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A marine propeller as a hydrokinetic turbine – CFD analysis of energy characteristics

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Abstract. It is sometimes economically unreasonable to develop specific turbines, but rather to use a well-known concept of a pump in the turbine mode – PAT (Pump As Turbine). Inspired by PAT, the scope of this paper covers the assessments of the possibility to use marine propellers as runners for the hydrokinetic turbines. Numerical analyses are conducted for selected propeller geometry with the aim of evaluating its energy characteristics in the turbine regime. The chosen propeller belongs to the conventional, thoroughly experimentally tested, Wageningen B-screw series. The experimental data were used for validation of the numerical results in propeller regimes. Based on the obtained results, the researched non-optimized turbine in its optimum regime achieves a satisfactory power coefficient which is close to the value of other types of contemporary hydrokinetic turbines efficiencies.

1. Introduction

With the development of technology and society lays the need for energy production. Due to the continuous pollution and global climate change, the energy production is shifted towards the usage of renewable resources. As there is a big amount of energy contained in the motion of water, hydrokinetic turbines (HKTs) are employed to harvest it. Unfortunately, many of these HKTs are not as cost efficient as other technologies.

On the other hand, on some conventional small hydropower plants, it is sometimes economically unreasonable to develop specific turbines, but rather to use a well-known concept of a pump in the turbine mode – PAT (Pump As Turbine). Pumps employed in the turbine regime with less than 0.5 MW usually have a capital payback period of up to two years [1]. Inspired by the concept of PAT, an assessment was made of the possibility to use marine propellers as runners for the hydrokinetic turbines.

The propeller's technical task is to provide as much thrust as possible using minimum torque for a desired rotational speed. However, in the turbine regime, the technical task is to provide as much angular



momentum as possible with a minimum axial force acting on the propeller for a desired rotational speed. Such conditions, in the turbine regime, require a careful choice of the propeller type.

Numerical analyses are conducted for the selected propeller geometry with the aim of evaluating its energy characteristics in the turbine regime [2]. The chosen propeller belongs to the conventional, thoroughly experimentally tested, Wageningen B-screw series. The experimental data [3] were used for validation of the numerical results in propeller regimes.

Recently published papers [4,5] illustrate fairly good tendency in validating propeller characteristics obtained by contemporary numerical simulations with experimental results. Thus, CFD analysis of propeller characteristics has become a common way for simulating and gathering reliable results. After validating the numerical results of energy parameters in the propeller regimes, the CFD analyses were performed in turbine modes. Such researched non-optimized turbine in its optimum turbine regime achieves a satisfactory power coefficient which is close to the value of other types of contemporary hydrokinetic turbines efficiencies.

2. Numerical model

2.1. Geometry

The propeller geometry and flow domain are presented in Figure 1.

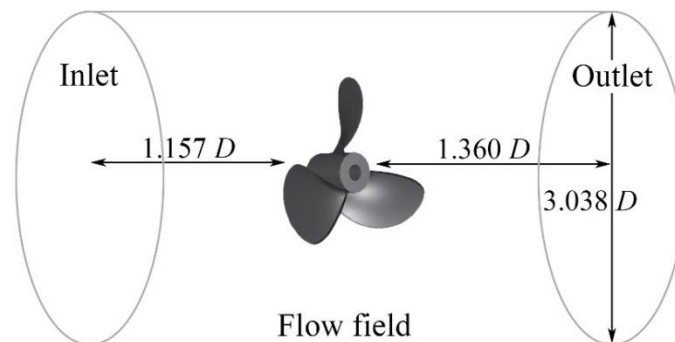


Figure 1. – The geometry of the numerical model

Since the propeller belongs to the Wageningen B-series, its profile geometry is standardized (details can be seen in [6]), and thus it is sufficient to define only several parameters (Table 1).

