Numerical Simulation of a Cross Flow Marine Hydrokinetic Turbine

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MSME Thesis Presentation

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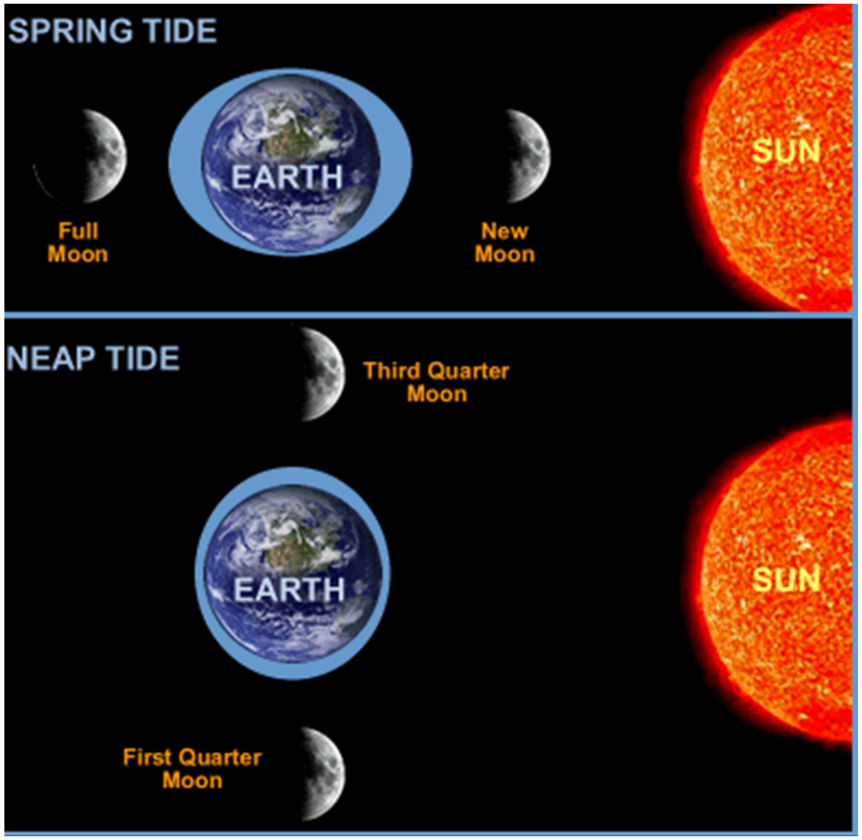
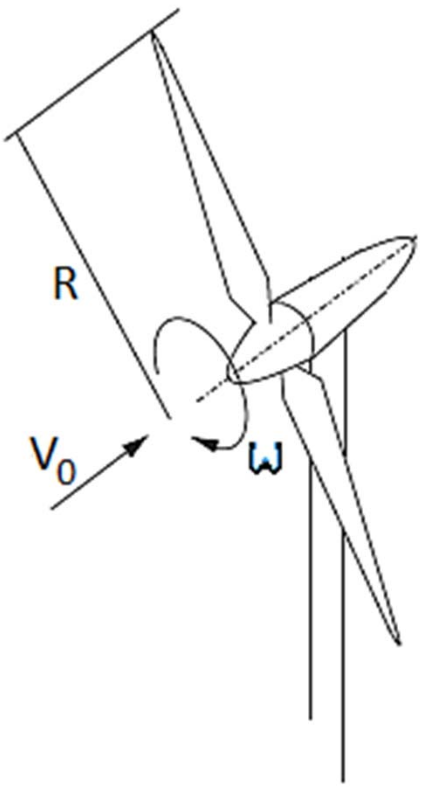
Tidal Energy

- Tidal energy resource realized
- Renewable
- Clean
- Predictable
- Many similarities to wind energy

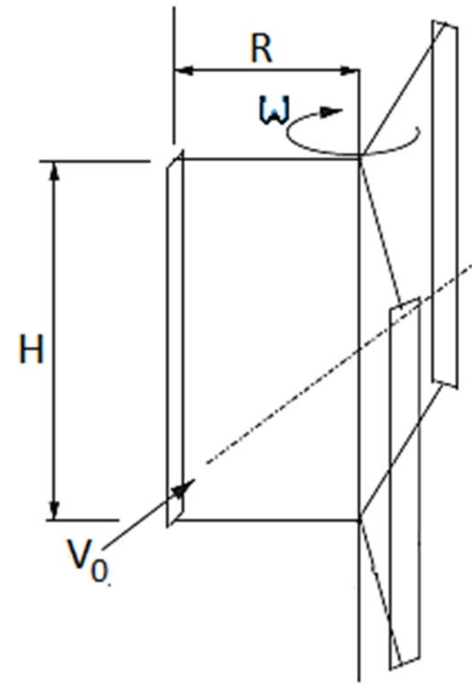


Turbine Classifications

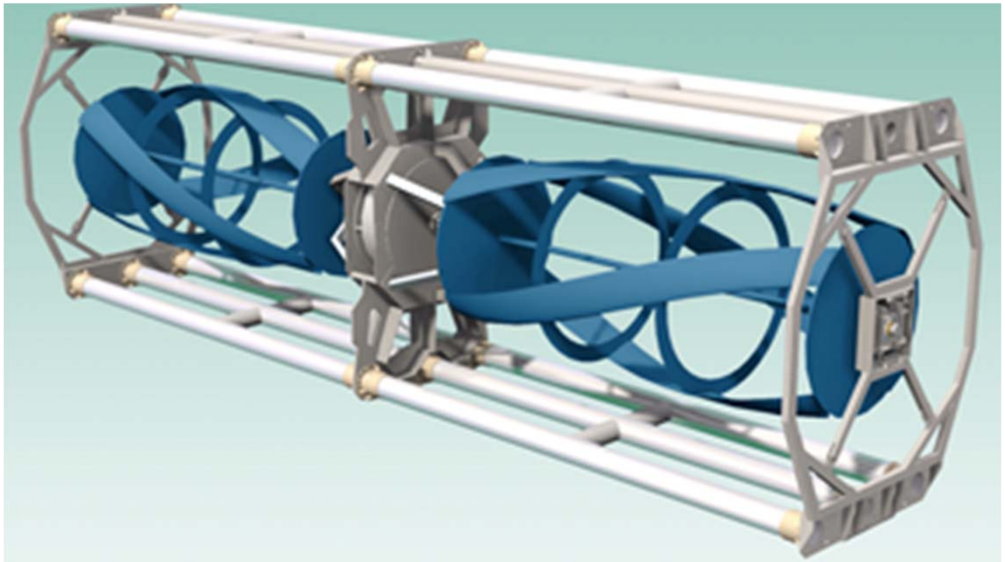
Axial Flow Turbines

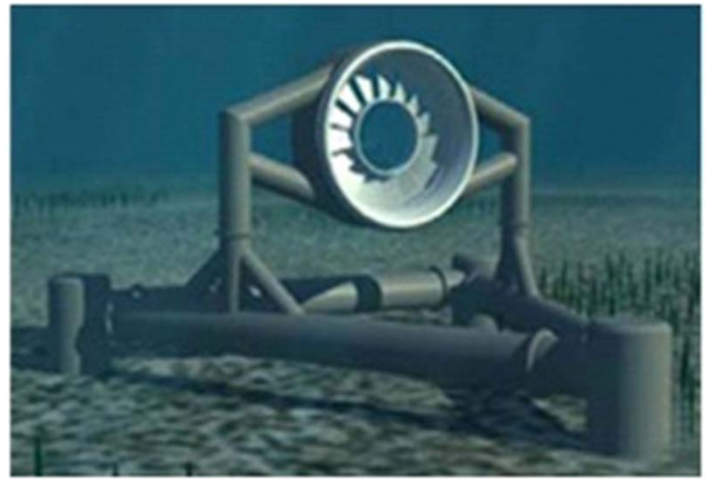


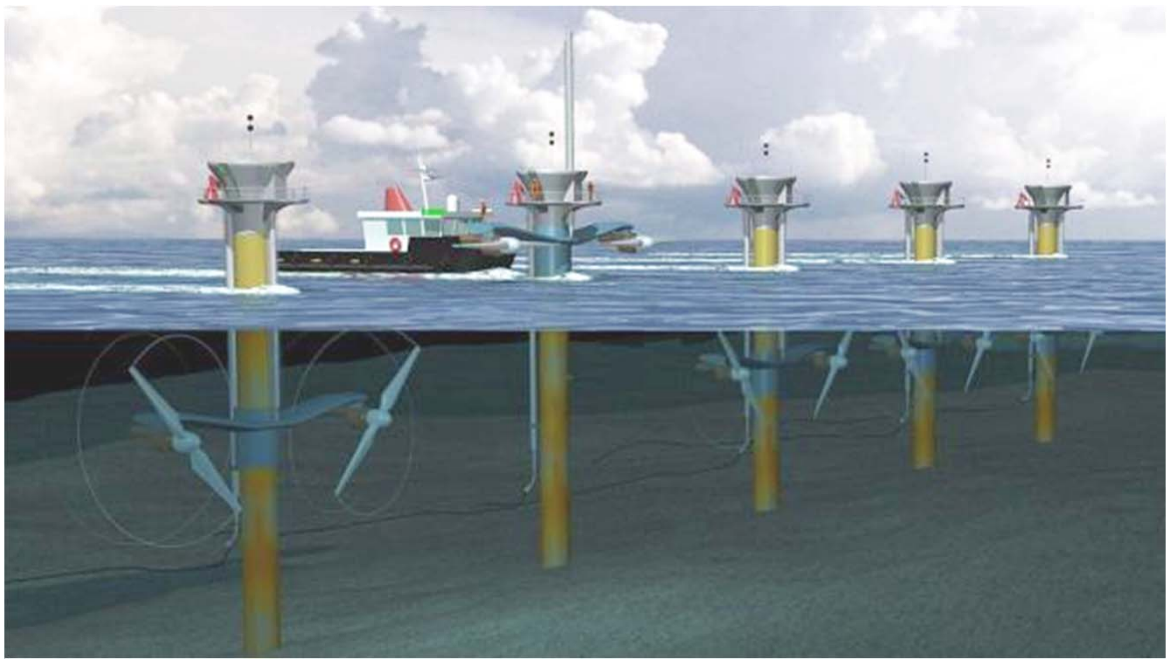
Cross Flow Turbines



Turbine Design Concepts

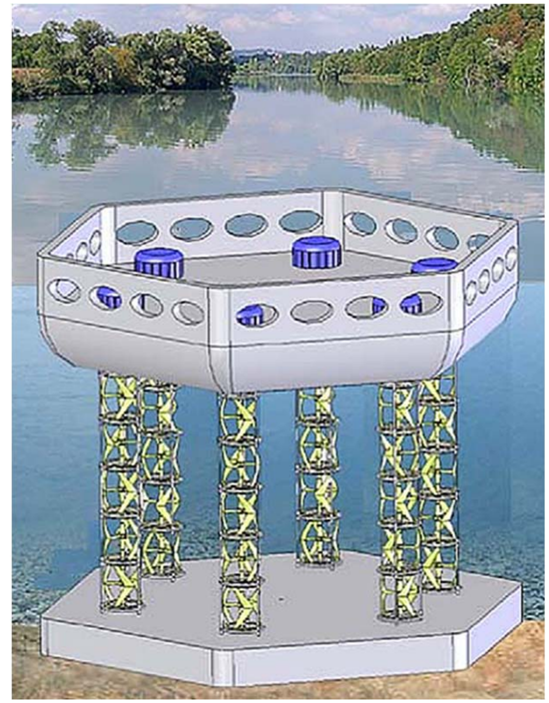


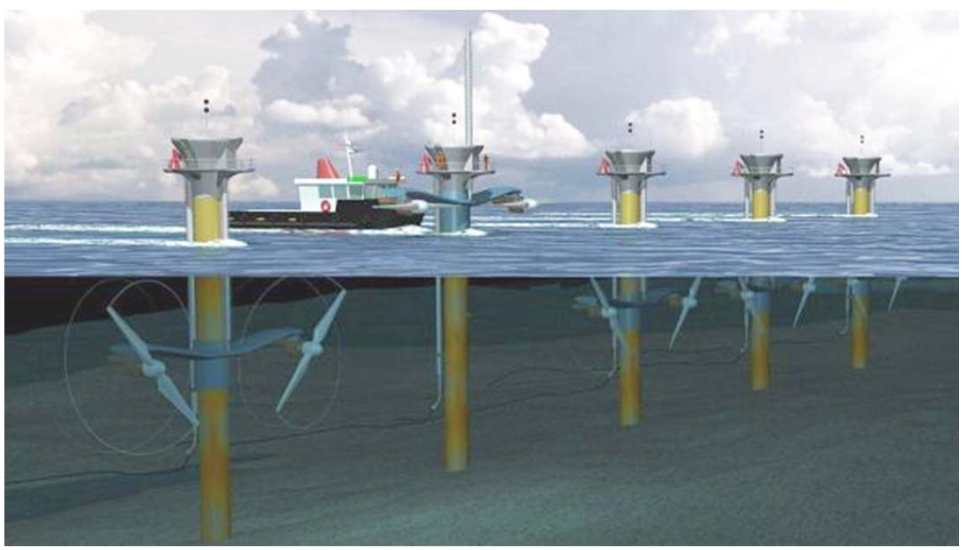


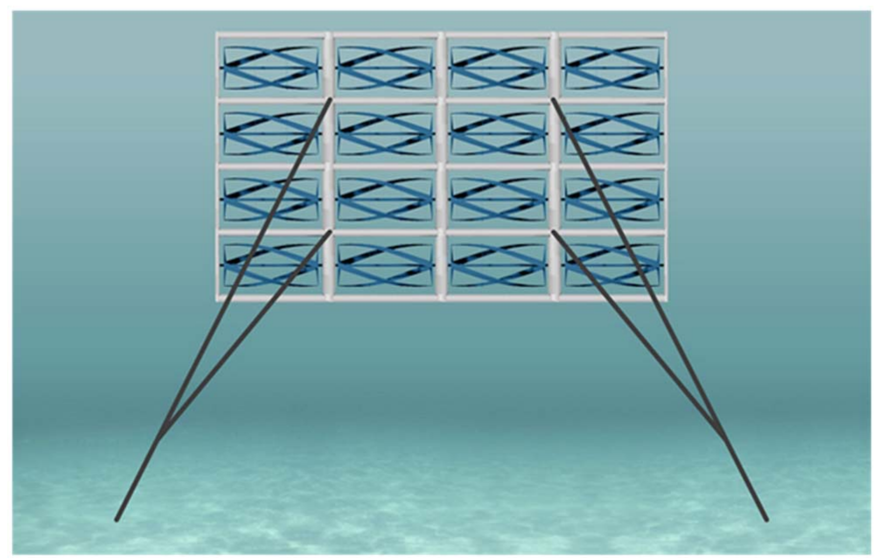


Cross Flow Turbine Advantages

- High energy density typically found in narrow constricted channels
- Packing critical to efficiency and economic feasibility
- Cross flow turbines can be stacked, efficiently utilizing limited space
- Vertical axis: works in any direction of flow







Motivation and Goals

- Recently realized advantages have ignited interest in cross flow hydrokinetic turbines (CFHT)
- Significant gaps in understanding and modeling capabilities
- Benefits of numerical models
  - Turbine performance for a variety of parameters
  - Influence of turbine surroundings: stacking, mooring, supports
  - Environmental impacts
  - Larger scale turbine performance
- Goals:
  - Gain a better understanding of the CFHT flow dynamics
  - Develop a numerical methodology for CFHT
  - Ultimate goal of developing a computational tool to aid in turbine design and array installation process

Hydrodynamics of a Cross Flow Turbine

- Many differences from axial flow turbines
- Unsteady and largely three-dimensional
- Interference between shed vortices and blades
- Can reach very high angles of attack
- Rapidly changing angles of attack
- Dynamic stall behavior

Hydrodynamics of a Cross Flow Turbine

Power available in the flow:

$$P_0 = \frac{1}{2}\rho S_{ref} {V_0}^3$$

$$S_{ref} = 2RH$$

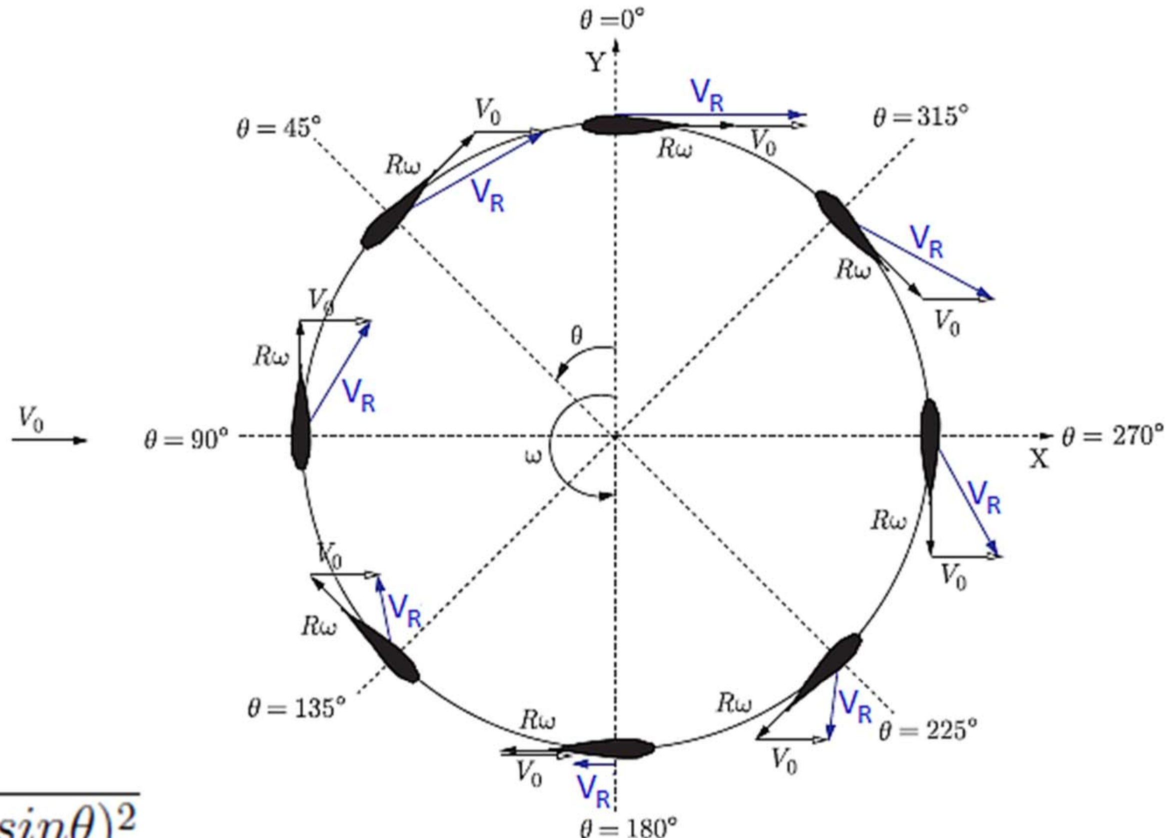
Free Stream Velocity: V_0

Tangential Velocity: $V_\theta = \omega R$

Relative Velocity:

$$V_R = \sqrt{(V_0 + V_\theta cos\theta)^2 + (V_\theta sin\theta)^2}$$

Tip Speed Ratio: $\lambda = \frac{\omega R}{V_0}$



Azimuthal Position: θ

Source: Antheaume

Hydrodynamics of a Cross Flow Turbine

Angle of Attack:

$$\alpha = tan^{-1}\left(\frac{sin\theta}{\lambda + cos\theta}\right)$$

Relative Reynolds Number:

$$Re_{rel} = \frac{\rho V_R c}{\mu}$$

Torque:

