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Theoretical and experimental methods for prediction the propeller jet hydrodynamic loads

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Abstract

The paper presents the theoretical and experimental methods used in scientific and operational practice to predict the hydrodynamic loads generated by propellers and thrusters on the hydrotechnical constructions. The influence of different parameters: pitch and rotational speed of the propeller, aft body form of the ship hull and shallow water effect on the velocity field are discussed.

Introduction

Prediction of the hydrodynamic loads generated by propellers and thrusters on the quays or seabed protection in ports is the case of still growing importance due to the increase of short sea shipping and number of high powered self-manoeuvring vessels. The main objectives in this case are the accurate data to design well protected hydrotechnical constructions and develop safe manoeuvring procedures [1, 2].

In the calculations of the propeller wash only 20–25% of the maximum installed engine power used per propeller is assumed with respect to operational restrictions in ports. In practice, the amount of power in different weather or ice conditions, used during manoeuvres, can be much greater [3].

The empirical formulae developed on the basis of the theoretical and experimental studies are used to predict the maximum design loads and distribution of the loads in time and space domains to determine reliability or safety functions for the port structures [4, 5, 6, 7]. They can be used for water depth optimization in berthing areas [8] and prediction of boundary weather conditions for the particular manoeuvers.

The major concern is the downstream propeller jet flow which lasts for the distance of several propeller diameters and has the axial velocity components whose magnitudes can exceed 10 m/s [9, 10]. In shallow water conditions the tangential and vertical velocity components can be the reason of propeller scouring under the vessel in the propeller plane [3]. The axial velocity distribution is different compared to the common design methods. The velocities are overestimated using the “Dutch method” and even more overestimated with the “German method” [11].

The results of the calculations based on two formulae proposed by and Blaauw and Van De Kaa [12] and Lam [13] in for efflux velocity for the open water propeller are compared with the measurements of mean jet velocities on the appended hull just behind the rudder.

Theoretical methods for prediction of hydrodynamic loads from propeller jet

PIANC [6] recommends Dutch and German methods for jet induced flow prediction for design of sea bed protection. These methods are valid only for a non-ducted propeller jets. They represent two different ways of computing the required flow velocities and are based on the axial momentum theory. However, both methods are based on the similar principles, the different empirical constants are used in them, therefore, mixing these two methods could lead to inaccurate results. The principles of

Dutch and German methods are presented in figure 1.

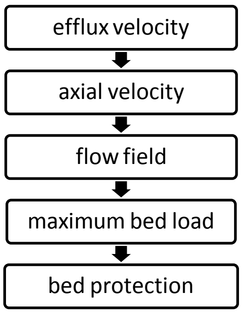

Fig. 1. Dutch and German method principles

There are serious limitations of the axial momentum theory regarding to propeller jets:

- the theory includes axial flow directions but omits the tangential and radial velocity which also occur;
- the velocities on each side of the propeller are not equal;
- the maximum axial velocity does not occur at any lateral section at the rotation axis.

The main parameters used in both, theoretical and experimental methods, for prediction of hydrodynamic loads from propeller jet cited in the paper are presented in table 1.

Dutch method for prediction of hydrodynamic loads from propeller jet

Blaauw and Van De Kaa [12] derived an equation for the estimated value of the efflux velocity. The flow directly behind the propeller is defined as a relation between rotational speed of the propeller, propeller diameter and propeller thrust coefficient. This equation is used in both Dutch and German method:

$$U_0 = 1.60\, n_p\, D_p \sqrt{K_t} \tag{1}$$

In some cases the propeller thrust coefficient is not known. For that reason, Blaauw and Van De Kaa [12] created another equation which includes engine power of the vessel, density of water and diameter of the jet just behind the propeller:

$$U_0 = 1.15 \left(\frac{P}{\rho_w D_0^2} \right)^{1/3} \tag{2}$$

The equation for the flow velocity along the jet axis depends on flow behind the propeller, located at maximal contraction of the jet, distance in axial direction from propeller and propeller diameter:

