Characteristics of fluid exciting force due to blade row interaction of a propeller turbine

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Abstract. As the use of renewable energy is promoted, much researches on propeller-type turbines are carried out. To improve reliability, it is important to evaluate the fluid exciting force due to blade row interaction that may cause vibration or fatigue fracture. In this paper, CFD(URANS) simulations and experimental studies were conducted to understand the characteristics of the fluid exciting force due to blade row interaction of a propeller turbine. To evaluate the fluid exciting force, strain gauges and pressure sensors were used in the closed-loop water channel. The fluid exciting force acting on the rotor due to the stator was measured by changing stator load type and the rotor-stator distance. Based on the difference in the attenuation tendency of potential interaction and wake interaction due to changes in the rotor-stator distance, the influences of these interactions could be distinguished. As results of experiments and computation, wake interaction affecting pressure fluctuation was hardly attenuated due to the swirling flow and changing the rotor-stator distance has little effect on the attenuation of blade row interaction.

Keywords: fluid exciting force, blade row interaction, axial flow turbine.

Introduction

In recent years, expectations for renewable energy has increased due to environmental considerations, and counter-rotating type horizontal axis tidal turbine have been developed. The fluid exciting force due to blade row interaction is generated, which causes vibration and increases risks of fatigue. Therefore, it is necessary to predict the fluid exciting force. Kemp [1] et al. studied the lift fluctuation due to blade row interaction and show that lift fluctuation due to blade potential interaction decreases exponentially with the blade to stator distance. Funazaki [2] et al. studied the relationship between the blade and stator distance and aerodynamic performance using an axial flow turbine and show that as distance between the blade and stator increased, a wall boundary layer flowing into the blade developed, and a strong...
secondary flow was induced. However, in hydraulic turbines, studies evaluating fluid exciting forces due to blade row interaction and the impact of design parameters modification are not numerous enough. In this paper, as a part of the NEDO (New Energy and Industrial Technology Development Organization) project, to evaluate the fluid exciting force due to blade row interaction of a propeller turbine that is the basis of counter-rotating type horizontal axis tidal turbine is evaluated. Experiments and CFD (computational fluid dynamics) simulations were conducted as a continuation of Ajiro [3] et al.’s research. The fluid exciting force was measured depending on the existence of swirl and the rotor-stator distance, experimentally and computationally.

1. Methods

1.1. Axial turbine experiment. An overview of the test apparatus is shown in Figure 1. This apparatus is a closed-loop water channel. The flow rate is adjusted by changing the rotational speed by manipulating the frequency of the inverter of the pump connected in parallel. An enlarged view of the axial flow hydro turbine is shown in Figure 2. The maximum flow velocity is 4 m/s and the rotor rotational speed can be varied from 600 to 1200 rpm. The rotor-stator distance is variable by steps of about 5% of the rotor diameter $D$ by using spacers. The rotor is a MEL 009 type hydrofoil type. The tow stator (one induces swirling and the other does not) are both NACA0012 hydrofoil type. The tip clearance is 0.5 mm. Other parameters of the axial turbine are given in Table 1. Since the swirl is generated by the loaded stator, the rotational speed is different compared to the unloaded stator to adjust the inflow angle of the rotor. A strain gauge was attached to the root of the rotor to measure the axial force, and a pressure sensor was attached to the mid pressure surface of the rotor to measure surface pressure on the rotor. A slip ring was used to convert signals from the rotating system to the stationary system. Each value is recorded by a universal recorder (EDX-200A).

![Figure 1. Overview of the test apparatus](image1)

![Figure 2. Enlarged view of the axial turbine](image2)

| Items                          | Values         |
|-------------------------------|----------------|
| Number of loaded stator blades $Z_{ls}$ | 8              |
| Number of unloaded stator blades $Z_{s}$ | 4              |
| Number of rotor blades $Z_{r}$   | 5              |
| Rotor tip Diameter $D$ m        | 0.2534         |
| Design point tip speed ratio $TSR$ | 3.68          |
| Design point flow rate $Q$ m$^3$/s | 0.165         |
| Rotational speed (Loaded) $N$ rpm | 1064          |
| Rotational speed (Unloaded) $N$ rpm | 900           |
| The rotor-stator distance $d/D$ | 0.0668-0.362  |

