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
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:
\[ \lambda = \frac{\omega R}{V_0} \]

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:
\[ R_{rel} = \frac{\rho V_R C}{\mu} \]

Torque:
\[ T = R(L \cos \alpha - D \sin \alpha) \]
\[ C_T = \frac{T}{\frac{1}{2} \rho V_0^2 S_{ref} R} \]

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

\[ Re_{\text{rel}} = \frac{\rho V_R C}{\mu} \]
Helical Cross Flow Turbine

Micropower generation project as benchmark study:
Adam Niblick and UW Mech. Eng. Capstone Design Team

<table>
<thead>
<tr>
<th>Blade Profile</th>
<th>NACA 0018</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Blades, $N$</td>
<td>4</td>
</tr>
<tr>
<td>Chord Length, $c$</td>
<td>0.040 m</td>
</tr>
<tr>
<td>Radius, $R$</td>
<td>0.086 m</td>
</tr>
<tr>
<td>Height, $H$</td>
<td>0.234 m</td>
</tr>
</tbody>
</table>

Source: Adam Niblick
Experiment/Simulation Parameters

Turbine and Channel Flow

- \( \text{Re}_C = \frac{\rho V_\infty C}{\mu} = 28,000 \)
- \( \text{Aspect Ratio} = \frac{H}{D} = 1.4 \)
- \( \text{Solidity Ratio} = \frac{N_C}{2\pi R} \)
  = 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
\[ \vec{U} = \bar{U} + \bar{u}' \]

Reynolds decomposition of a scalar variable
\[ \phi = \bar{\phi} + \phi' \]

Conservation of mass
\[ \nabla \cdot \bar{U} = 0; \quad \nabla \cdot \vec{u}' = 0 \]

Conservation of momentum
\[ \frac{D \bar{U}_i}{Dt} = \nu \nabla^2 \bar{U}_i - \frac{\partial u_i' u_j'}{\partial x_j} - \frac{1}{\rho} \frac{\partial p}{\partial x_i} \]

- Shear Stress Transport-\(k\omega\) turbulence closure model
- Default turbulence model coefficients
- Low-Reynolds number modeling
Channel Domain and Boundary Conditions

- **Channel Side Wall:** No Slip
- **Channel Free Surface:** Zero Shear
- **Channel Bottom Wall:** No Slip

Dimensions:
- Length: 23.6R
- Width: 8.7R
- Height (H): 2.8R
- Height (Z): 5.0R
Blockage Ratio: Matched to Experiments

\[
\text{Blockage ratio} = \frac{2RH}{\text{Channel Area}} = 0.12
\]
Simulation Cases

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

- **Rotating Turbine**
  - 1 blade
  - 4 blades
Labeling of Helical Blade Position

θ = 45°
Static Turbine: Single Blade

- Particle brake applies constant torque to hold turbine in stationary position: $\lambda=0$
- Torque cell measurements available from experiments
- Repeated at several azimuthal locations
Static Turbine: Single Blade
Flow fields for angles of attack ($\alpha$)
Modeling in Near Wall Region

Wall Functions

Near Wall Approach

Wall Function Approach

Near-Wall Model Approach
Modeling in Near Wall Region

Wall Functions

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

Near Wall Approach

- $y^+ < 1$
- $\frac{\text{First Length}}{\text{Chord Length}} = 0.0001$
- 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

Graph: Static Torque: 4 Blades

- C_Torque on the y-axis
- \( \theta \) (degrees) on the x-axis
- Data points represent experimental and CFD results.
Superposition of Single Blades vs. 4 Bladed Turbine
Superposition of Single Blades vs. 4 Bladed Turbine

CFD:

Experiments:
Single Blade
vs. 1 blade from 4 bladed turbine

VS.

VS.

VS.

VS.
Single Blade vs. 1 blade from 4 bladed turbine

Experiment Static Torque: Per Blade

- 4 bladed turbine, per blade
- 1 bladed turbine

θ (degrees)
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**
  - $\lambda=3.2$
  - $\lambda=3.6$ (no load)

- **CFD**
  - $\lambda=3.2$
  - $\lambda=3.6$
  - $\lambda=1.6$
Dynamic Torque: Single Blade

\[ C_{\text{Torque}} \]

\[ \theta \text{ (degrees)} \]

<table>
<thead>
<tr>
<th>( \lambda )</th>
<th>( C_{\text{avg Torque}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.6</td>
<td>0.047</td>
</tr>
<tr>
<td>3.2</td>
<td>0.065</td>
</tr>
<tr>
<td>3.6</td>
<td>0.057</td>
</tr>
</tbody>
</table>
Theoretical Angle of Attack

\[ \alpha, \text{Degrees} \]

Azimuthal Position, \( \theta \) (Degrees)

\( \lambda = 1.6 \quad \lambda = 3.2 \quad \lambda = 3.6 \)

Theoretical \( \text{Re}_{\text{REL}} \)

\[ \times 10^4 \]

Re\(_{\text{REL}} \)

Azimuthal Position, (Degrees)
Rotating Single Bladed Turbine: CFD vs. Experiment

Dynamic Torque for a Single-Bladed Turbine at $\lambda=3.2$
Oscillating Rotational Velocity

\[ \lambda_{avg} = 3.2 \]

\[ \lambda_{avg} = 3.6 \]
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

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

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

\[ T_{Hydrodynamic} \approx T_{ParticleBrake} \quad \text{if} \quad \frac{d\omega}{dt} \quad \text{and} \quad T_{SystemFriction} \approx 0 \]

\[ \int_{0}^{360^\circ} I \frac{d\omega}{dt} \, d\theta = 0 \]

\[ \frac{T_{Hydrodynamic}}{T_{ParticleBrake} + T_{SystemFriction}} = \frac{\int_{0}^{360^\circ} I \frac{d\omega}{dt} \, d\theta}{0} \]
Experiment Single Blade, $\lambda_{AVG} = 3.2$
Model Validation

Differences Attributed To:

• Constant vs. varying tip speed ratio

• System Frictional Torque

<table>
<thead>
<tr>
<th>$\lambda$</th>
<th>Experiment</th>
<th>CFD</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.2</td>
<td>0.032</td>
<td>0.065</td>
</tr>
<tr>
<td>3.6</td>
<td>0.007</td>
<td>0.057</td>
</tr>
</tbody>
</table>
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
  - $\lambda=1.3$ to 2.1

- CFD
  - $\lambda=1.3, 1.6, 2.0$
Rotating 4 Bladed Turbine
1 Bladed Turbine vs. 4 Bladed Turbine: Effect of Solidity for $\lambda=1.6$
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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