		2.2	2.2 3.7

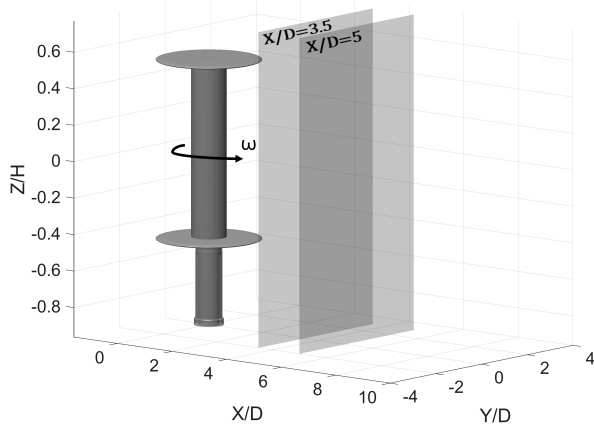
Table 3.6: Measurement positions X/D of the 7-hole probe in the xy -plane for each D_e/D and spin ratio α for both the stationary and moving probe method, with $Re = 1.5 \times 10^4$, acquisition frequency 1kHz. Two measurement methods were applied as discussed in section 3.5.

D_e/D	$\alpha = 0$			$\alpha = 1$			$\alpha = 3$		
0	0			0			0		
1.6	-0.5	0	0.5	-0.5	0	0.5	-0.5	0	0.5
3		0			0			0	

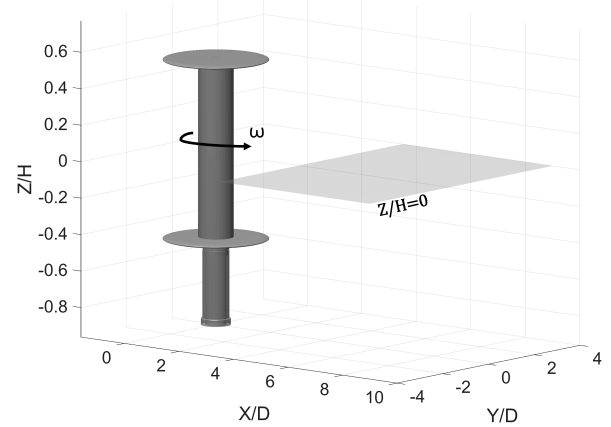
Table 3.5: Measurement positions Z/H of the pressure rake in the x, y -plane for each D_e/D and spin ratio α , with $Re = 1.5 \times 10^4$, acquisition frequency 2kHz and measurement time 5 seconds.

D_e/D	$\alpha = 0$	$\alpha = 0.5$	$\alpha = 1$	$\alpha = 2$	$\alpha = 3$
0					
1.6	1.8		1.8		1.8
3	1.8	1.8	1.8	1.8	1.8

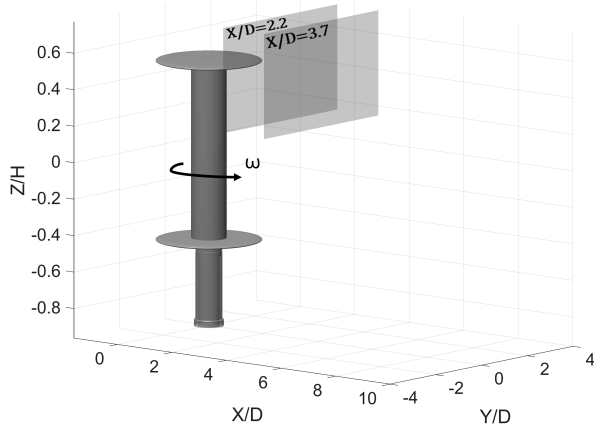
Table 3.7: Measurement positions X/D of the 7-hole probe along the y -axis for each D_e/D and spin ratio α , with $Re = 1.5 \times 10^4$, acquisition frequency 1kHz and measurement time 5 seconds.



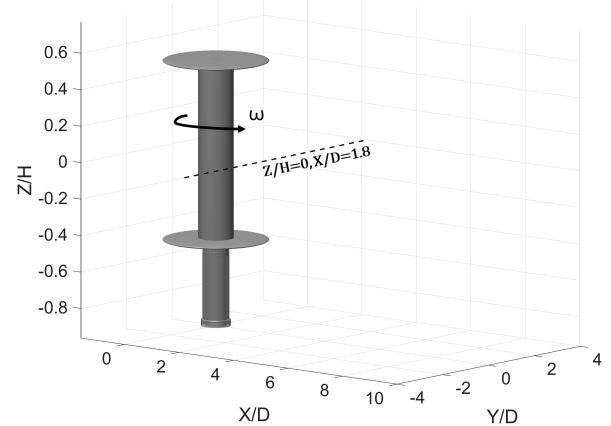
(a) Visualization of two measurement planes in the yz -plane for $X/D = [3.5, 5]$ for the pressure rake with the measurement matrix shown in Table 3.4.



(b) Visualization of one measurement plane in the xy -plane for $Z/H = 0$ for the pressure rake with the measurement matrix shown in Table 3.5.



(c) Visualization two measurement planes in the yz -plane for $X/D = [2.2, 3.7]$ for the 7-hole probe with the measurement matrix shown in Table 3.6.



(d) Visualization the measurement points along $Z/H = 0$ and $X/D = 1.8$ for the 7-hole probe with the measurement matrix shown in Table 3.7.

Figure 3.14: Visualizations of measurement planes for the rotor sail with $D_e/D = 3.0$.

3.4. Total pressure rake wake surveys

To characterize the deflection of the wake and the vortical development within the wake, pressure measurements were performed using a pressure rake. This section describes the measurement principle and, data acquisition, and subsequent data processing.

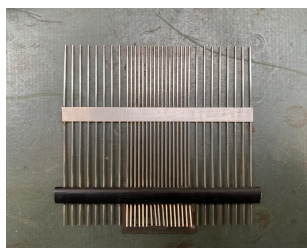
3.4.1. Measurement principle and data acquisition

The pressure rake setup consists of two separate rakes: one used to measure total pressure and the other for static pressure. Total pressure is measured at the tip of the tubes, while static pressure is measured through a hole on the side. Both rakes were mounted on a sting, which was attached to the traverse system. The static pressure rake had a 51 mm offset with regards to the total pressure rake in the positive z -direction. Figure 3.15 shows the layout of the pressure rake and the operational layout with the traverse systems. The pressure system is a custom made pressure scanner (NUB-System) with Honeywell HSC series differential pressure transducers. One system had a range of 2500 Pa with an accuracy of ± 25 Pa, and two other systems had a range of 10 inH₂O. All measurements were taken relative to the ambient pressure.

At each measurement point, the pressure rake was positioned using the traverse system. After arriving, a 2 second pre-trigger delay was applied to allow vibrations to settle. Then, a trigger was sent to start data acquisition, and pressures were recorded for 5 seconds. The measured pressures were averaged over this period. Following this, a 0.5-second post-trigger delay was used to reduce the influence of any movement before the rake moved to the next point. The acquisition frequency of the pressure rake was 2 kHz.

Two sets of measurements were performed. The first set of measurements was performed with the goal to get insights into the wake deflection and the second one to get insights into the vortical structures of the flow. The first set of measurements were performed in the xy -plane at the rotors mid-height ($Z/H = 0$) with reference to the total pressure rake. The schematic representation of the measurement plane is shown in Figure 3.14b. The full spanwise and streamwise range of the traverse system was used, with 5% overlap in the y -direction and 50 mm spacing in x -direction. These measurements were done for each end plate configuration: $D_e/D = [1.0, 1.6, 3.0]$ and for the spin ratios $\alpha = [0, 1, 3]$. For $D_e/D = 1.6$ measurements were also performed at $Z/H = -0.5$ and $Z/H = 0.5$ with respect to the total pressure rake. Table 3.5 presents the measurement matrix.

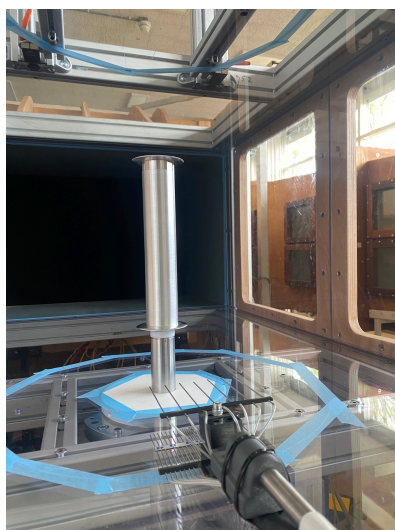
The second set of measurements was performed in the yz -plane at streamwise distances of the rotor $X/D = [3.5, 5, 9.5]$. The schematic representation of the measurement planes is shown in Figure 3.14a. In the spanwise y -direction, a distance of 200 mm was covered where the pressure rake was moved with a 5% overlap between measurement positions. The frame was moved with regards to the expected deflection of the wake. In the z -direction, measurements were taken at 2 mm intervals. This was done for all three end plate configurations and spin ratios $\alpha = [0, 1, 3]$. Table 3.5 presents the measurement matrix.



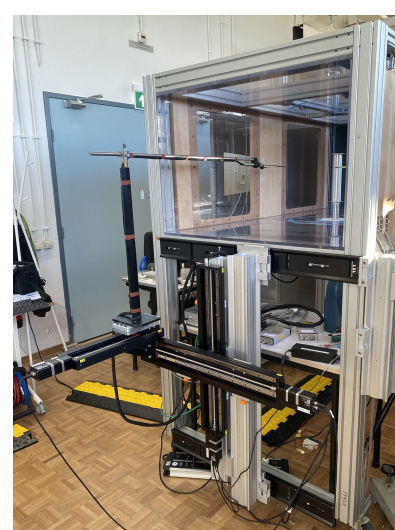
(a) Layout of total pressure tubes.



(b) Layout of static pressure tubes.



(c) Operational layout of the pressure rake



(d) Operational layout of traverse system

Figure 3.15: Pressure rake layout and traverse system

3.4.2. Data processing

This subsection describes the processing of the raw data obtained by the pressure rake, beginning with data filtering and followed by an explanation on the definition of the wake deflection angle.

Data filtering

The raw total pressure data shows spatial disturbances. This is especially clear when looking at freestream measurements in the streamwise direction, so a fixed z and x while varying y , where the total pressure should ideally remain constant. Instead, regular fluctuations are observed in space, suggesting disturbances that might be related to tube alignment. To reduce these effects, a spatial low-pass Butterworth bandpass filter is applied as a pre-processing technique to filter the data for subsequent analysis. An overview of the processing procedure is shown in Figure 3.17.

First, the data is interpolated onto an evenly spaced grid along the y -direction. To verify that the Butterworth filter properly removes unwanted spatial frequencies, a Fast Fourier Transform (FFT) is applied to the interpolated pressure data. This spectral analysis helps to identify the main spatial patterns and shows that high-frequency components are effectively filtered out. From an analysis of total pressure profiles across all test conditions, it is found that spatial structures with wavelengths shorter than $\lambda = 25$ mm are not physically relevant and can be suppressed. Figure 3.16 shows the pressures along the y -axis for this smallest physical relevant wavelength, with the filtered data conserving the pressure drop. Therefore, the cutoff frequency for the low-pass filter is set at this wavelength. The Butterworth filter requires cutoff frequencies to be specified in normalized form, relative to the Nyquist frequency. The Nyquist frequency represents the highest frequency component of a signal that can be accurately captured when sampled at rate f_s . In the spatial domain, the sampling and Nyquist frequencies are defined as:

$$f_s = \frac{1}{\Delta y}, \quad f_{\text{Nyquist}} = \frac{f_s}{2} \quad (3.2)$$

where Δy is the average spatial sampling interval.

The normalized cutoff frequency, $f_{\text{low,normalized}}$, corresponding to a desired minimum wavelength λ_{min} is given by:

$$f_{\text{low}} = \frac{1}{\lambda_{\text{min}}}, \quad f_{\text{low,normalized}} = \frac{f_{\text{low}}}{f_{\text{Nyquist}}}. \quad (3.3)$$

These normalized values are then used to define the passband of the Butterworth filter, allowing spatial features with wavelengths longer than λ_{min} to be preserved. A fourth-order Butterworth filter is used in this process as a balance between sharpness of cutoff and stability of the filtering. The data is filtered in both forward and reverse directions, eliminating phase distortion and preserving the correct spatial alignment of features. For all the time-averaged total pressure measurements using the pressure rake the described smoothing is adapted with the same filter frequency and settings.

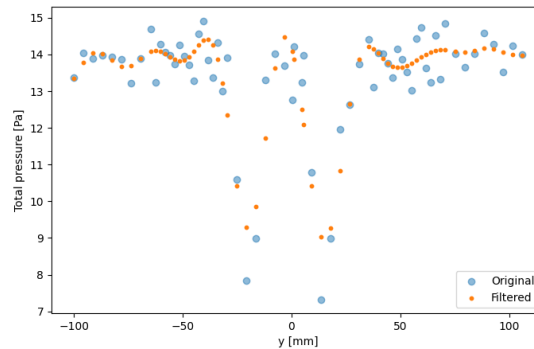


Figure 3.16: Comparison of original and Butterworth-filtered total pressure along the spanwise y -direction, using a minimum physical wavelength of $\lambda_{\text{min}} = 25$ mm. Blue markers $\bullet$ represent the original data, and orange markers $\bullet$ represent the filtered data. Data shown for the case $D_e/D = 1.6$, $\alpha = 0$, at $Z/H = 0.4$ and $X/D = 3.5$ with $Re = 1.5 \times 10^4$.

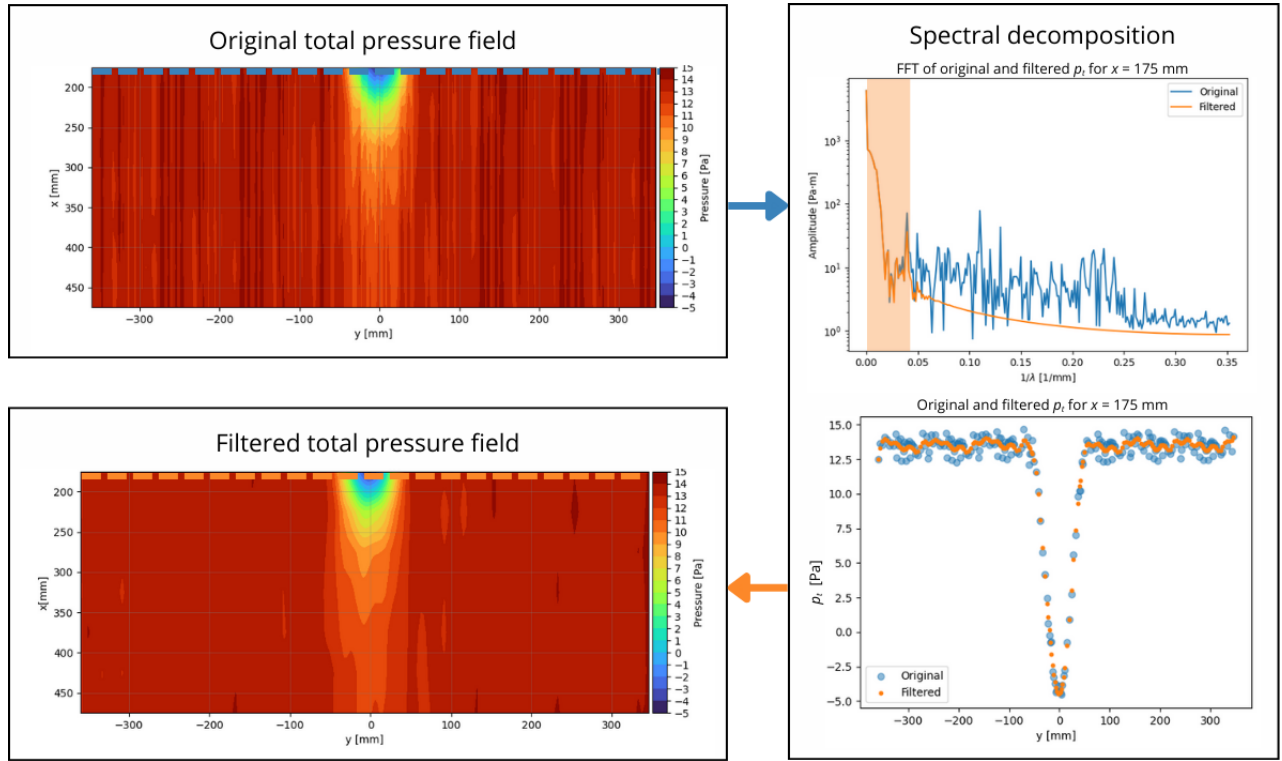


Figure 3.17: Schematic overview of spatial filtering of time-averaged total pressure using a low-pass filter with $\lambda_{min} = 25$ mm, showing the contour plot of the original and filtered total pressure field for the case $D_e/D = 1.6$, $\alpha = 0$, $Z/H = 0$ with $Re = 1.5 \times 10^4$.

Definition of wake deflection angle

To define the wake deflection angles, it is assumed that the lowest time-averaged total pressure occurs at the center of the wake. For each x -position, the y -coordinate corresponding to the minimum time-averaged (filtered) total pressure is identified. These points are then fitted with a straight line through the center of the cylinder ($x = 0$, $y = 0$). Using the slope of this fitted curve the deflection angle of the wake is determined. Although the wake deflection does not increase linearly, this result will give an indication of the overall wake deflection and allows for comparison across different spin ratios and end plate configuration. In Figure 3.18 an example of the determination of the deflection angle, β , is given for $D_e/D = 1.0$ and $\alpha = 3$.

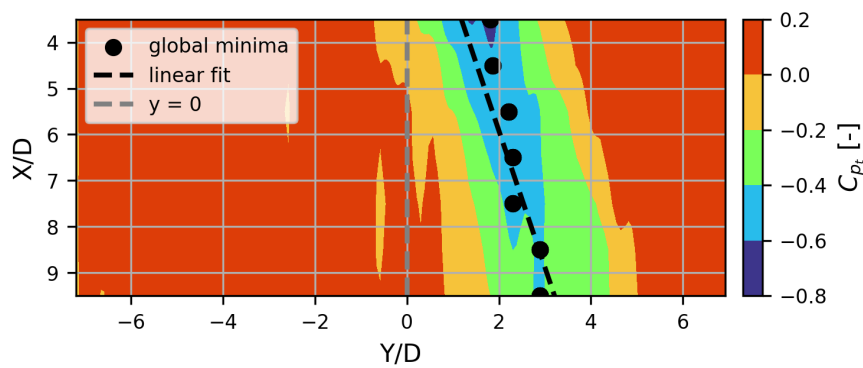


Figure 3.18: Contour plot of the time-averaged total pressure coefficient for $D_e/D = 1.0$ and $\alpha = 3$ with $Re = 1.5 \times 10^4$. The global minima are marked with $\bullet$. A linear fit is shown through these minima and the origin $(x, y) = (0, 0)$, illustrating the definition of the deflection angle β .

3.5. 7-hole probe wake velocity measurements

A 7-hole probe was used to measure the three-dimensional velocity components u , v and w in the wake of the rotor. These velocity fields make the identification of vortical structures possible using the Q-criterion and vorticity. Additionally, the direct digital measurement of the probe makes spectral analysis of the flow possible. This section firstly describes the measurement principle and secondly gives an overview on the data processing steps.

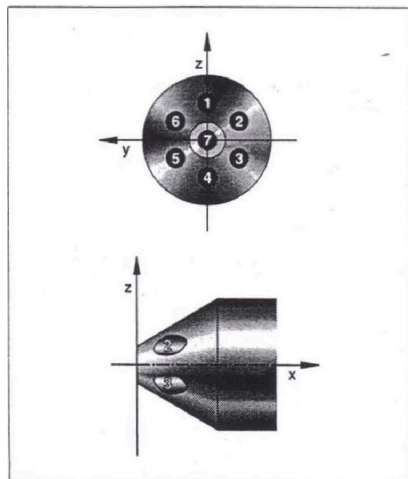
3.5.1. Measurement principle and data acquisition

As the name suggests, the probe measures the flow using seven pressure ports, as shown in Figure 3.19a. The recorded pressures were converted into velocity components using a calibration method. Like the pressure rake, the 7-hole probe was mounted on a sting, which was attached to the traverse system.

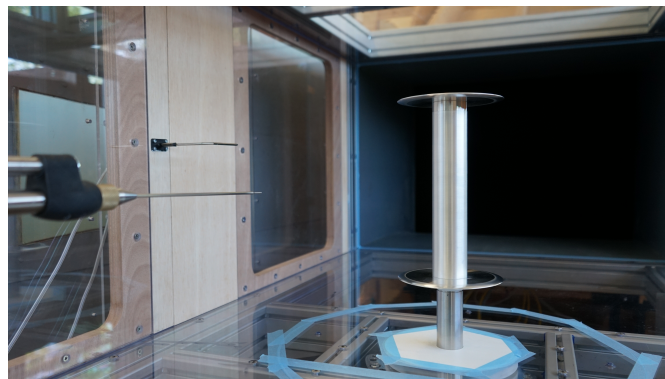
Three measurement methods were used with the 7-hole probe. The first two methods were used to identify vortical structures at the top end plate and measurements were taken in the yz -plane, as shown in Figure 3.14c. In y -direction approximately 125 mm was covered and in z -direction approximately 50 mm, covering the top part of the cylinder. The measurement plane was moved in the y -direction where the vortices were expected. The measurements were carried out for the end plate configuration $D_e/D = 1.6$ at $\alpha = 3$ at $X/D = 2.2$ and $D_e/D = 3.0$ for $\alpha = 3$ at $X/D = [2.2, 3.7]$, as well as $\alpha = 1$ at $X/D = 2.2$. Table 3.6 presents the measurement matrix.

In the first method, referred to as the moving method, the probe moved continuously at a speed of 7.5 mm/s in y -direction, while pressure data were acquired at 1 kHz. In z -direction a spacing of 3 mm was used. This results in a fine measurement grid. In the second method, referred to as the stationary method, the probe moved to fixed points with 10 mm spacing, where it remained stationary for 2.5 seconds to record pressures. The first 0.5 seconds of each measurement were discarded to minimize fluctuations caused by probe movement. This stationary method produces a coarser spatial resolution, but it eliminates disturbances from probe motion. The stationary data is used to correct or validate the continuously acquired measurements, as explained in section subsection 3.6.3.

The third method was used to identify vortex shedding frequencies. In this approach, the probe traversed along the y -direction from $-4.2 \leq Y/D \leq 4.2$, with a spacing of 5 mm between points, at a fixed streamwise position $X/D = 1.8$ and at the mid-height of the rotor, $Z/H = 0$. At each measurement point, pressure data was recorded for a duration of 5 seconds. The measurements were performed for $D_e/D = 1.6$ at $\alpha = [0, 1, 3]$ and $D_e/D = 3.0$ at $\alpha = [0, 0.5, 1, 2, 3]$. Table 3.7 presents the measurement matrix.



(a) Schematic layout of pressure ports of a 7-hole probe. Reproduced from Gameiro da Silva and Viegas (1994).



(b) Operational layout of the 7-hole probe.

Figure 3.19: Layout of the 7-hole probe.

3.5.2. Data processing

The output of the 7-hole probe is a file with raw pressure data for each of the 7 holes. This raw data is converted to a total velocity and the three velocity components using a calibration process. A block diagram of this process is shown in Figure 3.20. Firstly, it has to be determined if the 7-hole probe has been used in the right position or if a roll correction needs to be applied on the calibration file. This can be determined from the acceleration data in z -direction.

During the calibration process, p_i is converted into a nondimensional coefficient c_i obtained from the pressure at each hole i as

$$c_i = \frac{p_{max} - p_i}{p_{max} - p_{min}} \quad (3.4)$$

where p_i is the differential pressure recorded at hole i , and p_{max} and p_{min} are the maximum and minimum differential pressures recorded over all holes, respectively. In the calibration file, the pitch and yaw angles, ψ and θ respectively, and a stagnation differential pressure coefficient c_0 are given. The calibration file is probe specific and is therefore delivered with the probe. An interpolation process over a structured grid is used to return the closest matching flow angularity from the calibration file to the measurement data. With the matching flow angles then known, C_0 can be interpolated from the calibration data at those angles, returning the local stagnation pressure. With this local stagnation pressure the velocity magnitude can then be obtained from the stagnation pressure (Surrey Sensors Ltd., 2024). Lastly, the velocity components can be determined using the yaw and pitch angle in the reference coordinate system where u , v and w are the velocity components in x -, y - and z -direction, respectively.

