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HYDRODYNAMICS OF SHIPHANDLING TUGS

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ABSTRACT

The advent of omnidirectional drive tugs has enabled significant increases to be made in the efficiency of shiphandling operations, however the added manoeuvring ability has made evaluation of operating procedures and force prediction a more complex matter. Investigation of the tug's abilities when the ship is underway requires careful consideration of the hydrodynamic forces acting on the tug hull and those generated by the thrusters.

A series of physical model tests have been carried out to determine the hydrodynamic force acting on a typical Australian omnidirectional stern drive tug when orientated at any drift angle(or angle of attack). This has been combined with published thruster curves (Oosterveld, 1973) to enable the tug to be modelled for any configuration possible during the shiphandling manoeuvre.

Using this model, existing operating procedures are investigated to identify the technique which gives the maximum possible force from the tug for given ship speeds.

NOTATION

A_0 propeller disk area, $A_0 = \frac{\pi D^2}{4}$

B tug beam on waterline

C_M propeller moment coefficient

$$C_M = \frac{M}{1/2 \rho A_0 D (V_A^2 + (0.7\pi n D)^2)}$$

C_Q propeller torque coefficient

$$C_Q = \frac{Q}{1/2 \rho A_0 D (V_A^2 + (0.7\pi n D)^2)}$$

C_T propeller thrust coefficient

$$C_T = \frac{T}{1/2 \rho A_0 (V_A^2 + (0.7\pi n D)^2)}$$

D propeller diameter

Fn Froude Number, $Fn = \frac{V}{\sqrt{gL}}$

L tug length on waterline

n propeller revolutions per second

P tug/ship force, propeller pitch

V_A advance velocity of propeller

V velocity of tug/ship

T propeller thrust, draft of tug to baseline

(X^*,Y^*,N^*) forces in ship fixed coordinate system

(X,Y,N) forces in tug fixed coordinate system

X_H' hull surge force coefficient

$$X_H' = \frac{X_H}{1/2 \rho V^2 BT}$$

Y_H' hull sway force coefficient

$$Y_H' = \frac{Y_H}{1/2 \rho V^2 TL}$$

N_H' hull yaw moment coefficient

$$N_H' = \frac{N_H}{1/2 \rho V^2 TL^2}$$

(x,y) distances in tug fixed coordinate system

Greek symbols

α angle of rotation of propeller thrust vector from velocity vector

β_H drift angle of tug hull

β_P advance angle of propeller blade

$$\beta_P = \arctan \frac{V_A}{0.7\pi n D}$$

δ angle of rotation of thruster

ϕ angle of rotation of thrust vector

γ angle of rotation of tug/ship force vector

θ thruster angle of incidence

subscripts

H hull

P propeller

S ship

p port

s starboard

INTRODUCTION

The concept of the omnidirectional drive tug was first introduced by Baer (1954) of the Voith company, Germany. This tug was of the tractor configuration with twin Voith Schneider cycloidal propellers located underneath the forebody. Considerable work has been done to investigate the abilities of the *Voith Water Tractor*, in particular the influence of hydrodynamic hull forces, Baer (1971).

More recently, tugs of both tractor and stern drive configurations have being designed with azimuthing ducted screw propellers fitted. Invariably tugs of the stern drive type are propelled by azimuthing propellers and are the type solely used in Australia.

The ability of the azimuthing propellers to produce thrust in any direction means that generally both the propellers and hull are orientated at oblique angles to the direction of motion. Force predictions for this type of tug have been presented by Kose et al, (1987) where account has been taken of forces on the hull and propellers in oblique flow.

This paper presents details of a mathematical model using more general hydrodynamic data for hull and propellers. Consideration is also given to optimum propeller configurations and the tug fender and tow point geometry.

MATHEMATICAL MODEL

Tug Equilibrium

To investigate the forces available for shiphandling, equilibrium of the tug is considered. There are assumed to be three contributions to the forces acting on the tug namely:

* The reaction force from the ship acting either at the point of contact, or at the towing point;
* The hydrodynamic force acting on the tug hull at a particular drift angle; and
* The hydrodynamic forces acting on the thrusters operating at a particular revolutions and angle of rotation.

The influence of interactions, such as between the ship and the tug, and between the tug propellers and hull, have been neglected at this stage.