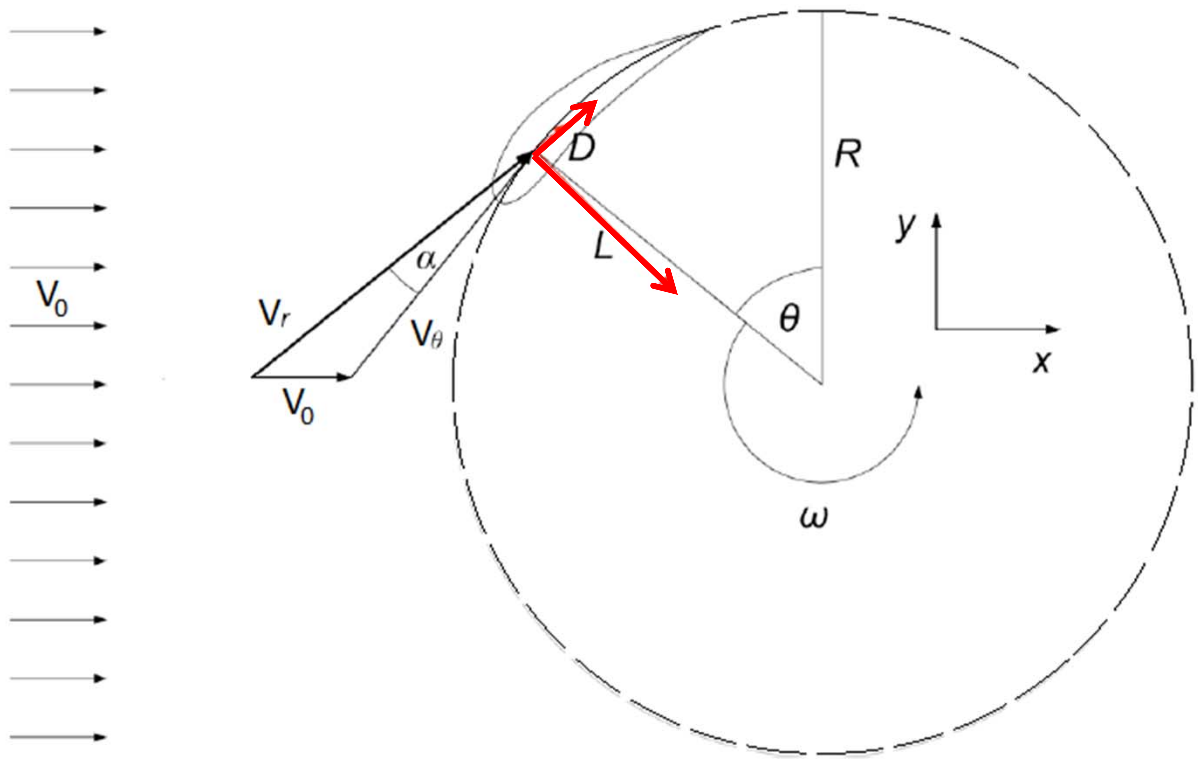
$$T = R(Lcos\alpha - Dsin\alpha)$$

$$C_T = \frac{T}{\frac{1}{2}\rho {V_0}^2 S_{ref} R}$$

Power:

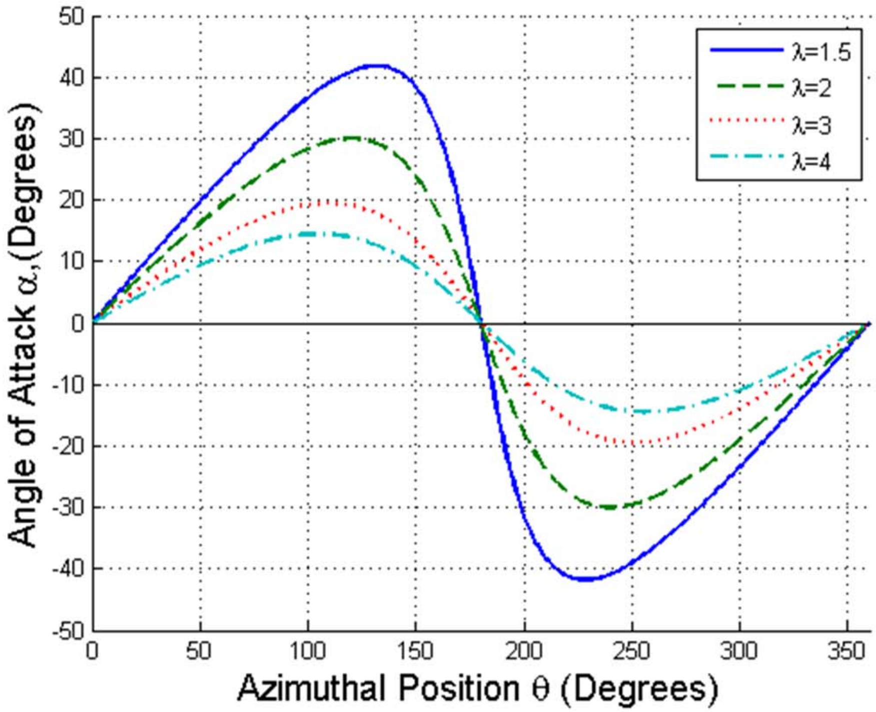
$$P = T\omega$$

$$C_P = \frac{P}{P_0} = \frac{P}{\frac{1}{2}\rho S_{ref} {V_0}^3} = \lambda C_T$$



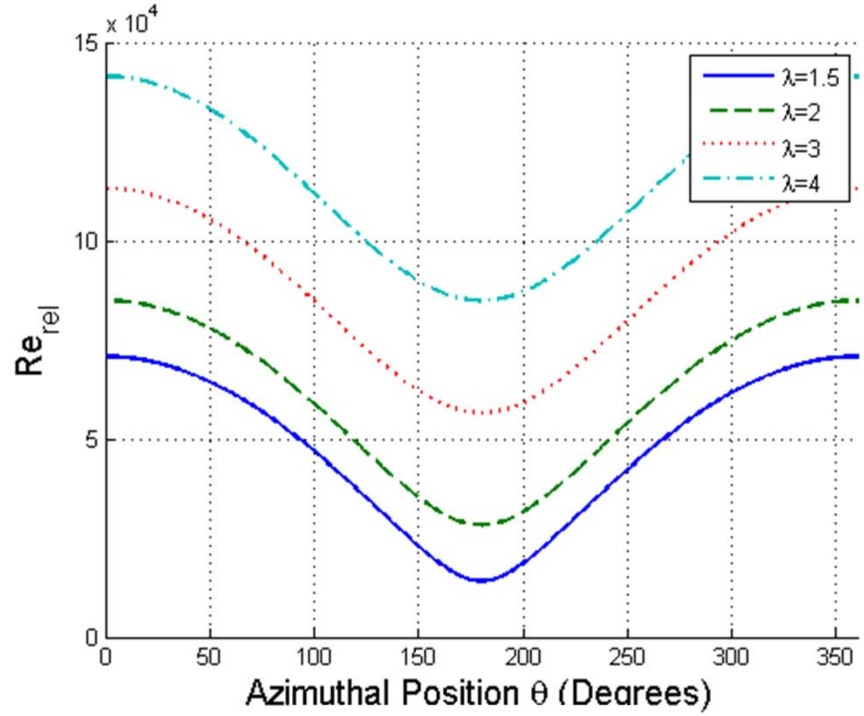
Hydrodynamics of a Cross Flow Turbine

Angle of Attack



$$\alpha = tan^{-1}\left(\frac{sin\theta}{\lambda + cos\theta}\right)$$

Relative Reynolds Number



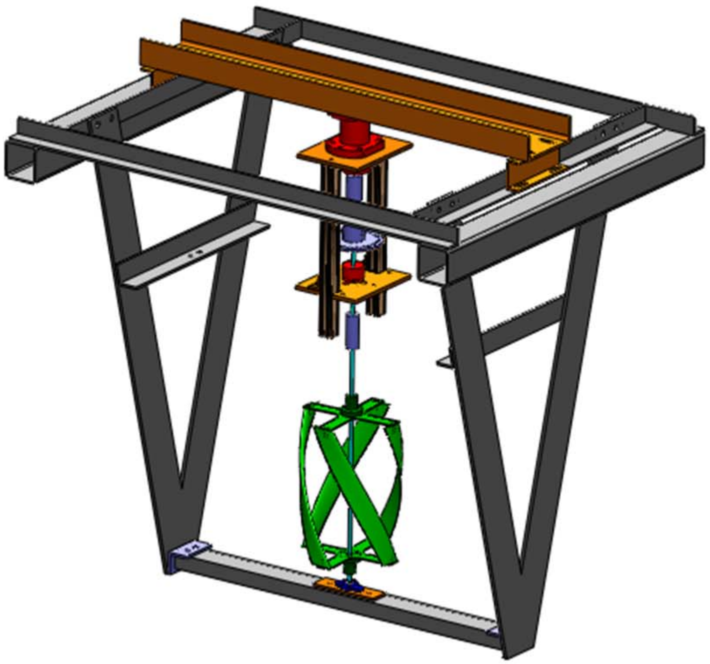
$$Re_{rel} = \frac{\rho V_R c}{\mu}$$

Helical Cross Flow Turbine

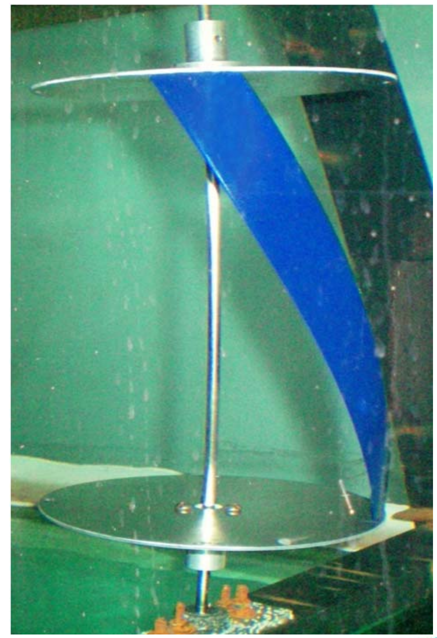
Micropower generation project as benchmark study:

Adam Niblick and UW Mech. Eng. Capstone Design Team

Blade Profile	NACA 0018
Number of Blades, N	4
Chord Length, c	0.040 m
Radius, R	0.086 m
Height, H	0.234 m



Source: Adam Niblick



Experiment/Simulation Parameters

Turbine and Channel Flow

- $\mathrm{Re}_{\mathrm{C}} = \frac{\rho V_{\infty} C}{\mu} = 28{,}000$
- $\text{Aspect Ratio} = \frac{H}{D} = 1.4$
- $\text{Solidity Ratio} = \frac{\mathrm{NC}}{2\pi R}$
  $= 0.075$ for 1 blade
  $= 0.3$ for 4 blades

Numerical Modeling

- CFD Software Fluent v12.0
- Reynolds-Average Navier-Stokes (RANS) equations

Reynolds Average Navier Stokes (RANS) Equations

Reynolds decomposition of velocity $\vec{U} = \overline{\vec{U}} + \vec{u\prime}$

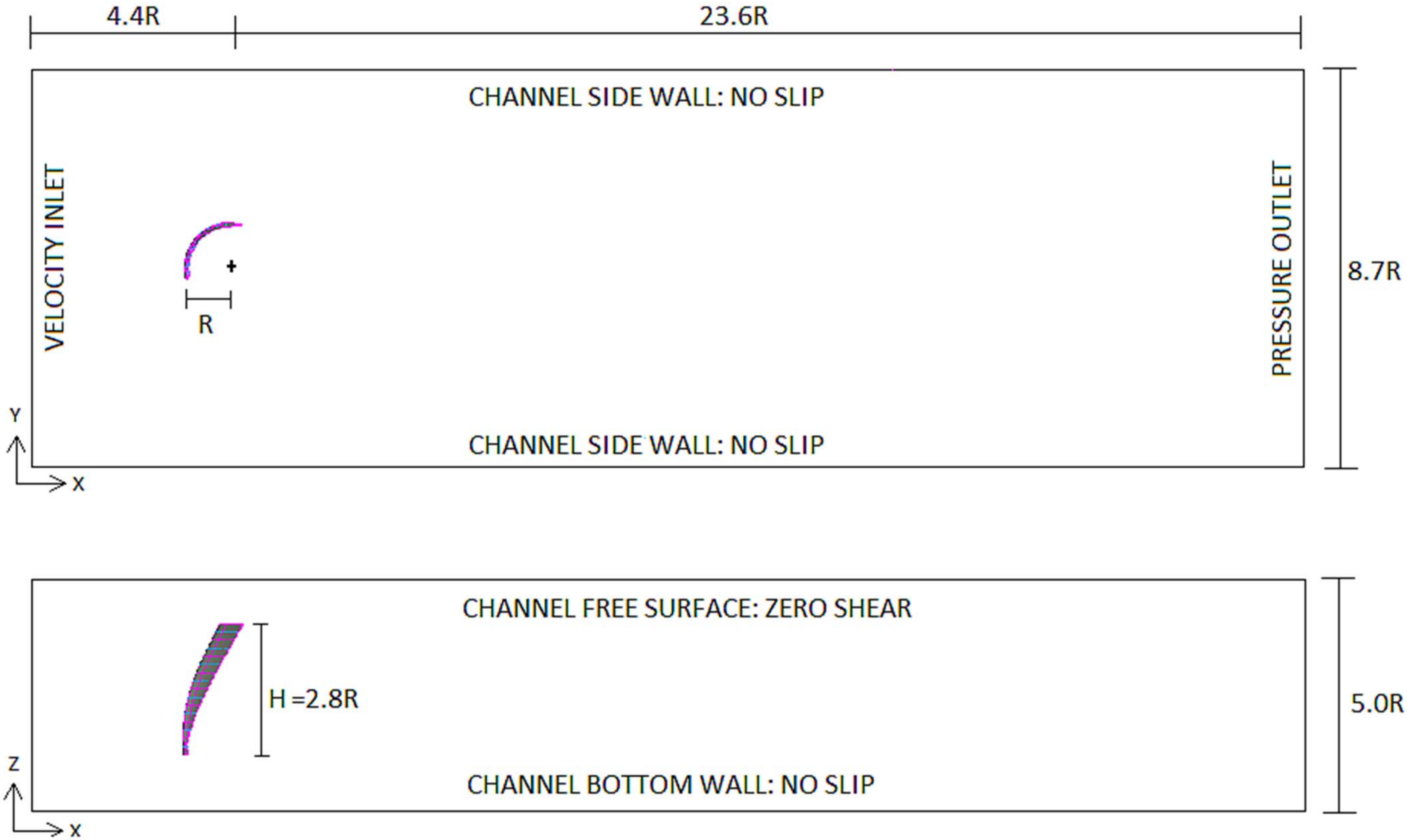
Reynolds decomposition of a scalar variable $\phi = \overline{\phi} + \phi\prime$

Conservation of mass $\nabla \cdot \overline{\vec{U}} = 0; \quad \nabla \cdot \vec{u\prime} = 0$

Conservation of momentum $\frac{D\overline{U_i}}{Dt} = \nu \nabla^2 \overline{U_i} - \frac{\partial \overline{u_i\prime u_j\prime}}{\partial x_j} - \frac{1}{\rho}\frac{\partial \overline{p}}{\partial x_i}$

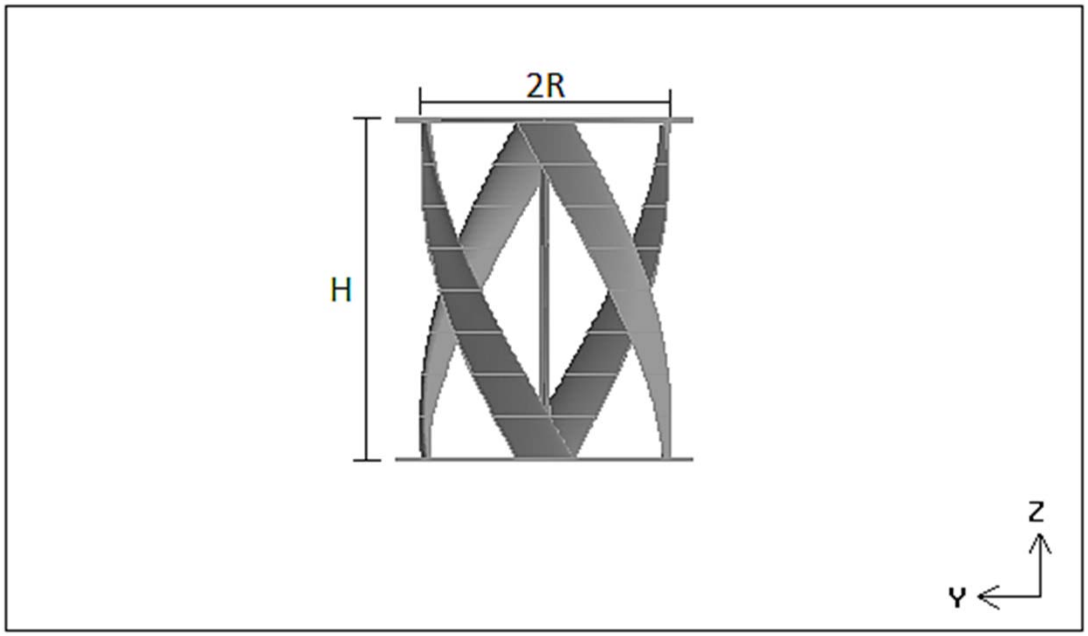
- Shear Stress Transport-kω turbulence closure model
- Default turbulence model coefficients
- Low-Reynolds number modeling

Channel Domain and Boundary Conditions



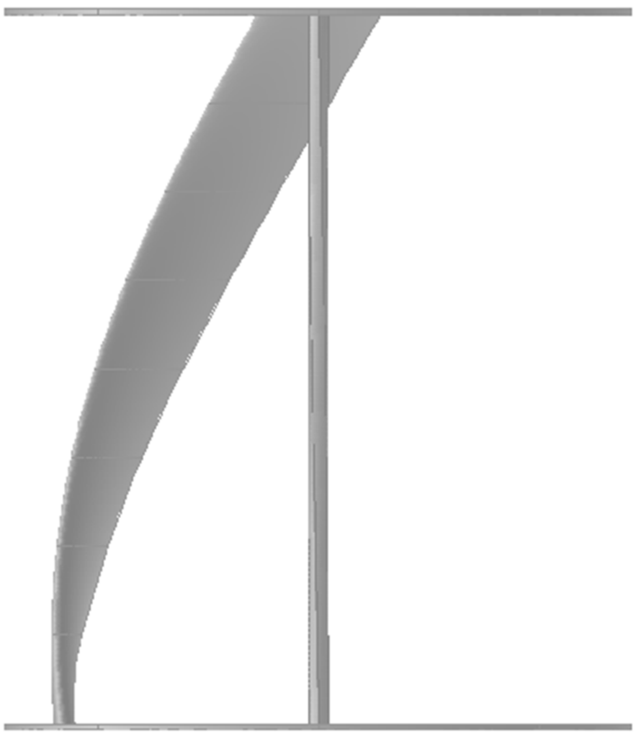
Blockage Ratio: Matched to Experiments

$$\text{Blockage ratio} = \frac{2RH}{Channel\ Area} = 0.12$$

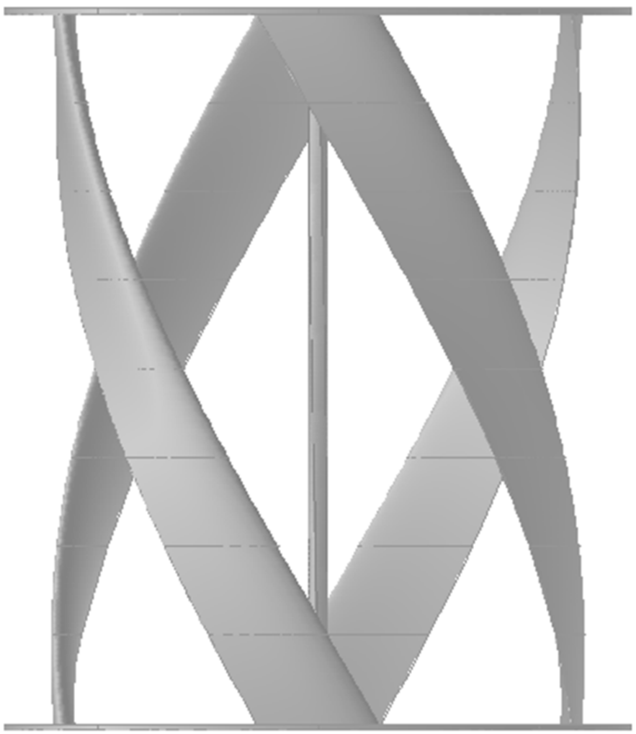


Simulation Cases

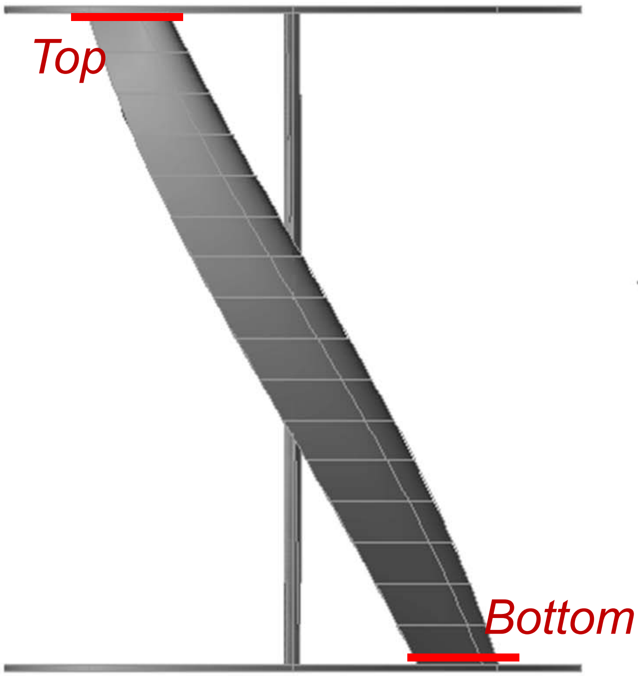
- **Static Turbine**
  - 1 blade
  - 4 blades