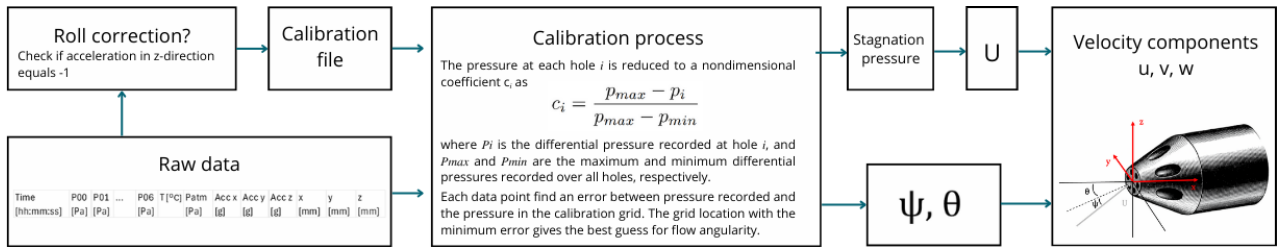
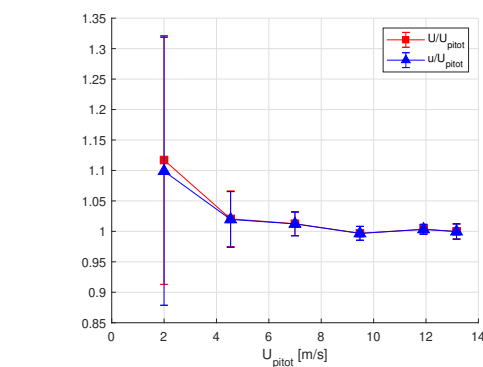


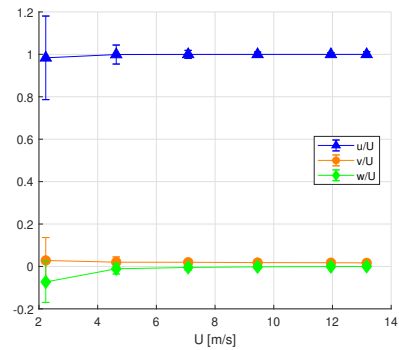
Figure 3.20: Block diagram of the 7-hole probe data processing. The 7-hole probe figure is readapted from Gameiro da Silva and Viegas (1994).

One verification measurement has been performed, to determine whether the 7-hole probe data is measured and converted correctly within the calibration process. During this verification measurement, the 7-hole probe was placed next to the pitot tube. Figure 3.13a shows a schematic overview of the placement of the pitot tube. Figure 3.21a compares the measured velocities from the 7-hole probe to the measured velocities of the pitot tube. The standard deviation of the measured velocities of the 7-hole probe are larger for lower freestream velocities. For higher freestream velocities, the average measured velocities converge and the standard deviation becomes smaller.

In the freestream the flow should be mainly in the streamwise direction. This confirmed by Figure 3.21b, where it is clear that velocity components v/U and w/U are very close to zero and u/U is equal to one. This means that the calibration works and that the probe has been mounted straight, and no further correction is necessary.



(a) Comparison of 7-hole probe velocities to pitot tube velocity with: $\blacksquare U/U_{pitot}$, $\blacktriangle u/U_{pitot}$, with the error bars showing the standard deviation of the 7-hole probe measurements.



(b) Comparison of velocity components from 7-hole probe to the total velocity from 7-hole probe: $\blacktriangle u/U$, $\bullet v/U$, $\blacklozenge w/U$ with the error bars showing the standard deviation of the velocity components of the 7-hole probe measurements.

Figure 3.21: Verification measurement of 7-hole probe for increasing freestream velocities.

Data processing for vortex identification

For the moving method, the data processing steps are shown in the block diagram in Figure 3.22. First, the measurement plane edges are cropped to remove artifacts due to the traverse movement. These occur as the probe moves upward to shift to a new z -position, following a zig-zag measurement pattern.

To obtain a time-averaged measurement, a spatial moving average was applied. A spatial average window is defined with a width of 5 mm and a height of 6 mm, covering two measurement rows that are time-uncorrelated. The blocks overlap by 50% in both the y - and z -direction, with each block including approximately 1000 measurements. Applying a moving average to the data produces a smoother velocity contour plot, as shown in Figure 3.20.

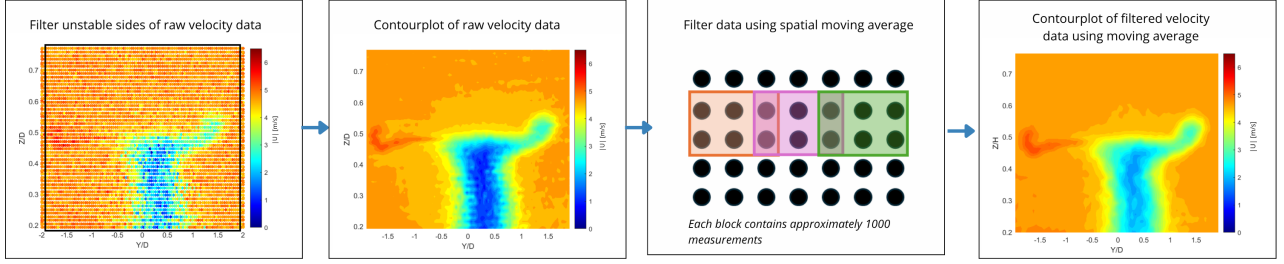


Figure 3.22: Block diagram of post processing velocity data of 7-hole probe.

To identify vortical structures, the Q -criterion is determined with the filtered data for the three velocity components, using

$$Q = \frac{1}{2} (\|\Omega\|^2 - \|\mathbf{S}\|^2), \quad (3.5)$$

where

$$\Omega = \frac{1}{2} [\nabla \mathbf{U} - (\nabla \mathbf{U})^T], \quad (3.6)$$

describes the rate of rotation for the pure rotational motion, and

$$\mathbf{S} = \frac{1}{2} [\nabla \mathbf{U} + (\nabla \mathbf{U})^T], \quad (3.7)$$

describes the rate of strain for the pure irrotational motion. So the Q -criterion shows whether the flow is rotation dominant ($Q > 0$) or shear dominant ($Q < 0$). For a 2D measurement in the yz -plane, the gradient tensor, can be simplified to (Chen et al., 2015)

$$Q = \frac{\partial v}{\partial y} \frac{\partial w}{\partial z} - \frac{\partial v}{\partial z} \frac{\partial w}{\partial y} - \frac{1}{2} \left(\frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)^2. \quad (3.8)$$

Additionally, the vorticity is plotted, to show the rotation direction of the vortices. The vorticity describes the local rotation of the fluid and is defined as the curl of the velocity field. For a yz -plane, the relevant component of the vorticity vector is in the x -direction and is computed as

$$\omega_x = \frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}. \quad (3.9)$$

For the current right-handed coordinate system, positive values of ω_x indicate counter-clockwise rotation, while negative values indicate clockwise rotation.

Data processing for analysis on shedding frequencies

To identify the vortex shedding frequencies in the flow, the total velocity data from the 7-hole probe was converted from the time domain to the frequency domain. It was found that the data was not sampled at consistent intervals matching the intended sampling rate. Consequently, classical Fourier-based methods for calculating the Power Spectral Density (PSD), such as Welch's method, could not be used because they require uniformly sampled data. The Lomb–Scargle method was therefore applied, as it is specifically intended for PSD estimation from unevenly sampled data (Van der Plas, 2018).

In this approach, a sinusoidal signal is fitted to the measured data for a range of trial frequencies, and the quality of each fit is quantified. By adjusting both the frequency and phase of the sinusoid, the method identifies the frequencies that best describe the measured variations, without requiring the data to be resampled onto a uniform time grid (Van der Plas, 2018).

The resulting power spectral density (PSD) was averaged into 200 frequency bins from 0 to 100 Hz, resulting in a resolution of 0.5 Hz per bin. This binning smooths out small variations and reduces the number of narrow spectral peaks, making the dominant frequency content easier to identify and interpret.

3.6. Measurement errors

This section discusses the limitations of the collected data and the influence of the measurement errors on the results. Firstly, the wind tunnel blockage effects are discussed. Thereafter, an uncertainty assessment is performed for both the pressure rake measurements and the 7-hole probe measurements.

3.6.1. Wind tunnel blockage

This chapter analyzes the potential blockage effects from the model on the pressure measurements. During the wind tunnel tests the pitot tube measured the static and total pressure of the freestream. In Figure 3.13a and Figure 3.24, the layout of the wind tunnel with the rotor and the pitot tube is shown. In the wind tunnel, the pressure is measured before and after the contraction. Through a calibration performed in the absence of the model, the dynamic pressure and velocity in the test section are determined from the difference between these pressures.

Due to the physical obstruction of the flow by the presence of the rotor, the flow in the wind tunnel is affected by solid blockage (Barlow et al., 1999). The blockage effects can be observed comparing the dynamic pressure of the pitot tube in the test section to the reference dynamic pressure of the wind tunnel. As the Reynolds number is kept constant during all tests, the dynamic pressures between different spin ratios and end plate configurations can be compared to each other. In Figure 3.23a the relative difference between the dynamic pressures of the pitot tube and the reference of the wind tunnel is plotted for each end plate configuration and spin ratio using

$$\Delta q = \frac{q_{\text{pitot}} - q_{\text{ref}}}{q_{\text{ref}}} \cdot 100\% \quad (3.10)$$

As the solid blockage ratio is only 3% this effect is relatively small and there is only a small offset between the dynamic pressure pitot tube and the reference of the wind tunnel for $\alpha = 0$ and $\alpha = 1$. It can be observed that when $\alpha = 3$ the dynamic pressure measured by the pitot tube increases in comparison the reference pressure of the wind tunnel.

Figure 3.23b and Figure 3.23c show the static and total pressure measured by the pitot tube, normalized by the dynamic pressure. It can be seen that the increase in dynamic pressure is caused by a decrease in static pressure and a slight increase in total pressure. This can be explained by the larger deflection of the wake due to increase in spin ratio. Figure 3.24 shows a colormap of the static pressure measurement in Pa for $D_e/D = 3.0$, $\alpha = 3$ at $Z/H = 0.2$ using the pressure rake. It can be seen that the deflection of the wake obstructs the freestream of the flow and thus leads to a larger blockage effect, resulting in a difference in static pressure on both sides of the rotor.

For bluff bodies, like a rotor, blockage correction methods exist and depend on the aerodynamic characteristics of the model, the model size relative to the tunnel's cross-sectional area, the freestream Mach number, and the tunnel type. For a rectangular cross-sectional, subsonic, closed wind tunnel, blockage correction methods exists, the most classical correction method is from Maskell (1963) and was later extended by Hackett (1996). In comparison to bluff bodies, the blockage correction for rotating cylinders is more complex. One of these complexities is the wake deflection interaction with the wind tunnel walls, as seen in Figure 3.23, which depend mainly on the spin ratio. In addition, the transition of the wake from unsteady to steady behavior at higher spin ratios and Reynolds numbers complicates the selection of an appropriate correction method.

Badalamenti and Prince (2008) analyzed the use of two correction methods, one for streamlined method, based on Lock (1929) and Hackett's method for bluff bodies for lift and drag coefficients. The tests were conducted in two different wind tunnels. It was found that applying the conventional streamlined body correction worsened the agreement with reference data. Hackett's method over-corrected lift values but improved drag agreement at high velocity ratios and sufficiently large end plates ($D_e/D \geq 1.5$). Due to the inconsistencies

between the corrected results and the good agreement between the uncorrected results, it was decided in their study not to use any correction methods.

These results show that wind tunnel corrections for rotating cylinders remain a complex topic as the usage of existing correction methods have not been successful. Moreover, most correction efforts have focused on force coefficients rather than wake measurements, which are the focus of this study. Combined with the lack of literature on correction methods for rotating cylinders, it is chosen to not correct the pressure results. It is important to keep this in mind when interpreting the results, as the measured pressures and velocities may be affected. However, the physical structures in the flow are expected to remain visible and largely unchanged, which is the primary focus of this study.

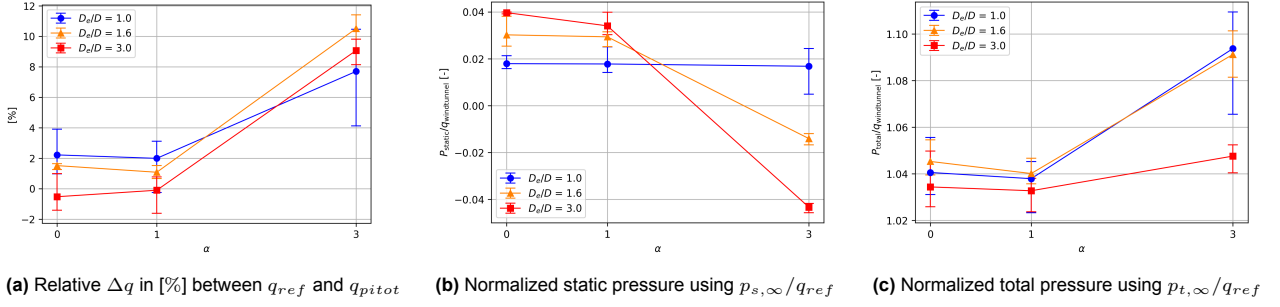


Figure 3.23: The mean dynamic, static, and total pressure measured by the pitot tube normalized by the dynamic pressure measured far upstream per spin ratio and end plate configurations: $\bullet$ $D_e/D = 1.0$ (no end plates), $\blacktriangle$ $D_e/D = 1.6$, and $\blacksquare$ $D_e/D = 3.0$. The mean is shown for three test runs for each end plate configuration and spin ratio, with error bars representing the minimum and maximum deviation from the mean.

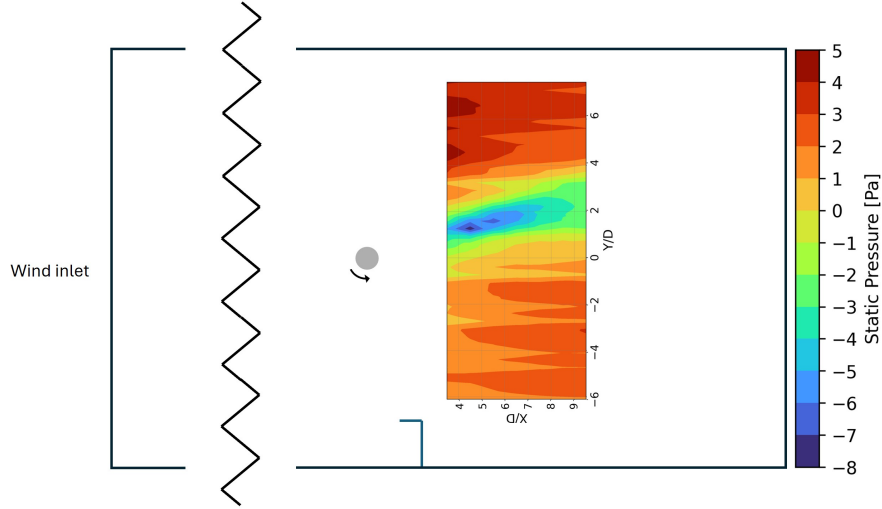


Figure 3.24: Scaled wind tunnel layout, with the pitot tube indicated by blue at height of $Z = 0.37H$, the rotor indicated in gray and the wind tunnel walls indicated in black. A contourmap of the time-averaged static pressure measurement in [Pa] for $D_e/D = 3.0$, $\alpha = 3$ at $Z/H = 0.2$ using the pressure rake is shown.

3.6.2. Uncertainty pressure rake measurements

For the measurements performed using the pressure rake, the standard error (SE) is calculated for the total and static pressure coefficient, $C_{p,t}$ and $C_{p,s}$. The SE quantifies the uncertainty in the time-averaged pressure value using the sample standard deviation, σ , and using the number of effective independent samples, N_{eff} (Taylor, 2022):

$$SE = \frac{\sigma}{\sqrt{N_{eff}}}. \quad (3.11)$$

The standard error therefore indicates how accurately the mean pressure is determined. Within the shedding region, the number of effective samples can be estimated using the shedding frequency, f_s (see section 4.3),

and the measurement duration, $t_m = 5$ s:

$$N_{\text{eff}} = t_m \cdot f_s, \quad (3.12)$$

For the freestream pressure signal, p_∞ , the number of effective samples is calculated using the autocorrelation function (Bayley & Hammersley, 1946; Zięba, 2010):

$$N_{\text{eff}} = \frac{N}{1 + 2 \sum_{k=1}^M \rho_k} \quad (3.13)$$

where N is the total number of measurements, ρ_k is the autocorrelation at lag k , and M is the maximum lag, determined by the first zero-crossing of the autocorrelation function.

For the freestream region, the number of effective samples was found to be approximately 10^3 , which is much larger than the number of effective samples within the unsteady wake. Furthermore, the standard deviation is significantly lower in the freestream than in the wake region, leading to the conclusion that $SE_{p_\infty} \ll SE_p$.

The standard error for the time-averaged total pressure coefficient C_{P_t} is calculated by propagating the standard errors in both the measured total pressure and the freestream total pressure, resulting in

$$SE_{C_{P_t}} = \frac{\sqrt{SE_{p_t}^2 + SE_{p_{t,\infty}}^2}}{0.5\rho U_\infty^2} \approx \frac{SE_{p_t}}{0.5\rho U_\infty^2}. \quad (3.14)$$

The standard error in the time-averaged static pressure coefficient C_{p_s} is calculated by propagating the standard errors in both the measured static pressure and the freestream static pressure, resulting in

$$SE_{C_{p_s}} = \frac{\sqrt{SE_{p_s}^2 + SE_{p_{s,\infty}}^2}}{0.5\rho U_\infty^2} \approx \frac{SE_{p_s}}{0.5\rho U_\infty^2}. \quad (3.15)$$

It is assumed that the standard error in ρ is negligible and therefore not included in the propagation of uncertainty.

The sample standard deviation, σ_p , is determined from the instantaneous pressure signals recorded for $D_e/D = 1.6$ at spin ratios $\alpha = 0$, $\alpha = 1$, and $\alpha = 3$. For the total pressure the measurements were taken at height $Z/H = 0$ and $Z/H = -0.5$ and for the static pressure the measurements were taken at $Z/H = 0.2$ and $Z/H = -0.3$ as the static pressure rake has an offset of 51 mm in the positive z -direction with respect to the total pressure rake.

Table 3.8 lists the highest observed standard deviation of the total pressure signal for each spin ratio at $Z/H = 0$ and $Z/H = -0.5$, respectively. The highest standard deviation is measured for $\alpha = 0$ at $Z/H = 0$, in the wake, close to the rotor. This is logical as this is a high turbulent region. Table 3.9 lists the highest values of the standard deviation for the static pressure, measured at $Z/H = 0.2$ and $Z/H = -0.3$. At $\alpha = 0$, no distinct shedding frequency was observed for $D_e/D = 1.6$ (see section 4.3). Therefore, the shedding frequency obtained for $D_e/D = 3.0$ was used as a reference in determining the SE.

α	$Z/H = 0$			$Z/H = -0.5$			f_s [Hz]
	σ_{p_t} [Pa]	p_t [Pa]	$X/D, Y/D$	σ_{p_t} [Pa]	p_t [Pa]	$X/D, Y/D$	
0	4.51	5.02	3.5, -0.56	3.75	3.23	3.5, -0.84	13.7
1	2.12	9.45	3.5, 0.008	2.88	8.68	3.5, -0.84	19.7
3	1.14	14.79	3.5, -5.02	3.62	2.71	3.5, 0.89	[-]

Table 3.8: σ_{p_t} values of the time-averaged total pressure, p_t , per spin ratio α with corresponding (X/D, Y/D) coordinates for $D_e/D = 1.6$, planes $Z/H = 0$ and $Z/H = -0.5$, and shedding frequency f_s .

α	$Z/H = 0.2$			$Z/H = -0.3$			f_s [Hz]
	σ_{p_s} [Pa]	p_s [Pa]	$X/D, Y/D$	σ_{p_s} [Pa]	p_s [Pa]	$X/D, Y/D$	
0	0.94	2.14	3.5, 0.56	1.87	-2.98	3.5, 0.84	13.7
1	1.46	-0.51	3.5, -0.14	1.19	-0.39	3.5, 0.54	19.7
3	1.08	1.72	5.5, -6.82	2.01	-8.20	3.5, -0.14	[-]

Table 3.9: σ_{p_s} values of the time-averaged static pressure, p_s , per spin ratio α with corresponding (X/D, Y/D) coordinates for $D_e/D = 1.6$, planes $Z/H = 0.2$ and $Z/H = -0.3$, and shedding frequency f_s .

For each coefficient, the largest standard error, is calculated. For $\alpha = 3$, the effective number of samples is determined using the autocorrelation function and found to be approximately 10^3 , resulting in a very low SE. For all coefficient calculations, a constant air density of $\rho = 1.208 \text{ kg/m}^3$ and freestream velocity of $U_\infty = 4.6 \text{ m/s}$ were used. The results of the highest standard error, representing the worst-case scenario, are listed in Table 3.10.

Coefficient	Value	Case
SE_{p_t}	0.54 Pa	$\alpha = 0, Z/H = 0$
SE_{p_s}	0.23 Pa	$\alpha = 0, Z/H = -0.3$
$SE_{C_{p_t}}$	0.042 [-]	$\alpha = 0, Z/H = 0$
$SE_{C_{p_s}}$	0.018 [-]	$\alpha = 0, Z/H = -0.3$

Table 3.10: Maximum standard errors for total pressure coefficient, static pressure coefficient.

3.6.3. Uncertainty 7-hole probe measurements

To assess potential measurement errors introduced by the moving probe method, the stationary dataset was used to validate the moving-probe measurements. Figure 3.25a and Figure 3.25b show the non-dimensionalized total velocity results over the grid of the stationary method and the moving method for the case of $D_e/D = 1.6$, $\alpha = 3$ and $X/D = 2.2$. Appendix D shows the results for the remaining cases. For the stationary data, the data at each point were time-averaged and the standard error is determined per grid point. This was done by using Equation 3.11 and Equation 3.13, where M is the maximum lag, determined by the first zero-crossing of the autocorrelation function.

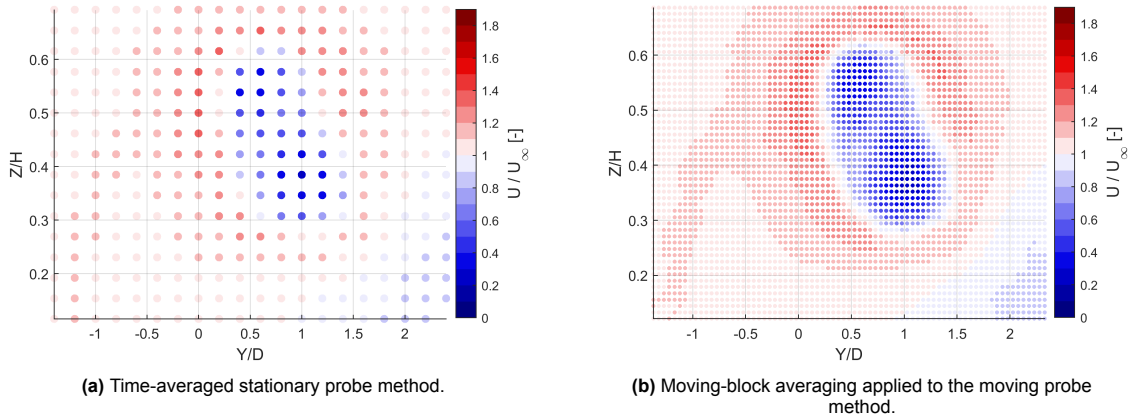


Figure 3.25: Non-dimensionalized total velocity grid, U/U_∞ , over the grid of the a) stationary measurement method and b) moving measurement method for $D_e/D = 1.6$, $\alpha = 3$, $X/D = 2.2$ and $Re = 1.5 \times 10^4$.