Table 1. – The propeller's geometry

Geometry parameter	Value
Propeller diameter D	344 mm
Hub-diameter ratio d_h/D	0.2
Number of blades z	3
Pitch ratio P/D	1.2
Blade area ratio A_e/A_o	0.5
Propeller skew angle θ_{sp}	16.5°
Propeller rake θ_r	15°
Hub length	230 mm

2.2 Mesh

The mesh consists of two sub-mesh domains – a cylindrical mesh used for the flow field domain around the propeller and the other one, used for the propeller domain. Values of absolute errors of the open water efficiency η_o were used to evaluate the mesh quality (Figure 2). The absolute error drastically drops for a mesh of more than 3 million elements and then slowly approaches zero. The chosen mesh has around 6 million elements, giving less than 1% absolute error, which was, for the purpose of evaluation, considered good enough.

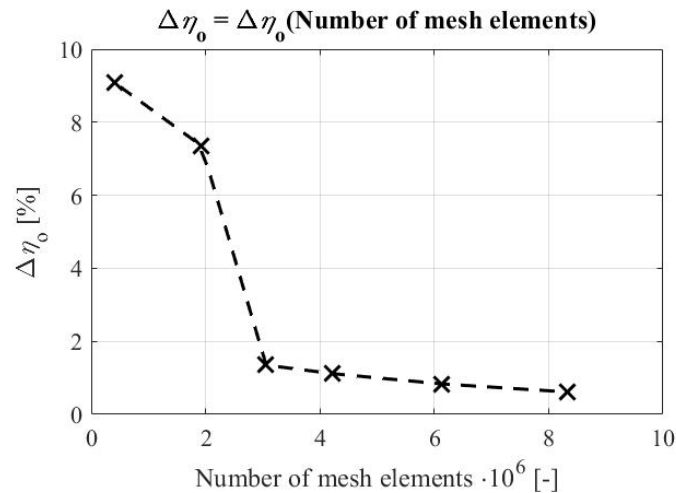


Figure 2. – Absolute error of the open water efficiency η_o as a function of mesh element number

2.3. Boundary conditions

The boundary conditions are setup separately for the propeller regime and for the turbine regime (Table 2). The only difference is that in the turbine regime, the main flow is in the opposite direction (the inlet becomes the outlet and vice versa) and the runner rotation is also in the opposite direction compared to the propeller regime.

Table 2. – Boundary conditions

Boundary	Given value in the propeller regime	Given value in the turbine regime
Inlet	Velocity	Pressure
Outlet	Pressure	Velocity
Shell (cylinder)		Pressure
Propeller domain		Rotating
Rotation	Clockwise	Counterclockwise

No measures were taken for optimizing the propeller for the turbine regime. Further details that define the numerical model are given (Table 3). They are chosen as to achieve compromise between computing time, simplicity and accuracy.

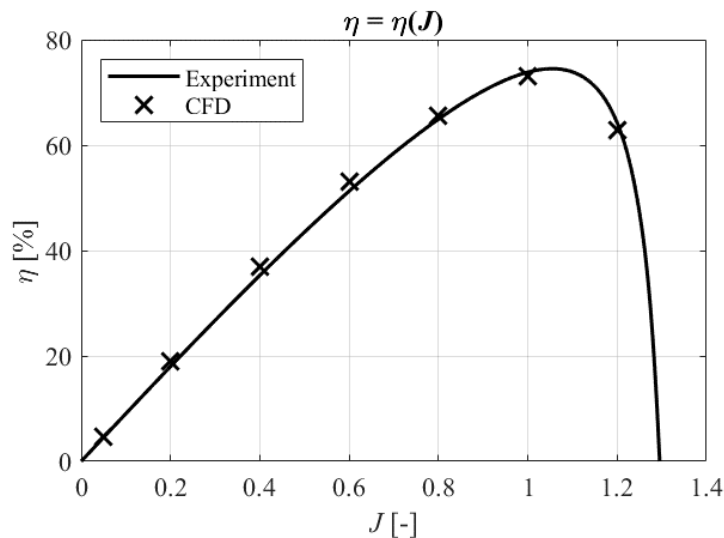
Table 3. – Numerical model details

Flow type	Stationary
Propeller domain interface	Frozen rotor
Working fluid	Water
Flow type	Isothermal
Turbulent model	SST