The coordinate systems are shown in Figure.1:

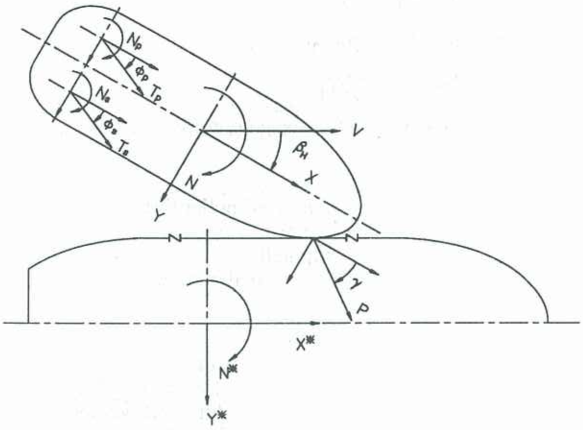


Figure.1 Coordinate systems

For equilibrium in the horizontal plane the following equations must be satisfied

$$X_H + X_P + X_S = 0 \qquad (1)$$
$$Y_H + Y_P + Y_S = 0 \qquad (2)$$
$$N_H + N_P + N_S = 0 \qquad (3)$$

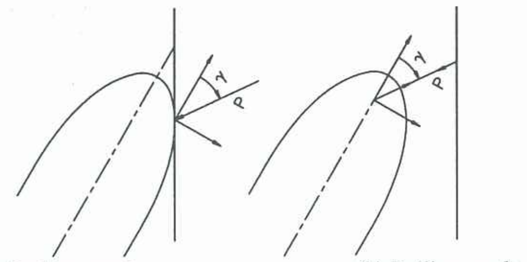
These equations are satisfied according to the following scheme:

* The force required to be imparted to the ship P and its direction γ is nominated and hence $(X,Y,N)_S$ can be calculated
* For a given velocity and drift angle the hydrodynamic force $(X,Y,N)_H$ acting on the hull may be calculated; and
* From the above, the hydrodynamic forces $(X,Y,N)_P$ required to be generated by the propellers for equilibrium may be calculated, and hence the required revolutions n, torque Q and angle of rotation δ.

The above procedure may be repeated to solve for the tug force available for maximum power or some intermediate value based on revolutions and torque. This method of solution was chosen since it allows the required direction of the tug force to be nominated from which the magnitude may be calculated depending on engine load.

Reaction Forces from the ship

The reaction force from the ship, which balances that from the tug, may act at either the position of the towing point, if the tug is operating in the pulling mode, or at the point of contact on the bow fender if operating in the pushing mode as shown in Figure.2. Although the former position remains fixed the latter is a function of the angle between the tug and ship centrelines. This also depends on the tug bow fender profile.



(a) Pushing mode (b) Pulling mode

Figure.2 Reaction force representation

Hydrodynamic force acting on the tug hull

The hull hydrodynamic force has been determined from a series of tank tests carried out in the 60m towing tank in the Ship Hydrodynamics Centre of the Australian Maritime College. The scale of the model used was 1:25, particulars of the full scale tug are listed in Appendix-A. Tests were carried out on the bare hull only, forces measured were those in the horizontal plane with the model free in pitch, heave and roll. The model was tested at a range of velocities corresponding to between 3 - 10 knots full scale and drift angles ranging between 0 - 360 degrees. The results are shown in Figure.3, it can be seen that the coefficients are essentially Fn independent.

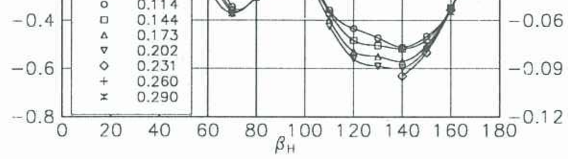


Figure.3 Experimentally determined hydrodynamic forces on tug hull

Tug propeller forces

The propeller forces $(X,Y,N)_P$ required for equilibrium are calculated from equations (1) - (3) as the other forces are known. The transformation of these to the actual propeller locations $(T,\phi)_p$ and $(T,\phi)_s$ is statically indeterminate to the first degree. However a fourth equation may be formulated using the condition that $(T_p + T_s)$ = minimum, resulting in the lowest total thrust, hence the following transformations:

for $\left|\frac{X_M}{X_P}\right| < 1$, Mode 1

$$T_p = \frac{1}{2}\left(1 + \frac{X_M}{X_P}\right)\sqrt{Y_P^2 + X_P^2}, \qquad \phi_p = \arctan\frac{Y_P}{X_P} \qquad (4)$$

$$T_s = \frac{1}{2}\left(1 - \frac{X_M}{X_P}\right)\sqrt{Y_P^2 + X_P^2}, \qquad \phi_s = \arctan\frac{Y_P}{X_P}$$