α	D_e/D	X/D	SE [-]
3	3.0	3.7	0.035
3	3.0	2.2	0.031
1	3.0	2.2	0.022
3	1.6	2.2	0.030

Table 3.11: Maximum standard error per measurement case for the stationary method for the total velocity U/U_∞ .

The standard error of the stationary method was found to be small and therefore the stationary data was taken as the ground truth. The measurement error of the moving method was assessed by calculating the velocity difference between the obtained average for the moving block method, as described in subsection 3.5.2, to the time-averaged stationary measurement. For the comparison, the midpoint of the moving block closest to the stationary point was used as reference velocity. These grid points had an offset of approximately 1–2 mm.

The results for case $D_e/D = 1.6$, $\alpha = 3$ and $X/D = 2.2$ of the total velocity are shown in Figure 3.26. An overview of the results for the remaining cases is provided in Appendix D. The velocity difference is generally small and spatially inconsistent. The largest differences occur in vortical regions, suggesting that the variation is not caused by a systematic offset of the probe movement.

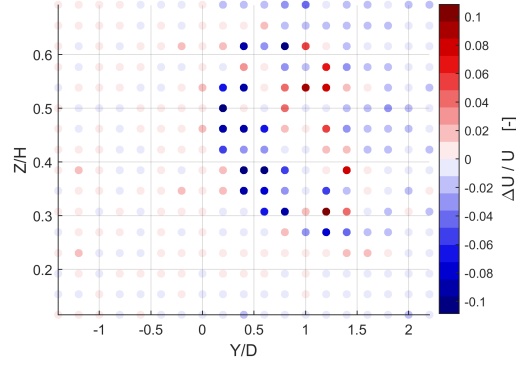


Figure 3.26: Comparison of total velocity difference, $\Delta U/U_\infty$, between stationary and moving probe measurements for $D_e/D = 1.6$, $\alpha = 3$, $X/D = 2.2$ and $Re = 1.5 \times 10^4$.

To analyze the cause of the velocity difference, the standard deviation for the obtained grid of the stationary method and the moving method was analyzed, as shown in Figure 4.5. In general the standard deviation from the moving method is higher than that from the stationary method. For both methods, the highest standard deviation is found within the vortex cores, where the velocity difference between the two methods is also highest. This is expected, as the flow is more unsteady within a vortex. These observations suggest two plausible explanations for the velocity difference, both caused by the data averaging within a moving block.

The first possible cause is that the moving method computes a spatial average within each block. This increases the error in regions where large velocity gradients exist over small distances, such as in vortices.

Another contributing factor may be time independence of measurements within a moving block. Each moving block consists of two rows in the z -direction, as shown in Figure 3.27. These rows are time-independent because the measurements were taken in a zig-zag pattern across the grid. The lack of time correlation between the two rows means that temporal variations in the flow are included within the average of the block. This can lead to a higher standard deviation and a larger velocity difference compared to the stationary method.

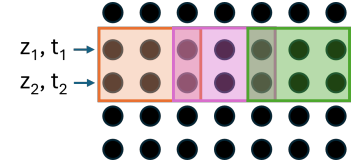


Figure 3.27: Schematic overview of moving average method for the moving dataset with z_1 and z_2 being two time independent measurements.

Finally, the velocity difference could also result from the difference in measurement time. The stationary probe measurements have a duration of 2 seconds, whereas in the moving method each block consists of two 5 mm rows. With the probe moving at 7.5 mm/s, the total measurement time per block was approximately 1.3 seconds. This difference in recording time could contribute to the observed velocity difference.

Due to the small and spatially inconsistent velocity differences, and since a finer grid captures the physical flow phenomena more effectively with the moving probe (see Appendix C), the moving method was used without correction.

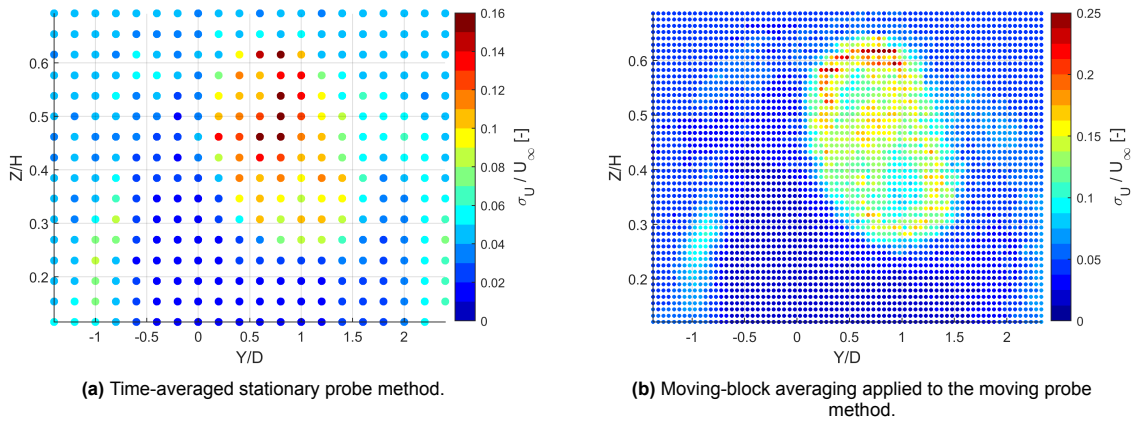


Figure 3.28: Non-dimensionalized standard deviation, σ_U/U_∞ , for a) the stationary probe method and b) the moving probe method for $D_e/D = 1.6$, $\alpha = 3$, $X/D = 2.2$ and $Re = 1.5 \times 10^4$.

This chapter provides the results on the aerodynamic characterization of the scaled rotor sail. Firstly, in section 4.1, the wake deflection is examined. Secondly, in section 4.2, the development of tip vortices is analyzed. Lastly, in section 4.3 the characterization of the vortex shedding is presented.

4.1. Wake deflection

Figure 4.1 shows the time-averaged total pressure coefficient field in the xy -plane at mid-height ($Z/H = 0$). A schematic overview of the measurement plane is given in Figure 3.14b. Energy losses, caused by flow separation, turbulence, and vortex shedding, lead to a decrease in total pressure. These losses are prominent in the wake region, where the direction of the mean flow becomes visible through the pressure deficit.

For $\alpha = 0$, the wake of the static rotor is observed. In the case of $D_e/D = 3.0$, shown in Figure 4.1g, the wake flow appears more two-dimensional and is less affected by tip vortices compared to $D_e/D = 1.0$ and $D_e/D = 1.6$, shown in Figure 4.1a and Figure 4.1e, respectively. Symmetric lower-pressure regions on both sides of the wake indicate the presence of a time-averaged Von Kármán vortex street. When $\alpha = 3$, the vortex shedding is suppressed for both $D_e/D = 1.6$ and $D_e/D = 3.0$, as shown in Figure 4.1f and Figure 4.1i, respectively. However, for $D_e/D = 1.0$, shown in Figure 4.1c, this suppression is not clearly evident. The observed pressure loss in that case may be attributed to vortical structures rather than vortex shedding of the wake.

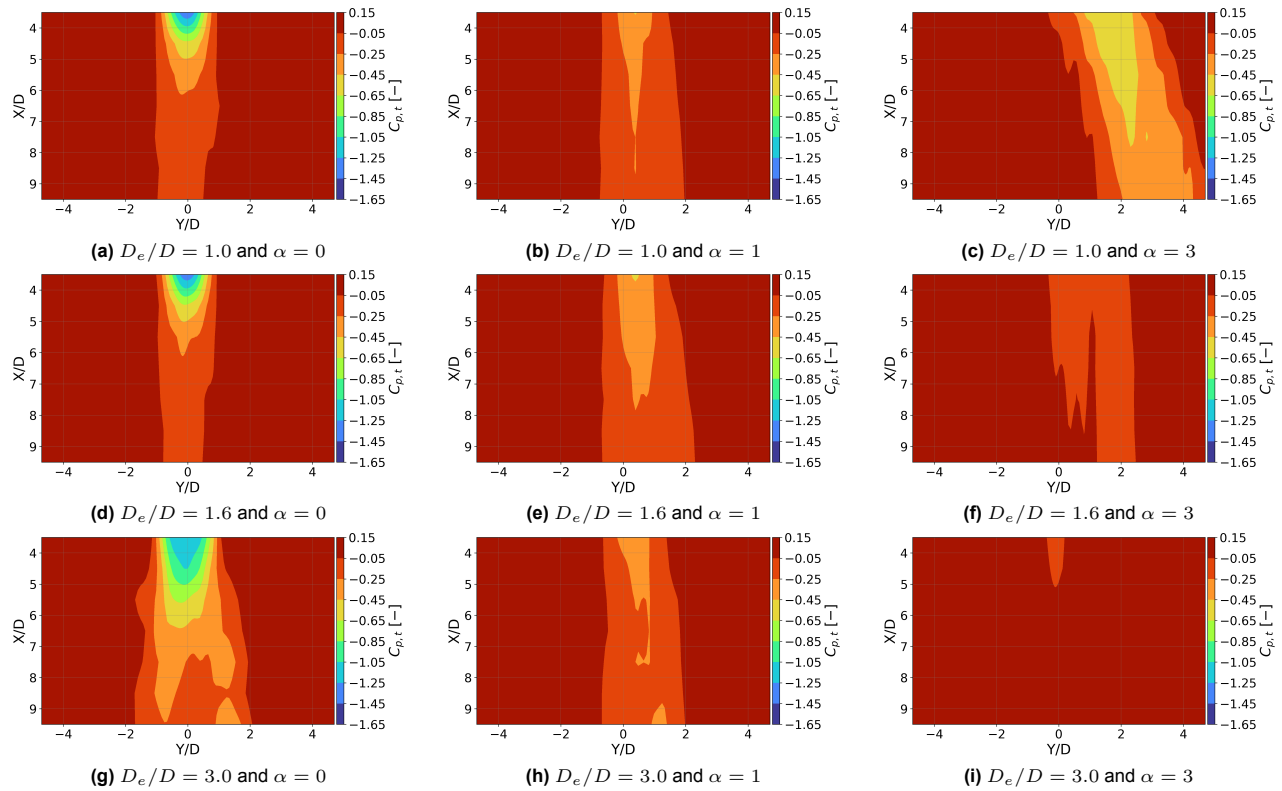


Figure 4.1: Time-averaged total pressure coefficient for $D_e/D=1.0, 1.6$ and 3.0 and $\alpha = 0, 1$ and 3 at the cylinder's midheight ($Z/H = 0$) with $Re = 1.5 \times 10^4$.

Figure 4.2 gives an overview of the deflection angles per configuration and spin ratio. The definition of the deflection angle is given in subsection 3.4.2. The results show that an increase in spin ratio leads to an increase in wake deflection. This is in agreement with results from Bordogna et al. (2020). For $\alpha = 0$, the wake should be symmetrical around the y -axis and no deflection should be present. The angle observed at $\alpha = 0$ likely results from inaccuracies in the determination of the wake deflection. For $\alpha = 1$, the wake is slightly deflected. It is notable that the deflection angle for $D_e/D = 3.0$ is higher than for $D_e/D = 1.0$ and $D_e/D = 1.6$. For $\alpha = 3$ the wake is largely deflected as can be seen from the wake deflection angles for $D_e/D = 1.0$ and $D_e/D = 1.6$. For $D_e/D = 3.0$ no realistic wake deflection angle could be determined using the time-averaged total pressure field as the vortex shedding was suppressed and no total pressure deficit was present.

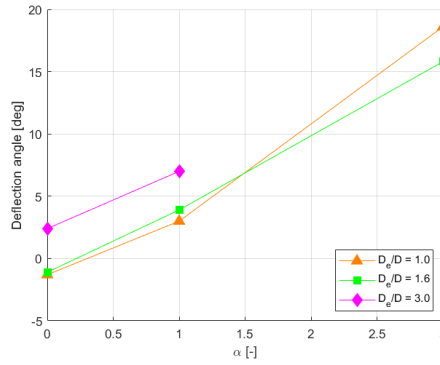


Figure 4.2: Deflection angle (β) as a function of D_e/D and α determined using a line fit through the global minima for the time-averaged total pressure for each X/D with $Re = 1.5 \times 10^4$.

Figure 4.3 and Figure 4.4 show the non-dimensionalized time-averaged total velocity and the velocity component v along the y -axis for $Z/H = 0$ and $X/D = 1.8$. Figure 3.14d provides a schematic representation of this measurement line obtained using the 7-hole probe. In Figure 4.3a the velocity profile is given for $D_e/D = 3.0$ comparing different spin ratios. It can be seen that the wake width decreases for increasing spin ratio, and from $\alpha = 3$, a steady-state flow is visible, as also observed in Figure 4.1i. This steady-state flow is confirmed by the standard deviation of the total velocity, shown in Figure 4.5, as no increase in standard deviation is observed within the wake region, instead, a slight decrease can be seen. The cause of the peak in the standard deviation at $Y/D = 1.95$ is unknown.

Figure 4.3b, Figure 4.3c, Figure 4.4b, and Figure 4.4c present the velocity profiles for end plate configurations $D_e/D = 1.6$ and $D_e/D = 3.0$ at streamwise position $X/D = 1.8$. The velocity contours for $\alpha = 1$ are nearly identical for both configurations (Figure 4.3b and Figure 4.4b), suggesting that the higher deflection angle observed for $D_e/D = 3.0$ in Figure 4.2 may result from inaccuracies in angle determination. For $\alpha = 3$, the total velocity profiles (Figure 4.3c) show lower peak velocities for $D_e/D = 1.6$ compared to $D_e/D = 3.0$, whereas the v -component profiles (Figure 4.4c) show the opposite trend, with higher peak velocities for $D_e/D = 1.6$. For $D_e/D = 1.6$, both velocity profiles are shifted further in the positive y -direction compared to $D_e/D = 3.0$, indicating greater wake deflection. This would indicate that, for higher spin ratios, the deflection angle decreases as the end plate size increases.

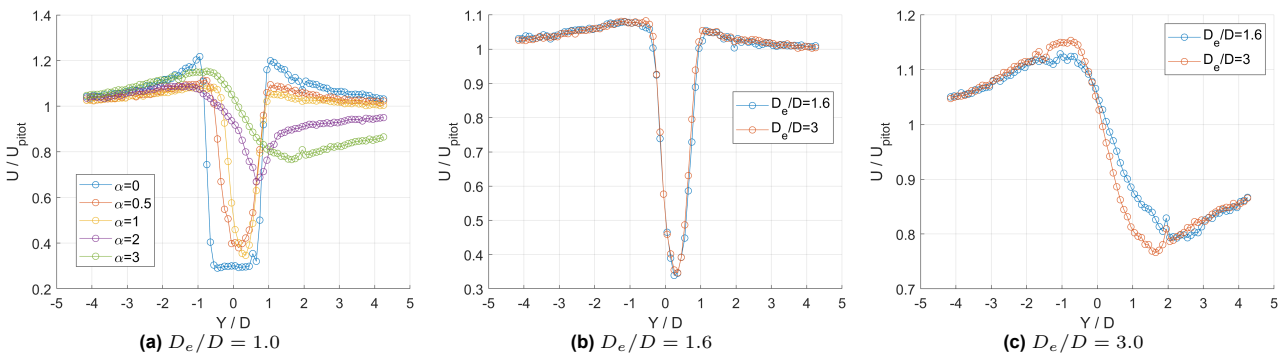


Figure 4.3: Non-dimensionalized time-averaged velocity, U/U_{pitot} , along the y -axis at the cylinder's midheight ($Z/H = 0$) and $X/D = 1.8$ for $Re = 1.5 \times 10^4$ obtained using the 7-hole probe.

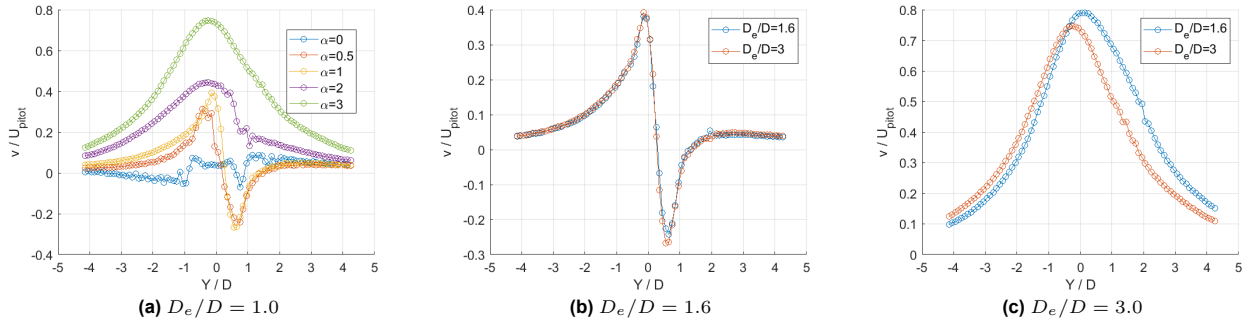


Figure 4.4: Non-dimensionalized time-averaged velocity component, v/U_{pitot} , along the y -axis at the cylinder's midheight ($Z/H = 0$) and $X/D = 1.8$ for $Re = 1.5 \times 10^4$ obtained using the 7-hole probe.

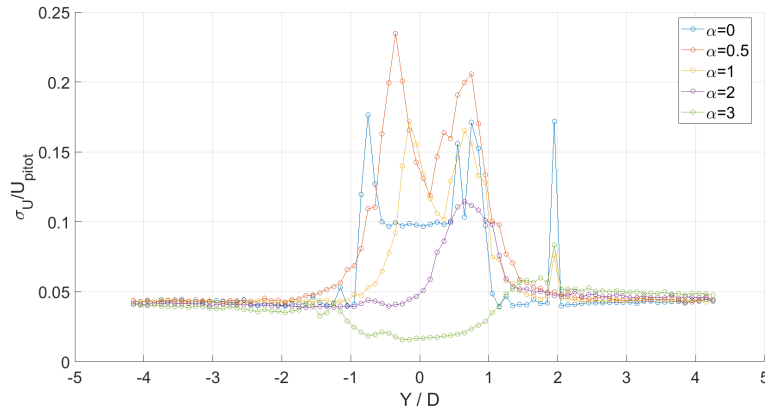


Figure 4.5: Standard deviation, σ_U , of time-averaged total velocity measurement, shown in Figure 4.3a, along the y -axis at the cylinder's midheight ($Z/H = 0$) and $X/D = 1.8$ with $D_e/D = 3.0$ for $Re = 1.5 \times 10^4$ obtained using the 7-hole probe.

4.2. Development of tip vortices

This section presents the vortical topology and wake development of the flow. The time-averaged velocity contour plots are shown in the yz -plane. For the pressure rake measurements the total velocity is defined as

$$U = \sqrt{\frac{p_t - p_s}{0.5 \rho}}. \quad (4.1)$$

The total velocity is non-dimensionalized by the freestream velocity (U/U_∞). The results for the time-averaged total and static pressure coefficients are shown in Appendix C. Results are presented for the three end plate configurations: $D_e/D = [1.0, 1.6, 3.0]$, three spin ratios $\alpha = [0, 1, 3]$ at streamwise locations $X/D = [3.5, 5, 9.5]$. The measurement matrix is presented in Table 3.4 and a schematic overview of the yz measurement planes for the pressure rake is shown in Figure 3.14a.

The velocity contour plots indicate how the flow accelerates or decelerates due to the cylinder's rotation and the formation of vortical structures. From the velocity contour plots, it can be seen that the wake weakens further in streamwise direction as the lower velocity regions diminish in intensity.

Furthermore, it can be seen that the flow is highly three-dimensional around its free ends. The low-velocity zones near the the bottom of the rotating cylinder are generally weaker than those at the top, likely due to reduced end effects caused by the presence of a stationary cylinder beneath.

The three-dimensional velocity components, obtained from the 7-hole probe measurements in the yz -planes, are visualized the top part of the rotor. The measurement matrix is presented in Table 3.6 and a schematic overview of the yz - measurement planes for the 7-hole probe is provided in Figure 3.14c. Furthermore, the Q-criterion is determined, which is a method to identify and visualize vortical structures within a flow field. A more detailed explanation on the Q-criterion is given in subsection 3.5.2. A positive Q-criterion confirms

that the low-pressure regions, seen in the total pressure contours, are vortical structures, which is expected looking at results from Liu, Ma, Zheng, et al. (2024).

Additionally, contour plots for the vorticity are shown, revealing the rotational direction of the vortices. A more detailed explanation on the vorticity calculation is given in subsection 3.5.2. The contour lines for positive Q-values are shown in the same graph as the vorticity results.

The results are presented and discussed per spin ratio.

Development of vortices for $\alpha = 0$

For $\alpha = 0$ and $D_e/D = 1.0$, shown in Figure 4.6, the wake is not clearly visible in the upper half of the domain due to three-dimensional effects. This phenomenon is also observed in the study of Adaramola et al. (2006), shown in Figure 2.6, where the strong downwash for a static circular cylinder of $AR = 5$ is visible. Although the tip vortices induce a strong downwash that changes the flow direction, the velocity magnitude remains nearly constant, which keeps the total velocity largely unchanged in the upper region of the flow ($Z/H > 0.1$). In the case of $D_e/D = 1.6$, shown in Figure 4.7, two disk vortices are clearly visible below the top end plate ($Z/H = 0.4$), while similar structures are not distinguishable near the bottom end. This contradicts the findings of Badalamenti and Prince (2008), who stated that the vortical structure only become present for $\alpha \geq 1$. For $D_e/D = 3.0$, shown in Figure 4.8, two distinct disk vortices are observed at the top and bottom end plates. The vortical structures appear near the height of the end plates ($Z/H \pm 0.5$), from which can be concluded that the downwash has stopped and the wake is visible in the upper region of the rotor ($Z/H > 0.25$).

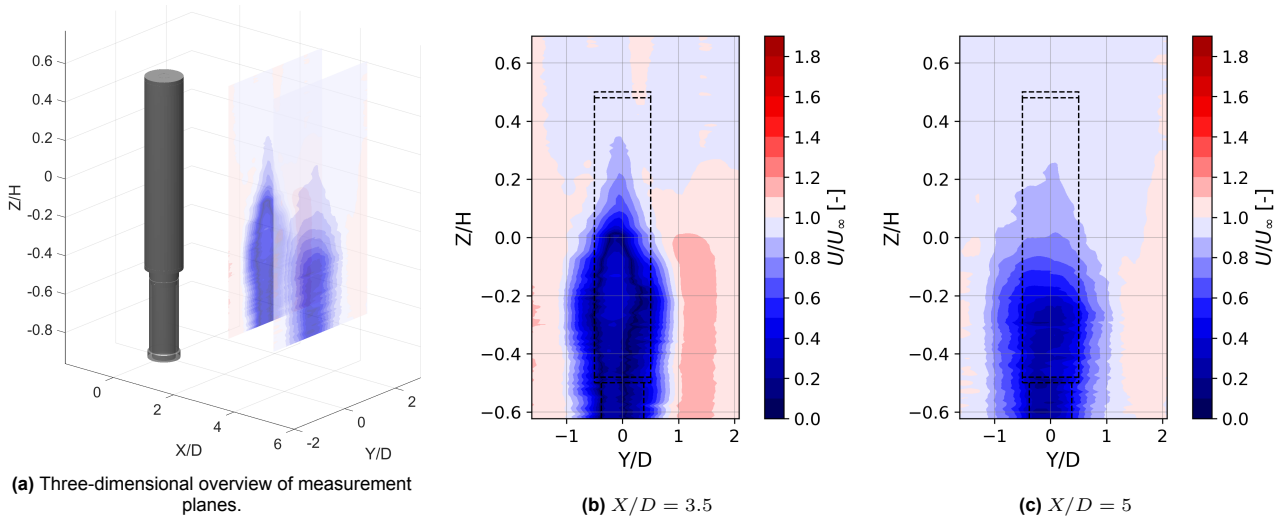


Figure 4.6: Contour plots for non-dimensionalized time-averaged velocity, U/U_∞ , for $D_e/D = 1.0$, $\alpha = 0$, $Re = 1.5 \times 10^4$ and two streamwise distances. The rotor sail is indicated by dashed lines.