1.2. CFD Simulation. Flow in the axial turbine is investigated by CFD simulation with the commercial software ANSYS CFX 17.2. URANS (Unsteady Reynolds-Averaged Navier-Stokes equation)
simulations using the SST k-ω (Shear Stress Transport) turbulence model. Simulations are carried out to evaluate the time fluctuation of pressure and the axial force on the rotor induced by the rotor-stator interaction. According to Ino et al. [4,5], the SST k-ω turbulence model is suitable for prediction of the flow near the blade surface of pumps and turbines. The analysis domain is shown in Figure 3. The model includes the rotor, stator, the struts to hold the rotor and stator and an L-shaped elbow located at the rotor outlet. The meshing is mainly made of tetrahedra elements. Near the wall surface, prism layer meshing is used to simulate the boundary layer. Information on the meshing is given in Table 2. The created meshing is shown in Figure 4. The CFD conditions are summarized in Table 3. The influence of gravity is not considered in this simulation.

### Table 2. Meshing details

| Domains | Nodes (k) | y+ |
|---------|-----------|----|
| Rotor   | 5212      | 2.5|
| Stator  | 3526      | 2.5|
| Strut   | 1549      | 10 |
| Total   | 13218     |    |

**Figure 3.** View of CFD domains

**Figure 4.** Meshing configuration

### 2. Results and Discussion

#### 2.1. Comparison of experiments and CFD results

In this chapter, the comparison results of the tendency of the rotor-stator blade row interaction by changing tow factors. They are the rotor-stator distance and stator load type that induces swirling flow or not. The reason for the comparison between the two types of stators is that the unloaded stator does not create a pressure field, so it is possible to grasp mainly the wake interaction. Then, both the wake interaction and the potential interaction can be grasped by the loaded stator that creates a pressure field.

#### 2.1.1. Unloaded stator

The axial force and the rotor surface pressure at the design point flow rate were measured, and FFT (Fast Fourier Transform) processing was performed at each the rotor-stator distance. Figure 5,6 shows the time history waveform of the axial force fluctuation rate and pressure fluctuation during one rotation of the rotor of comparison between experiment and CFD result at d/D=0.067. The axial force fluctuation is the value divided by the average value and pressure fluctuation takes average value to 0.
Figure 5. Waveform of the axial force fluctuation rate (Unloaded)

Figure 6. Waveform of pressure fluctuation (Unloaded)

Figure 5, 6 both show short-period fluctuations that occurs four cycles per rotation. In addition, there are long-period fluctuations that occurs during one cycle per rotation for pressure fluctuation. In short-period fluctuation, there are slowly varying components due to the potential interaction and rapidly varying components due to wake interactions. In Figure 5, the high-frequency fluctuation of the experimental results at the time of the fluctuation rate rise is an amplitude due to Karman vortex, and it is considered that it does not exist in the waveform of the CFD result because of RANS. The Karman vortex will be described in this chapter later. So that the experimental and simulated waveforms qualitatively coincided. On the other hand, in Figure 6, the waveform is different between the experimental value and the CFD result value. When considering the gravitational field on long-period fluctuation is considered, these occur because the water depth changes depending on the position of the rotor when it rotates. In other words, this is the static pressure fluctuation due to the gravitational field, and the theoretical value of the fluctuation is obtained as follows.

\[ \Delta P = \rho g \Delta h = 1.4 \text{ kPa} \]  

This value corresponds to the long-period fluctuation amplitude. When considering that the gravity is ignored in the CFD simulations, the rapid decrease in pressure fluctuation due to wake interaction is quantitatively consistent between the analytical and experimental values, but the recovery process of the decrease is different. This is due to underestimation of wake interaction due to the overestimation of velocity defect. The decrease in pressure due to the wake interaction was quantitatively consistent. In fact, the waveform was different in pressure recovery because the wake interaction was not correctly represented. Since the pressure recovery tendency is consistent, it is considered that potential interaction is represented.

Figure 7, 8 shows comparisons of experimental and simulated FFT results at \( d/D = 0.067 \).

Figure 7. FFT of the axial force (Unloaded)

Figure 8. FFT of the pressure (Unloaded)
The non dimensional frequency is obtained by dividing the frequency by the rotor rotational frequency \( N \). When the rotational frequency is \( N \) and the number of stator blades is \( Z_s \), the amplitude decreases as \( NZ_s \) become higher-order. It can be confirmed that the amplitude value around the \( 7NZ_s \) component is large only in experiments. This is a peak due to the Karman vortex. The frequency of the Karman vortex is calculated the Strouhal number.

\[
St = \frac{fD}{U}
\]

In this experiment, the Strouhal number when the Karman vortex is generated is about 0.2, and \( D = 0.0016 \text{ m} \) and \( U = 3.4 \text{ m / s} \), so that \( f = 425 \text{ Hz} \) and \( f/N = 28.33 \). So, it can be confirmed that the peak of \( 28N (= 7NZ_s) \) is also affected by the Karman vortex.