$$U_{x,\mathrm{axis}} = 2.8\, U_0 \frac{D_p}{X} \tag{3}$$

Table 1. Main parameters used in theoretical and experimental methods

Parameter	Description	Units
a	coefficient = 2.8	–
A	$1.88 \cdot \exp(-0.092\, H_{pb}/D_p)$ – no rudder, for $(0.9 \le H_{pb}/D_p \le 9)$ $1.88 \cdot \exp(-0.092\, H_{pb}/D_p)$ – with central rudder, for $(0.9 \le H_{pb}/D_p \le 8)$ 2.6 – unobstructed jets	–
C_4	0.25 – two propellers 0.30 – restriction by a transverse wall 1.62 – jet reflected at a quay wall 0.60 – with a restriction from bed and water level 1.00 – no restrictions	–
D_0	diameter of the jet just behind the propeller, located at the point of maximal contraction $D_0 = D/\sqrt{2} \approx 0.71\, D_p$ – applied to thrusters without tunnel $D_0 = 0.85 \cdot D_p$ – applied to propeller jet in a tunnel $D_0 = 1.00 \cdot D_p$ – applied to ducted thrusters	m
D_p	propeller diameter	m
E	0.71 – for sea going vessels equipped with a rudder 0.42 – for sea going vessels not equipped with a rudder 0.25 – for inland vessels with a tunnel stern and twin rudder configuration	–
H_{pb}	distance between the bed and the propeller axis	m
K_t	propeller thrust coefficient	–
n_L	scale factor for length	–
n_p	rotational speed of the propeller	s^{-1}
n_u	scale factor for flow velocity	
r	radial distance from the propeller axis	m
P	engine power	W
R_m	radius of the maximum axial velocity	m
R_p	propeller radius	m
R_h	propeller hub radius	m
ρ_w	density of water	kg/m³
U_0	flow directly behind the propeller, situated on the maximal contraction of the jet	m/s
$U_{b,\max}$	maximum flow velocity along horizontal bed	m/s
$U_{x,\mathrm{axis}}$	axial flow velocity in the centre of a free non ducted jet	m/s
$U_{x,\max}$	maximum axial flow velocity	m/s
$U_{x,r}$	axial flow velocity at radius r from the axis	m/s
X	axial distance	m

The equation for flow velocity distribution includes flow velocity along jet axis, distance in axial direction from the propeller and radial distance to the propeller axis:

$$U_{x,r} = U_{x,\mathrm{axis}} \exp\left[-15.4 (r/X)^2\right] \tag{4}$$

The values between 0.1–0.25 obtained by Verheij [11] results in:

$$U_{b,\max} = 0.3\, U_0 \, \frac{D_0}{H_{pb}} \tag{5}$$

German method for prediction of hydrodynamic loads from propeller jet

This method is established by Fuehrer, Römisch and Engelke [5]. The basic assumptions are exactly the same as in the Dutch method. Flow directly behind the propeller, situated on the maximal contraction of the jet, is calculated using equation (1) or (2).

The equation for flow velocity along jet axis in comparison to Dutch method uses values of constant A and exponent a:

$$U_{x,\text{axis}} = A\, U_0 \left(\frac{D_p}{X} \right)^{a} \tag{6}$$

The flow velocity distribution equation:

$$U_{x,r} = U_{x,\text{axis}} \exp\left[-22.2 \left(\frac{r}{X} \right)^{2} \right] \tag{7}$$

In the equation for maximum flow velocity constant values and C_4 are used:

$$U_{x,\max} = U_0 \, A \left(\frac{X}{D_p} \right)^{-C_4} \tag{8}$$

The formula for the maximum velocity at the bed (9) is developed for the different types of ships: sea going vessels equipped with a rudder, sea going vessels not equipped with a rudder, inland vessels with a tunnel stern and twin rudder configuration, expressed in the form of constant E values given in table 1.

$$U_{b,\max} = E\, U_0 \left(\frac{h_{pb}}{D_p} \right)^{-1} \tag{9}$$

Experimental methods for prediction of hydrodynamic loads from propeller jet

The experimental methods for prediction of hydrodynamic loads from propeller jet are mostly based on physical scale model tests which include average flow velocities and turbulence.

One of the major limitations of the model tests is the scale effect due to the difficulties in obtaining accurately dynamic and geometrical similarity of fluid flow. Mainly due to the influence of viscosity scale effect it is not possible to obtain proper scaling of all dynamic forces which act on the real seagoing vessel and transfer it to the physical scale model.

The Froude criterion with a high Reynolds number is used to obtain dynamic similarity and minimizing viscous scale effects. The gravity acceleration is exactly the same in the prototype and model for the Froude number $n_u = (n_L)^{1/2}$ and the Reynolds number $n_u = n_L^2$.

Reynolds number for a physical model is about 100 times smaller than it is for real ships [2]. It is recommended by the ITTC'78 (International Towing Tank Conference) to input empirical amendments which include viscosity. For the power-propulsion research, the extrapolation methods can