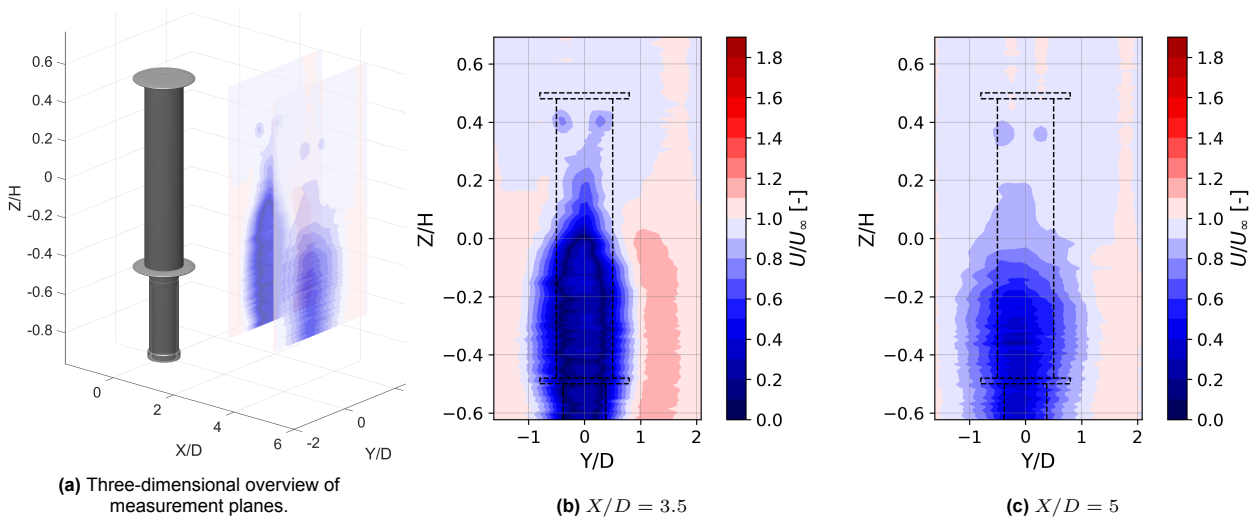


Figure 4.7: Contour plots for non-dimensionalized time-averaged velocity, U/U_∞ , for $D_e/D = 1.6$, $\alpha = 0$, $Re = 1.5 \times 10^4$ and two streamwise distances. The rotor sail is indicated by dashed lines.

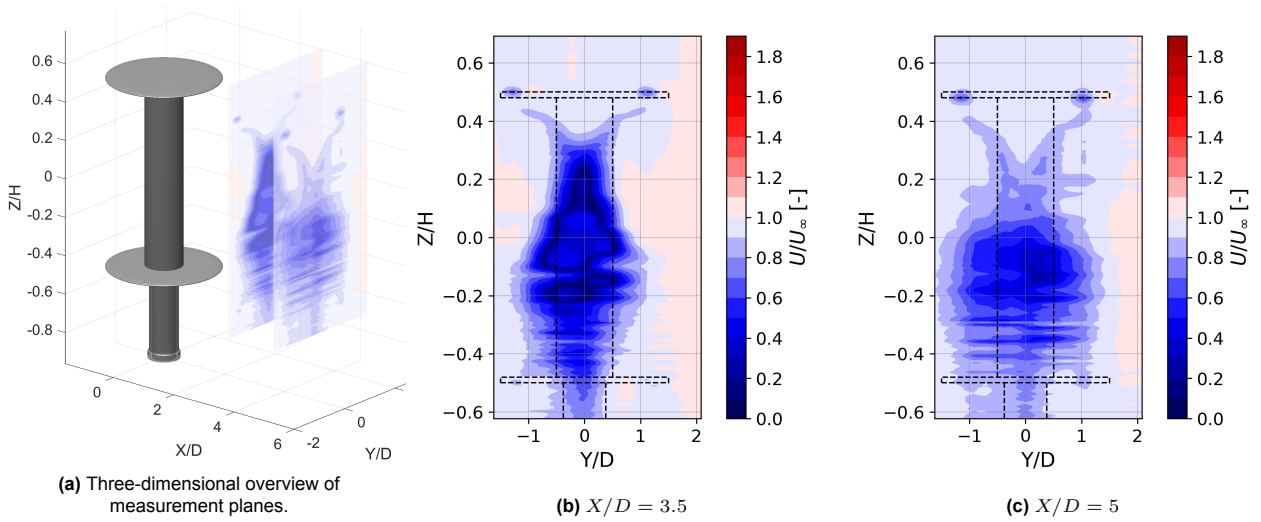


Figure 4.8: Contour plots for non-dimensionalized time-averaged velocity, U/U_∞ , for $D_e/D = 3.0$, $\alpha = 0$, $Re = 1.5 \times 10^4$ and two streamwise distances. The rotor sail is indicated by dashed lines.

Development of vortices for $\alpha = 1$

For a spinning rotor at $\alpha = 1$, the wake starts deflecting towards the right. For the no end plate case, shown in Figure 4.9, a low velocity region is visible below the tip. For $D_e/D = 1.6$, shown in Figure 4.10, the low velocity region is weaker than for the no end plate configuration. For the largest end plate size, $D_e/D = 3.0$, shown in Figure 4.11, a disk vortex is visible at the outside diameter of both the top and bottom end plates on the retreating side. For $D_e/D = 3.0$ the velocity components are shown in Figure 4.11d for $X/D = 2.2$ for the top part of the rotor. At the height of the end plate, the downward flow disappears. This indicates that the end plates are effective in suppressing the downwash. Figure 4.11e shows, on the advancing side, a clear disk vortex on top of the end plate, rotating in the counterclockwise direction. On the retreating side, a disk vortex on top of the end plate, rotating in clockwise direction, and underneath the end plate, rotating in counterclockwise direction, can be seen. This suggest the start of the vortical development as it is seen for $\alpha = 3$.

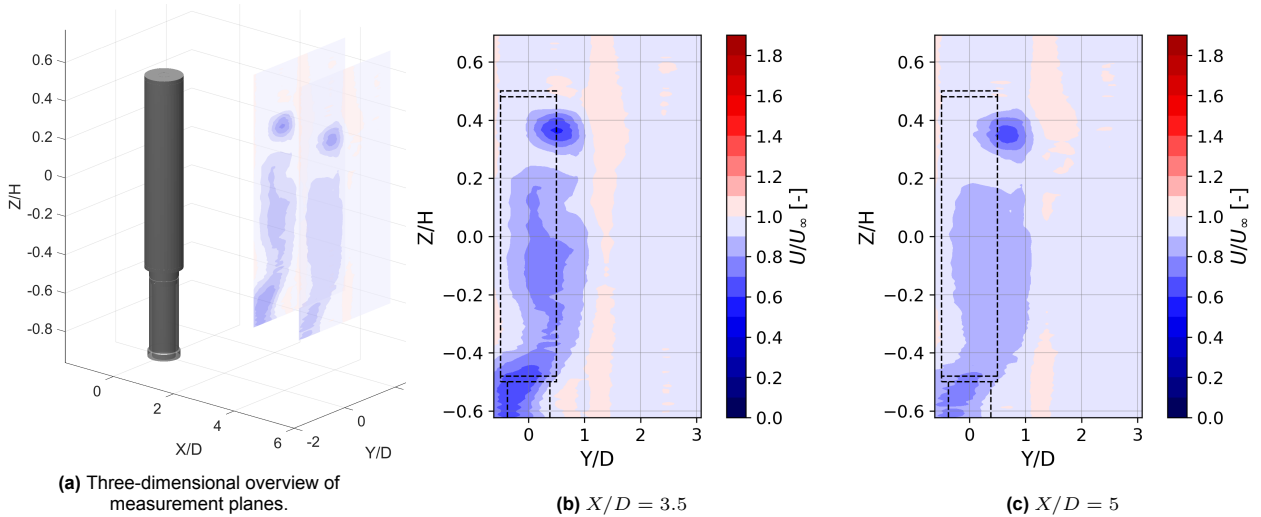


Figure 4.9: Contour plots for non-dimensionalized time-averaged velocity U/U_∞ , for $D_e/D = 1.0$, $\alpha = 1$, $Re = 1.5 \times 10^4$ and two streamwise distances. The rotor sail is indicated by dashed lines.

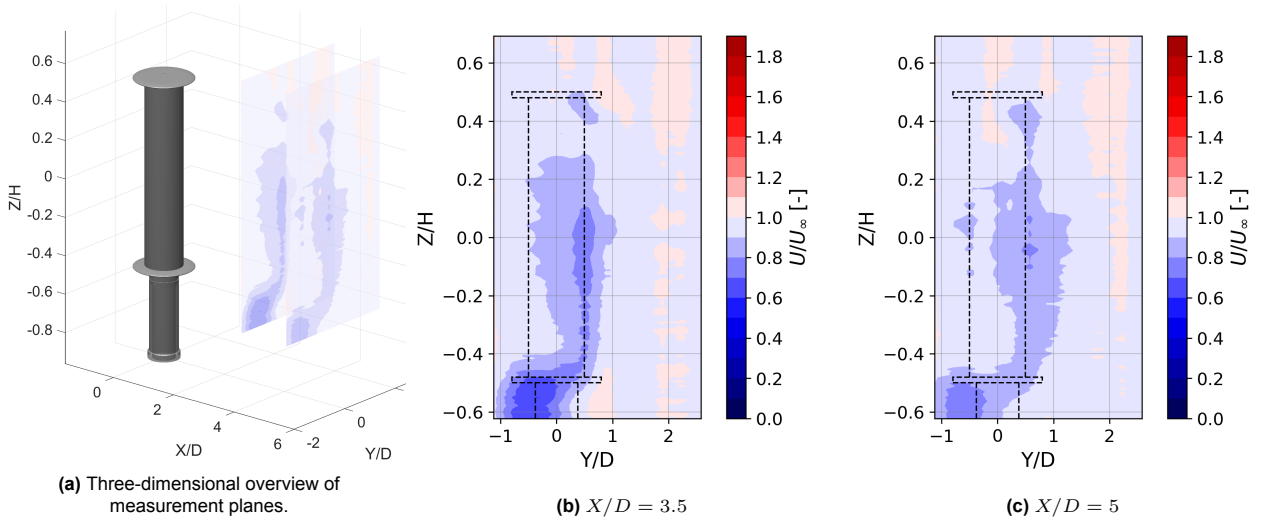


Figure 4.10: Contour plots for non-dimensionalized time-averaged velocity, U/U_∞ , for $D_e/D = 1.6$, $\alpha = 1$, $Re = 1.5 \times 10^4$ and two streamwise distances. The rotor sail is indicated by dashed lines.

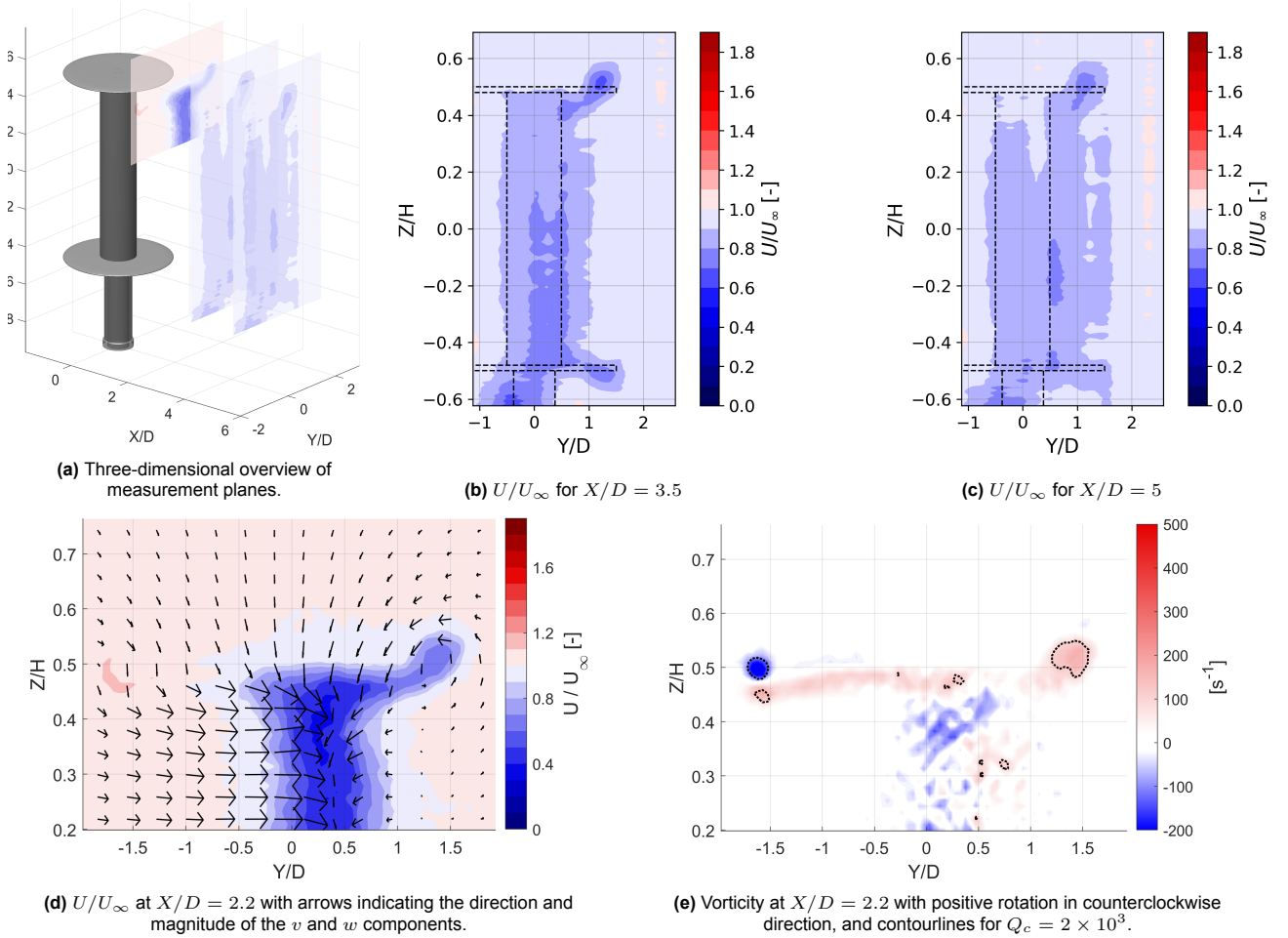


Figure 4.11: Contour plots for non-dimensionalized time-averaged velocity, U/U_∞ , for $D_e/D = 3$, $\alpha = 1$, $Re = 1.5 \times 10^4$ and three streamwise distances. The rotor sail is indicated by dashed lines in (d) and (e).

Development of vortices for $\alpha = 3$

As the spin ratio increases to $\alpha = 3$, the vortices grow in size and intensity. In comparison to the results from $\alpha = 1$ for $D_e/D = 1.0$ and $D_e/D = 1.6$, the downwash becomes more pronounced for $\alpha = 3$ as the vortices shift further towards the rotor's mid-height. For $D_e/D = 1.0$, shown in Figure 4.13b, two cores can be observed for the top tip vortex. The two vortices merge downstream into one core as can be seen in Figure 4.13c. For $D_e/D = 1.6$, shown in Figure 4.14, the inwards movement of the vortices is reduced relative to the no end plate case.

Figure 4.14e shows the v and w velocity components for $D_e/D = 1.6$ at streamwise position $X/D = 2.2$ for the top part of the rotor. The vorticity is shown in Figure 4.14f. These results clearly show that the cores of the two disk vortices of the top end plate rotate in counterclockwise direction. This confirms the findings of Thouault et al. (2012), who concluded, for $\alpha = 2$, $D_e/D = 2$ with comparable Reynolds number, that the two vortices have identical signs. From the velocity components, and the results at different streamwise locations it can be seen that the top vortices spiral in counterclockwise direction, while the bottom vortices spiral in clockwise direction. The separate cores of the two disk vortices from the top end plate can be observed up to streamwise position $X/D = 5$. For $X/D = 9.5$, shown in Figure 4.14d, the two cores cannot be distinguished and the vortices appear merged. This confirms the findings of Thouault et al. (2012), who stated that the two vortices merge further downstream. Figure 4.12 visualizes the measurement locations from the current experiments, added to the original flow sketch from Thouault et al. (2012), with the vortical merging between $5 < X/D < 9.5$.

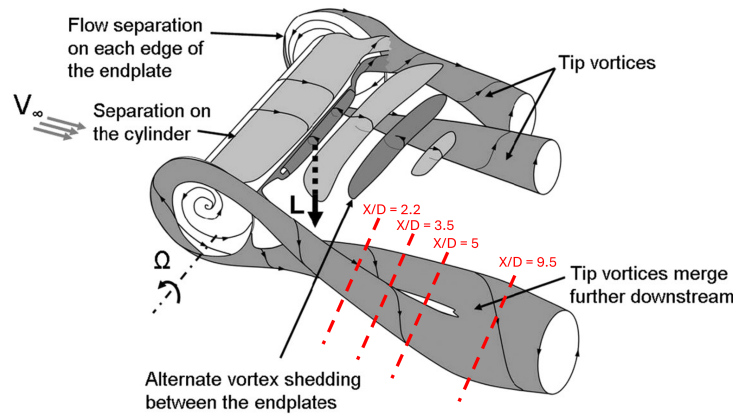


Figure 4.12: Flow topology showing the streamwise measurement locations from the current experiments. *Readapted from Thouault et al. (2012).*

For the largest end plate $D_e/D = 3.0$, the vortical structures of the end plate completely change. In Figure 4.15e two vortical structures are visible at each end: one outside the end plate on the advancing side and one inside the end plate on the retreating side, both rotating in counterclockwise direction.

The results from the 7-hole probe measurements in Figure 4.15f and the results from the pressure rake measurements in Figure 4.15b can be compared, as they correspond to nearly the same streamwise location ($X/D = 3.7$ and $X/D = 3.5$, respectively). The vortex core on the advancing side appears larger in Figure 4.15b compared to Figure 4.15f. This difference might be a result of the shorter measurement duration used for the 7-hole probe measurements in comparison to the pressure rake measurements. It could also result from spatial inaccuracies introduced for the results of the pressure rake, when converting the total and static pressure tube readings to spatial coordinates. This could be influenced by the lower spatial resolution of the static pressure measurements.

The velocity components in v and w direction, are shown in Figure 4.15f and Figure 4.15d. Together with the analysis from the vortical structures at different streamwise locations, $X/D = [2.2, 3.5, 5]$, it can be concluded that the two vortices at the top end plate spiral in counterclockwise direction. Figures Figure 4.15b and Figure 4.15c indicate clockwise spiraling of vortices at the bottom end plate.

Comparing the spanwise position of the vortices for the three end plate configurations for $\alpha = 3$ and $X/D = 5$, shown in Figure 4.13c, Figure 4.14c and Figure 4.15c, it is apparent that the configuration with $D_e/D = 1.0$ shows the largest deflection toward the positive y -direction. The low-velocity region from the bottom vortex

extends up to $Y/D = 3$ (see Figure 4.13c), while for $D_e/D = 1.6$ it only reaches approximately $Y/D \approx 2.5$ (see Figure 4.14c). For $D_e/D = 3.0$, the vortex on the top end plate extends furthest to approximately a maximum of $Y/D \approx 1.5$ (see Figure 4.15c). These observations are consistent with the deflection results discussed in section 4.1, indicating that the deflection decreases with increasing end plate size.

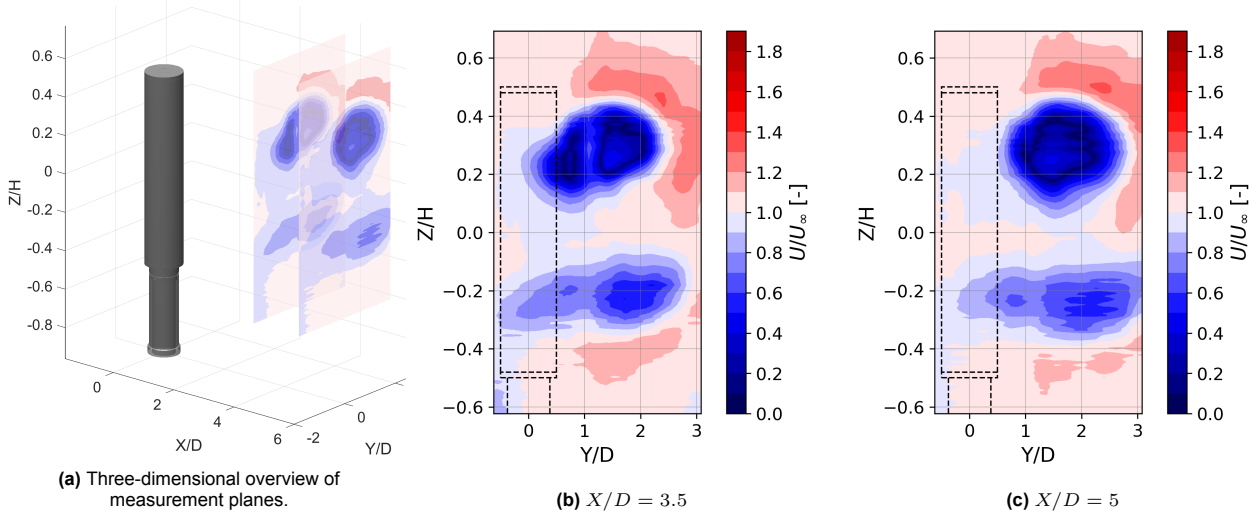


Figure 4.13: Contour plots for non-dimensionalized time-averaged velocity U/U_∞ , for $D_e/D = 1.0$, $\alpha = 3$, $Re = 1.5 \times 10^4$ and two streamwise distances. The rotor sail is indicated by dashed lines.

Connection between vortical behavior and force coefficients

The differences in the topology of the wake and the tip vortices for the various end plate configurations can help explain trends observed for the lift and drag coefficient observed in Figure 2.15, from the study of Badalamenti and Prince (2008). This study was performed at similar Reynolds numbers and aspect ratios to those in the present study. An analysis of the vortical structures indicates that the larger end plate, $D_e/D = 3.0$, has a different vortical topology compared to the $D_e/D = 1.6$ and no end plate cases, which show more similar flow patterns.

The large end plate, $D_e/D = 3.0$, shows two stages of vortex development: at low spin ratios, $\alpha = 1$, weak vortices form outside the end plate, while at high spin ratios, $\alpha = 3$, vortical structures of the large end plate case start developing, creating two strong disk vortices. In contrast, the no end plate configuration shows merging tip vortices at both low and high spin ratios. This could explain the lower drag of rotors with large end plates at low spin ratios, as well as the sharp increase in drag at higher spin ratios in comparison to no end plates.

The higher lift coefficient, reported in Figure 2.15a for configurations with larger end plates, may result from reduced downwash effects. The end plates suppress downwash, as seen from Figure 4.11d, increasing the effective aspect ratio of the rotor and thereby increasing lift. These observations are consistent with the conclusions drawn by Badalamenti and Prince (2008).