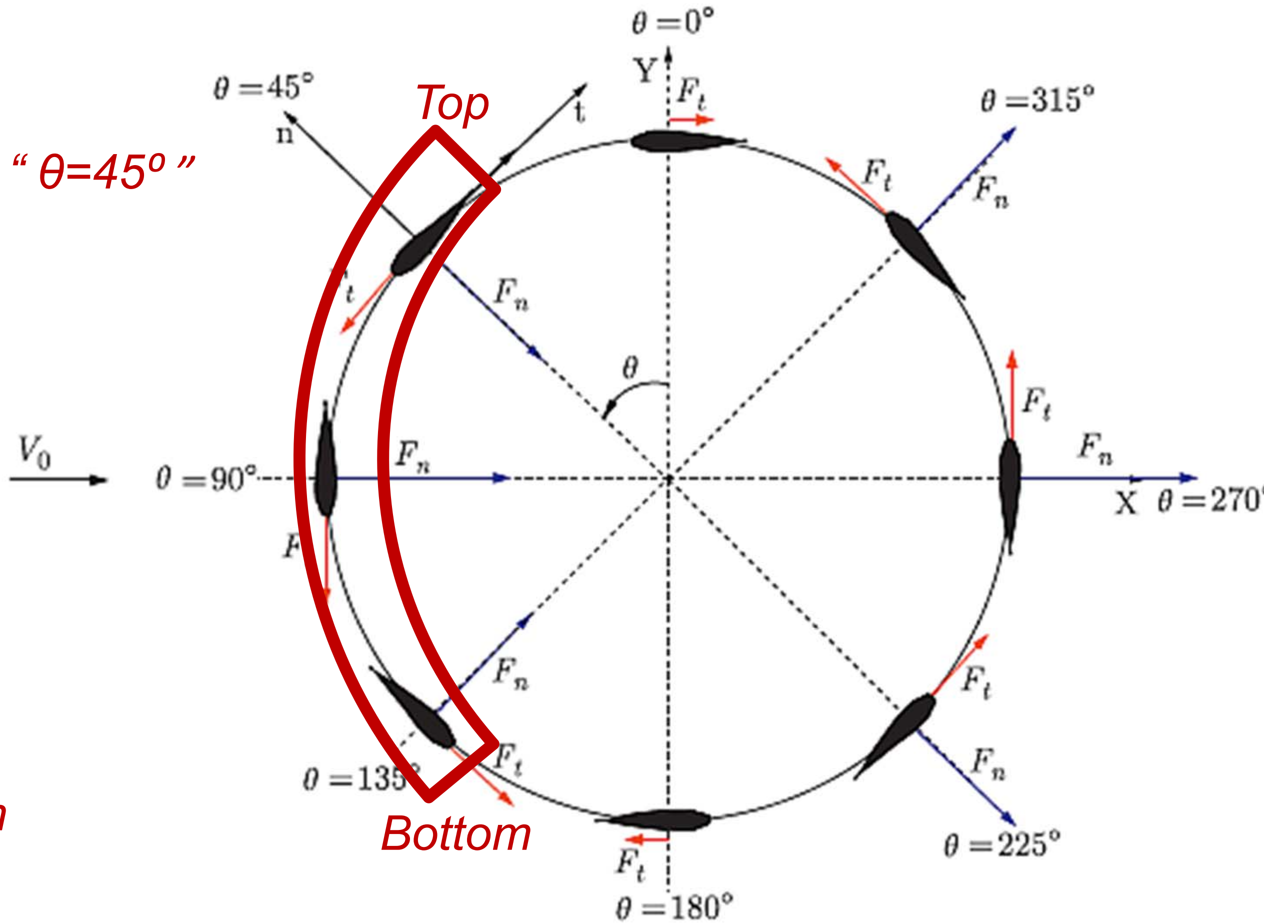


- **Rotating Turbine**
  - 1 blade
  - 4 blades

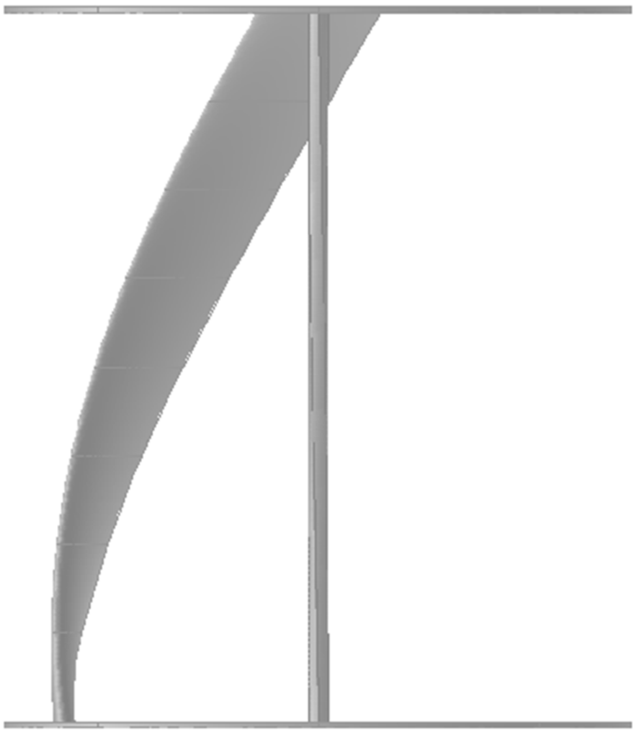


Labeling of Helical Blade Position



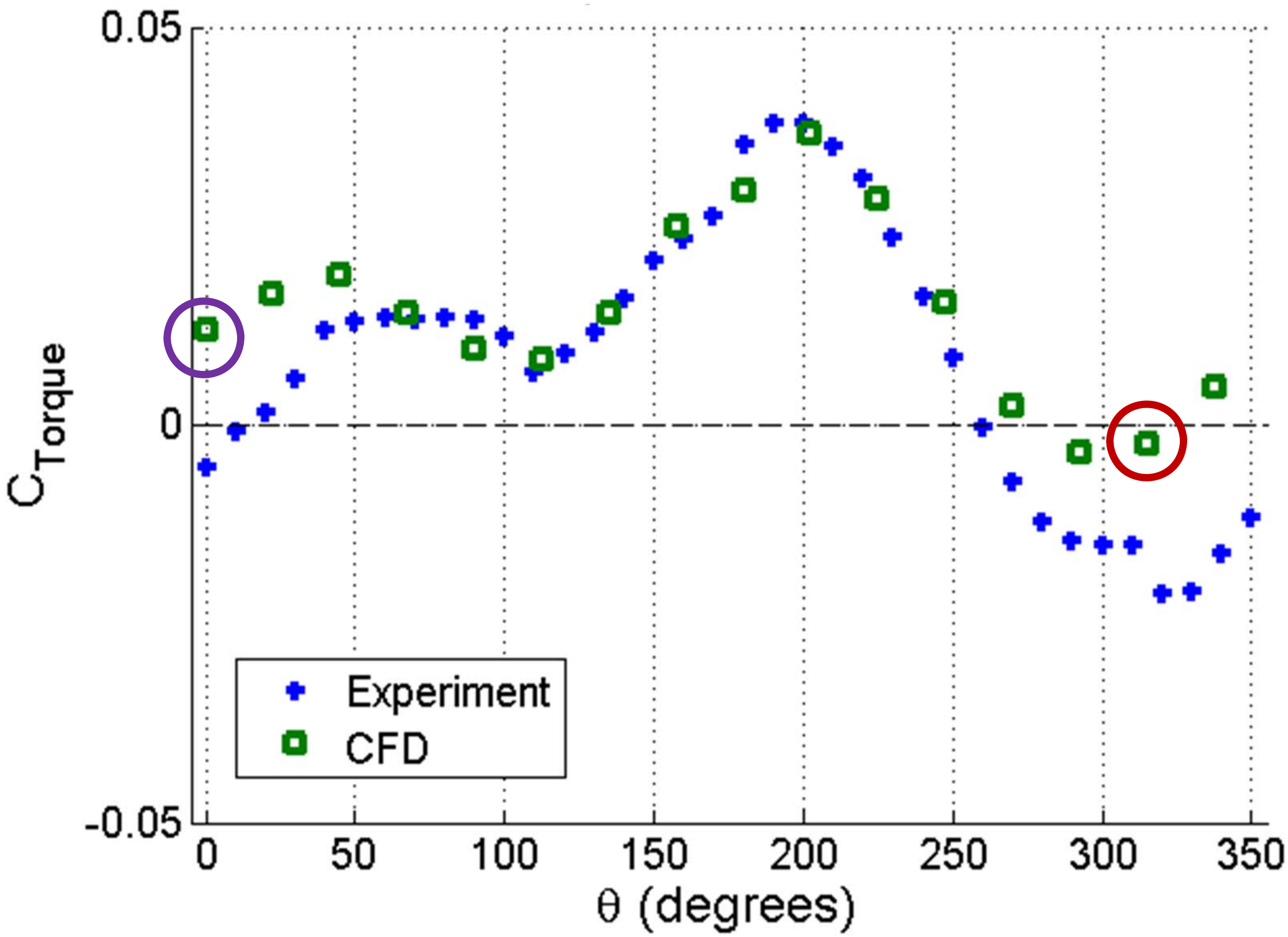


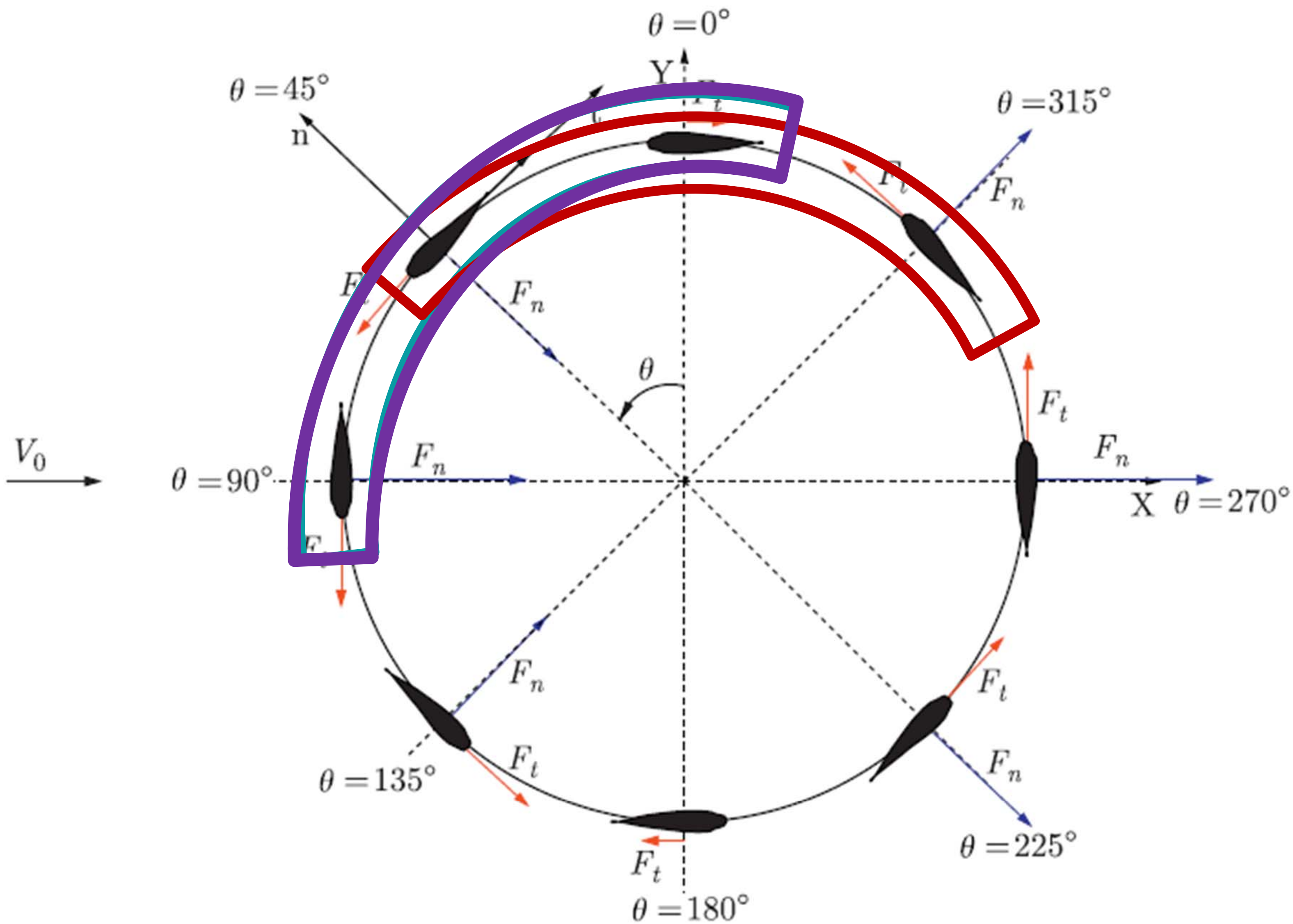
Static Turbine: Single Blade



- **Particle brake applies constant torque to hold turbine in stationary position: λ=0**
- **Torque cell measurements available from experiments**
- **Repeated at several azimuthal locations**

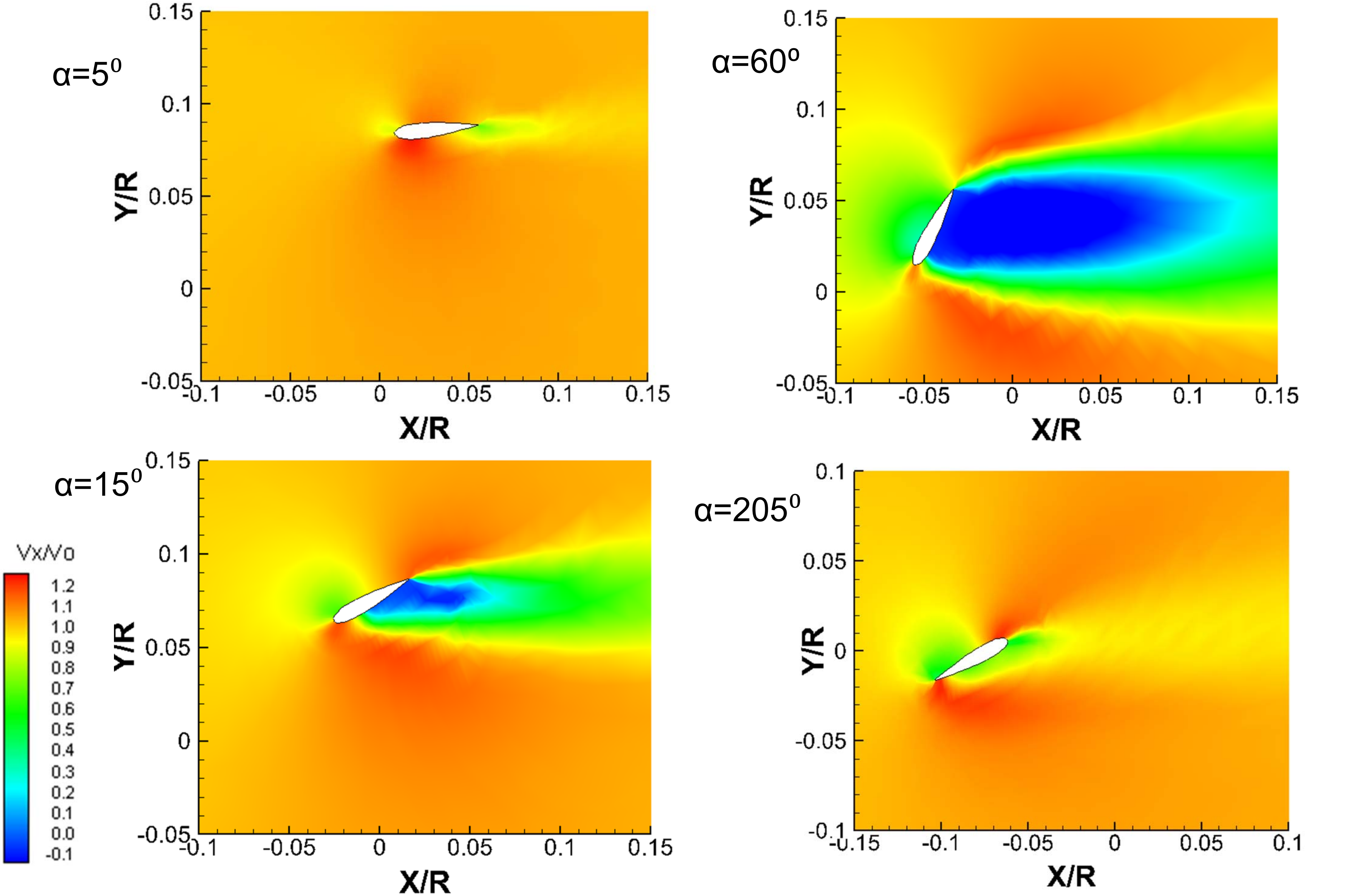
Static Turbine: Single Blade





$\theta = 0°$
Y
$\theta = 45°$
n
$\theta = 315°$
F_t
F_n
θ
V_0
$\theta = 90°$
X
$\theta = 270°$
$\theta = 135°$
$\theta = 225°$
$\theta = 180°$