3. Results

3.1. Propeller regime

The numerical results of open flow propeller efficiency, defined as $\eta_o = \frac{K_T J}{K_Q 2\pi}$, are validated by the experimental results (Figure 3). The numerically calculated torque coefficient, defined as $K_Q = \frac{Q}{\rho n^2 D^5}$, does not quite fit the experimental results in wide operating range, but only for higher values of the advance coefficient (Figure 4). Advance coefficient is defined as $J = \frac{v_a}{nD}$, where Q [Nm] is the torque, ρ $\left[\frac{\text{kg}}{\text{m}^3}\right]$ is the water density, n $[\text{s}^{-1}]$ is the rotational speed, D [m] is the propeller diameter, and v_a $\left[\frac{\text{m}}{\text{s}}\right]$ is the speed of advance. The same trend of agreement follows the thrust coefficient $K_T = \frac{T}{\rho n^2 D^4}$, where T [N] is the thrust.

**Figure 3.** – Open flow efficiency: comparison of the CFD and experimental results

Deviations of the torque coefficient are most probably caused because the experimental results were performed on a prototype propeller that has a cap on. This tends to distribute the more uniform velocity field, and consequently the pressure field.

The obtained results were found satisfying enough to continue numerical testing of the propeller in the turbine regime.

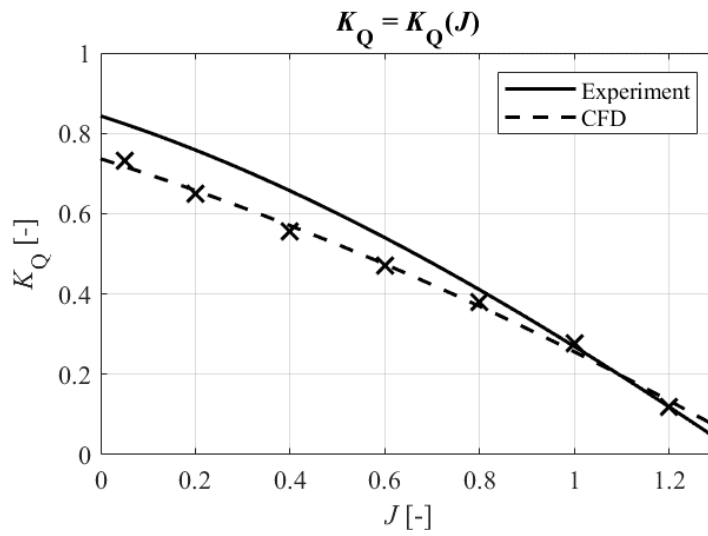


Figure 4. – Torque coefficient: comparison of the CFD and experimental results