Wake deflection of stationary cylinder

An interesting additional finding is the opposite deflection of the wake of the static cylinder compared to that of the rotating cylinder. This phenomenon is best visible from the results of the time-averaged total pressure (see Appendix C), as the measurement domain reaches the height of the static cylinder. Figure 4.16 shows the results for the three end plate configuration of the time-averaged total pressure coefficient for $\alpha = 3$ and $X/D = 3.5$. For these three results, the opposite wake deflection of the static cylinder is clearly visible. A plausible explanation involves the boundary layer behavior around the cylinder, as discussed in subsection 2.3.1. As the separation of the flow on the retreating side of the rotating cylinder occurs later, the flow is accelerated on this side, resulting in a lower pressure zone. For the static cylinder, where the flow is not driven by rotation, the air chooses the path of the least resistance, flowing from high pressure zones to low pressure zones. This results in a boundary layer that develops in the opposite direction compared to that of the rotating case, as illustrated in Figure 4.17, resulting in an opposite wake deflection. Future studies with detailed flow measurement techniques, like surface pressure measurements, are needed to characterize this effect and confirm the

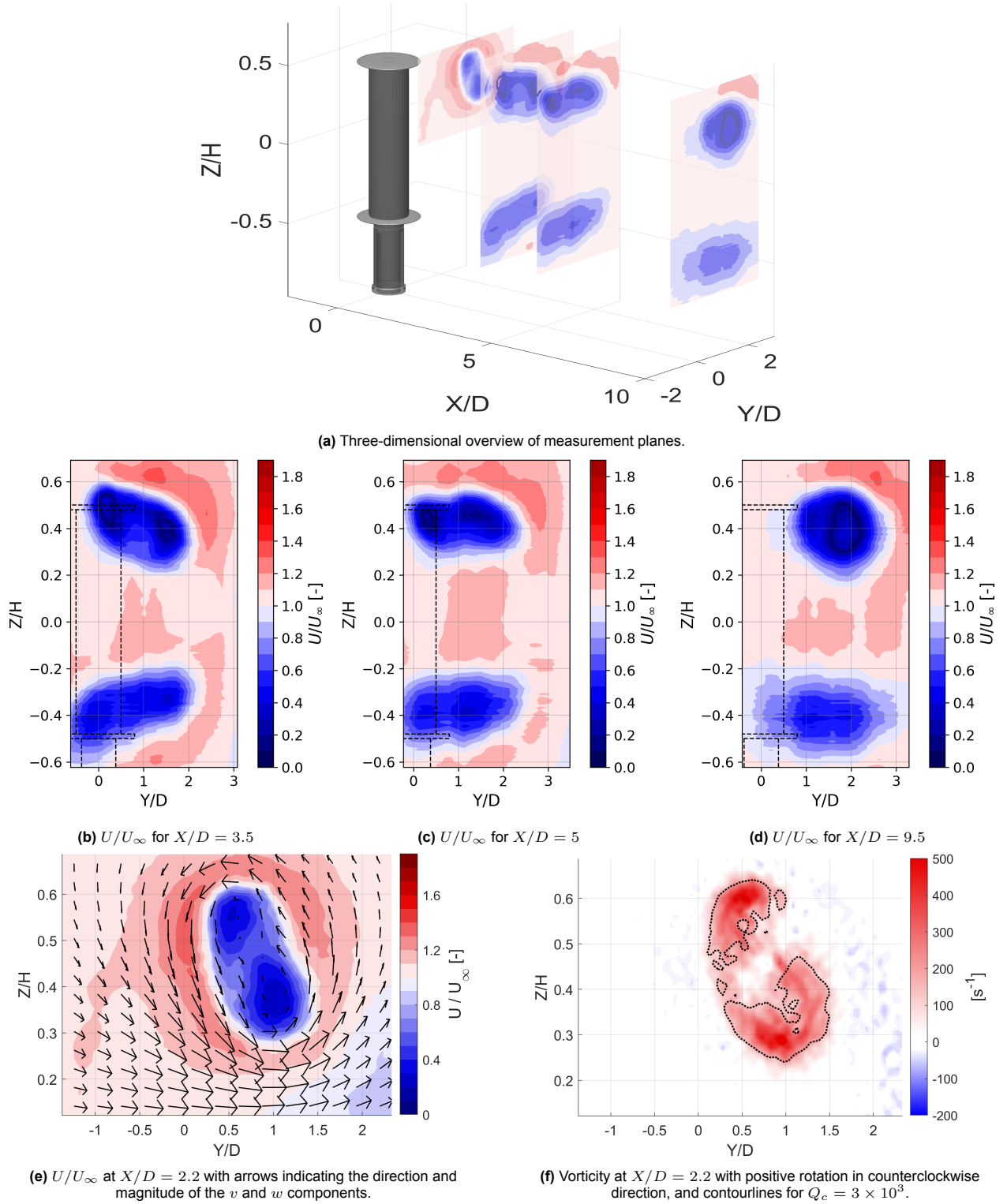


Figure 4.14: Contour plots for non-dimensionalized time-averaged velocity, U/U_∞ , for $D_e/D = 1.6$, $\alpha = 3$, $Re = 1.5 \times 10^4$ and three streamwise distances. The rotor sail is indicated by dashed lines in (d) and (e).

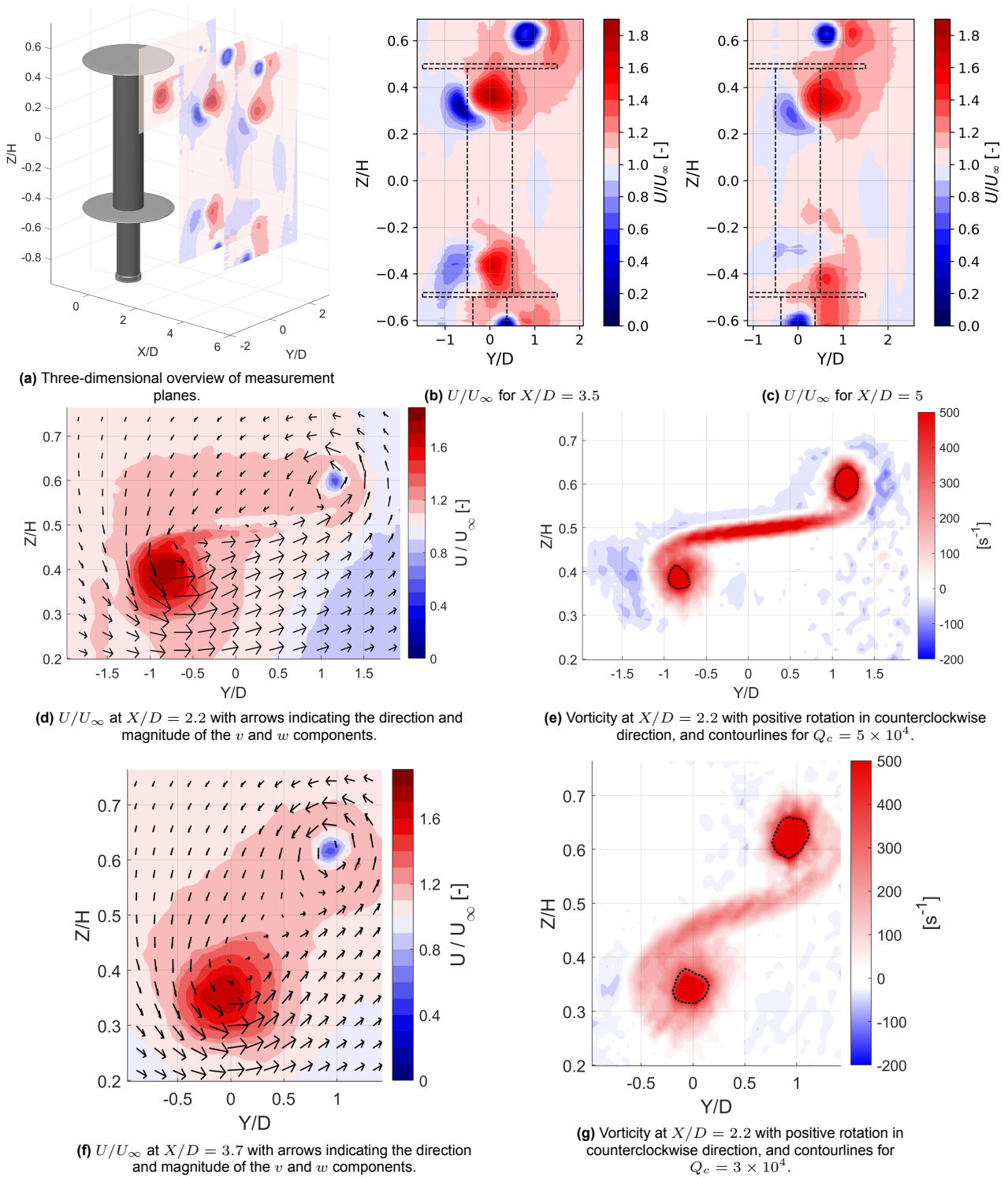


Figure 4.15: Contour plots for non-dimensionalized time-averaged velocity U/U_∞ , for $D_e/D = 3.0$, $\alpha = 3$, $Re = 1.5 \times 10^4$ and three streamwise distances. The rotor sail is indicated by dashed lines in (d) and (e).

explanation above.

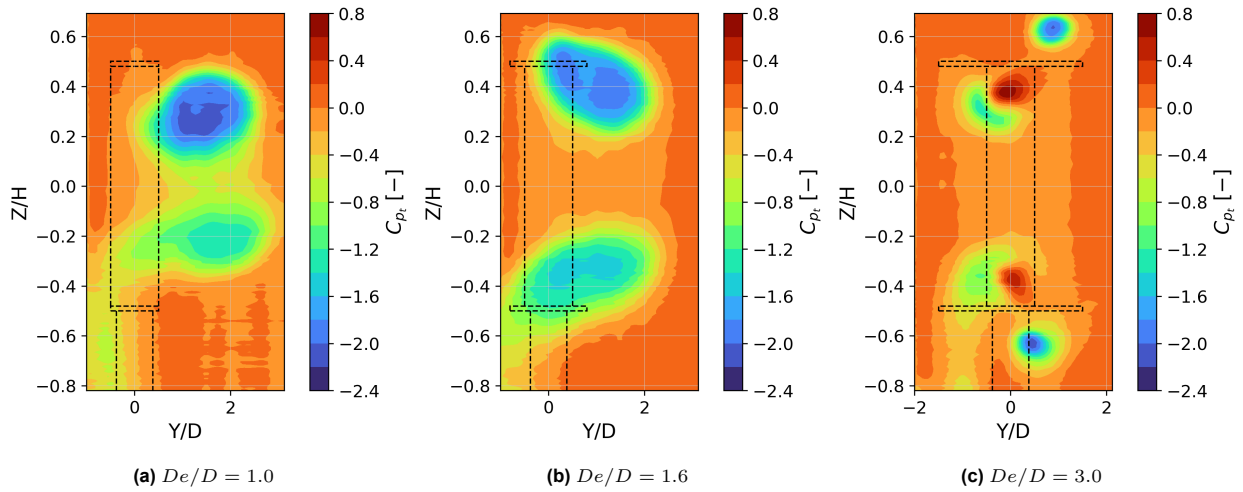


Figure 4.16: Contour plots of time-averaged total pressure coefficient, C_{pt} , for three end plate configurations for $\alpha = 3$ and $X/D = 3.5$ with $Re = 1.5 \times 10^4$.

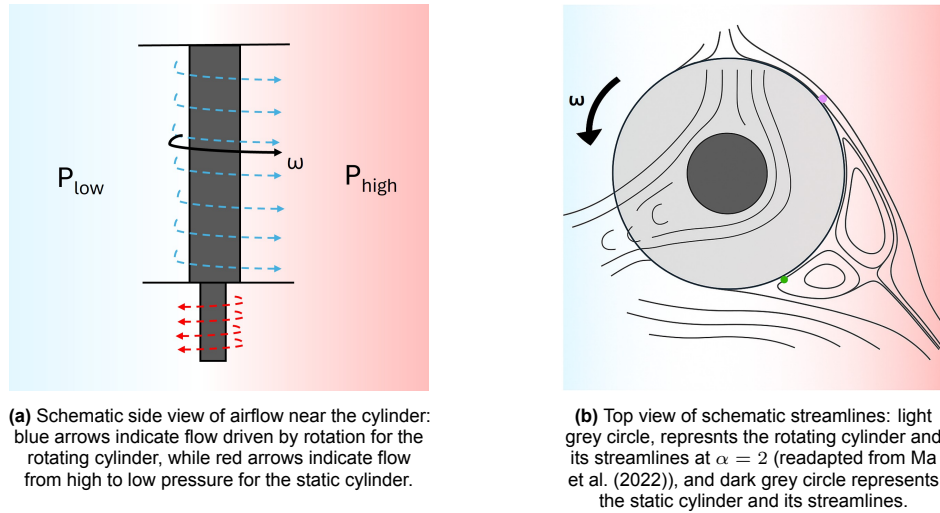


Figure 4.17: Illustration of the flow mechanism proposed to explain the deflection opposite to the rotation direction of the cylinder.

4.3. Spectral analysis of vortex shedding

This section presents the results of the experiments aimed at identifying vortex shedding frequencies. Measurements were conducted using the 7-hole probe for two end plate configurations: $D_e/D = 1.6$ with spin ratios $\alpha = [0, 1, 3]$ and $D_e/D = 3.0$ with spin ratios $\alpha = [0, 0.5, 1, 2, 3]$ at streamwise position $X/D = 1.8$ and cylinder's mid-height $Z/H = 0$.

Figure 4.20 and Figure 4.19 show contour plots of the normalized PSD amplitudes for each Strouhal number along the y -axis. The vortex shedding is most clearly observed in the shear layer of the wake. As the spin ratio increases, the wake deflects toward positive y -values, causing a gradual shift of the shear layer in that direction. Despite this deflection, the dominant frequencies on either side of the wake remain similar, indicating that the vortex shedding frequency is not affected by the flow deviation.

For $\alpha = 0$, shown in Figure 4.19a and Figure 4.20a for $D_e/D = 1.6$ and $D_e/D = 3.0$, respectively, the observed peaks for the low Strouhal number ($St = 0.037$) correspond to the wind tunnel fan rotational frequency (204 RPM = 3.4 Hz). The static rotor results for $D_e/D = 1.6$ do not show a clear trend in Strouhal number, while the results for $D_e/D = 3.0$ do show a clear trend in Strouhal number, although some fluctuations in dominant

Strouhal number can be seen along the y -axis. For $\alpha = 0.5$, $\alpha = 1$ and $\alpha = 2$, the dominant Strouhal number appears much more steady. Zdravkovich (1997) stated for $\alpha = 0$ and $AR = 5$ and $Re = 1.1 \times 10^5$, that the Strouhal number varied ($0.15 \leq St \leq 0.19$) confirming the irregularity seen in present results along the y -axis for the static rotor case with $D_e/D = 3.0$. At $\alpha = 3$, no dominant frequency is found, suggesting that vortex shedding is suppressed at this condition.

When comparing the flow-induced spectra to the mechanical vibration spectra presented in subsection 3.2.2, it is evident that the shedding frequencies originate from the flow and are not mechanically induced. No peaks are present in the mechanical spectra for the identified shedding frequencies, with the exception of the $D_e/D = 3.0$ and $\alpha = 2$ case. Here, a clear peak is observed in Figure 3.12d, possibly due to the shedding frequency being close to the resonance frequency of the setup.

Figure 4.18 presents a comparison between the Strouhal numbers obtained in this study, for $D_e/D = 3$, $X/D = 1.8$ and $Y/D = 1.25$, and those reported by Badalamenti and Prince (2008) and Ma et al. (2022), who conducted experiments at similar Reynolds numbers ($Re = 4.1 \times 10^4$ and 4.6×10^4 , respectively) and end plate ratios ($D_e/D = 2$). While the exact Y/D measurement positions varied across the studies, previous work by Badalamenti (2010) and Ma et al. (2022), along with the findings in Figure 4.20a and Figure 4.19a, suggest that the Strouhal number does not vary on both sides of the wake, and is thus insensitive to wake deflection. This implies that the results can be compared to each other. The results exhibit the same increase in Strouhal number with increase in spin ratio. The results also have similar orders of magnitude to previous work, although they are generally slightly lower. For $\alpha = 1$, the Strouhal number for both end plate sizes ($D_e/D = 1.6$ and $D_e/D = 3.0$), can be compared for $X/D = 1.8$, $Y/D = 1.25$ and $Z/D = 0$. A slight increase in Strouhal number is found for the larger end plates ($St = 0.23$) compared to the smaller end plates ($St = 0.21$), confirming the findings of Badalamenti (2010).

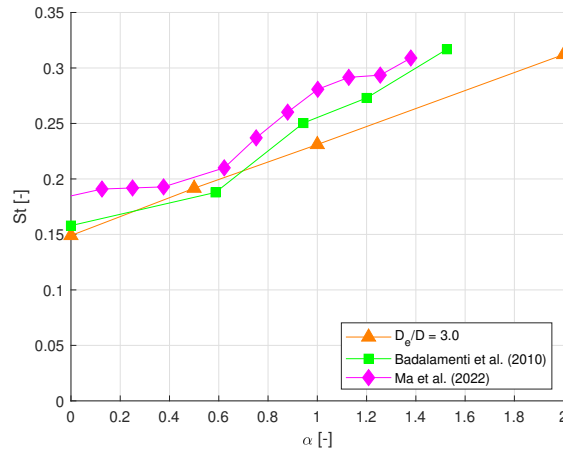


Figure 4.18: Strouhal number as a function of spin ratio α with: $\blacktriangle$ $D_e/D = 3.0$, $Re = 1.5 \times 10^4$, $X/D = 1.8$, $Y/D = 1.25$ and $Z/D = 0$, $\blacksquare$ data from Badalamenti (2010) ($AR = 5.1$, $Re = 4.1 \cdot 10^4$, $D_e/D = 2$), where ($X/D = 2$, $Y/D = 0.6$ and $Z/D = 0$) $\blacklozenge$ data from Ma et al. (2022) ($AR = 10$, $Re = 4.6 \times 10^4$, $D_e/D = 2$), where $X/D = 3.5$, $Y/D = 1$ and $Z/D = 0$.

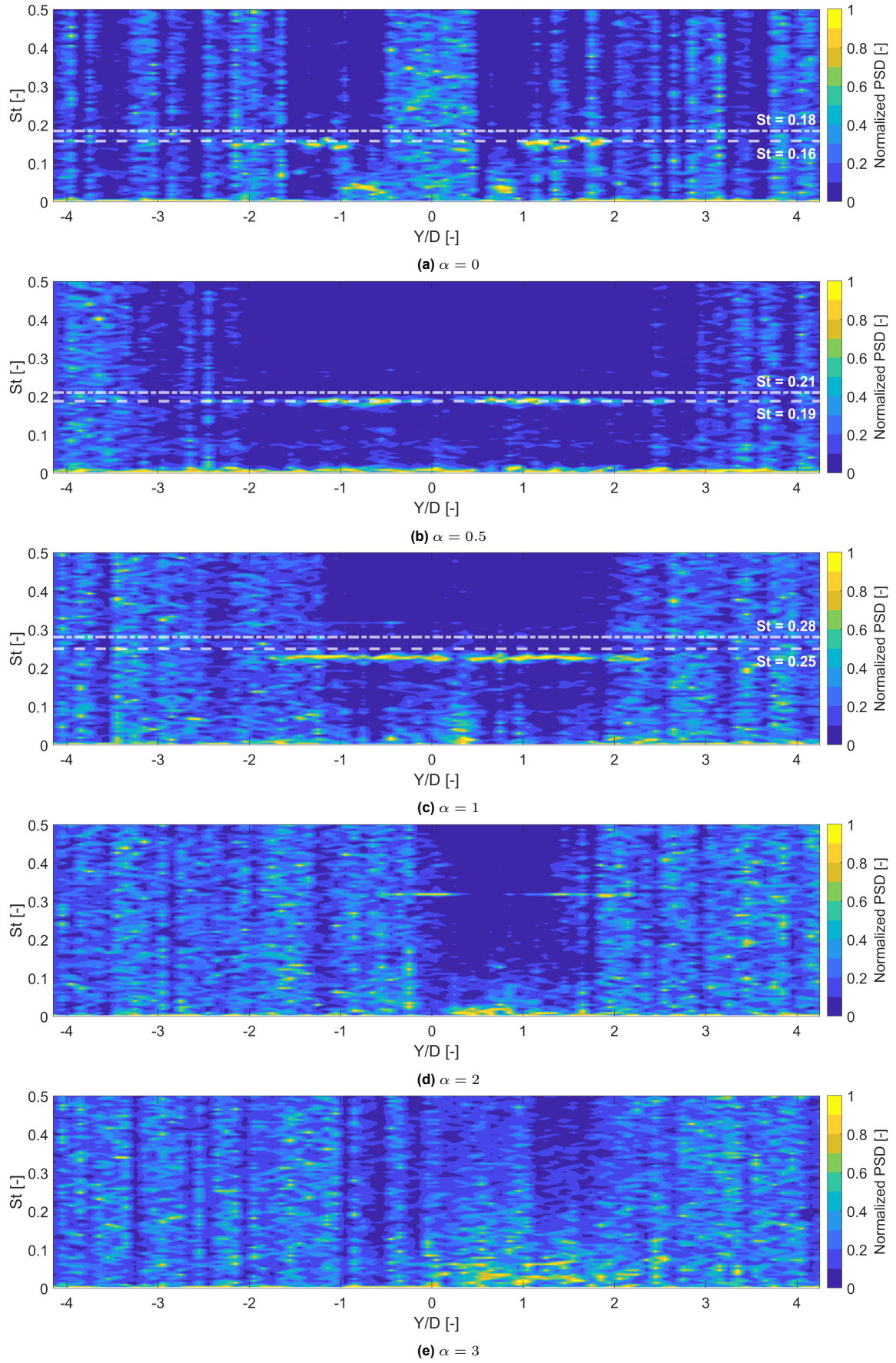


Figure 4.19: Contour plot for normalized PSD amplitudes for $D_e/D = 3.0$ at $X/D = 1.8$, $Z/H = 0$, $Re = 1.5 \times 10^4$ for spin ratios $\alpha = 0$, $\alpha = 1$, and $\alpha = 3$. The white lines represent Strouhal numbers reported by ((- · -)) Ma et al. (2022) and (--) Badalamenti (2010).

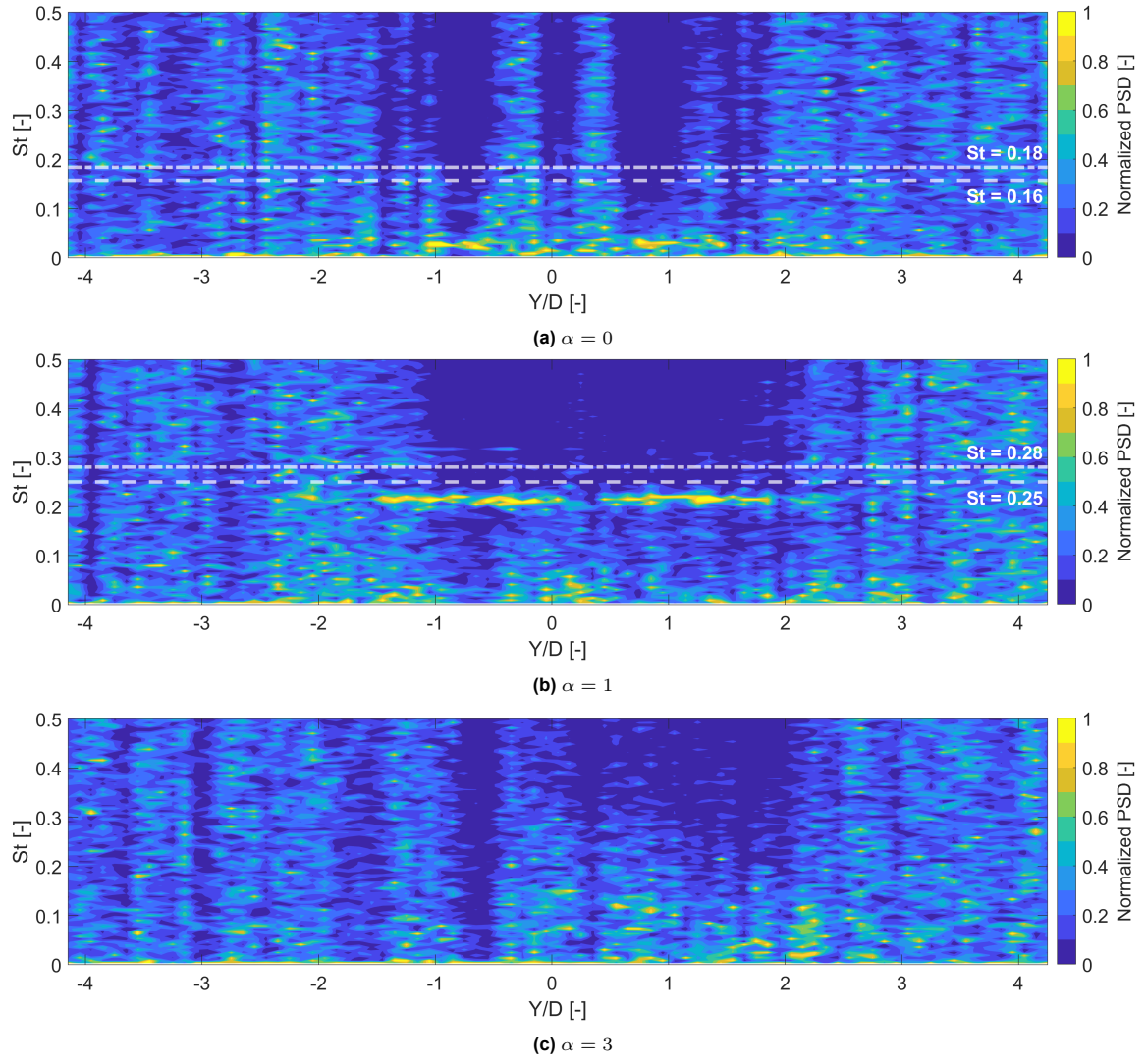


Figure 4.20: Contour plot for normalized PSD amplitudes for $D_e/D = 1.6$ at $X/D = 1.8$, $Z/H = 0$, $Re = 1.5 \times 10^4$ for spin ratios $\alpha = 0$, $\alpha = 1$, and $\alpha = 3$. The white lines represent Strouhal numbers reported by ((-.-)) Ma et al. (2022) and (--) Badalamenti (2010).