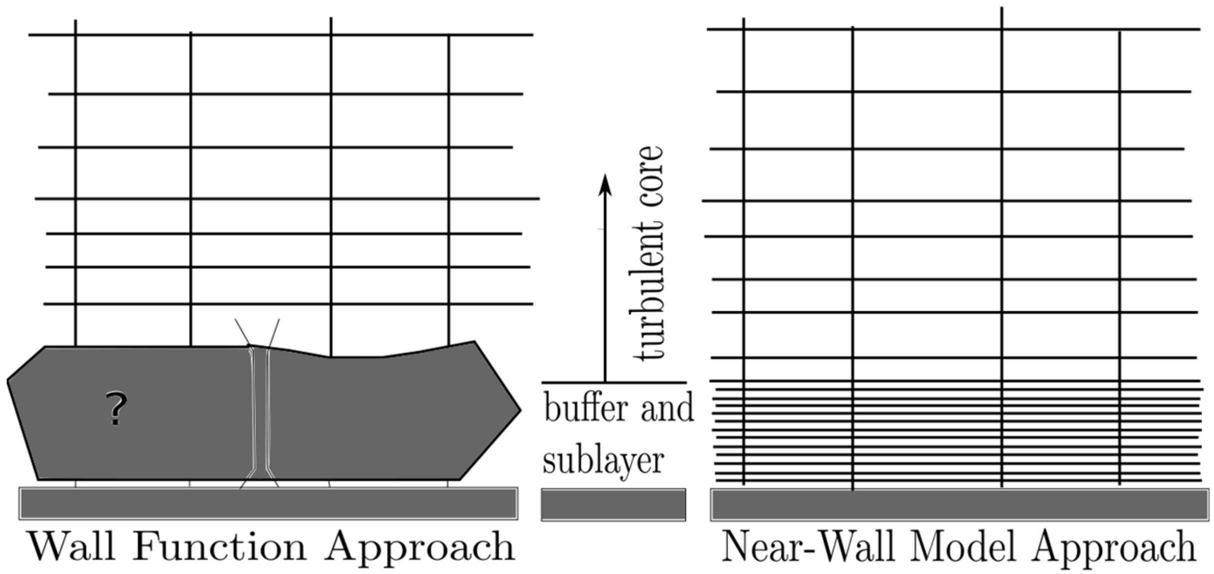
Flow fields for angles of attack (α)



Modeling in Near Wall Region

Wall Functions

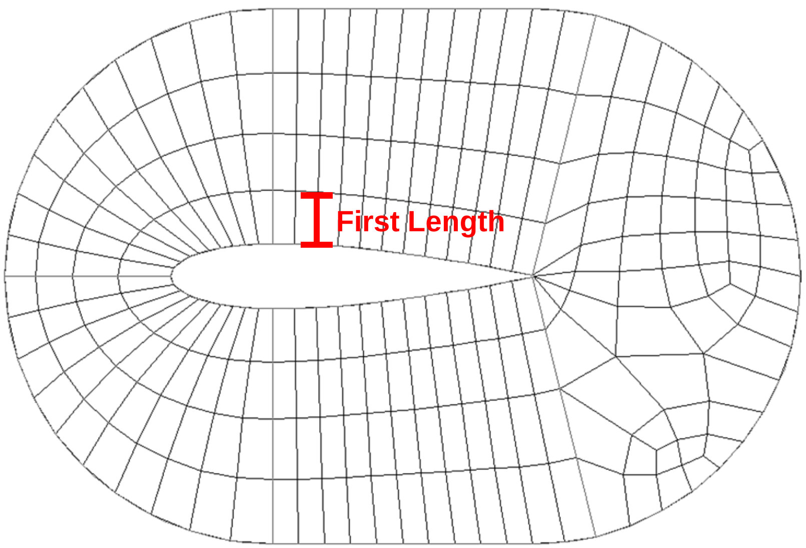
Near Wall Approach



Modeling in Near Wall Region

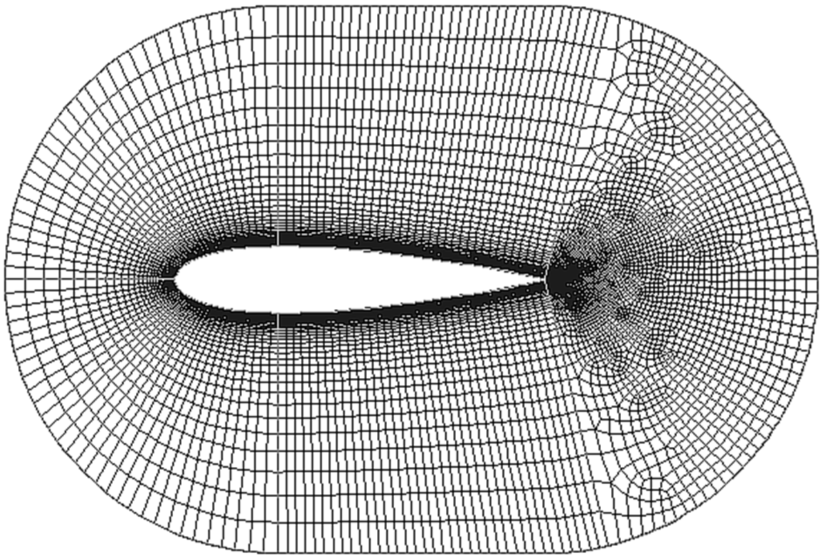
Wall Functions

- $30 < y^+ < 300$
- $\frac{First\ Length}{Chord\ Length} = 0.15$
- Total Elements = 185,000



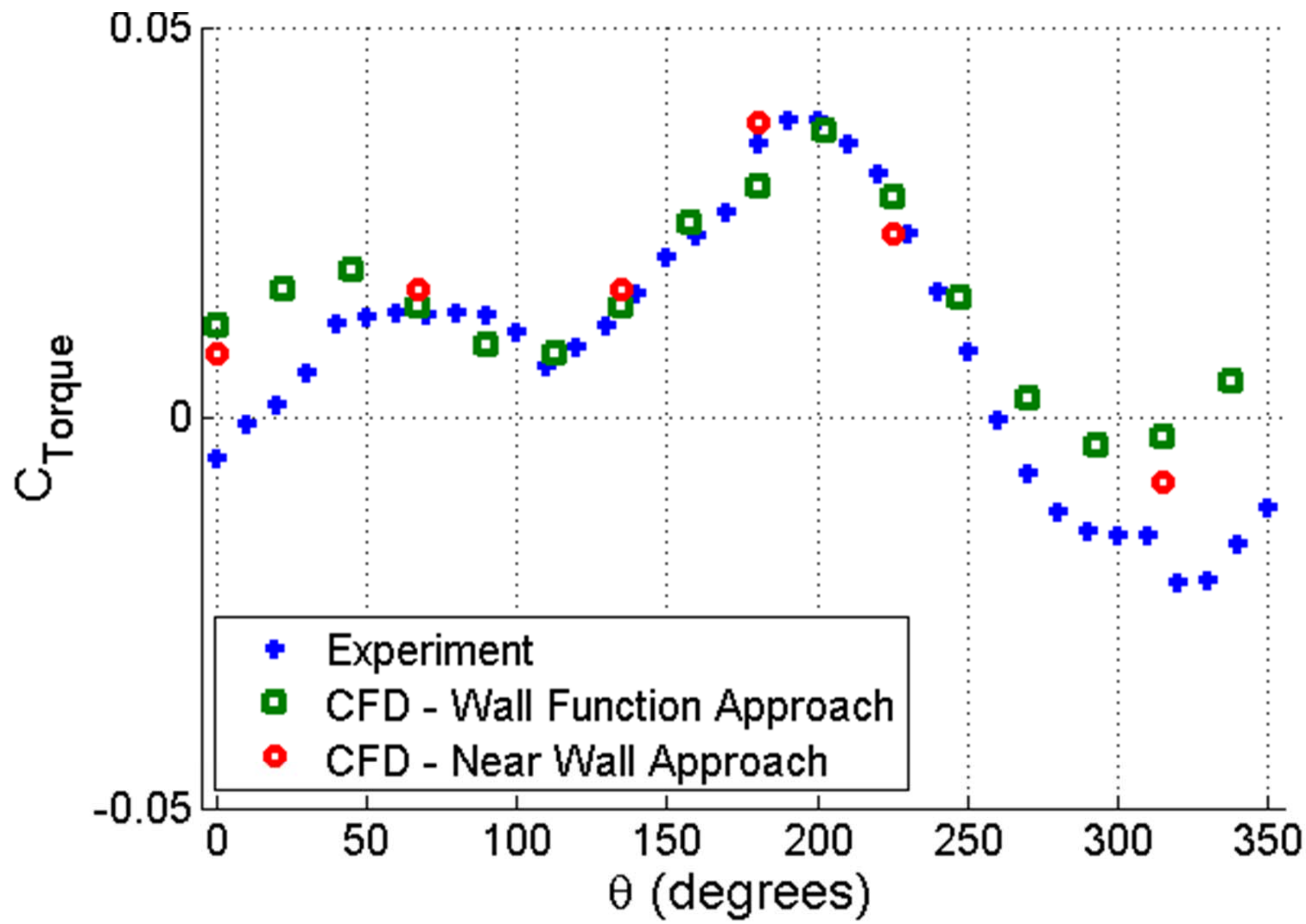
Near Wall Approach

- $y^+ < 1$
- $\frac{First\ Length}{Chord\ Length} = 0.0001$
- Total Elements = 4.0 million

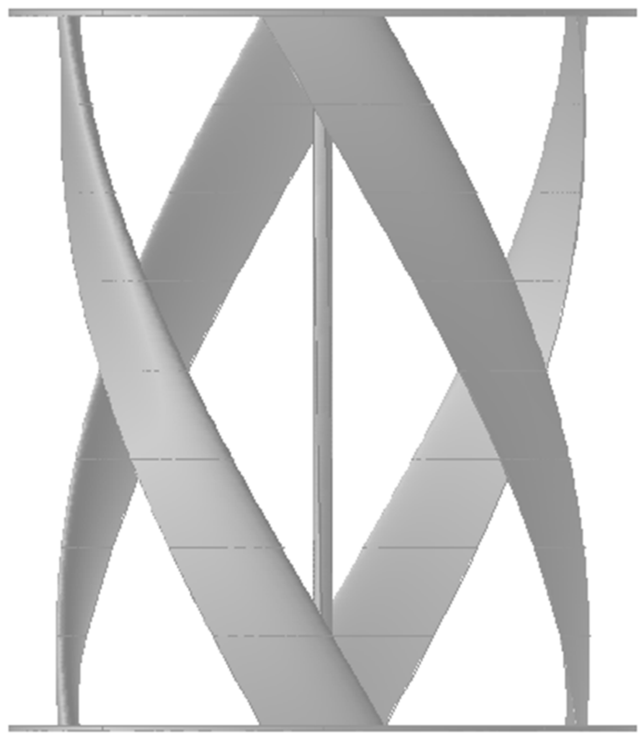


Near Wall: 20x computation time, 20-25% reduction in error

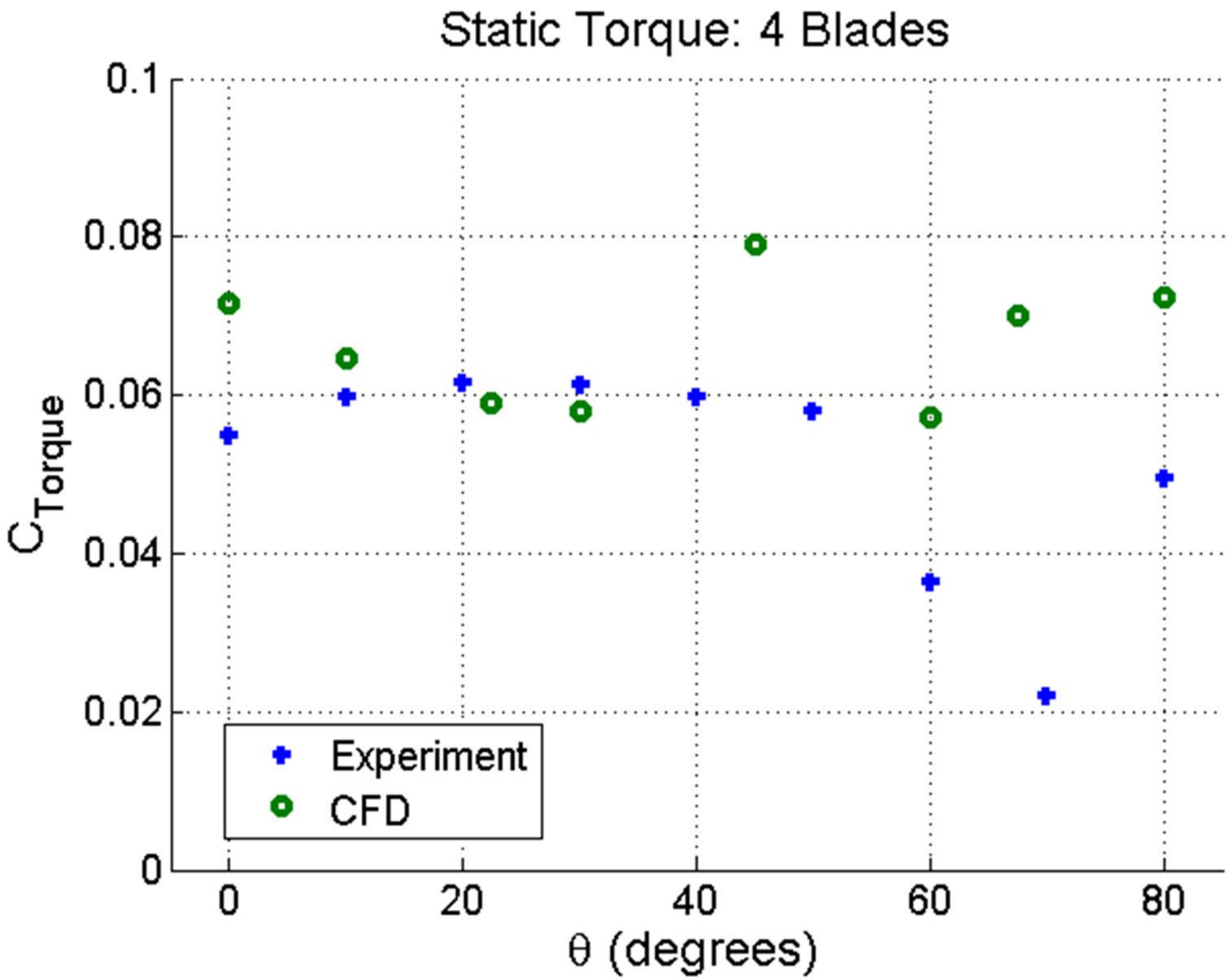
Static Turbine: Single Blade with near-wall modeling approach



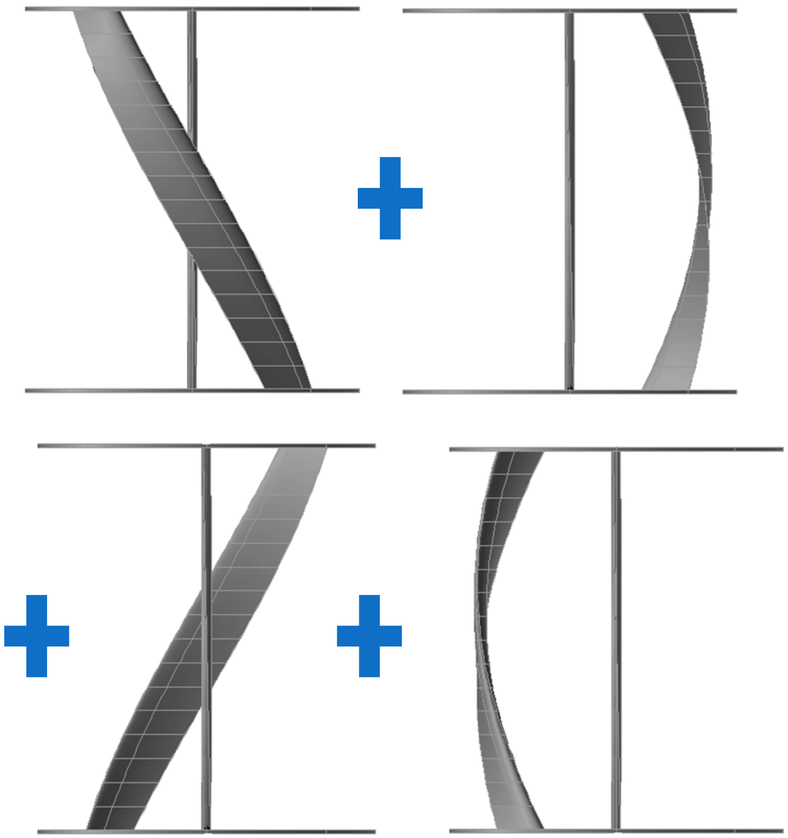
Static Turbine: 4 Blades



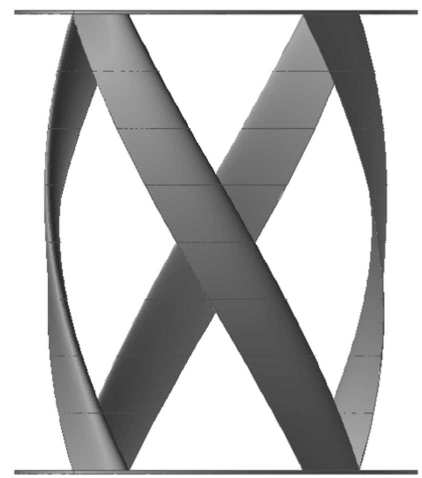
Static Torque: 4 Bladed Turbine



Superposition of Single Blades vs. 4 Bladed Turbine



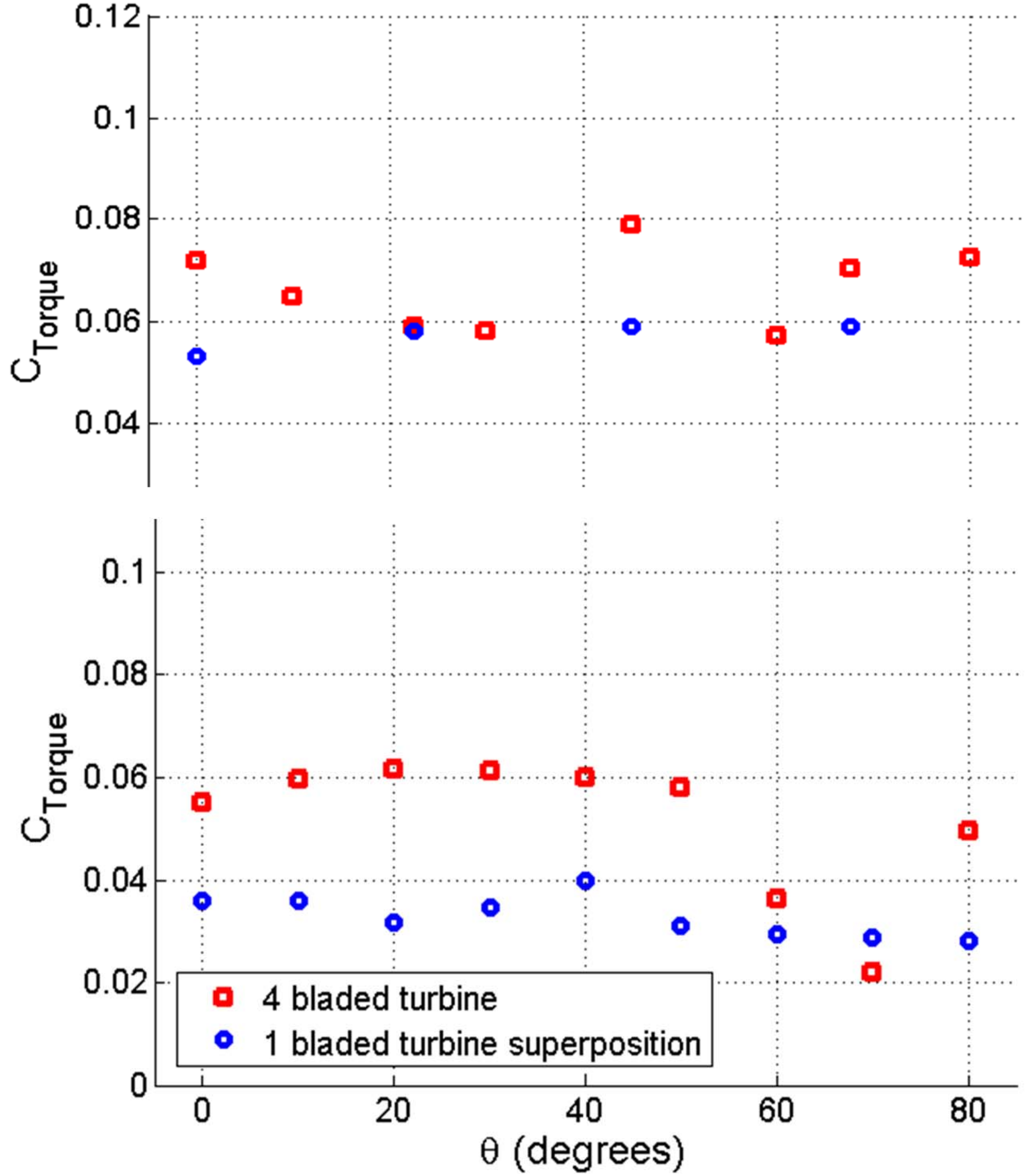
vs.