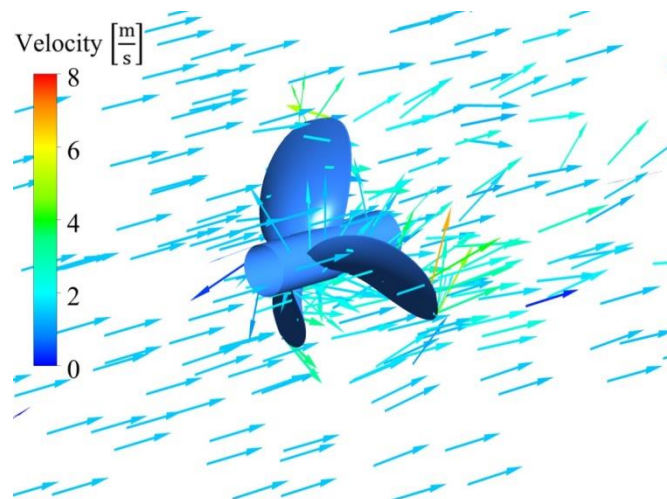


Figure 5. – A velocity field in the propeller regime

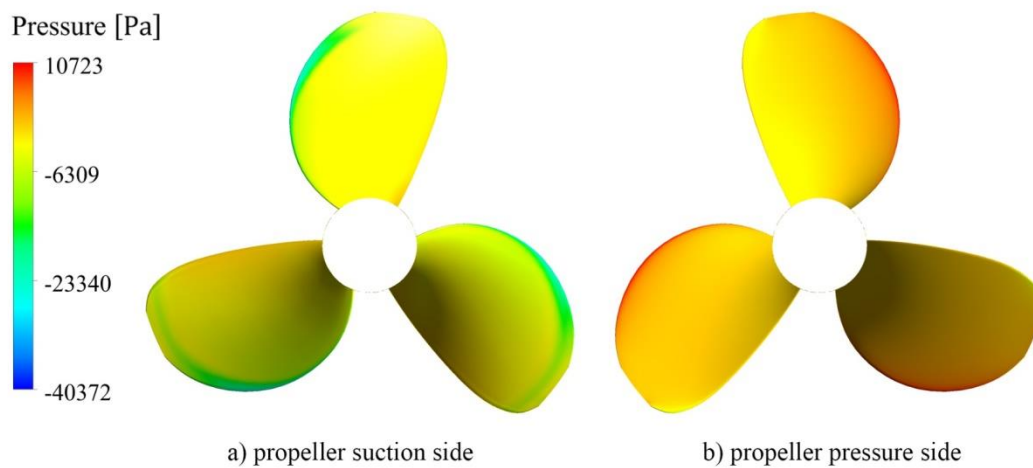


Figure 6. – Pressure distribution on the propeller's blades

3.2. Turbine regime

The CFD analysis in the turbine modes is performed for various values of the rotational speed and runner inlet velocity.

Energy characteristics of hydrokinetic turbines are commonly given as the power coefficient $C_p = \frac{P}{\frac{\rho}{2}Av^3}$ in the function of the tip-speed ratio $\lambda = \frac{u}{v}$, where u $\left[\frac{m}{s}\right]$ is the blade tip speed, v $\left[\frac{m}{s}\right]$ is the free flow velocity, A is the runner disk surface area.

The maximum theoretical value of the power coefficient for the unshrouded axial kinetic turbine is known as the Betz limit and has the value of $C_p = 59.3\%$. This value is important as it gives an upper bound on the power coefficient, independent of the runner geometry. However, Betz analyzed a heavy idealized turbine and many assumptions were introduced in the pursue of this value, such as uniform velocity at the runner inlet (neglecting the hub influence on the velocity distribution), non-existing swirl on the outlet, no losses in the process of energy transmission and conversion, etc. The result of such assumptions is that some turbines have a much lower C_p value than the Betz limit.