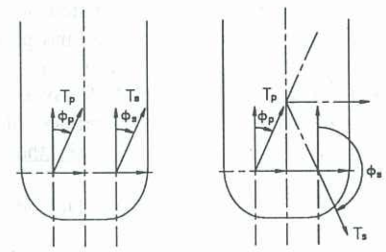
for $\left|\frac{X_M}{X_P}\right| > 1$, Mode 2

$$T_p = \frac{1}{2}\left(1 + \frac{X_P}{X_M}\right)\sqrt{Y_P^2 + X_M^2}, \qquad \phi_p = \arctan\frac{Y_P}{X_M} \qquad (5)$$

$$T_s = \frac{1}{2}\left(1 - \frac{X_P}{X_M}\right)\sqrt{Y_P^2 + X_M^2}, \qquad \phi_s = \arctan -\frac{Y_P}{X_M}$$

where, $$X_M = \frac{N_P + x_P Y_P}{y_P} \qquad (6)$$

The force term X_M, represents the force in the X direction required by each propeller to balance $(Y,N)_P$ and depending on the magnitude of $|X_M/X_P|$ the forces will be configured in one of the two modes as shown in Figure.4.



Mode 1, $\phi_p = \phi_s$ Mode 2, $\phi_s = 180 - \phi_p$

Figure.4 Configurations of propeller forces

The moments N_p and N_s which occur on each of the propellers as shown in Figure.5, have been found to have a negligible effect and are ignored.

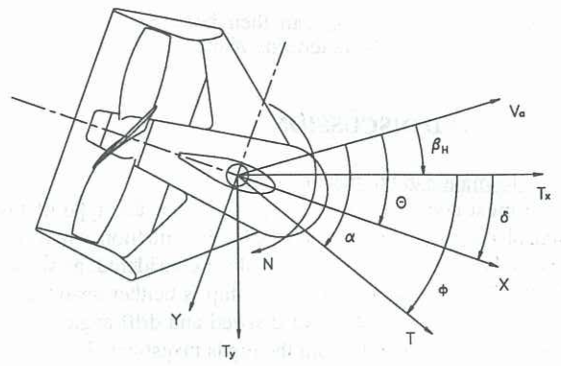
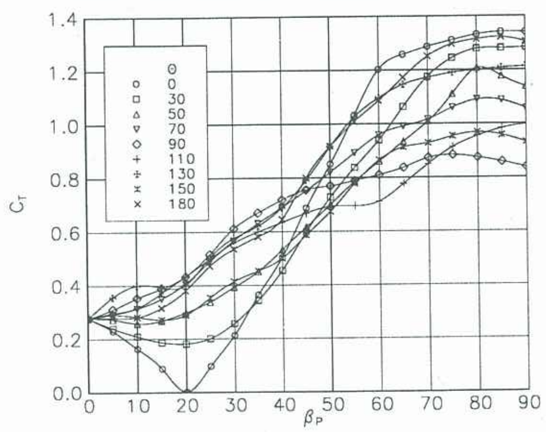


Figure.5 Propeller forces and sign convention

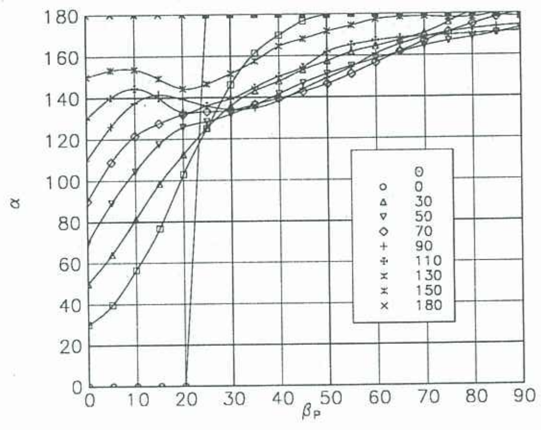
As $(T,\phi)_{p,s}$ are known the required propeller rps n, torque Q and angle of rotation δ are calculated using a model developed from tests carried out at MARIN (Oosterveld, 1973). The results are for a MARIN Ka 4-70 series screw with P/D = 1 in nozzle no. 19A. Curves of C_T, α, C_Q and C_M - β_p, for θ varying between 0 - 180° are shown in Figure.6.