Conclusion

The results in this study indicate that the end plate size has a large influence on the wake flow characteristics of a rotor sail. These results contribute to the fundamental understanding of flow development around a rotor sail and provide insights into the aerodynamic performance benefits reported in literature regarding end plate configuration.

Within this research an experimental campaign has been carried out using a scaled rotor sail in the SLT wind tunnel of the TU Delft. Pressure measurements were performed in the wake of the model using both a pressure rake and a 7-hole probe, making this the first experimental study to report results on all three velocity components. This study aimed to identify the influence of end plate configuration on the wake flow characteristics of a finite rotating circular cylinder by analyzing three main wake characteristics: 1) flow direction of the wake, 2) vortical topology of the wake flow and 3) shedding of the flow.

The experimental results confirmed that wake deflection increases with spin ratio, showing an especially large increase in wake deflection for $\alpha = 3$. Interestingly, for this spin ratio, the end plate configuration was found to influence the deflection angle: larger end plates reduced the extent of wake deflection. Furthermore the wake width significantly decreases by increasing the spin ratio from $\alpha = 0$ up to $\alpha = 2$. From $\alpha = 3$ vortex shedding is suppressed, leading to a steady-state flow.

The analysis on the vortical topology indicates that the end plate size has a large influence on the development of vortices around the rotor's free ends. For the static case, $\alpha = 0$, without end plate, a strong downwash effect is observed. By the addition of end plates, disk vortices are created and the downwash effect is reduced. For the largest end plates, $D_e/D = 3$, the downwash has diminished and the disk vortices remain at the height of the end plates. At $\alpha = 1$, low-pressure regions form near the top end plate. For $D_e/D = 1.6$ these pressure regions are weaker than for the no end plate case. For $D_e/D = 3$, a clear difference in vortex behavior can be seen: on the advancing side a vortex forms above the end plate, rotating counter clockwise, while on the retreating side one vortex forms above the end plate, rotating clockwise, and another below the end plate, rotating counter clockwise.

As the spin ratio increases to $\alpha = 3$, the influence of the end plate becomes even more visible. Without end plates, the two tip vortices are fully merged by streamwise location $X/D = 5$, while for $D_e/D = 1.6$ at this streamwise location, the vortices start to merge but the two vortical cores remain distinguishable. For $D_e/D = 3$ the vortical pattern looks completely different as the two vortical cores are separated: one outside the end plate on the advancing side and one inside the end plate on the retreating side, both rotating in counter clockwise direction. These results indicate that an increase in end plate size leads to a delay in the merge of the tip vortices.

These observed differences in vortical behavior between the end plate configurations provide further insight into the observed trends in lift and drag coefficient in previous studies. The large end plate, $D_e/D = 3.0$, shows two stages of vortex development: at low spin ratios, the vortices are relatively weak, while at high spin ratios, strong tip vortices form. In contrast, the no end plate configuration shows a clear tip vortex at both low and high spin ratios. This behavior coincides with the lower drag of rotors with large end plates at low spin ratios, as well as the sharp increase in drag at higher spin ratios observed in the studies of Badalamenti and Prince (2008) and Chen et al. (2023). The higher lift coefficient observed for larger end plates is likely due to downwash suppression and an increase in effective aspect ratio.

While the aerodynamic role of the top and bottom end plates appears similar, both reducing downwash and increasing the effective aspect ratio of the rotor, differences in vortex behavior are observed. Firstly, the disk vortices, present for $\alpha = 3$ with $D_e/D = 1.6$ and $D_e/D = 3.0$, exhibit a spiraling pattern, rotating counter-

clockwise at the top and clockwise at the bottom. Secondly, the low-velocity regions near the bottom of the rotating cylinder, formed by these vortices, are generally weaker than those at the top, likely due to reduced end effects caused by the presence of a stationary cylinder beneath. These results indicate that the top and bottom end plate influence the flow differently. To know the exact performance benefit of a bottom end plate compared to a top end plate, further research is needed.

Both this study and the study of Badalamenti and Prince (2008) show that increasing end plate size reduces the inward movement of the tip vortices. However, in the study of Badalamenti and Prince (2008), who performed wake measurements under similar conditions, part of the wake lay outside the measurement domain. In contrast, the present study captures the complete velocity field, including all three velocity components, providing a more detailed view of the vortical structures. Furthermore, the experimental setup, used within this study, represents more accurately a rotor sail. The addition of a static cylinder positioned beneath the rotor, leads to distinct top and bottom vortex behavior. Overall, the trends observed here are consistent with those reported by Badalamenti and Prince (2008), but the present work offers a more comprehensive characterization of the vortical topology of a rotor sail, thereby making a contribution to the existing literature.

The spectral analysis on velocity fluctuations in the wake of the rotor confirms results from previous studies (e.g. Badalamenti (2010) and Ma et al. (2022)): an increase in spin ratio leads to an increase in Strouhal number. A slightly higher Strouhal number was observed for larger end plate sizes, which confirms the findings of Badalamenti (2010).

Another observation during the study was the opposite deflection of the wake for the stationary cylinder. The following hypothesis has been formulated: as the separation of the flow on the retreating side of the rotating cylinder occurs later, the flow is accelerated on this side, resulting in a low-pressure zone. For the static cylinder, where the flow is not driven by rotation, the air chooses the path of least resistance, flowing from high- to low-pressure zones. This results in the development of the flow in the opposite direction compared to that of the rotating case, ultimately reversing the direction of wake deflection.

Recommendations

This research gives an initial overview of wake characteristics around a rotor sail for various end plate sizes. However, to obtain more insights into this topic, recommendations for future research are proposed:

- First of all, due to time constraints, only limit cases have been performed for the 7-hole probe measurements. It would be valuable to perform measurements on the entire wake region of the rotor or at least conduct measurements near the bottom end plate as it might show different vortical behavior.
- Secondly, due to time limitations, no configurations were tested with asymmetric end plate configurations (e.g., larger top end plate and smaller bottom end plate). This could be interesting, since in practice many rotor sails operate with only a top end plate. Based on the results in this study, the vortical structures seem less intense on the bottom side, yet they suggest an increase in the effective aspect ratio. Therefore, it would be valuable to investigate the exact performance benefit on the aerodynamic force coefficient from the addition of a bottom end plate. Also, the addition of a load cell to the setup would make it possible to directly measure lift and drag forces and better evaluate the performance.
- Furthermore, only a limited number of spin ratios were tested. More spin ratios would give a better understanding of the development of vortices, especially at higher spin ratios. Additionally, more points between $\alpha = 0$ and $\alpha = 2$ could help to better identify the trend of the Strouhal number. In previous studies, like Mittal and Kumar (2003), Stojković et al. (2002), and Badalamenti (2010), a second vortex shedding regime was found at higher velocity ratios. Future research could try to identify this regime and study its impact on lift and drag.
- To improve the spatial understanding of vortical topology, additional measurements in the streamwise x -direction should be taken. This would offer better insight into the development and merging of spiraling structures over space and time. Alternatively, advanced techniques such as 3D Particle Image Velocimetry (PIV) could be used to provide a more comprehensive view of vortex evolution.
- In addition, surface pressure measurements on the rotor would help determine how end plates influence pressure distribution and could be directly compared to force measurements to better understand aerodynamic performance.
- It is also important to consider that rotor sails operate in real-world conditions within an atmospheric boundary layer. This may influence flow development and vortex dynamics. Investigating the effects of boundary layer profiles would therefore be a valuable extension of the current work.
- Furthermore, the influence of Reynolds number has not been studied in this research. It could be useful to study how the vortical topology changes at different, especially higher, Reynolds numbers, since rotor sails operate in the supercritical regime. Reaching these higher Reynolds numbers with this setup would require a different motor or gearbox to obtain higher rotation speeds.
- Moreover, it would be valuable to measure the power consumption of the rotor for various end plate configurations. These results could be compared to force coefficients. In this way the configuration with the optimal energy efficiency could be determined.
- An additional uncertainty lies in the effect of wake blockage on the results, which remains unclear due to the complexity and limited study of wind tunnel correction methods for rotating cylinders. Further investigation into suitable correction approaches is recommended.
- Lastly, from the mechanical eigen frequency tests it would be an addition to confirm the resonance frequencies found in these tests, by performing additional simulations, for example with ANSYS. Furthermore, conducting a dynamic analysis could provide insights into the system's behavior under operational loads and help assess the risk of resonance during rotor operation.

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Technical specifications

A.1. Festo servo motor - EMMT-AS-80-HKR-HS-RMB

Informatieblad

FESTO

 Condition of use

Artikelnummer : 4595815
 Bestelcode 1 : EMMT-AS-80-HKR-HS-RMB
 Bestelcode 2 :
 Order-ID:

Technische gegevens

Benaming eigenschap	Waarde
Type code	EMMT-AS
Omgevingstemperatuur	\-15 - 40 °C
Opmerking bij omgevingstemperatuur	Up to 80 °C with derating of -1.5% per degree Celsius
Stoekteemperatuur	-20 - 70 °C
Relatieve luchtvochtigheid	0 - 90%
Volgens de norm	IEC 60034
Temperatuur monitoring	Digitale motortemperatuuroverdracht via EnDat 2.2
Inbouwpositie	naar keuze
Beschermingsgraad	IP65
Interface code motor out	80P
Opmerking bij materiaal	RoHS conform
Corrosiebestendigheidsklasse CRC	0.0 - geen corrosiebelasting
Toelating	RCM compliance mark
CE-markering (zie conformiteitsverklaring)	volgens EU-EMV-richtlijn
Nominale bedrijfsspanning DC	325
Wikkelingsschakel type	Stern innen Ster binnenkant
Spitzendrehmoment	4.3 Nm
Nominaal draaimoment	3.4 Nm
Piekdraaimoment	13.5 Nm
Nominale rotatiesnelheid	3000 1/min
Max. rotatiesnelheid	7820 1/min
Langdurige stilstandstroom	4.8 A
Nominale motorspanning	3.8 A
Piektroom	21.7 A
Motorconstante	0.9
Spanningsconstante, fase-fase	61.4
Wikkelingsweerstand fase-fase	2.21 Ohm
Wikkelingsinductiviteit fase-fase	10.7
Totaal traagheidsmoment van drive output	2.43
Productgewicht	4.75 g
Toelaatbare axiale as-belasting	120 N
Toelaatbare radiale as-belasting	620 N
Rotorpositie encoder	Absolute multi-turn encoder
Rotorpositie encoder interface	EnDat 2.2
Rotorpositie encoder meetprincipe	inductief
Houdmoment rem	7 Nm
Bedrijfsspanning DC-rem	24 V
Rem stroomverbruik	15
Massatraagheidsmoment, rem	0.459
Schakelcyclus, houdrem	10 million idle actuations (without friction work!)

Figure A.1: Festo EMMT-AS-80-HKR-HS-RMB technical specifications

A.2. Mobac magnetic disk coupler - MTD-5

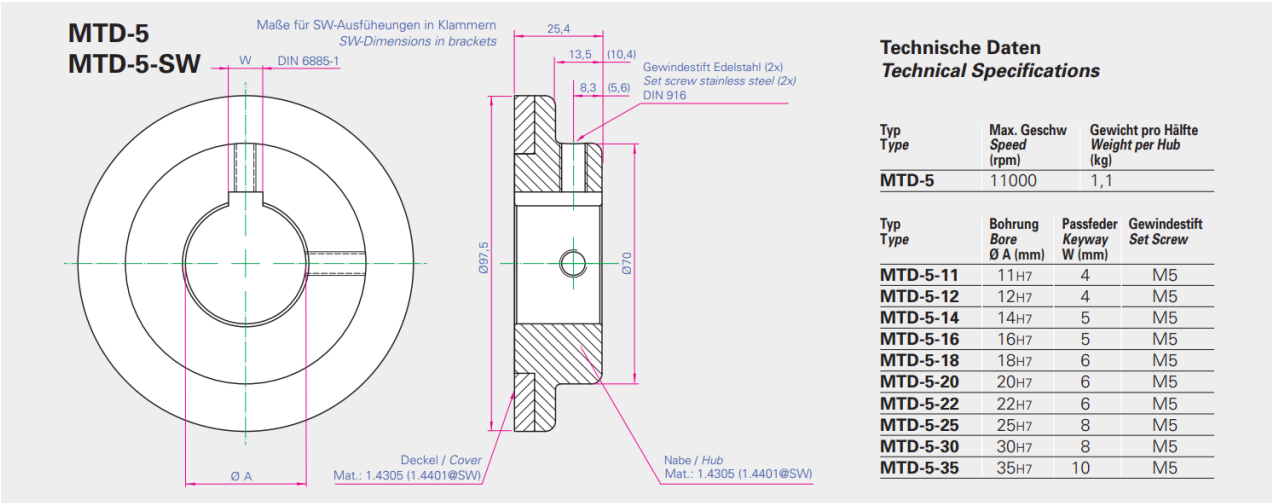


Figure A.2: Mobac MTD-5 dimensions and technical specifications

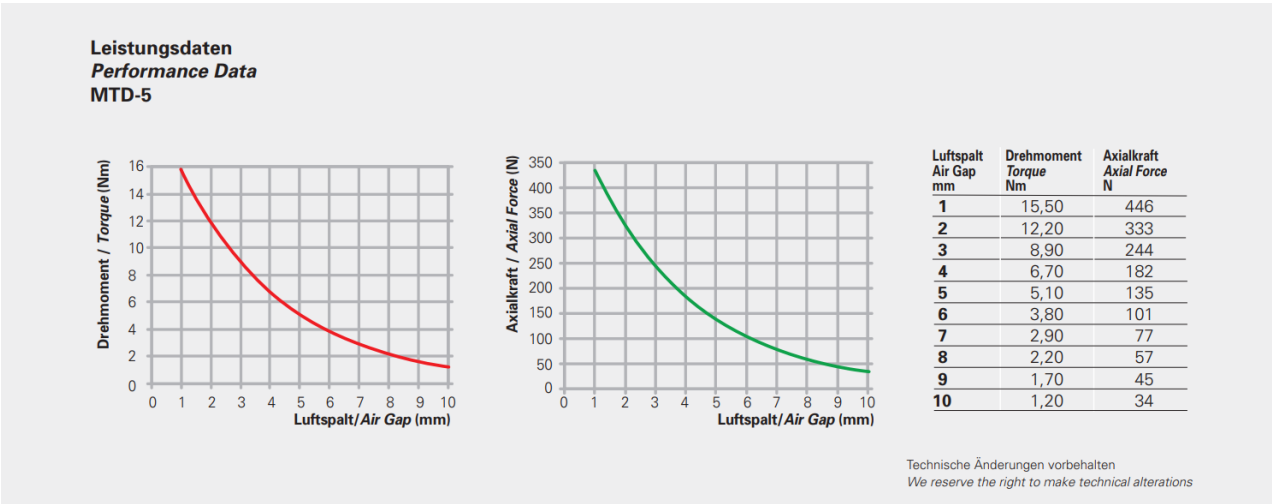


Figure A.3: Mobac MTD-5 performance data

A.3. Components list

Category	Amount	Part	Material	Mass [kg]	Total mass [kg]
Support rods	6	Support Rods	1.4021 (X20Cr13)	0.43	2.60
	4	Support Rod Motor $L=25\text{mm}$	1.4021 (X20Cr13)	0.53	2.12
	3	Rods Base	1.4021 (X20Cr13)	0.25	0.74
Support plates	1	Support Base	AL7075-T6	5.90	5.90
	1	Support Motor	1.4021 (X20Cr13)	1.44	1.44
	1	Support Plate Test Section	RVS316 Stainless steel	5.40	5.40
	1	Support Motor Frame	RVS316 Stainless steel	4.50	4.50
Cylinder	1	Shaft	1.4021 (X20Cr13)	1.64	1.64
	1	Bottom End Plate $D_e=150\text{mm}$	7075-T6 (SN)	0.03	0.03
	1	Top End Plate $D_e=150\text{mm}$	7075-T6 (SN)	0.24	0.24
	1	Bottom End Plate $D_e=80\text{mm}$	7075-T6 (SN)	0.03	0.03
	1	Top End Plate $D_e=80\text{mm}$	7075-T6 (SN)	0.18	0.18
	1	Bottom End Plate $D_e=50\text{mm}$	7075-T6 (SN)	0.03	0.03
	1	Top End Plate $D_e=50\text{mm}$	7075-T6 (SN)	0.12	0.12
	1	Rotating Outer Cylinder $D=50\text{mm}$	6061-T6 (SS)	0.56	0.56
	1	Static Cylinder	1.4021 (X20Cr13)	0.40	0.40
	1	Cylinder Base	7075-T6 (SN)	2.69	2.69
	1	Cylinder base insert	Delrin 2700 NC010	0.01	0.01
	1	Cap Top End Plate 3D Print	-	-	-
	1	Balancing Ring	1.4021 (X20Cr13)	0.08	0.08
Bearing block assembly	3	CS Stage Shaft Upper Bearing (SKF 6206-2Z)	-	0.20	0.60
	3	CS Stage Shaft Lower Bearing (SKF 6205-2Z)	-	0.13	0.39
	2	CS Nut (SKF KM5)	-	0.03	0.06
	5	CS Washer (SKF MB 5)	-	0.01	0.03
	1	CS Stage Shaft Sleeve	AISI 4140	0.04	0.04
	1	CS Bearing Lower Cap	AISI 4140	0.08	0.08
	1	CS Bearing Plate Top	AISI 4140	6.27	6.27
	1	CS Stage Shaft	1.4021 (X20Cr13)	0.62	0.62
Motor & coupler	1	Magnetic Coupler Bore 20 mm (Mobac MTD-5)	-	0.07	0.07
	1	Magnetic Coupler Bore 19 mm (Mobac MTD-5)	-	0.07	0.07
	1	Motor (Festo Servo motor EMMT-AS-80-HKR-HS-RMB)	-	4.80	4.80
Shock absorbers	4	Shock Absorber (Paulstra 2052797)	-	0.04	0.17
TOTAL MASS					41.90

Table A.1: Bill of materials for scaled rotor sail assembly with mass breakdown.

Results pressure rake measurements

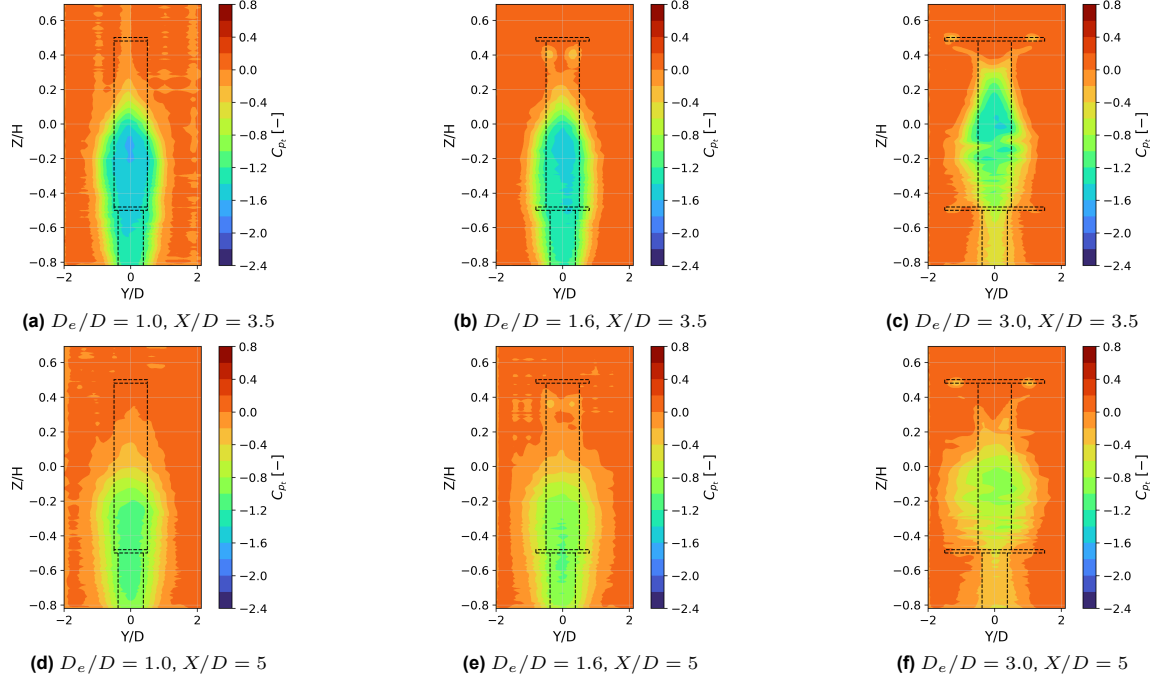


Figure B.1: Contour plot of time-averaged total pressure coefficient, $C_{p,t}$, for three spin ratios and two streamwise distances for $\alpha = 0$ with $Re = 1.5 \times 10^4$.

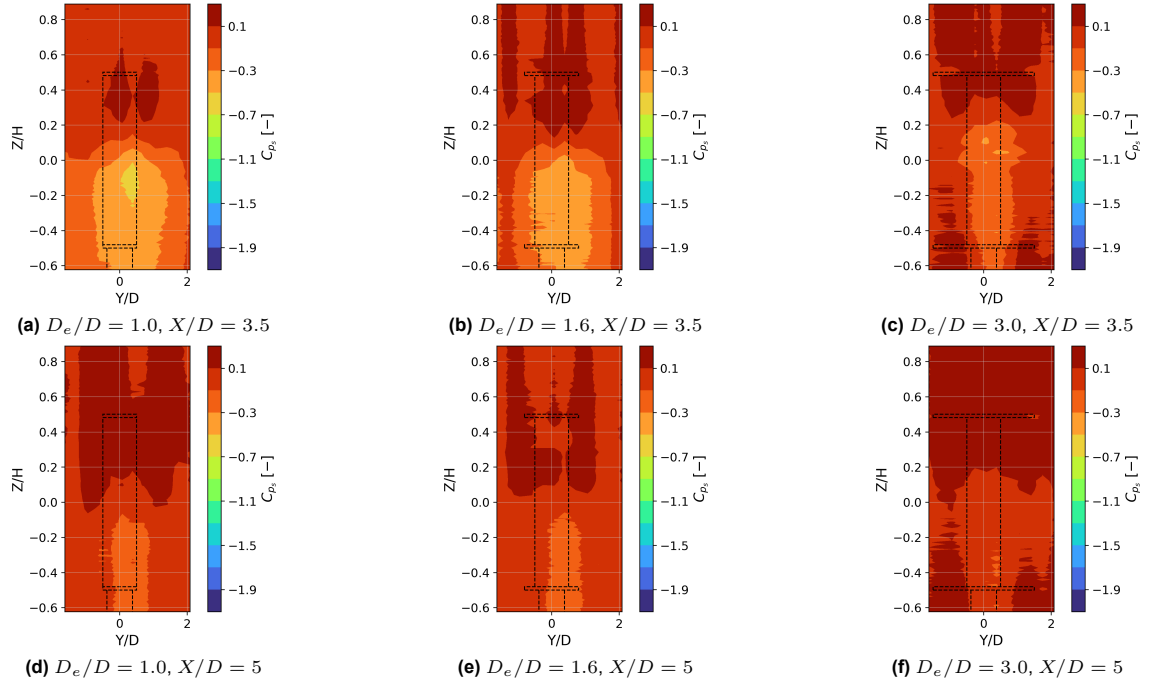


Figure B.2: Contour plot of time-averaged static pressure coefficient, $C_{p,s}$, for three spin ratios and two streamwise distances for $\alpha = 0$ with $Re = 1.5 \times 10^4$.