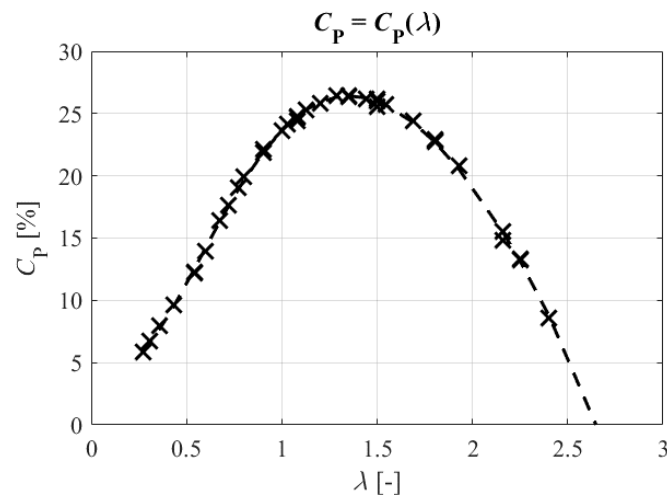


Figure 7. – The dependence of the power coefficient on the tip-speed ratio

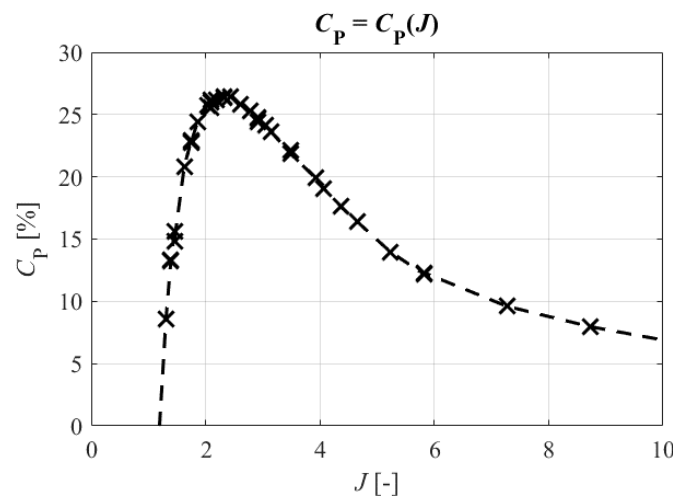


Figure 8. – The power coefficient vs. the advance coefficient

The results (Figure 7) show a peak value of the power coefficient of about $C_p = 26\%$ for $\lambda = 1.35$. For the interval $\lambda \in [0.8, 1.8]$, the power coefficient does not decrease below 20%.

Turbine energy parameters are also given in the form $C_p = C_p(J)$ (Figure 8). The torque coefficient $K_Q = K_Q(J)$ is monotonously rising and crosses the J axis at $J = 1.19$, which corresponds to the crossing of the function $C_p = C_p(J)$.

The influence of the angle of flow α , defined as the angle between the runner's axis of rotation and the free flow velocity, on the power coefficient C_p is relatively small (Figure 9).

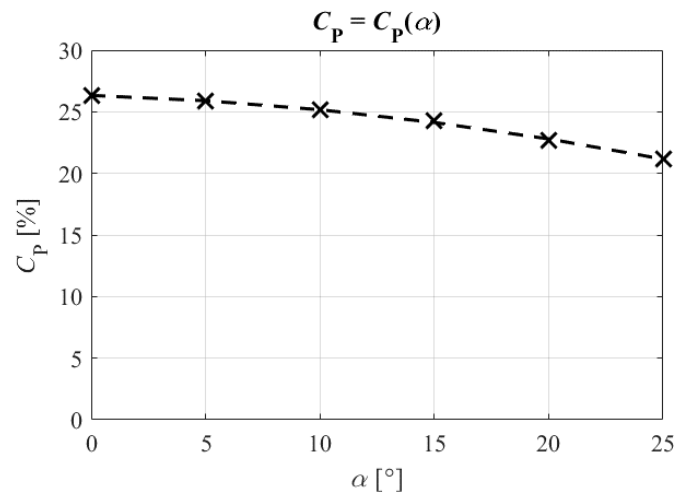


Figure 9. – The influence of the flow angle α on the power coefficient

The results of the runner rotating in the opposite direction as well as the opposite direction of the free flow velocity (like in the propeller regime) have similar trends to the showed results, but smaller peak values of the torque and power coefficient, thus they are not interesting and not given.

4. Discussion

The speed of rotation in the optimum turbine regime (Figure 7) is relatively small compared to other hydrokinetic turbine types which commonly have the values of $\lambda = 2 \div 4$ in their optimum regime [7], [8]. Such small and even smaller tip-speed ratios are common for Savonius type turbines [9].

The value of around 26% for the power coefficient in the optimum regime is a relatively good-one, given that most conventional hydrokinetic turbines, such as Darius, Savonius and Gorlov, have an optimum power coefficient of (20 ÷ 45)%. The runner is not optimized, so this leaves room for even higher value of power coefficient.