There are two type of blade row interactions, wake interaction and potential interaction. Generally, wake interaction affects \( nNZ \) components and potential interaction affects only the \( NZ \) component. Attention is paid to the amplitude values of the \( NZ_s \), \( 2NZ_s \), \( 3NZ_s \), and \( 4NZ_s \) components, and the attenuation tendency when the \( NZ_s \) components become higher is compared. Figures 9,10 show the comparison.

\( NZ_s \) component attenuation of CFD result is larger for both the axial force and the pressure, and the amplitude value attenuates more rapidly from the \( NZ_s \) component to the \( 2NZ_s \) component and later than the experimental value. It is also confirmed here that the simulated wake interaction is weaker than the experimental value. The experimental values and the analytical values are qualitatively consistent since the attenuation tendency after the \( 2NZ_s \) component is almost the same.

The attenuation tendency of the \( NZ_s \) component when the rotor-stator distance is changed is compared. Figure 11,12 show the comparison.
In Figure 11, CFD result value exceeds experiment values at all distances, but the attenuation tendencies of the amplitude values are like the experimental values, they are qualitatively consistent. In Figure 12, the CFD result value was lower than the experiment value, but the tendency of the change of the amplitude value was similar, and a qualitative agreement is confirmed. Amplitude value of the axial force fluctuation decreased as the distance increased, whereas amplitude value of fluctuation increased once as the distance increased and then decreased thereafter. This tendency is considered that the influence of the wake interaction of the strut becomes stronger as the stator approaches the strut as the rotor-stator distance increases. Since the experimental value and the analytical value agree qualitatively, it is considered that the phenomenon also occurs in the CFD result value.

2.1.2. Loaded stator: Figure 13, 14 shows the time history waveform of the axial force fluctuation and pressure fluctuation on one rotation of the rotor of comparison between experiment and CFD result at $d/D=0.067$ alike that of unloaded stator.

First, consider the results of each experiment. Figure 13 shows there are two types of fluctuations, fluctuation that occurs eight cycles per rotation and fluctuation that occurs four times per rotation shown in green. Fluctuation that occurs eight cycles is the number of stator blades, so they are wake interaction components of the stator. The green fluctuation is the wake interference component due to the wake interaction of the strut. Figure 14 shows there are two types of fluctuations that occurred in four cycles per one rotation: a large change in pressure, indicated by a black circle, and a small change, indicated by a green circle. They occur at approximately equal intervals, and when combined, the fluctuations are equally spaced for eight cyclers per rotation. This waveform was obtained as a result of the strut wake interaction overlapping four of the eight stator blades. The reason why the waveforms differed despite the influence of the struts was due to the attached position of the sensor and the shape of the loaded.
The surface pressure shows the pressure waveform at the mid because the attached position is at the mid. On the other hand, the axial force waveform represents the axial force applied to the entire rotor blade, and the force at the tip is strongly affected. Loaded stator are shaped so that turning becomes stronger from hub toward tip as shown in Figure 15,16. In mid, the strut wake interaction overlaps with the stator wake interaction, but in tip, the strut wake interaction passes between the stator and does not overlap with the stator wake interaction.

![Image](strut_stator_mid.png) ![Image](strut_stator_tip.png)

**Figure 15.** Strut wake at mid surface  
**Figure 16.** Strut wake at tip surface

Next, the experiment results are compared with the CFD result. In figure 13, there was no agreement between the waveforms of the experiment value and the CFD result value. In the wake interaction component of stator, the experiment values show that there are eight variations of the same magnitude for the number of stator blade, while the CFD result values show that the variations of the two types of magnitude decrease every four times. In the experiment, the wake interaction of the strut passed between the stator blade and did not interfere with the stator, but in the CFD, the wake width is wider than the actual phenomenon and overlaps the stator blade. In Figure 14, the waveform is quantitatively consistent between the experiment value and the CFD result value considering the non-setting of the gravitational field in the analysis.