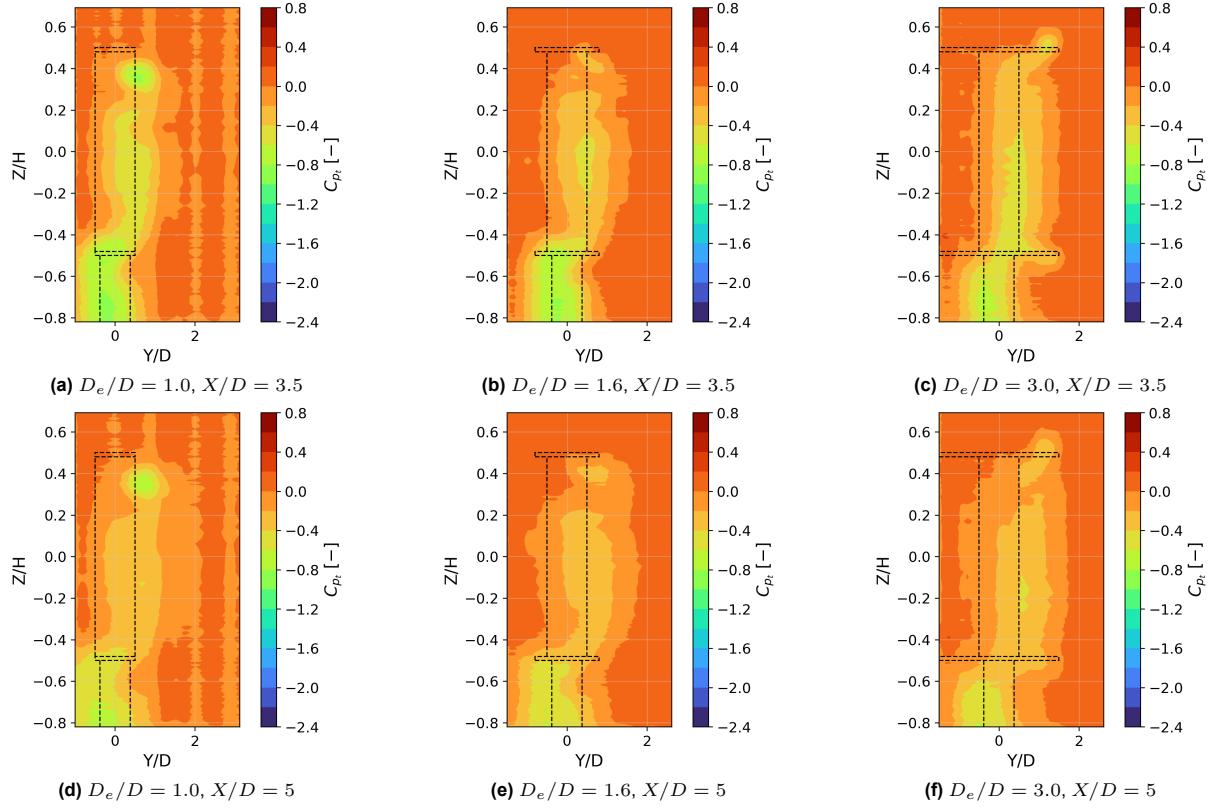


Figure B.3: Contour plot of time-averaged total pressure coefficient, $C_{p,t}$, for three spin ratios and two streamwise distances for $\alpha = 1$ with $Re = 1.5 \times 10^4$.

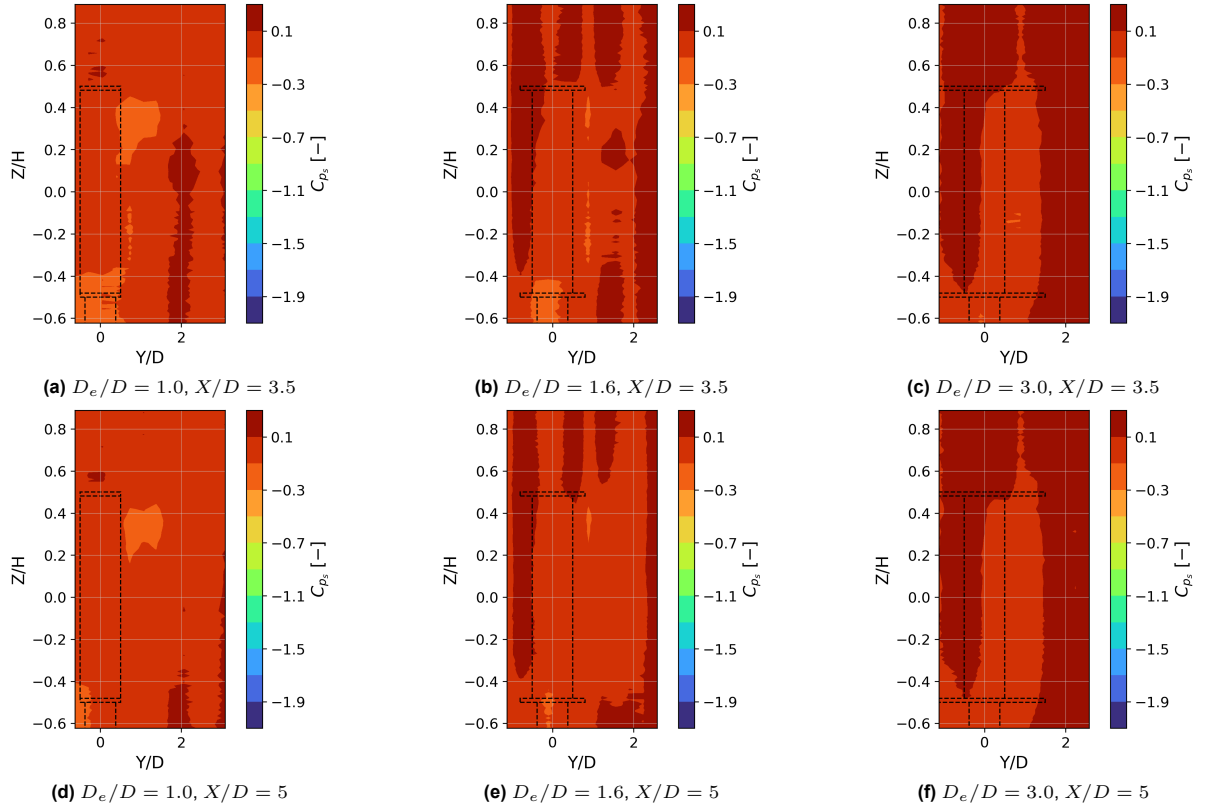


Figure B.4: Contour plot of time-averaged static pressure coefficient, $C_{p,s}$, for three spin ratios and two streamwise distances for $\alpha = 1$ with $Re = 1.5 \times 10^4$.

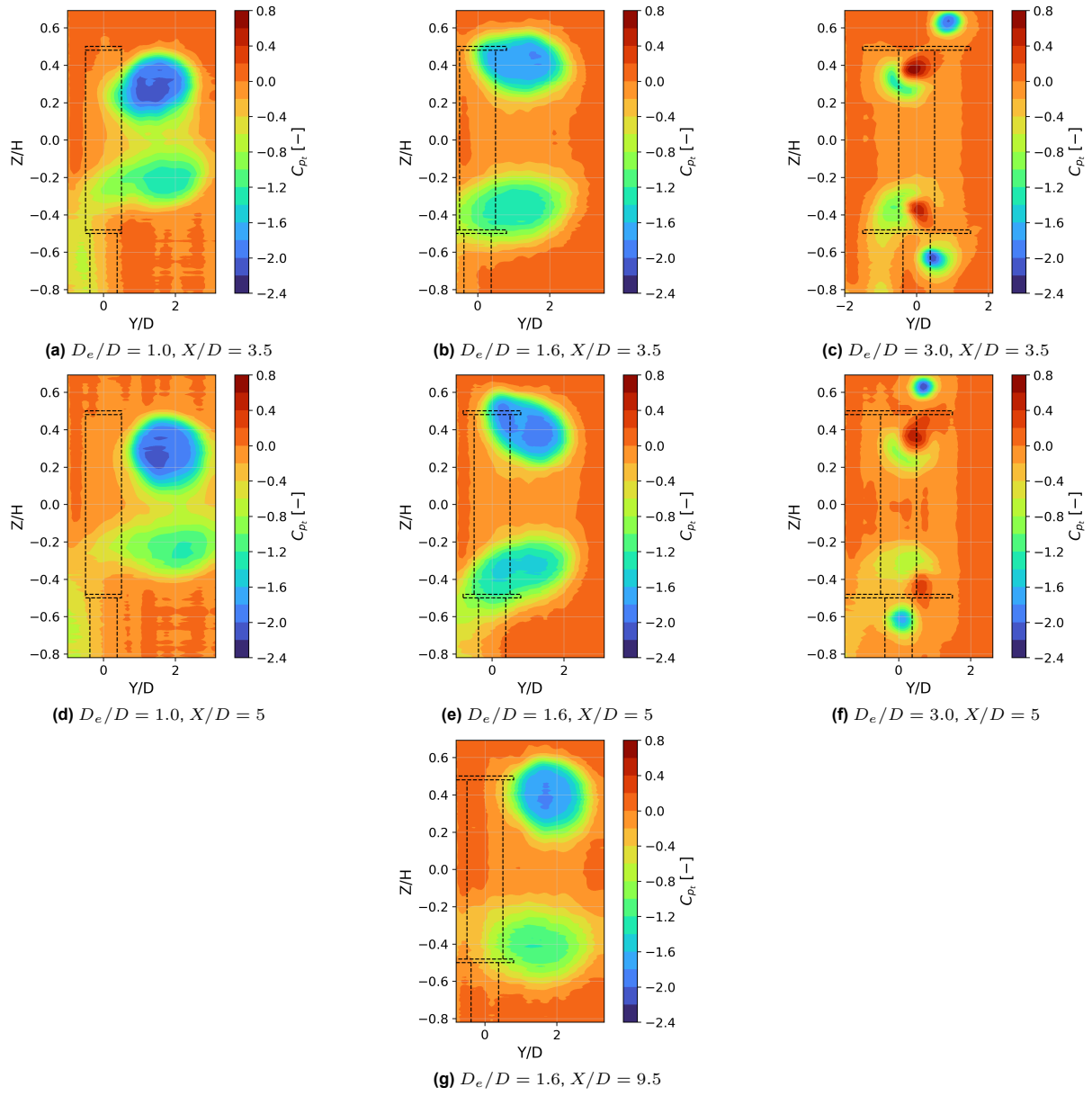


Figure B.5: Contour plot of time-averaged total pressure coefficient, $C_{p,t}$, for three spin ratios and three streamwise distances for $\alpha = 3$.

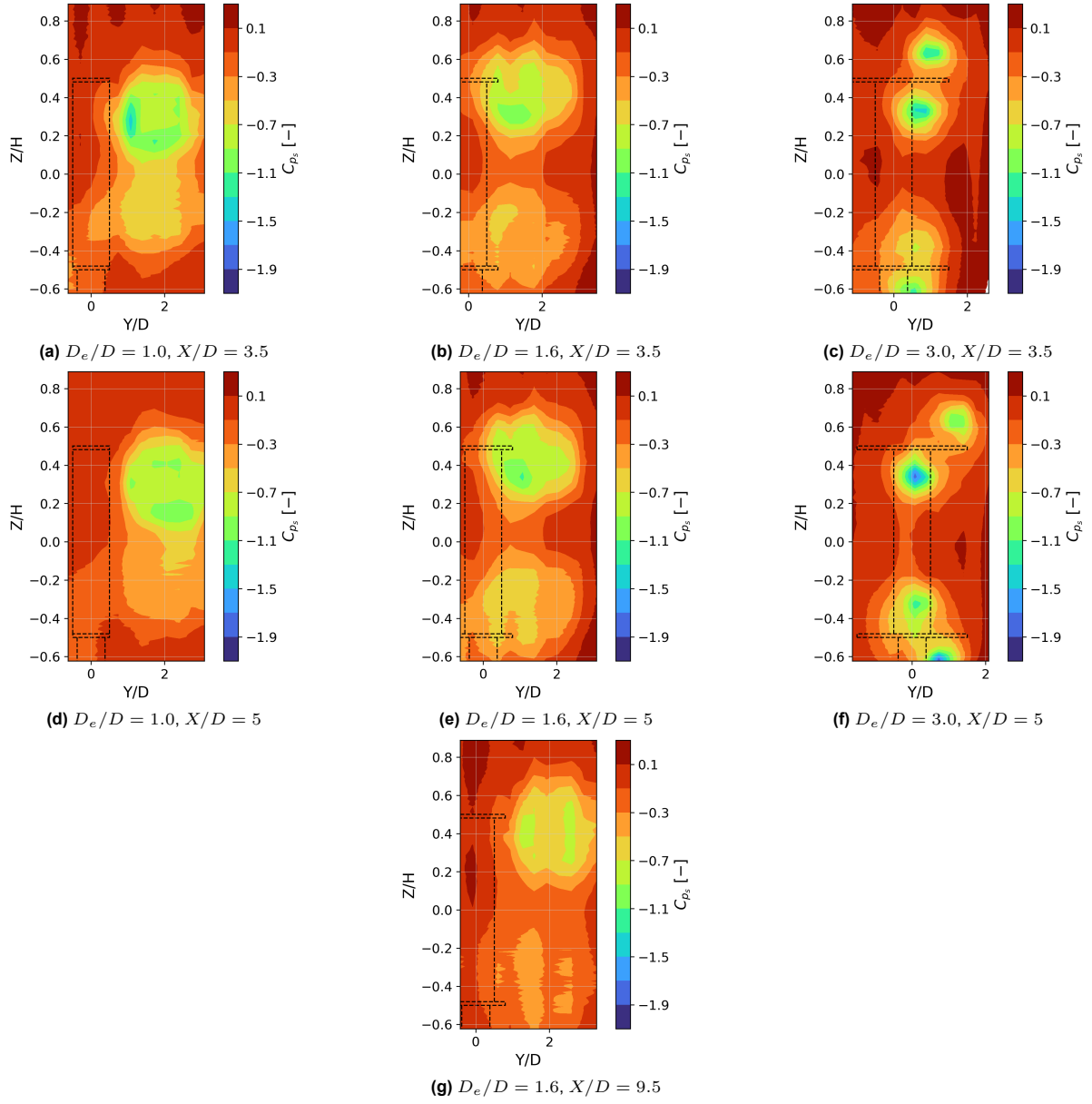


Figure B.6: Contour plot of time-averaged static pressure coefficient, $C_{p,s}$, for three spin ratios and three streamwise distances for $\alpha = 3$.

C

Results 7-hole probe measurements

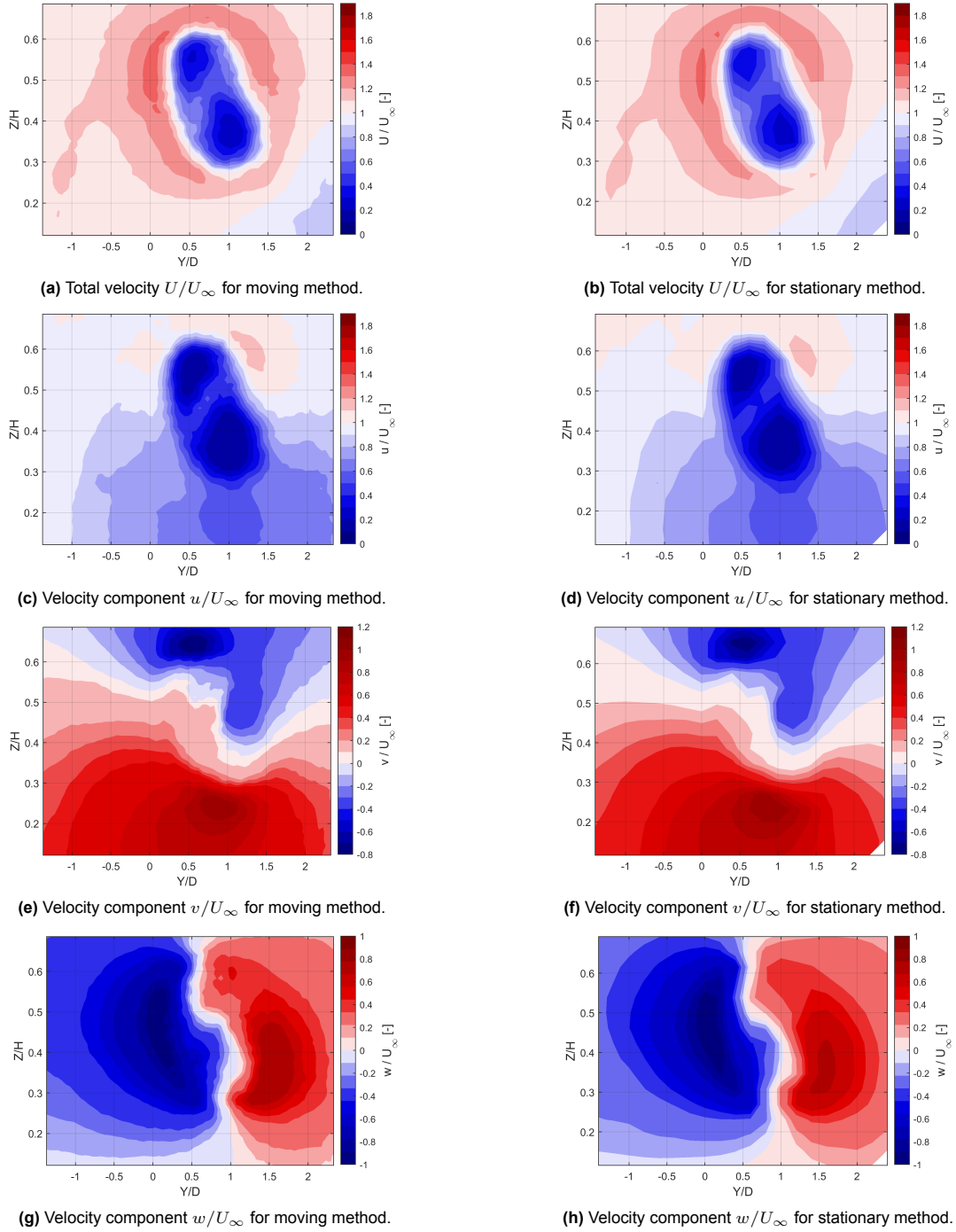


Figure C.1: Contourplots for non-dimensionalized velocities for $D_e/D = 1.6$, $\alpha = 3$, $X/D = 2.2$ and $Re = 1.5 \times 10^4$ comparing results from the 7-hole probe moving method (left) and stationary method (right).

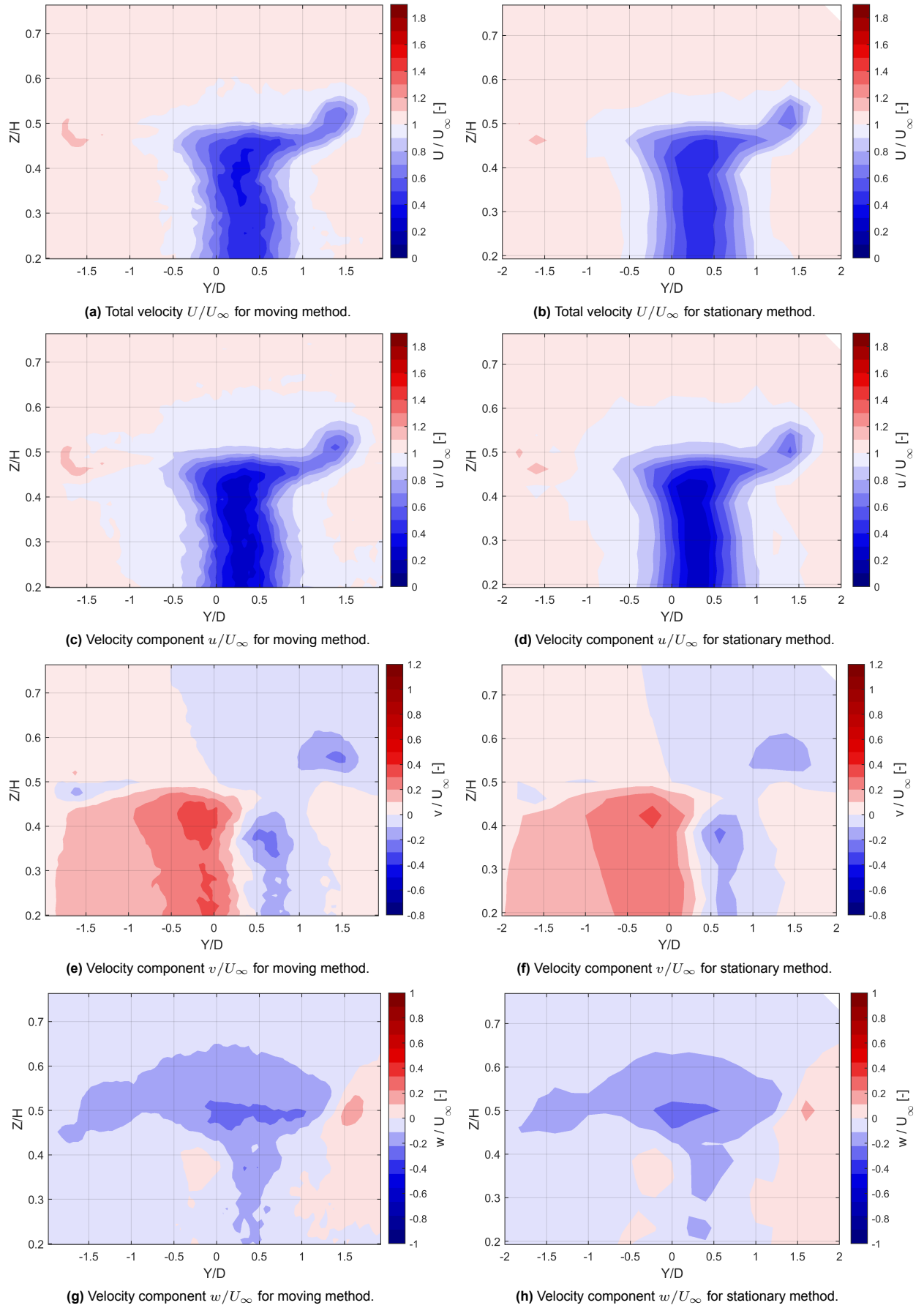


Figure C.2: Contourplots for non-dimensionalized velocities for $D_e/D = 3.0$, $\alpha = 1$, $X/D = 2.2$ and $Re = 1.5 \times 10^4$ comparing results from the 7-hole probe moving method (left) and stationary method (right).