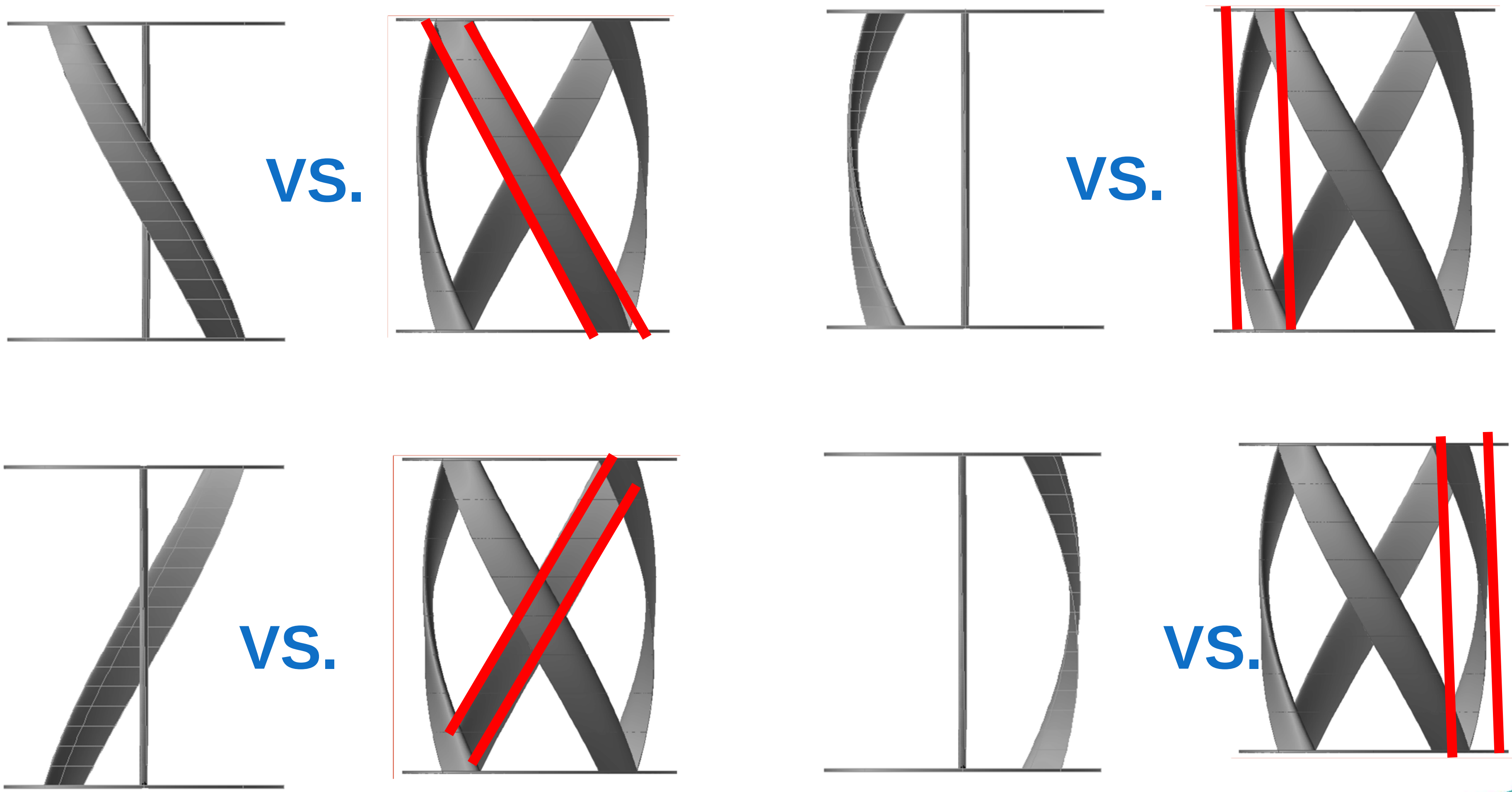
Superposition of Single Blades vs. 4 Bladed Turbine

CFD:

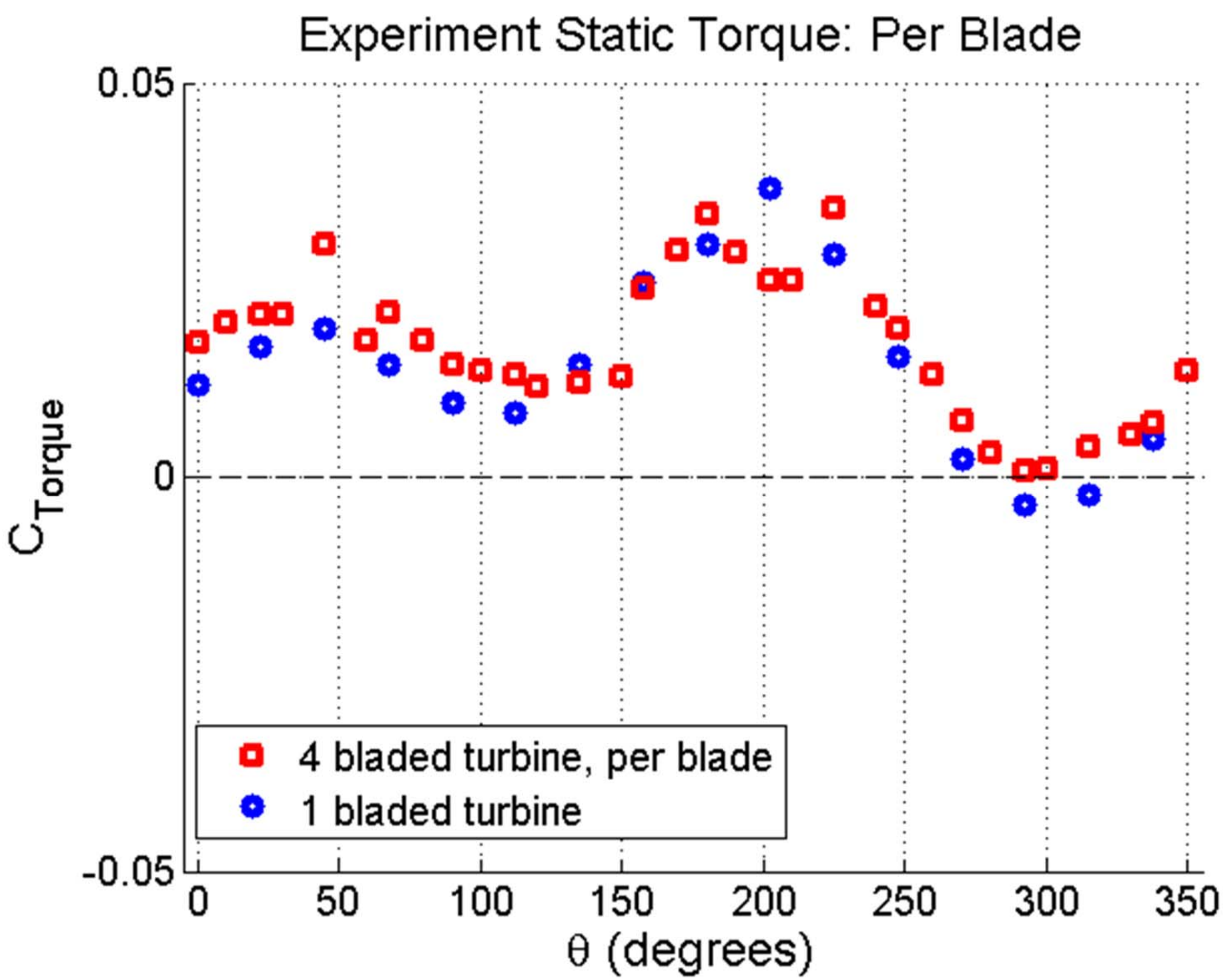
Experiments:



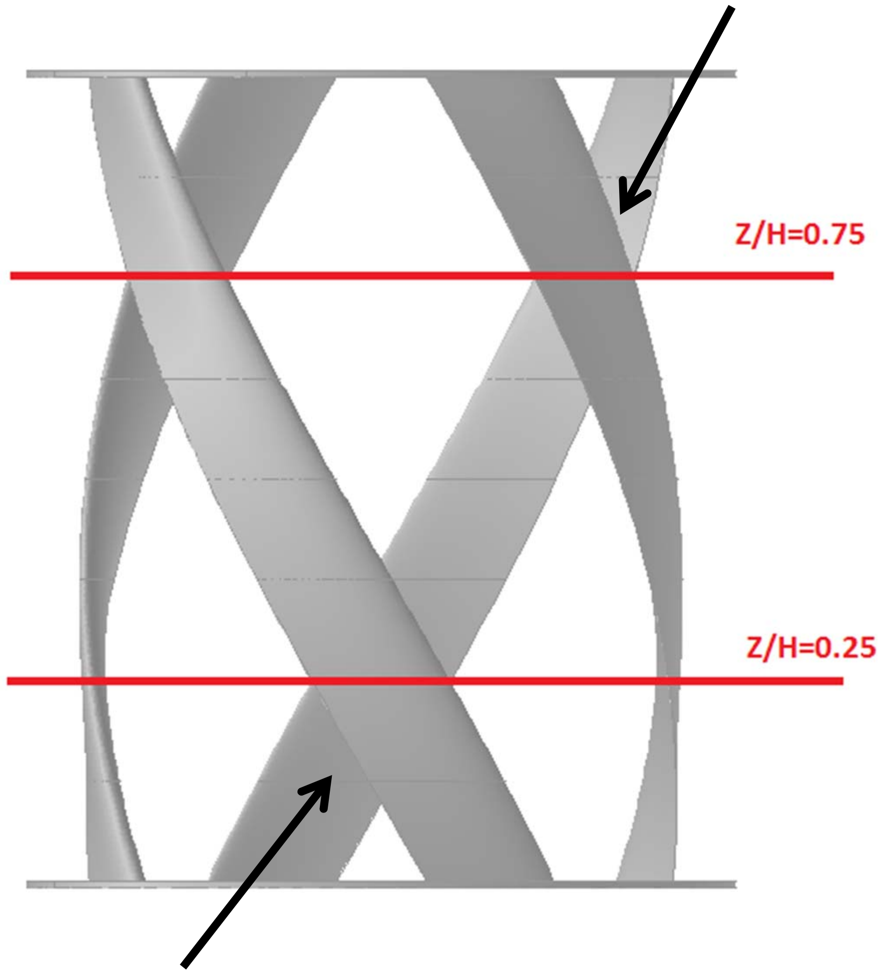
Single Blade
vs. 1 blade from 4 bladed turbine

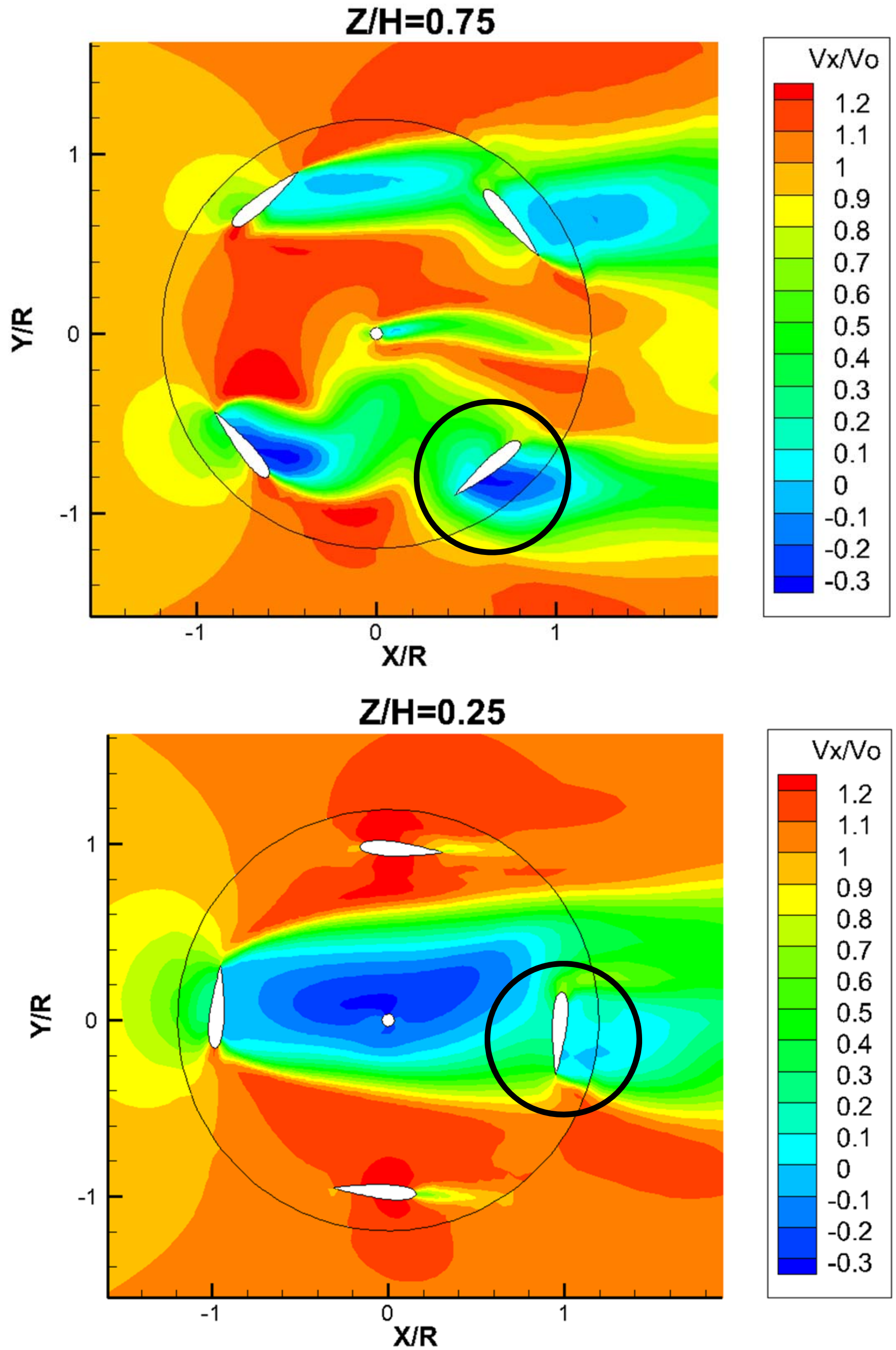


Single Blade vs. 1 blade from 4 bladed turbine



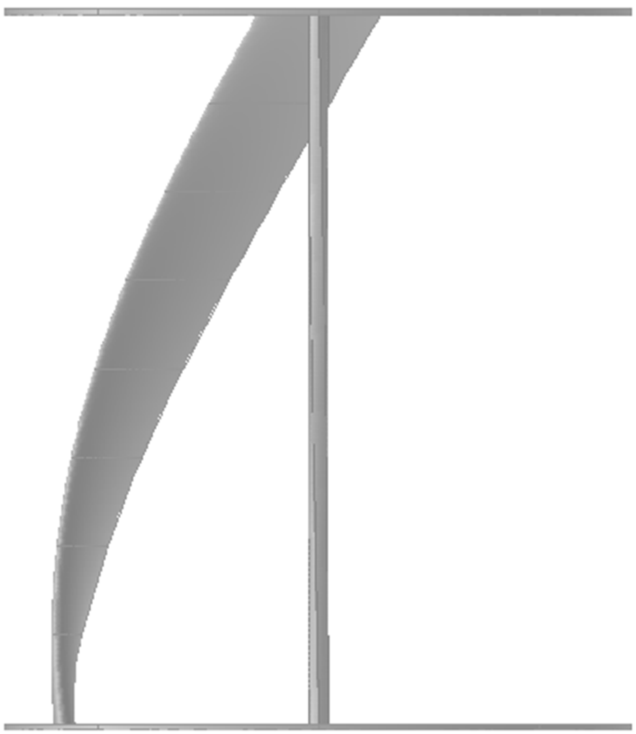
Wake Interactions

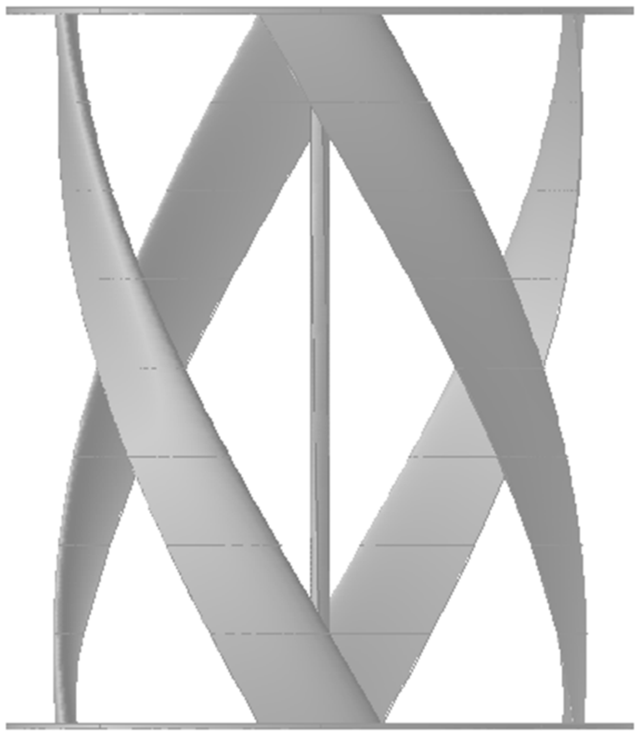




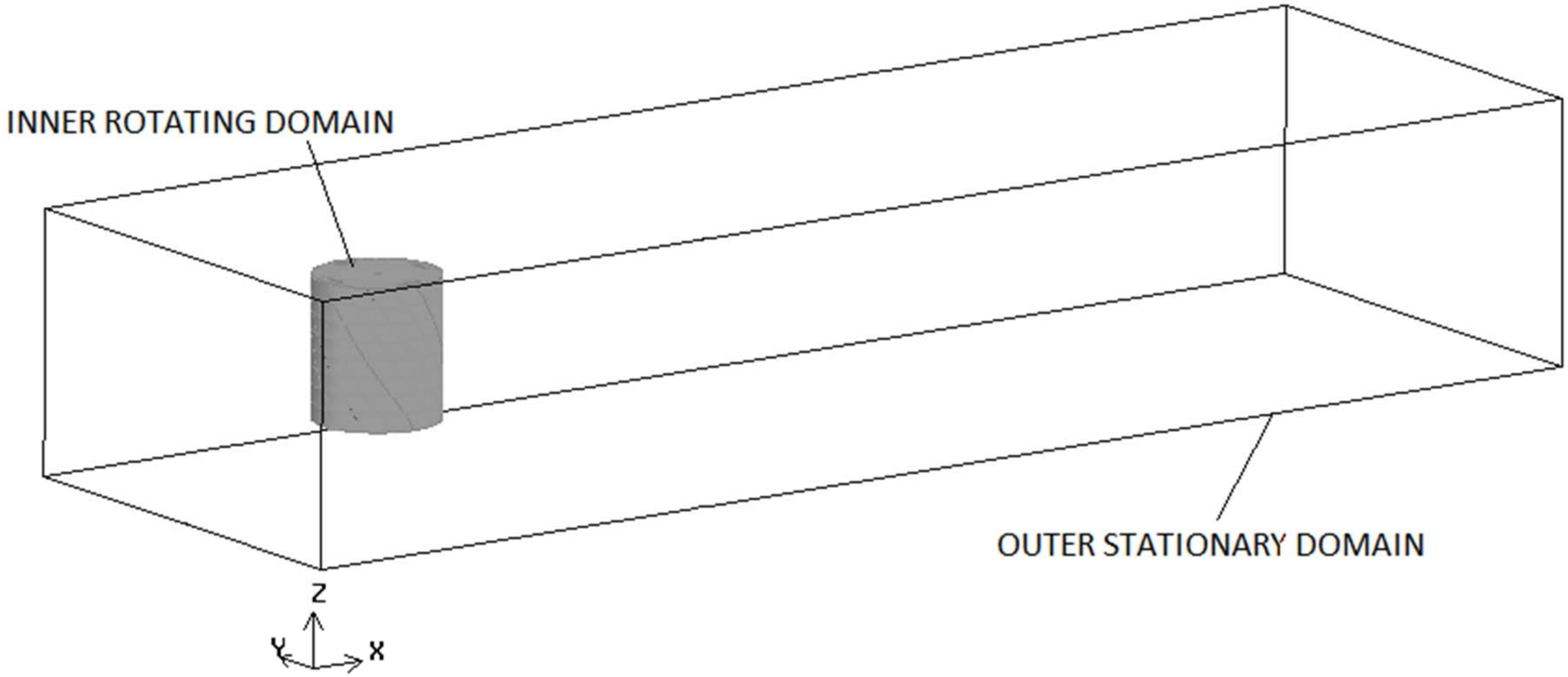
Simulation Cases

- Static Turbine
  - 1 blade
  - 4 blades
- **Rotating Turbine**
  - 1 blade
  - 4 blades