Figure 17,18 show comparison of FFT results about experiment and CFD result at $d/D=0.067$.

![Image](fft_comparison.png)

**Figure 17.** Comparison of the axial force FFT (Loaded)  
**Figure 18.** Comparison of pressure FFT (Loaded)

Loaded stator has eight blades and there are peaks at $0.5NZls$ that is affected by strut, $NZls$, $2NZls$, $3NZls$ and $4NZls$. In Figure 17, it was confirmed that the variation appeared four cycles in one rotation and the variation occurred eight cycles in Figure 13 were superimposed. In figure 18, it decreases rapidly when the $NZls$ component becomes higher-order. It is confirmed that the loaded stator has a large potential interaction component because it is similar to the attenuation tendency of general potential interaction.
In addition, it can be confirmed that the Karman vortex effects near the $3NZls$ component as in the case of unloaded stator.

Attention is paid to the amplitude values of the $NZls$, $2NZls$, $3NZls$, $4NZls$ and $0.5NZls$ components, and the attenuation tendency when the $NZls$ components become higher is compared. Figures 19,20 show the comparison results.

**Figure 19.** FFT amplitude of the axial force fluctuations at each non dimensional frequency (Loaded)

In Figure 19, the amplitude value of $NZls$ and $0.5NZls$ is different between experiment and CFD result, but the attenuation tendency from $NZls$ component to higher-order component is qualitatively consistent. Attenuation of CFD result from $2NLs$ component to higher-order component is smaller than that of experiment, this is because the amplitudes are small. In Figure 20, the effect of the Karman vortex is visible, but the attenuation is qualitatively consistent.

The attenuation tendency of the $NZls$ component when the rotor-stator distance is changed is compared. Figure 21,22 show the comparison.

**Figure 20.** FFT amplitude of pressure fluctuations at each non dimensional frequency (Loaded)

In Figure 21, CFD result value falls below experiment values at all distances, but the attenuation tendencies of the amplitude values are like the experimental values, they are qualitatively consistent. In Figure 22, the CFD result values have almost the same attenuate tendency as the experiment values that hardly attenuate even if the rotor-stator distance and are qualitatively consistent.

**Figure 21.** FFT amplitude of the axial force fluctuations at the frequency equal to $NZls$

**Figure 22.** FFT amplitude of pressure fluctuations at the frequency equal to $NZls$
2.2. Comparison of unloaded stator and loaded stator. In the previous chapter, the experiment and CFD result were in qualitative agreement. In this chapter, the attenuation tendency due to swirling flow are compared. Figure 23,24 shows the amplitude values of the $NZ$, $2NZ$, $3NZ$, and $4NZ$ components of the pressure at $d / D = 0.067$ and the amplitude values of the $NZ$ component of pressure when the rotor-stator distance was changed.

![Figure 23](image1.png)  
**Figure 23.** FFT amplitude of pressure fluctuations at each non dimensional frequency

![Figure 24](image2.png)  
**Figure 24.** FFT amplitude of pressure fluctuations at the frequency equal to $NZls$

In Figure 23, comparing the attenuate tendency from the $NZ$ component to the $2NZ$ component, the value of loaded stator is larger than that of unloaded stator. This is because the potential interaction becomes stronger due to the swirling flow in the interaction of the loaded stator, and the potential interaction component is stronger than that of the unloaded stator. In Figure 24, the amplitude value changes when there is unload stator due to the change in the rotor-stator distance but does not change when the stator is unloaded. This is because the swirling flow caused strong interaction of stator wake interaction and attenuate blade row interaction a little.

3 Conclusions

(1) In the unsteady analysis of unloaded stator, the wake interaction component became smaller than the experiment value due to the overestimation of the velocity defect for both the axial force and pressure.

(2) In the unsteady analysis of the loaded stator, the experimental and computational waveforms of the axial force were different due to the velocity defect and overestimation of the wake width of the strut wake. The attenuation of the CFD results value was larger than the experiment value. However, a qualitative agreement was obtained. On the other hand, pressure fluctuation was quantitatively consistent. This is because the swirling flow makes it difficult for the wake interaction of stator to be attenuated.

(3) As the rotor-stator distance increased and the stator approached the strut, the axial force decreased and the pressure at the unloaded stator decreased, but pressure of the loaded stator did not decrease.

(4) Due to the swirling flow, the attenuation of the pressure from the $NZls$ component to the $2NZls$ component increased, and it shows that potential interaction was dominant.

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