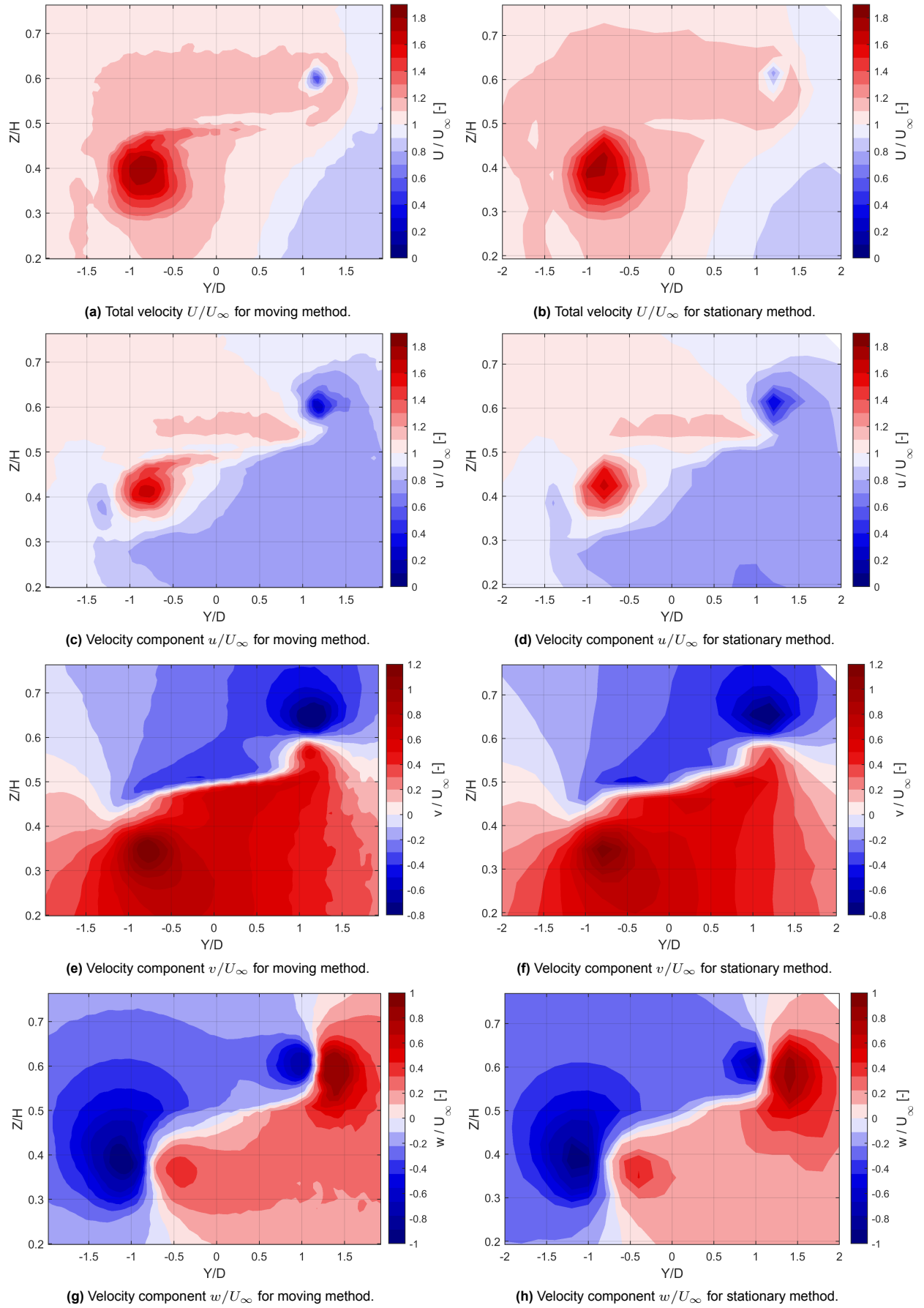


Figure C.3: Contourplots for non-dimensionalized velocities for $D_e/D = 3.0$, $\alpha = 3$, $X/D = 2.2$ and $Re = 1.5 \times 10^4$ comparing results from the 7-hole probe moving method (left) and stationary method (right).

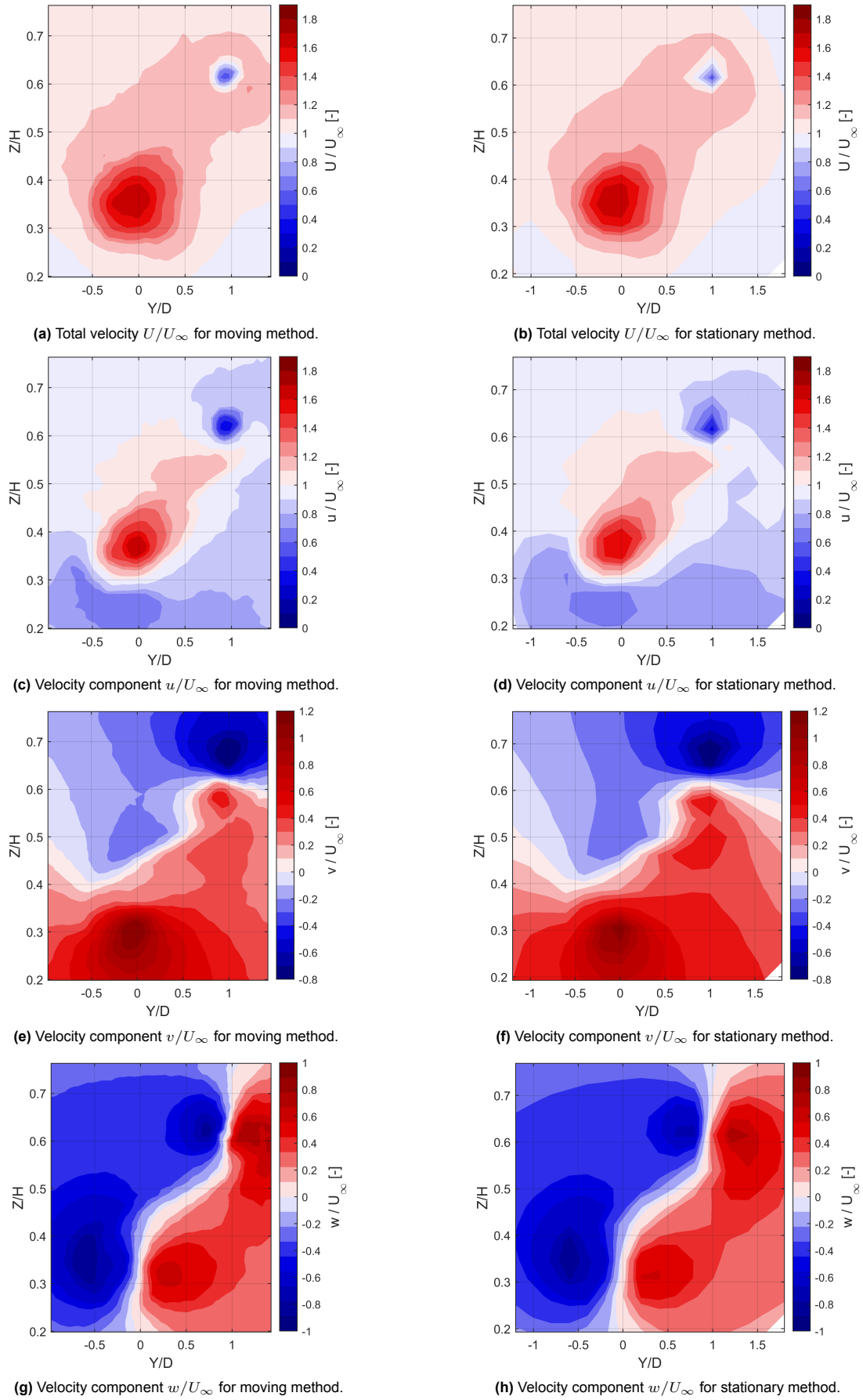


Figure C.4: Contourplots for non-dimensionalized velocities for $D_e/D = 3.0$, $\alpha = 3$, $X/D = 3.7$ and $Re = 1.5 \times 10^4$ comparing results from the 7-hole probe moving method (left) and stationary method (right).

Validation 7-hole probe measurements

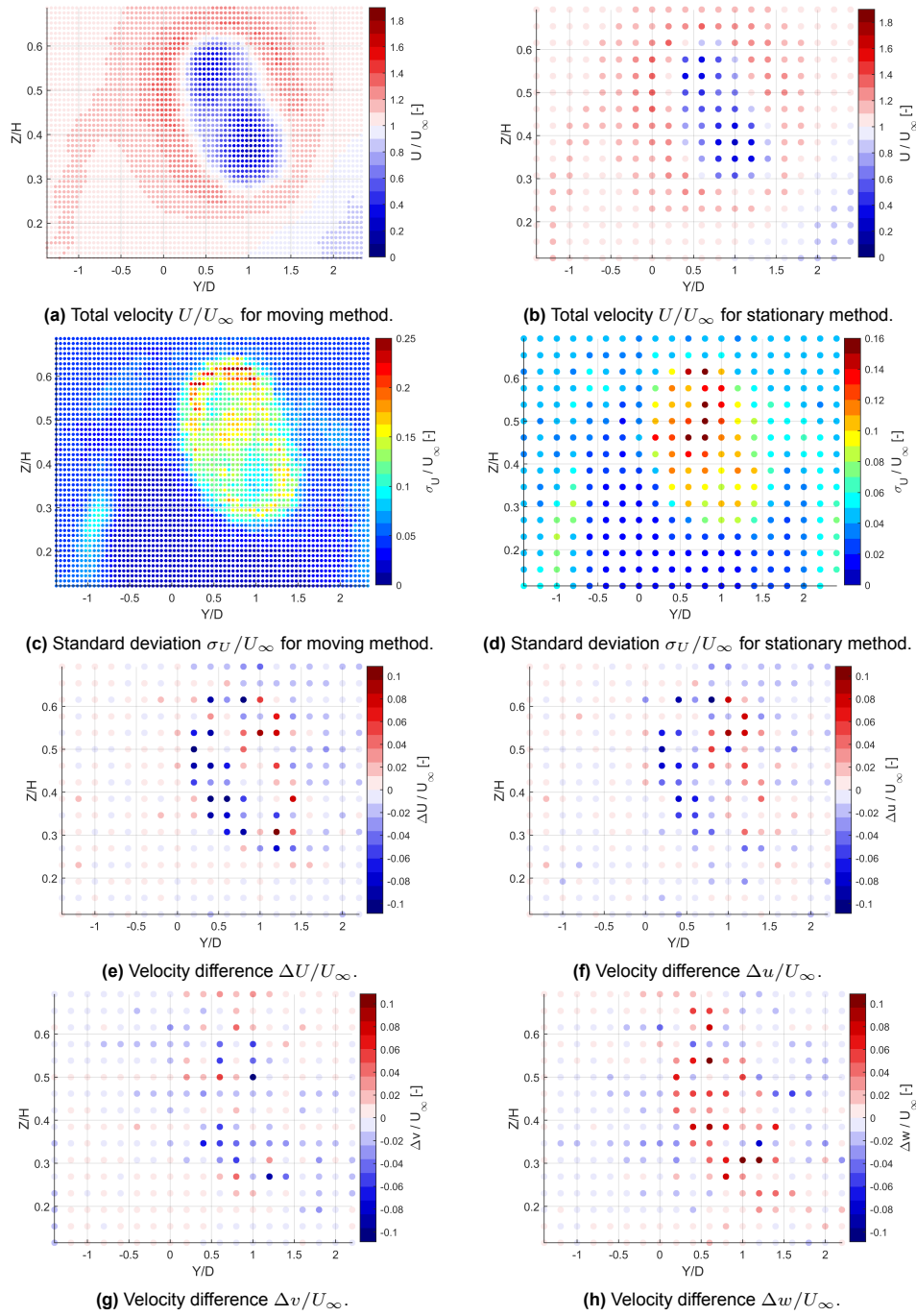


Figure D.1: Results for $D_e/D = 1.6$, $\alpha = 3$, $X/D = 2.2$, and $Re = 1.5 \times 10^4$: (a, b) non-dimensionalized total velocity U/U_∞ ; (c, d) standard deviation σ_U/U_∞ for the grid points measured with the moving and stationary 7-hole probe methods; and (e–h) velocity differences between the methods: $\Delta U/U_\infty$, $\Delta u/U_\infty$, $\Delta v/U_\infty$, and $\Delta w/U_\infty$, respectively.

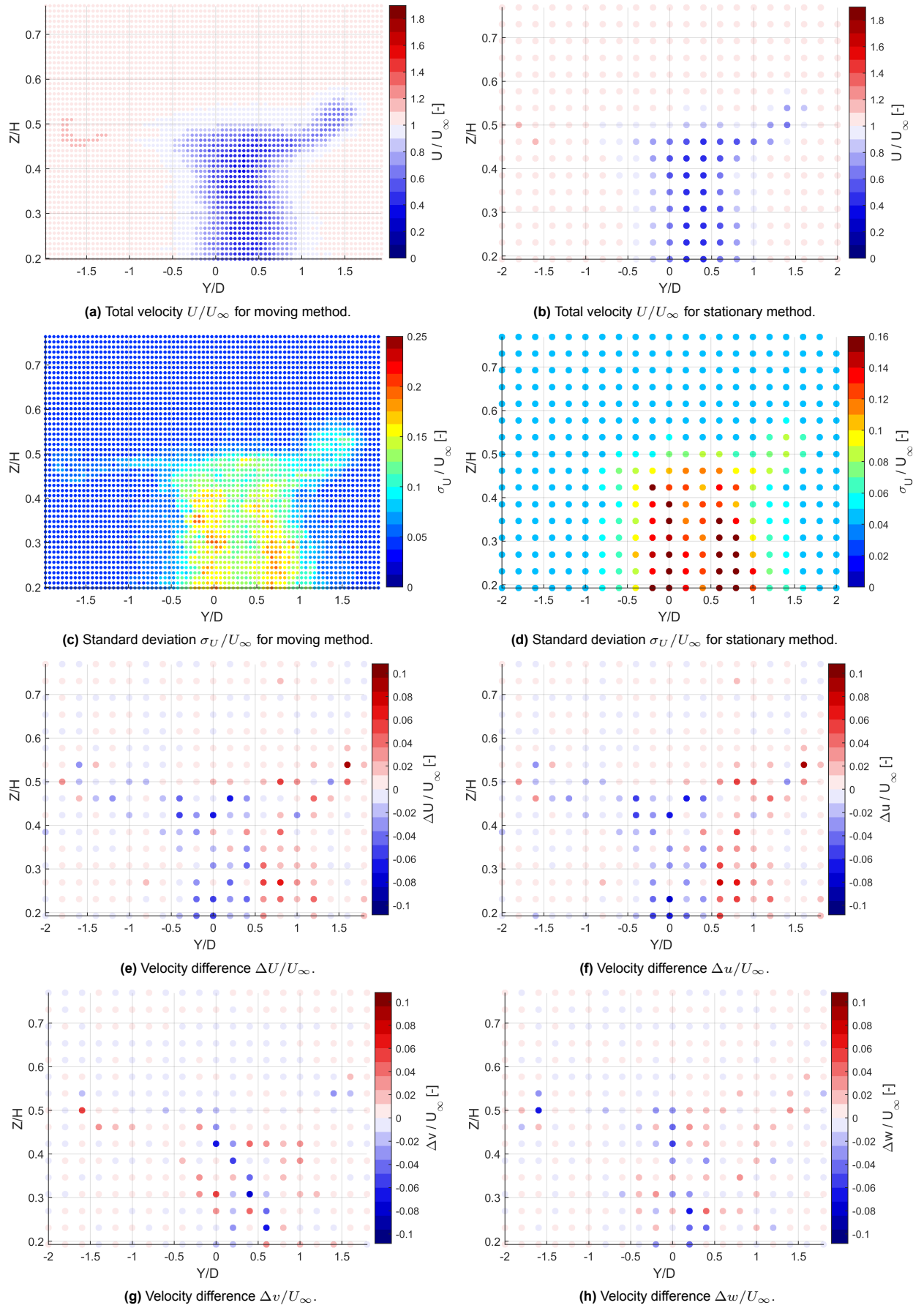


Figure D.2: Results for $D_e/D = 3.0$, $\alpha = 1$, $X/D = 2.2$, and $Re = 1.5 \times 10^4$: (a, b) non-dimensionalized total velocity U/U_∞ ; (c, d) standard deviation σ_U/U_∞ for the grid points measured with the moving and stationary 7-hole probe methods; and (e–h) velocity differences between the methods: $\Delta U/U_\infty$, $\Delta u/U_\infty$, $\Delta v/U_\infty$, and $\Delta w/U_\infty$, respectively.

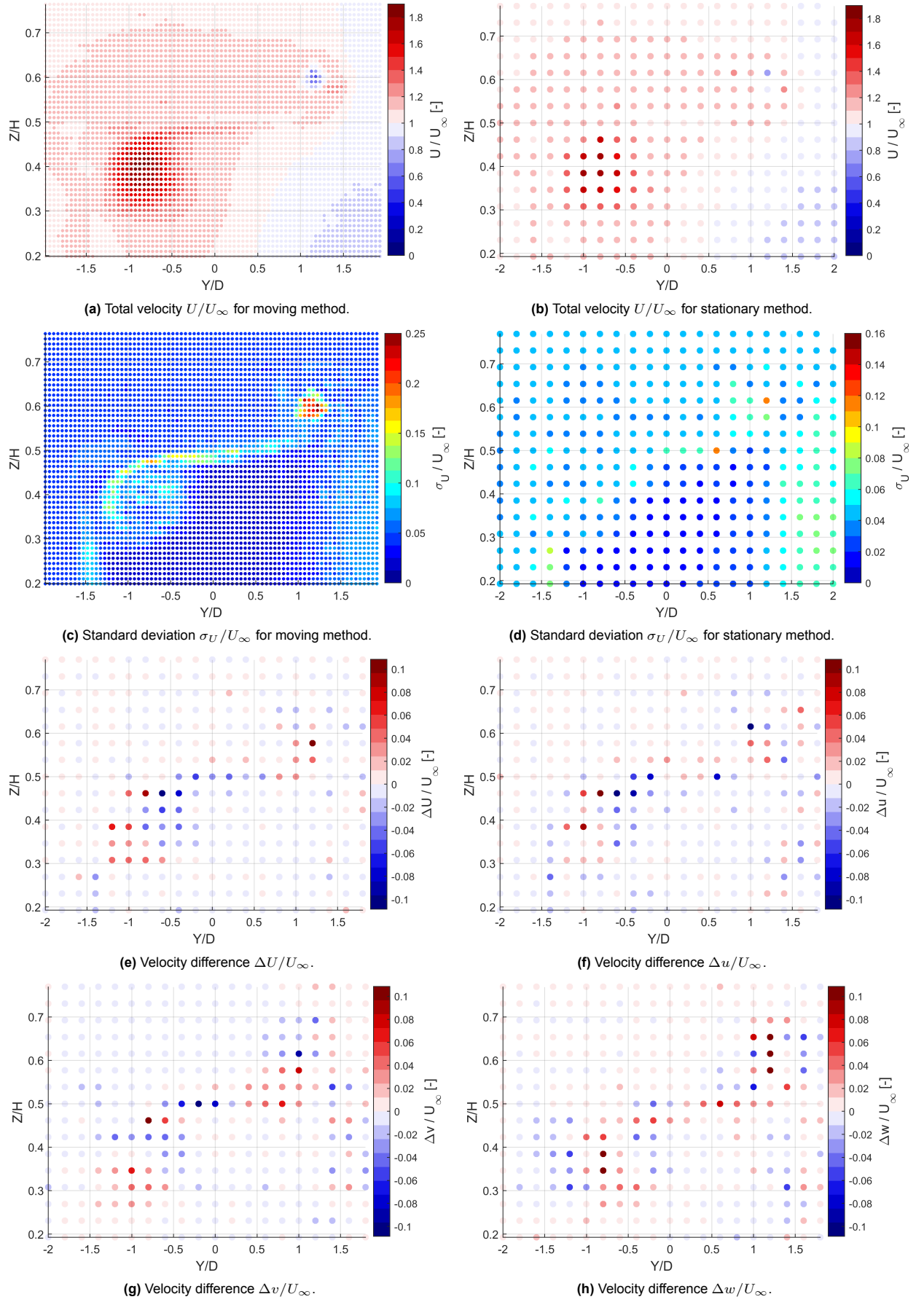


Figure D.3: Results for $D_e/D = 3.0$, $\alpha = 3$, $X/D = 2.2$, and $Re = 1.5 \times 10^4$: (a, b) non-dimensionalized total velocity U/U_∞ ; (c, d) standard deviation σ_U/U_∞ for the grid points measured with the moving and stationary 7-hole probe methods; and (e–h) velocity differences between the methods: $\Delta U/U_\infty$, $\Delta u/U_\infty$, $\Delta v/U_\infty$, and $\Delta w/U_\infty$, respectively.

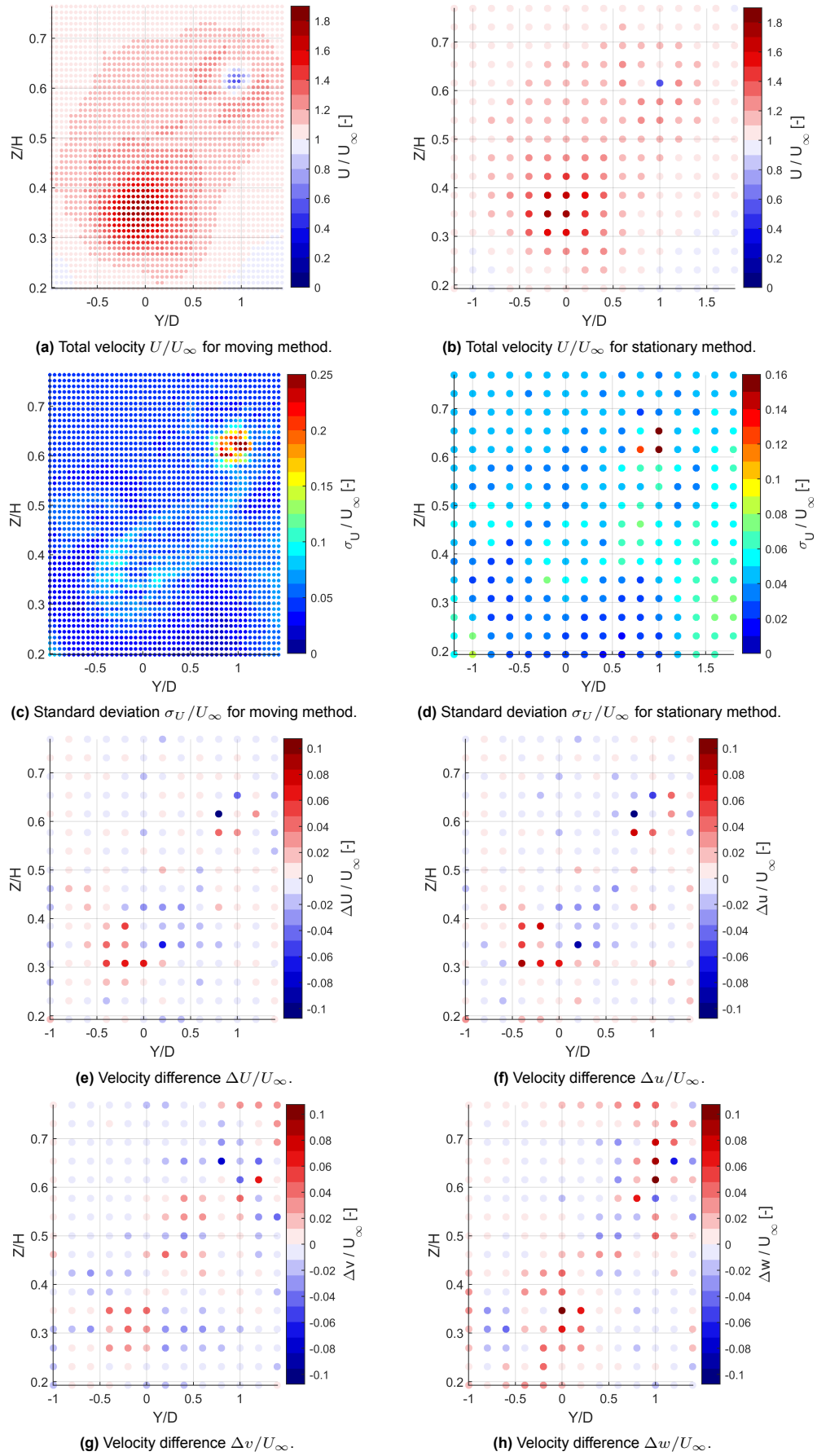


Figure D.4: Results for $D_e/D = 3.0$, $\alpha = 3$, $X/D = 3.7$, and $Re = 1.5 \times 10^4$: (a, b) non-dimensionalized total velocity U/U_∞ ; (c, d) standard deviation σ_U/U_∞ for the grid points measured with the moving and stationary 7-hole probe methods; and (e–h) velocity differences between the methods: $\Delta U/U_\infty$, $\Delta u/U_\infty$, $\Delta v/U_\infty$, and $\Delta w/U_\infty$, respectively.