Sliding Mesh Technique for Rotating Turbine

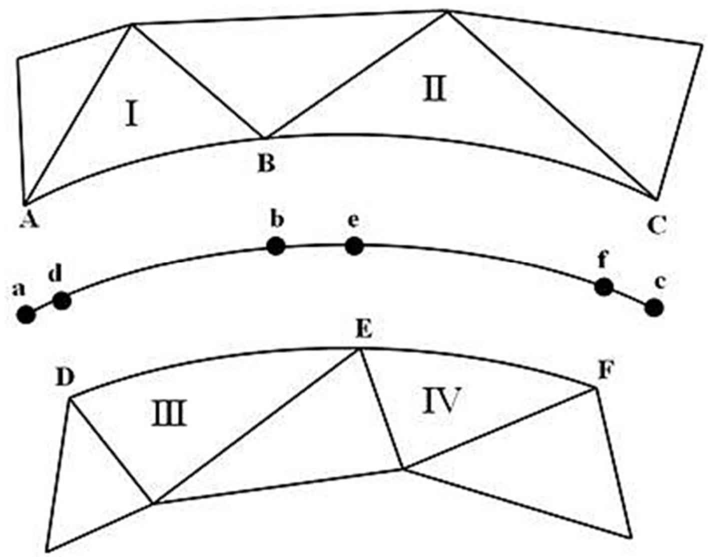


Set RPM of rotating domain

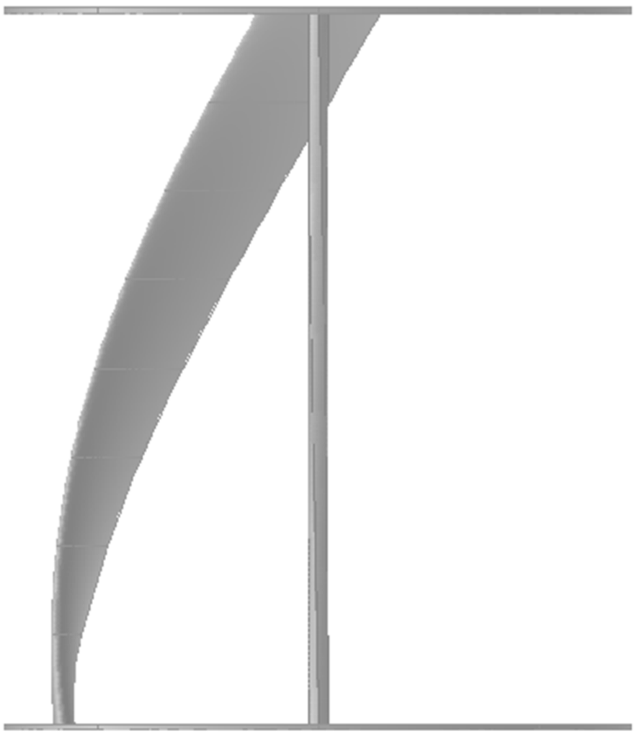
Time step =1.2 degree rotation

At each time step:

- Non-conformal boundary
- Determine interfaces
- Calculate flux

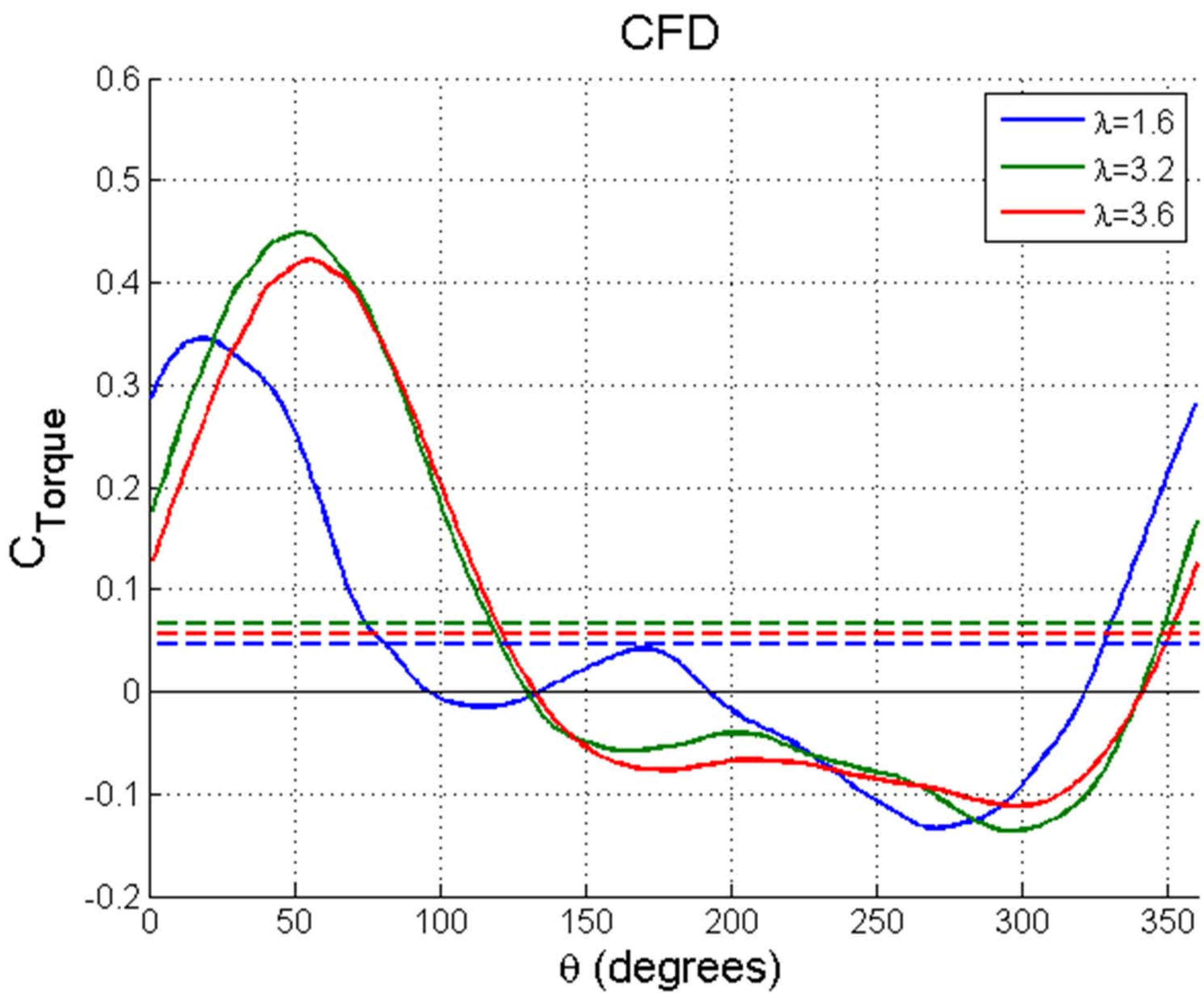


Rotating Turbine: Single Blade

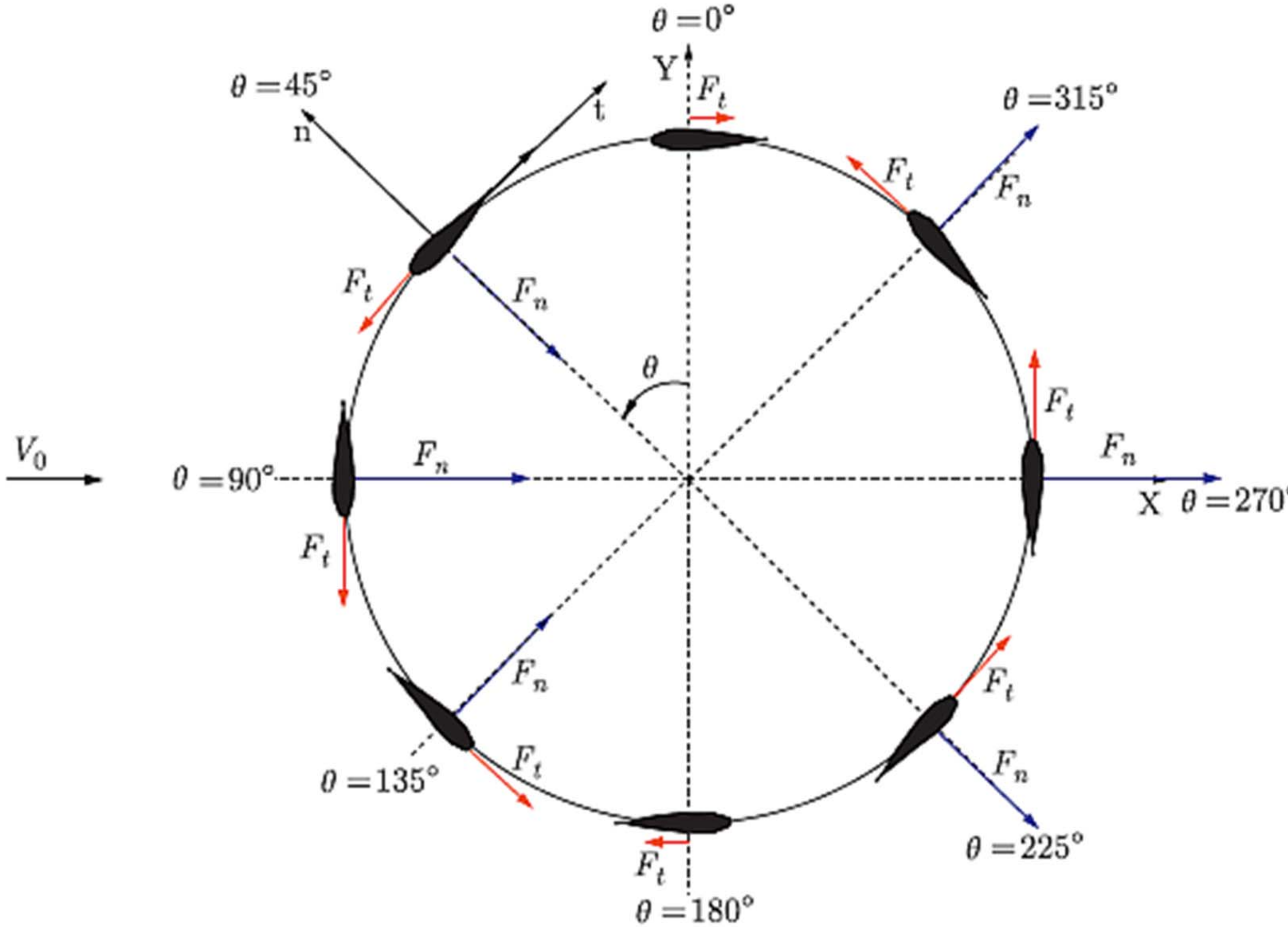


- **Experiment**
  - — λ=3.2
  - — λ=3.6 (no load)
- **CFD**
  - — λ=3.2
  - — λ=3.6
  - — λ=1.6

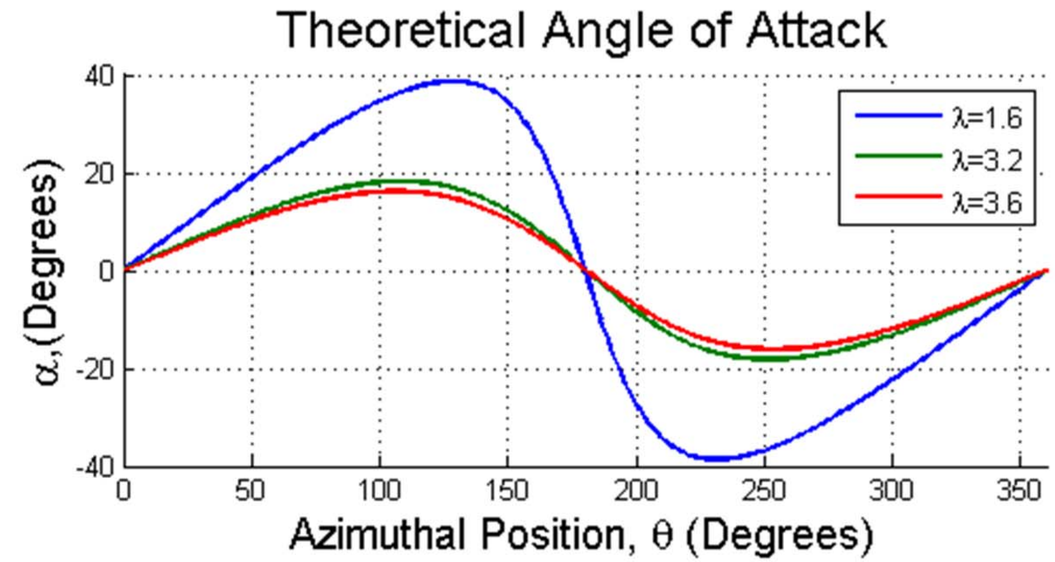
Dynamic Torque: Single Blade

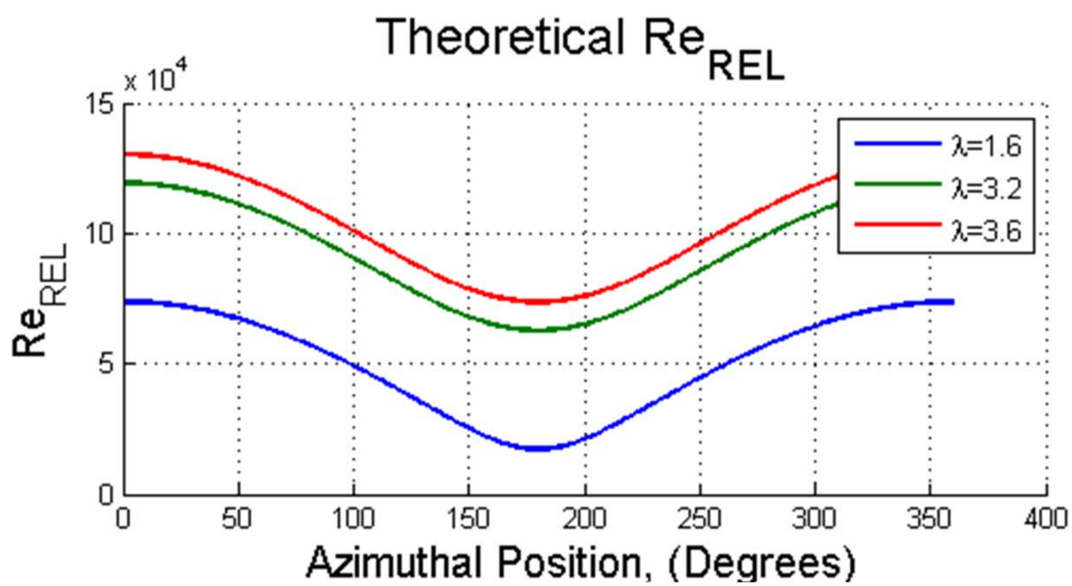


λ	$C_{avgTorque}$
1.6	0.047
3.2	0.065
3.6	0.057

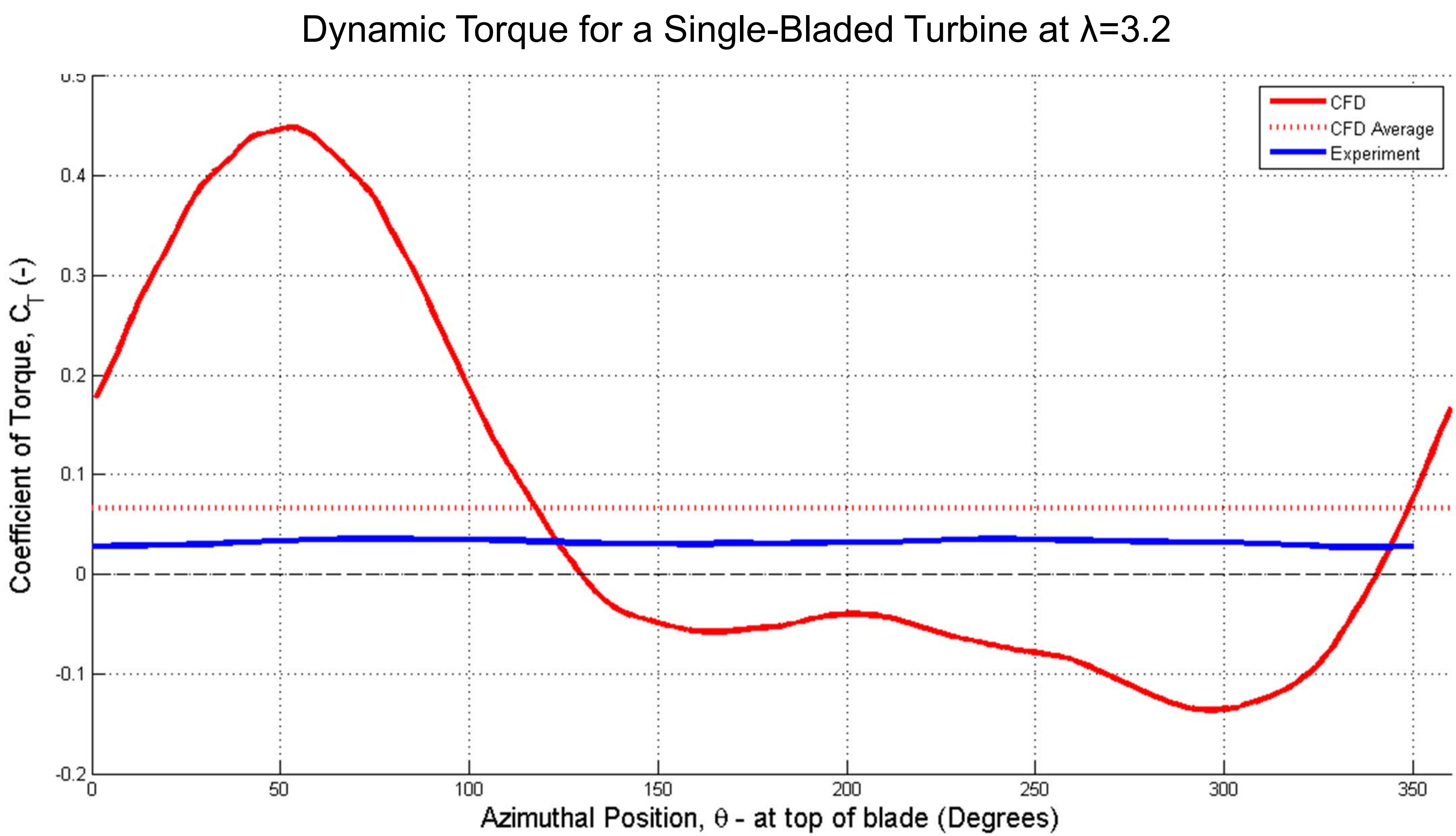

θ =0°
Y
F_t
θ =45°
n
t
θ =315°
F_t
F_n
F_t
F_n
θ
F_t
V_0
θ =90°
F_n
F_n
X θ =270°
F_t
F_n
F_t
θ =135°
F_t
F_n
F_t
θ =225°
θ =180°

