Scale effect and undersea noise of counter-rotating propellers installed in tidal stream power unit

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Abstract. More attentions are paid to the renewable energies in face of global warming and depletion of fossil fuel. Tidal energy as one of the renewable energies has fruitful advantages as predictable and sustainable resources. The authors have proposed a counter-rotating type tidal stream power unit and investigated preliminarily the effect of the Reynolds number on the output, namely scale effect, and undersea noise of the counter-rotating propellers. The hydraulic loss increases at the low Reynolds number with accompanying the boundary layer separation. The noise has the dominant frequencies corresponding to the blade passing frequencies in counter-rotating propellers.

1. Introduction

With the rapid growth of fuel consumption, renewable energy especially ocean resource more than 2x10³ TW which can be exploited has been attracting human’s attention. Horizontal-axis tidal stream turbines have been proposed to effectively get tidal energy from the ocean, for instance Reference [2]. The optimum design of counter-rotating propellers installed in the tidal stream power unit has also been proposed and investigated the performances [3-5].

It is important to guarantee the efficiency, namely the output coefficient, of the prototype conversion from the model test results [6]. That is, in this paper, the effects of the Reynolds number on the hydraulic loss and the flow condition around a two dimensional blade are predicted preliminarily by CFD. Besides, the undersea noise is also investigated preliminarily by the experiment in the water tunnel because the counter-rotating propellers bring the flow interaction [7].

2. Effects of Reynolds number

2.1. Flow simulation

The two dimensional blade originates from the mid-span section of the front propeller blade on a counter rotating propellers, which is used in the undersea noise test in the water tunnel. The chord length of the blade element is 22.35 mm. The computational domain surrounding the blade element in the numerical simulation is shown in Figure 1. The dimensions are 1000 mm in the stream direction and 700 mm in the vertical direction (wide), while the blade element is placed at the center of the domain where the leading edge is placed at 400 mm downstream from the inlet/datum section of the domain.

The flow around the blade element is simulated by the numerical commercial code Ansys 17.2. The domain area is filled with structured quadrilateral grids by employing blocks method in ICEM CFD. The meshes around the blade element are refined to make the Y⁺ acceptably small as possible. The number of mesh nodes is about 2.09 million. The SST k-ω turbulence model in ANSYS Fluent was
applied to solve the incompressible Reynolds-average Navier-Stokes equations, and the energy equation is ignored in all the simulations. The uniform velocity distribution and the outflow boundary conditions were given at the inlet/datum section and other sections surrounding the domain. The no-slip and smooth walls were imposed on the surfaces of the blade element. The absolute convergence criterion was set no more than $10^{-3}$ to all the residuals values in the turbulent model in this code.

![Computational domain](image)

**Figure 1.** Computational domain

### 2.2. Characteristics of blade element

Figure 2 presents the lift-to-drag ratio $C_l/C_d$ against Reynolds number $Re = \frac{\rho UL}{\mu}$, $\rho$: fluid density, $U$: inlet velocity, $L$: blade chord, $\mu$: fluid viscosity) in various with angle of attack $\alpha$. The ratio increases with the increase of the Reynolds number at the range of $Re$ from $10^3$ to $10^6$, where the smaller Reynolds number maybe in the transition or the laminar flow conditions. Figure 3 presents the characteristics of lift and drag coefficients against angle of attack at $Re=10^5$ and $Re=5 \times 10^5$. The drag coefficient increases slightly with the increase of the angle of attack and the lift coefficient increases at the angle of attack smaller than about 16 degrees. At the angle larger than 16 degrees, the drag immediately increases with accompanying stall, especially at $Re=10^5$. The stall brings the decrease of the lift coefficient. The lift coefficient is higher and the drag coefficient is lower at the larger Reynolds number.