Research articles [10] and [11] have shown that the tip-speed ratio of the Darrieus-Savonius turbine optimum regimes have values of around $\lambda = 1.3 \div 1.5$. Turbine energy characteristics have a similar tip-speed of around $\lambda = 1.35$ that corresponds to the optimum regime. Furthermore, the propeller as turbine has optimum values of the power coefficient close to those of the Darrieus-Savonius turbine.

Figure 10 gives a comparison of optimum values of C_p between different types of axial HKTs and the propeller as turbine. The compared turbines and their references are given in Table 4. It can be seen that the obtained result for the propeller as turbine can be compared to other turbine types, as it achieves a satisfactory value.

Recommended values of the advanced coefficient for usage of the runner are $J = 1.6 \div 3.8$, where the power coefficient is $C_p \geq 20\%$. Regarding the slope (Figure 8), it is better to utilize the runner at higher values of the interval, e.g. for $J = 2.8$.

The minimum free flow velocity required to turn the runner is determined when the power coefficient a/o torque coefficient are equal to zero (Figure 8), and that is for $J = 1.19$. Other elements of the turbine were not accounted for, for example bearings and friction generated in them. Thus, a higher value of the advanced coefficient J would be needed to turn the turbine.

The flow angle α has little impact on the power coefficient, at least for values of up to 20° .

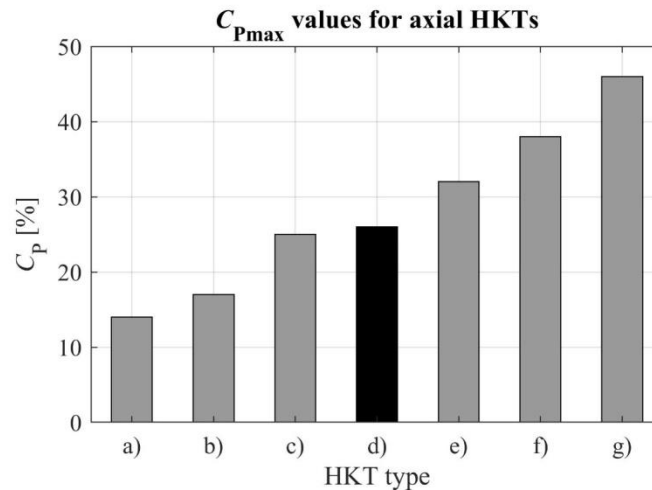


Figure 10. – Comparison of optimum values of C_p for different HKT types

Table 4. – Hydrokinetic turbine types

Label	Type	Reference
a)	Low solidity turbine with 2 blades	[12]
b)	High solidity turbine (Tyson turbine) with 7 blades	[12]
c)	Bare HKT with 3 blades	[13]
d)	Wageningen B-screw series propeller with 3 blades	-
e)	Low solidity turbine with 4 blades	[12]
f)	Unshrouded axial hydrokinetic turbines with 3 blades	[14]
g)	Low solidity turbine with 3 blades	[15]

5. Conclusion

Propellers that are found out of order (e.g. that suffered damage from not so severe cavitation) could be used as runners for hydrokinetic turbines as this eases the initial costs.

The conducted numerical simulations have shown that utilizing the Wageningen B-screw series propeller as a runner for HKTs is possible and provides satisfactory results in terms of energy efficiency. Further research will be directed towards experimental verification of the obtained results, while an economic analysis will confirm the applicability of such a concept.

Also, the Wageningen series propellers with a wider variety of different geometric parameters should be analyzed, as well as other propeller series in order to establish a general view of the concept of using ship propellers as turbine runners.

Besides, ideas for further research involve using such runners inside suitable ducts to evaluate the increase in the efficiency. In river or channel applications, the influence of multiple turbines, placed along the width of river, on a single turbine could lead to an increase in power coefficient of a turbine.

Acknowledgments

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