W OSU

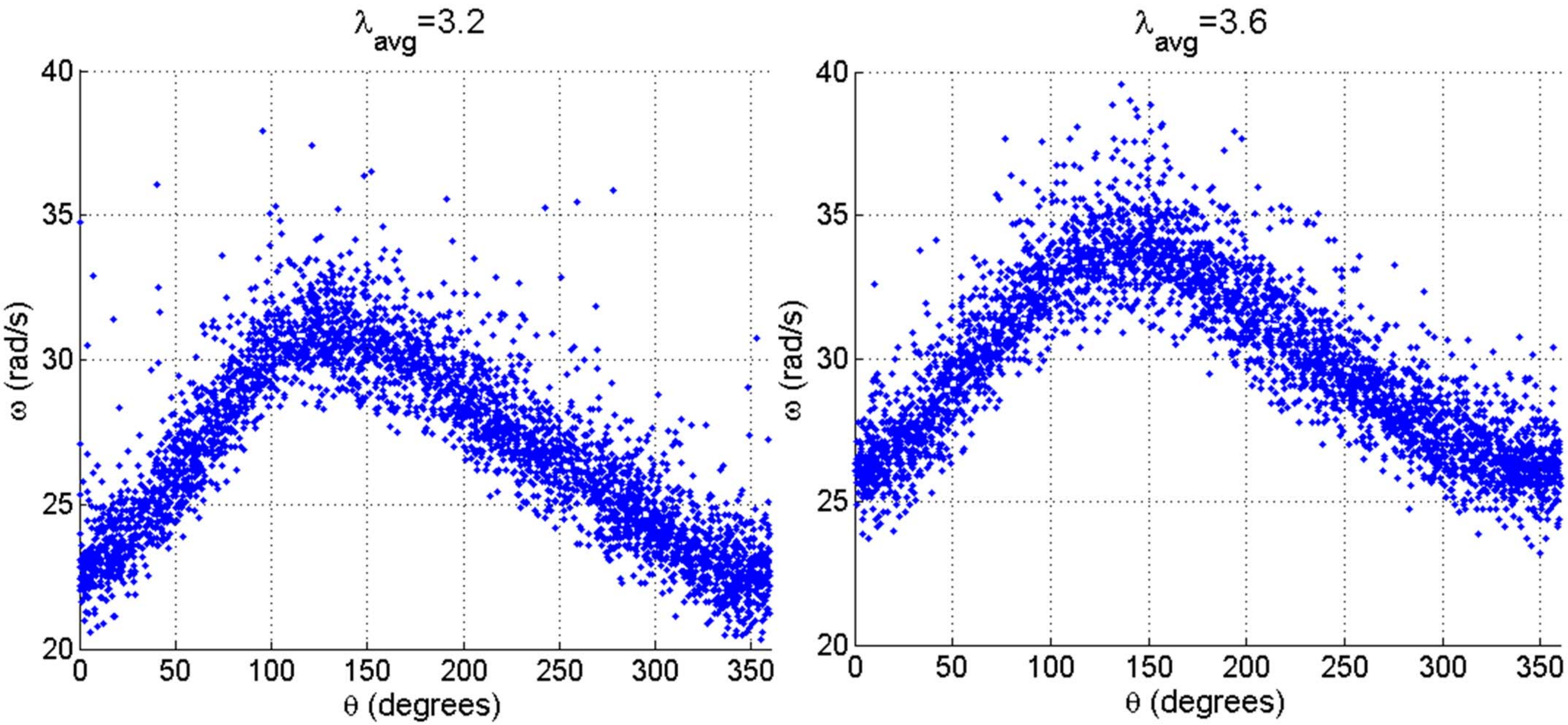

Theoretical Angle of Attack
α,(Degrees)
Azimuthal Position, θ (Degrees)
λ=1.6
λ=3.2
λ=3.6
40
20
0
-20
-40
0
50
100
150
200
250
300
350


Theoretical Re_REL
x 10^4
Re_REL
Azimuthal Position, (Degrees)
λ=1.6
λ=3.2
λ=3.6
15
10
5
0
0
50
100
150
200
250
300
350
400

Rotating Single Bladed Turbine: CFD vs. Experiment

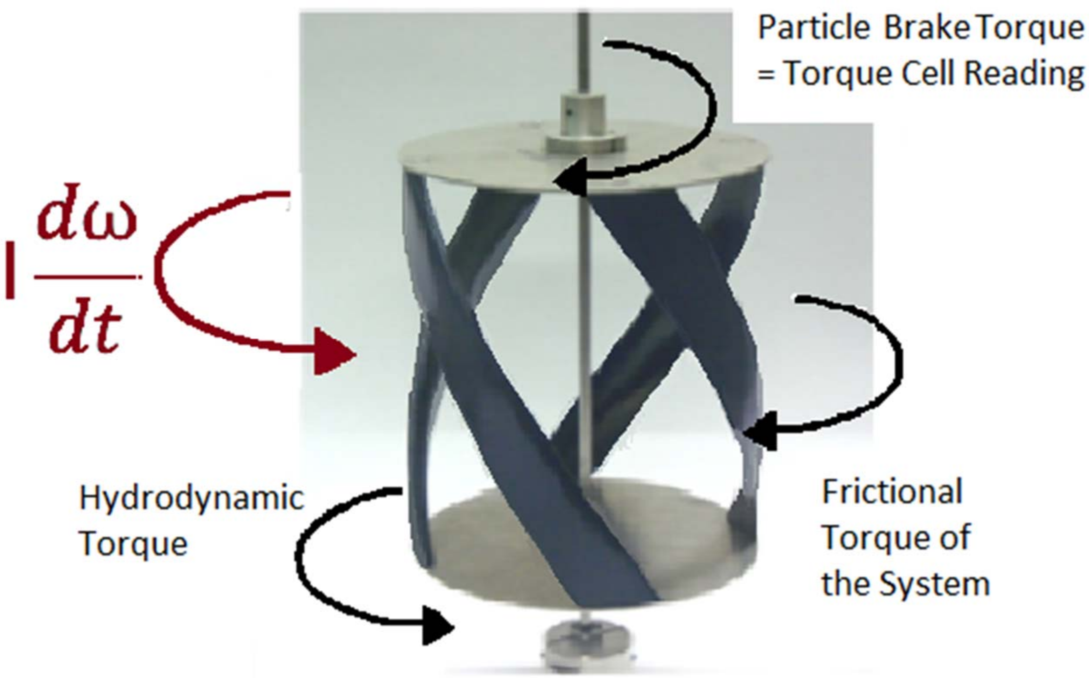


Oscillating Rotational Velocity



System Dynamics

$$I\frac{d\omega}{dt} = \sum M_{ext} = T_{Hydrodynamic} - T_{ParticleBrake} - T_{SystemFriction}$$



- CFD models the Hydrodynamic Torque
- Experiments measure the Particle Brake Torque

Torque Calculations

$$I\frac{d\omega}{dt} = \sum M_{ext} = T_{Hydrodynamic} - T_{ParticleBrake} - T_{SystemFriction}$$

$$T_{Hydrodynamic} - T_{SystemFriction} = T_{ParticleBrake} + I\frac{d\omega}{dt}$$

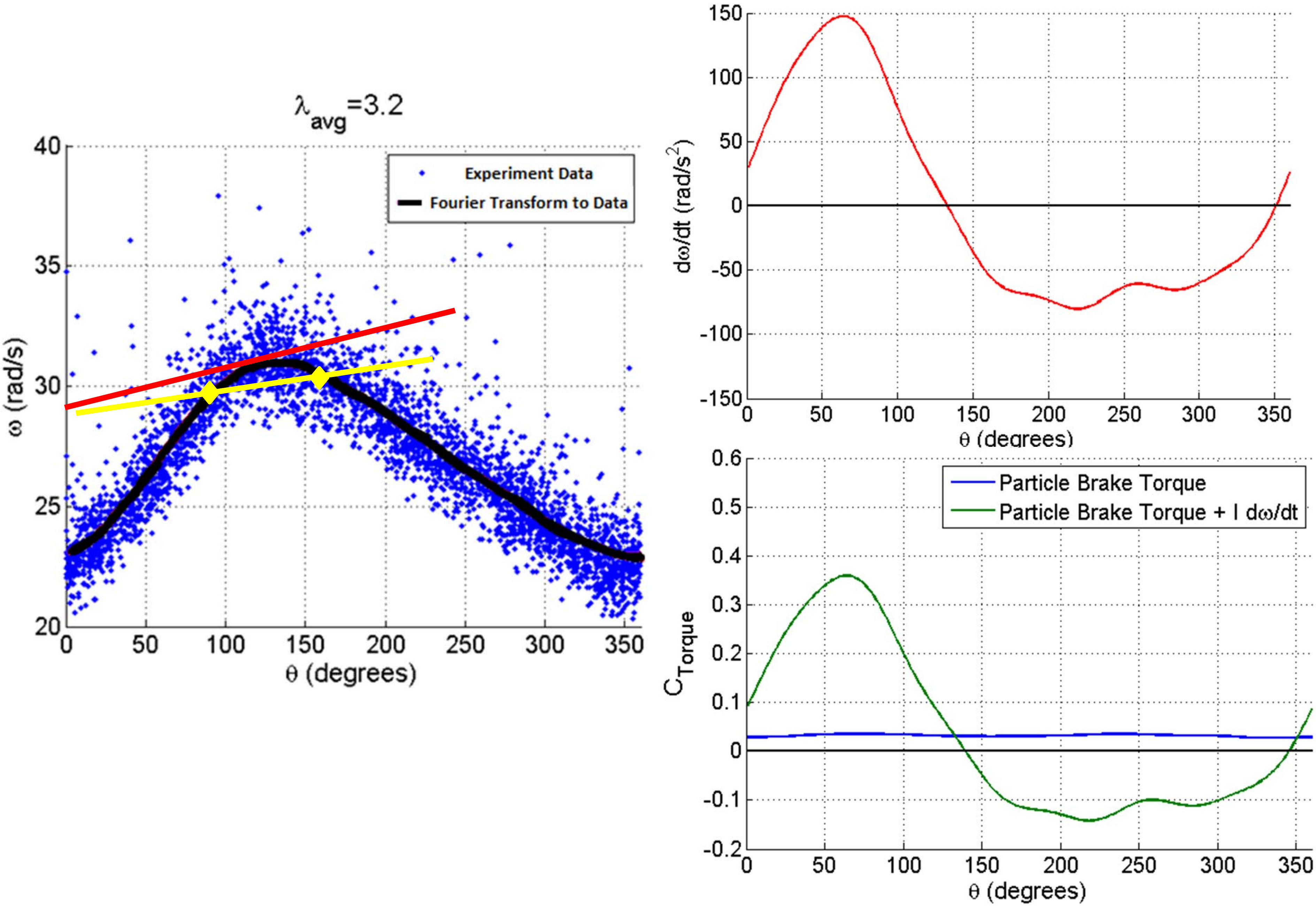
$$T_{Hydrodynamic} \approx T_{ParticleBrake} \qquad if\ \frac{d\omega}{dt}\ and\ T_{SystemFriction} \approx 0$$

$$\int_{0^\circ}^{360^\circ} I\frac{dw}{dt}\ d\theta = 0$$

$$\overline{T_{Hydrodynamic}} = \overline{T_{ParticleBrake}} + \overline{T_{SystemFriction}}$$

W OSU

Experiment Single Blade, $\lambda_{AVG} = 3.2$

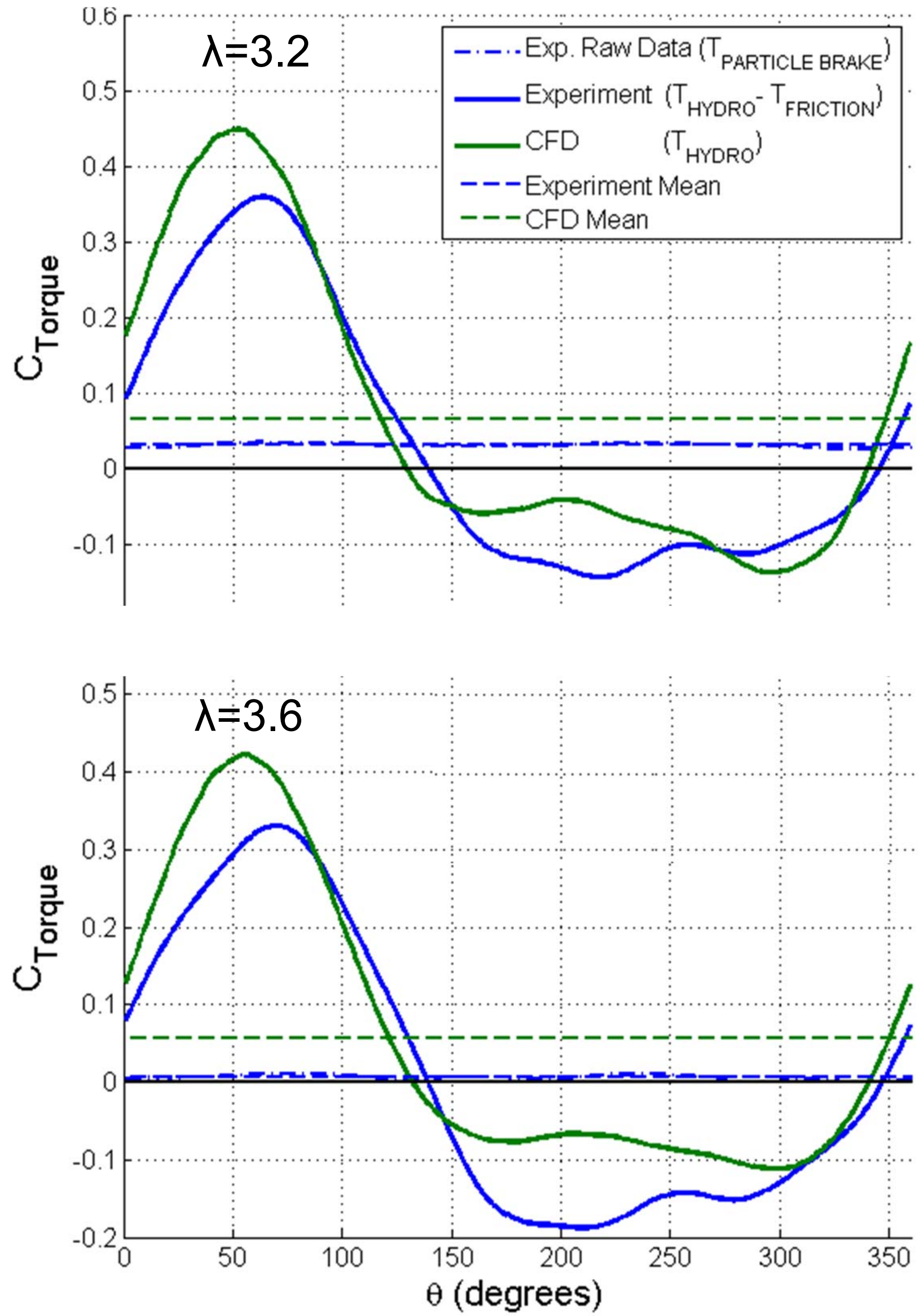


Model Validation

Differences Attributed To:

- Constant vs. varying tip speed ratio
- System Frictional Torque

λ	Experiment	CFD
3.2	0.032	0.065
3.6	0.007	0.057



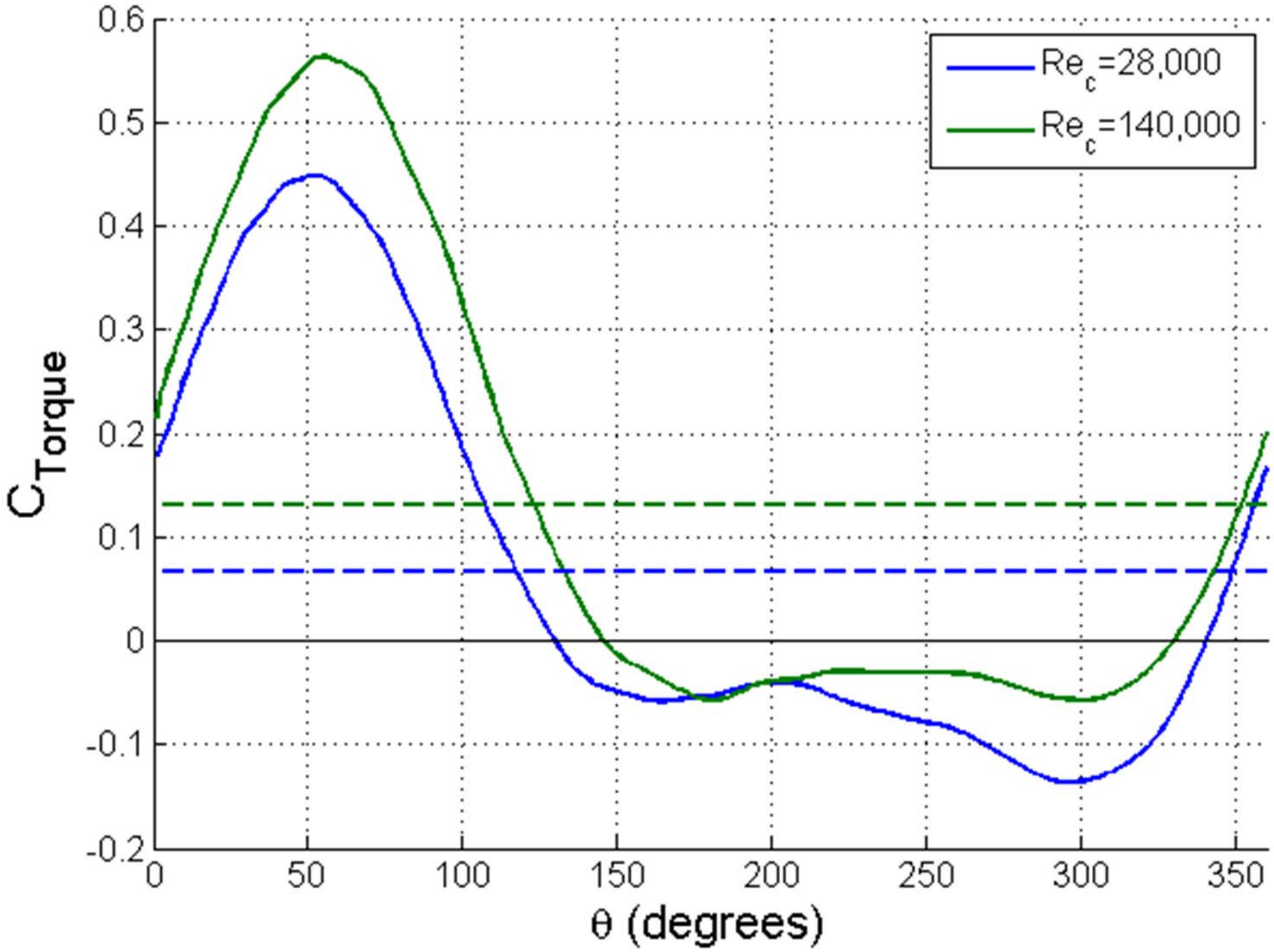
Possible Comparison Improvements

— Run simulations at several different tip speed ratios : create a composite result
— Run experiments with a variable load to keep a constant rotation speed