![Effect of the Reynolds number on the lift-to-drag ratio](image)

**Figure 2.** Effect of the Reynolds number on the lift-to-drag ratio

![Effect of the angle of attack on the lift and drag coefficients](image)

**Figure 3.** Effect of the angle of attack on the lift and drag coefficients
2.3. Total pressure loss

Figure 4 presents the total pressure loss versus the Reynolds number at different angles of attack, where the total pressure loss is estimated with the following equation:

$$\zeta = \frac{p_1 - p_2}{\rho U^2}$$

(1)

$p_1$ and $p_2$ is mass-weighted average total pressures at the inlet and the outlet sections of computational domain shown in Figure 1. The total pressure loss decreases as the Reynolds number increases and the decrement is sensitive at the angle of attack $\alpha = 12$ degrees, though the actual flow may be in the laminar or the transition at the smaller Reynolds number. The total pressure losses are scarcely affected by the angle of attack $\alpha$ at the larger Reynolds number $Re$ but affected obviously by $\alpha$ at smaller $Re$. Figure 5 shows the mixing loss coefficient $\zeta_m$ estimated at the downstream of the blade, where $p_1$ was changed from the inlet of the computational domain to the section of the trailing edge. The losses against the $Re$ have similar profiles, that is, the mixing loss may contribute to the total pressure loss.

As is well known, the output efficiency is grossly affected by the Reynolds number, which is called “Scale effect”. In the numerical simulation, the Reynolds number is changed easily with the velocity $U$, the dimension $L$ or the kinematic viscosity $\nu$. Figure 6 and 7 show the total loss coefficient $\zeta$ and the mixing loss coefficient $\zeta_m$ against the Reynolds number by changing the dimension $L$, where $L_0$ is the chord of the original blade element shown in Figure 1. At the same Reynolds number, while keeping the $LU$ constant, the loss coefficients of $L/L_0 = 0.5$, namely $U/U_0 = 2$, has the lowest value and they are increasing a little bit at $L/L_0 = 2$. Furthermore, the kinematic viscosity is also in consideration of variable and the results show the loss keeps constant in the same Reynolds number. These results suggest that the scale effect must be discussed carefully while evaluating the numerical results.

![Figure 4. Effect of the Reynolds number on the total pressure loss coefficient](image1)

![Figure 5. Effect of the Reynolds number on the mixing loss coefficient](image2)

![Figure 6. Total pressure loss coefficient in various with the blade chord](image3)

![Figure 7. Mixing loss coefficient in various with the blade chord](image4)
2.4. Flow conditions
Figure 8 shows the effect of Reynolds number $Re$ on the velocity distributions around the blade element at the angle of attack $\alpha=8$ degrees, where the flow near the trailing edge is enlarged with square view port. The reverse flow appears at the suction surface in close to the trailing edge, and large $Re$ promotes more or less to develop the vortex which scarcely affects the loss.

3. Undersea noise
The model power unit with tandem propellers was provided for the water tunnel having the cross section with 1000 mm (width) x 700 mm (depth), where the diameters of the front and the rear propeller are 250 mm with three blades and 237.5 mm with five blades. The hydrophone OST2130 is set at the same as the depth of the axis of the counter-rotating propellers and 500 mm horizontally crossways from the center of the front propeller. The signal is amplified and transmitted to the FFT system.

Figure 9 shows the sound pressure level distributions of the counter-rotating propellers at the stream velocity 1 m/s, where the counter-rotational speeds of the front and the rear propellers are 239 min$^{-1}$ and 237 min$^{-1}$. The circle points in Figure 9 (b) are the dominant and the harmony frequencies of the counter-rotating propellers. They are derived from the Hanson equation \[f = |mZ_RN_R + kZ_FN_F|\] (2)
where, $f$ is the frequency, $m$ and $k$ is integral numbers, $Z$ and $N$ is the blade number and the rotating speed of propellers (subscripts F and R denote the front and the rear). The peak sound pressure levels are almost focus on the 60Hz and its times numbers but they may be ignored due to the frequency of the...
grid system in Japan while evaluating the undersea noise. The blade passing frequency and the synthesized frequencies predicted by Hanson formula scatter widely and it cannot discuss clearly the noise from the counter-rotating propellers, at this stage.

![Sound pressure level of counter-rotating propellers](image)

**Figure 9.** Sound pressure level of counter-rotating propellers

4. **Concluding Remarks**

This paper investigates preliminarily the effects of the Reynolds number on the hydraulic losses in the two dimensional blade and the undersea noise of the counter-rotating propellers. The scale effect must be discussed carefully while evaluating the numerical results, because the Reynolds number can be changed easily with the velocity, the dimension and the kinematic viscosity while taking the same
Reynolds number. The blade passing frequency scatters widely but it cannot discuss clearly the noise from the counter-rotating propellers, at this stage

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