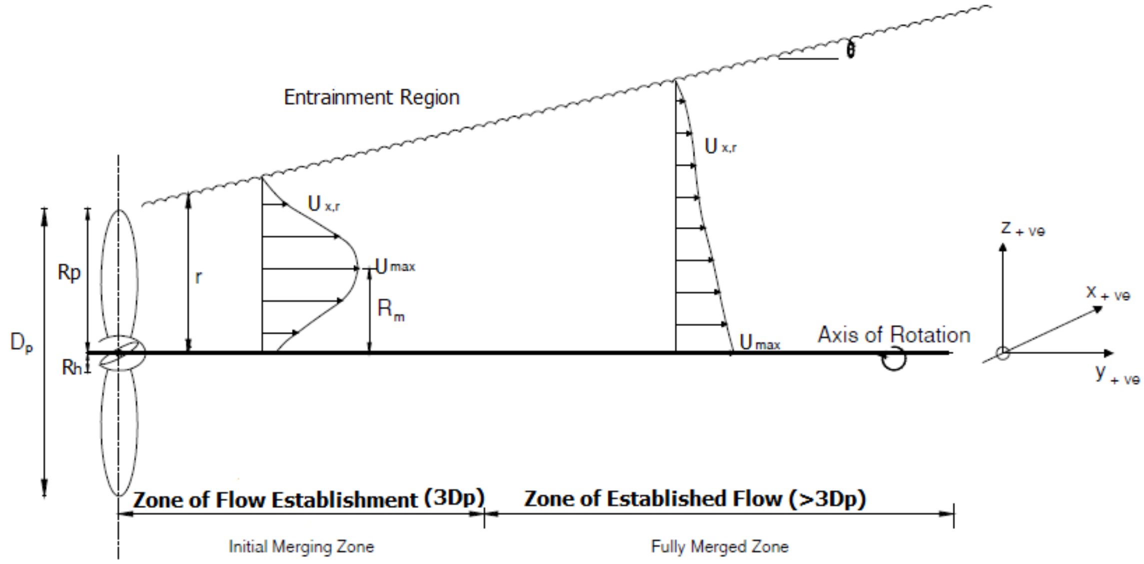

Fig. 2. Distribution of the axial velocity by Lam

be used. The Reynolds number for physical model of the propeller should be larger than the critical Reynolds number.

The experiments are conducted to establish equations for efflux velocity, maximum flow velocity, flow velocity distribution and maximum velocity over the sea bed. Lam et al. [14] presented a maximum tangential velocity which stands for 82% of the maximum axial velocity. The tangential velocity has two peaks lying in between the rotation axis and the jet boundary, at the efflux plane. Additionally, he showed that the contraction at the efflux plane is trifling.

Figure 2 [14] illustrates that the position of the maximum velocity within the zone of flow establishment remains at the constant location r/R_p = 0.53 from the rotation axis and for the zone of established flow remains at the rotation axis $r/R_p = 0$.

The distribution of the axial velocity is not axisymmetric. It is influenced by the rudder, aft body shape, free surface and seabed [2, 3, 15].

The flow in the stern region of a fully appended hull was analyzed by Muscari et al. [15] by both computational and experimental fluid dynamics. The study was focused on the velocity field induced by the rotating propellers. Measurements have been performed by laser Doppler velocimetry (LDV) on the vertical midplane of the rudder and in two transversal planes behind the propeller and behind the rudder.

The mean axial velocity U measured for deep and shallow water on the appended physical model, in geometrical scale 1:16, just after the rudder, $0.18D_p$ ($D_p = 0.319$ m) from the propeller plane has been presented in [3].

The result of the measurements behind the rudder compared with efflux velocity values calculated using the semi-empirical equation based on the actuator disc theory proposed by Blaauw and Van De Kaa – efflux velocity U_0 (1) [12] and equation proposed by Hamil – efflux velocity U_{01} (9), refined through several experimental investigations [13] are presented in table 2.

$$U_{01} = 1.33\, n_p\, D_p \sqrt{K_T} \qquad (9)$$

Table 2. Mean axial velocities

Propeller settings			U [m/s]		Open water propeller	
θ [°]	n_p [1/s]	K_T	$h/T = 3$	$h/T = 1.2$	U_0 [m/s]	U_{01} [m/s]
5	13.63	0.012	0.284	0.307	0.633474	0.762074
10	15.88	0.032	0.444	0.616	1.205224	1.449894
15	17.08	0.079	0.725	0.907	2.036778	2.450259
19	17.08	0.114	0.917	0.992	2.446711	2.943411

The mean velocity results from the integration of flow speed over the circle area of 0.304 m in diameter, equal to the maximum range of the pressure probe used for the measurements.

The calculated values are over twice greater than the measured mean values. However, the presented mean values are about 50% less than the maximum measured axial velocities.

The thrust coefficient K_T for the corresponding propeller pitch angle θ, propeller rotational speed n_p was calculated using equation (10).

$$K_T = \frac{T}{\rho \cdot n_p^2 \cdot D_p^4} \qquad (10)$$

The velocity field was measured behind the port rudder, for the propeller settings presented in table 2, for rudder angle 0° and two depth to draft ratios, $h/T = 1.2$ for shallow water and $h/T = 3$ for deep water.

Conclusions

The knowledge of hydrodynamic loads is essential for better understanding of the forces affecting the hydrotechnical structures. In spite of many investigations there are still some areas in design and experimental methods which should be improved for better protection of quay and bed constructions.

Coefficients in the German method include restrictions of walls, seabed and water level. The German method used in the Hamburg harbour resulted in heavier bed protection, compared to the Dutch method in the Rotterdam harbour, although in the Rotterdam harbour occurs no extensive damage level to bed protection [11].

The Dutch method is still considered leading, however, the results of model tests performed on the appended physical models show the underestimated initial jet velocity, more than twice less than the calculated on the basis of the axial momentum theory. The proper prediction of the initial velocity is the most important problem as it is a basic variable used in design methods.

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