Four- bladed turbine

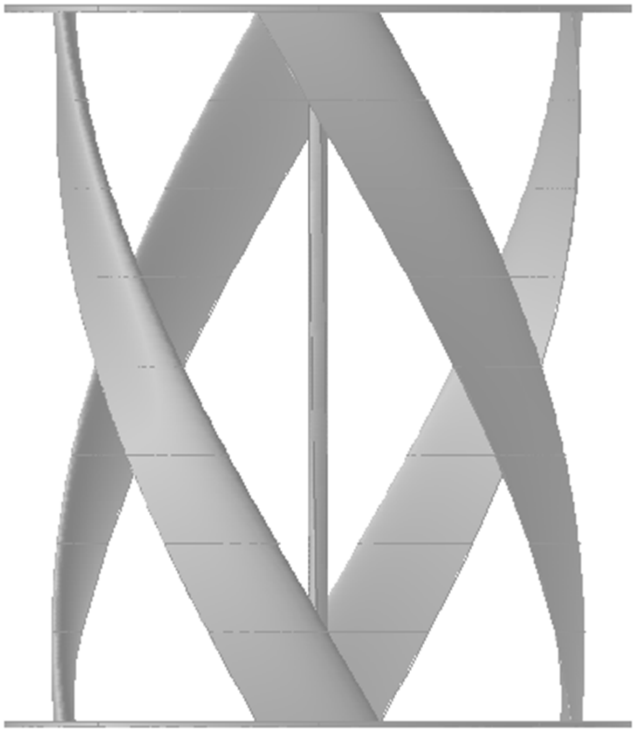
- Hydrodynamic torque undergoes much smaller fluctuations
- Much higher moment of inertia
- Leads to much smaller oscillations in the rotational velocity

Increased Reynolds Number Simulation



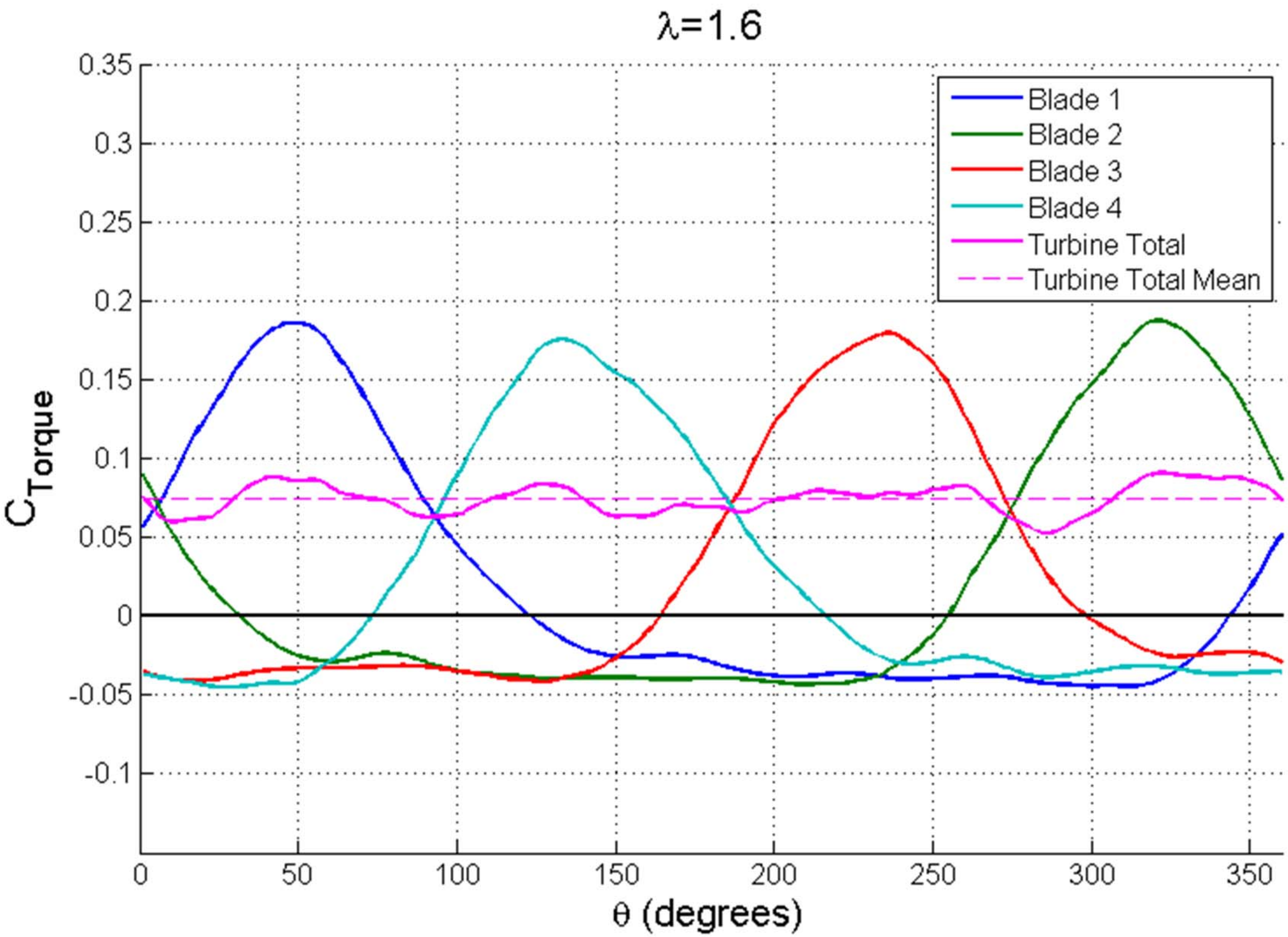
- Decreased viscosity to achieve higher Re
- Required finer mesh
- Increase in turbine efficiency from 21 to 42%

Rotating Turbine: 4 Blades

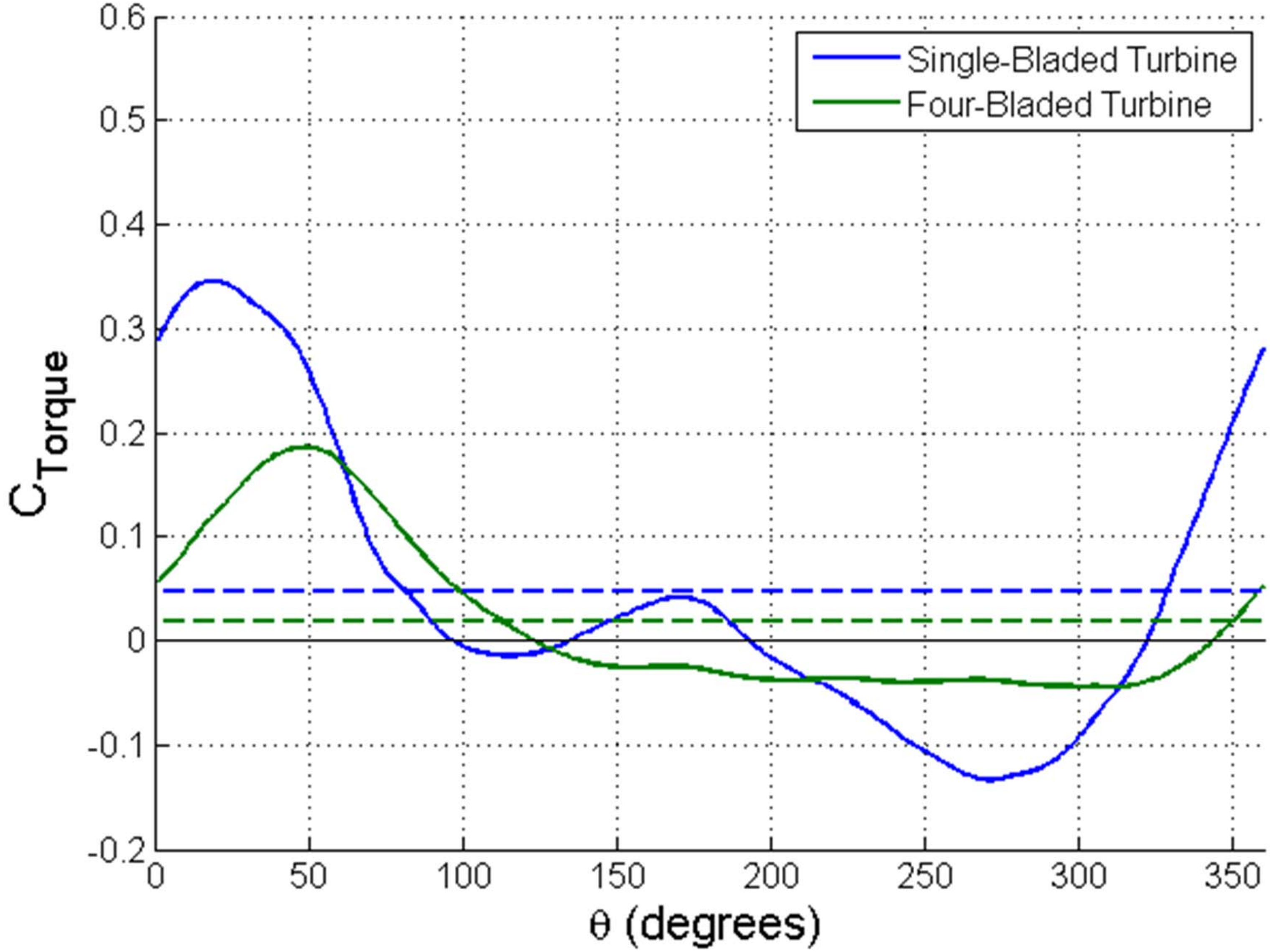


- **Experiments**
  - λ=1.3 to 2.1
- **CFD**
  - λ=1.3, 1.6, 2.0

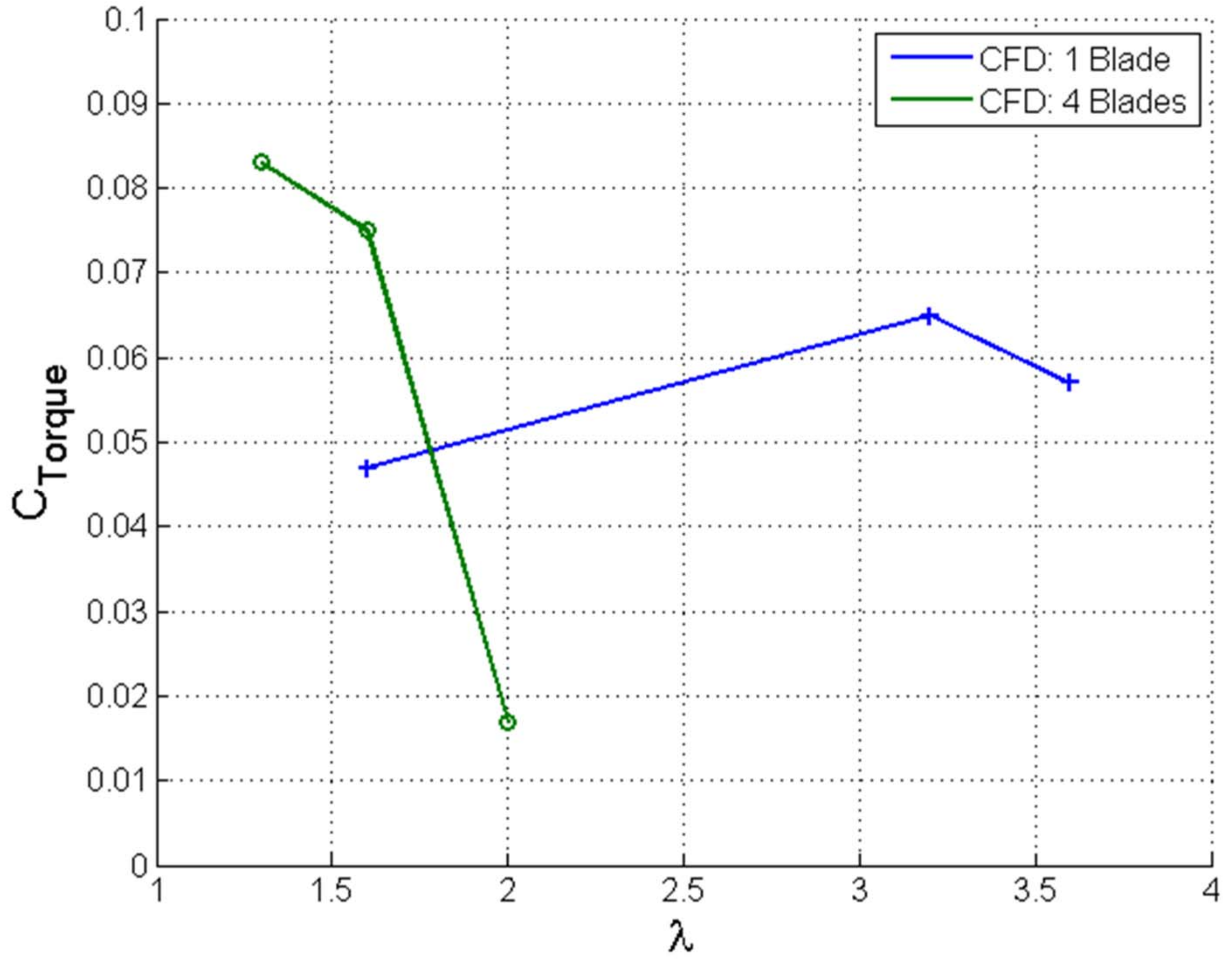
Rotating 4 Bladed Turbine



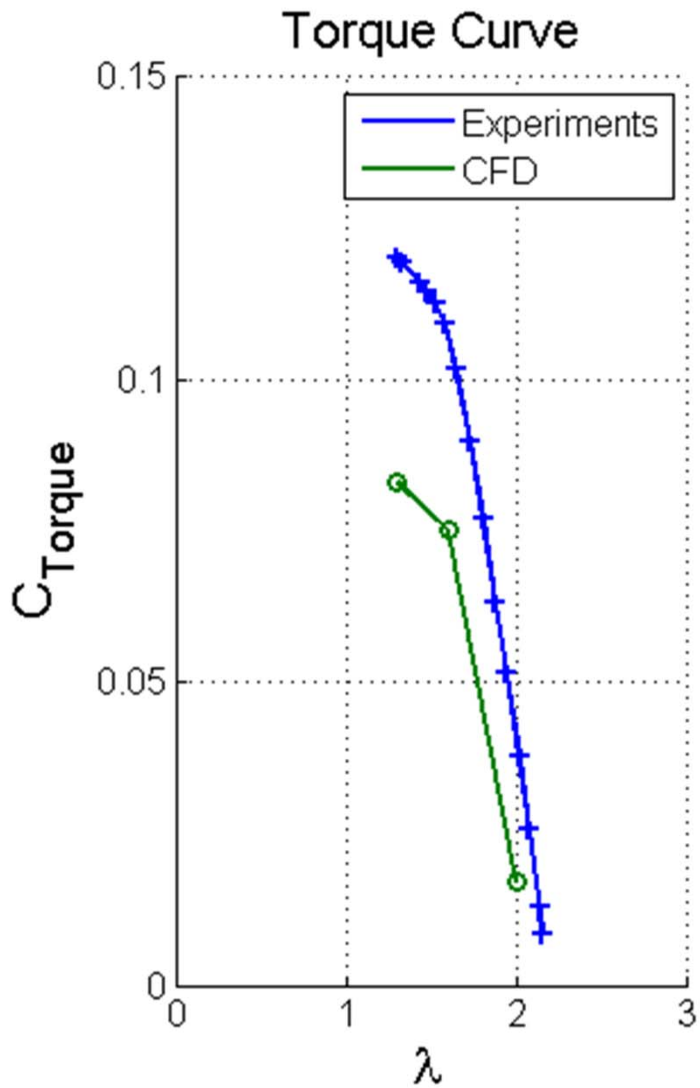
1 Bladed Turbine vs. 4 Bladed Turbine: Effect of Solidity for λ=1.6

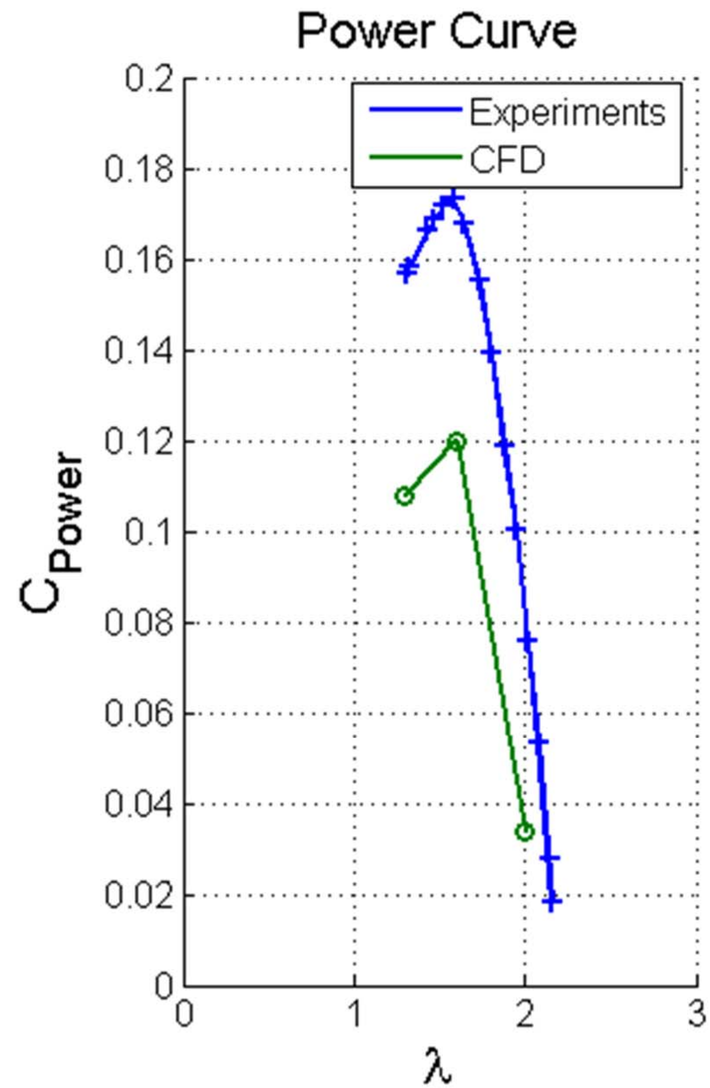


1 Bladed Turbine vs. 4 Bladed Turbine: Effect of Solidity



4 Bladed Rotating Turbine: CFD vs. Experiments





$$C_P = \frac{T}{\frac{1}{2}\rho V_\infty^3 S_{ref} R}$$

$$C_P = \frac{P}{\frac{1}{2}\rho V_\infty^3 S_{ref}} = C_T \lambda$$

Summary and Conclusions

Static Turbine Analysis

- Methodology can accurately model the starting torque of the turbine
- Limitations associated with predictions of separated flow in RANS simulations

Rotating Turbine Analysis

- Direct comparisons between the model and experiments were challenging for a 1-bladed turbine with high levels of experimental angular acceleration
- Limitations modeling dynamic stall
- Simulations for 4 blades show great qualitative agreement
- Promising results for using the methodology for future turbine performance predictions

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