Simultaneous solution of the following equations for β_P and θ then enables the calculation of δ, n, Q and M:

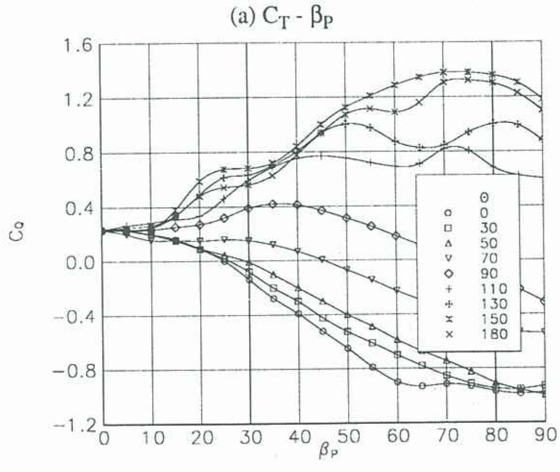
$$C_T(\beta_P,\theta)\, V_A^2 - \frac{T}{1/2\rho A_0}\, \mathrm{Sin}^2\beta_P = 0 \qquad (7)$$



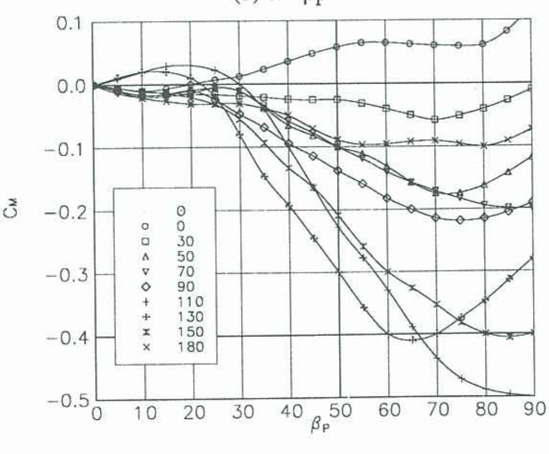
(a) C_T - β_P



(b) α - β_P



(a) C_Q - β_P



(b) C_M - β_P

Figure.6 Propeller curves

$$(\phi + \beta_H) - \alpha(\beta_P, \theta) = 0 \qquad (8)$$

$$\beta_P(C_T) - \beta_P(\alpha) = 0 \qquad (9)$$

The values of n and Q can then be compared with the engine characteristics to determine if the equilibrium position is possible.

RESULTS AND DISCUSSION

Tug performance in the pushing mode

The most common mode of operation for this type of tug is that of pushing. For this example, the situation where the tug is applying a transverse force only is considered, as shown in Figure.7. It is assumed that the ship is neither swaying or yawing, and the effect of forward speed and drift angle on the maximum force available from the tug is investigated.

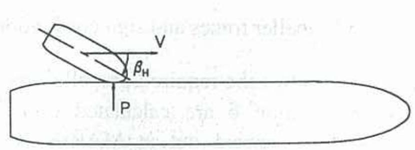


Figure.7 Tug and ship arrangement for pushing investigation

As shown earlier for the optimum use of propeller forces there are two modes in which the propeller forces may be arranged. Mode 2 is considered an impractical operating condition when shiphandling and is only used for berthing and unberthing. A convenient operating condition that satisfies the requirements of mode 1 is simply that of equal revolutions and angle of rotation for each propeller. Ignoring propeller hull/interaction parallel thrust vectors of equal magnitude will result.

Using the above criteria the effect of speed for a fixed drift angle can be investigated, this has been done for a range of drift angles as shown in Figure. 8. From this family of curves an envelope can be drawn for the maximum revolutions condition. It can be seen that the maximum force available does not vary significantly as would perhaps be expected, the value being approximately 40 tonnes throughout. This is because, as the speed increases the hydrodynamic force increases to compensate for the loss of direct thrust from the thrusters. A larger portion of this can be transferred to the ship because as the speed increases the drift angle is decreased and the lateral centre of pressure moves forward.

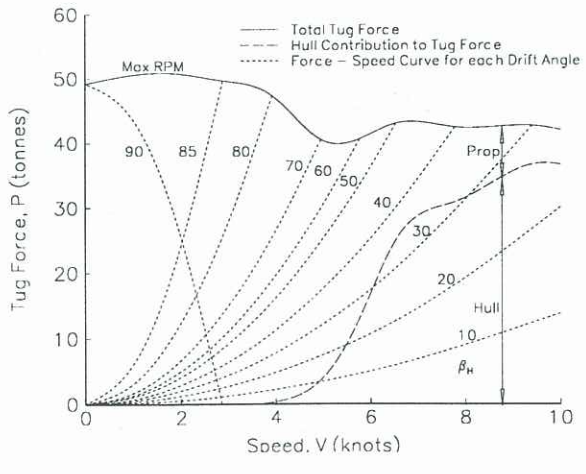


Figure.8 General operating envelope for tug in pushing mode

CONCLUSIONS

By making use of the mathematical model developed it has been shown that the omnidirectional stern drive tug is capable of imposing forces on a ship of the order of its static pull across the entire range of shiphandling velocities if used in a way such that hull forces supplement the propeller forces.

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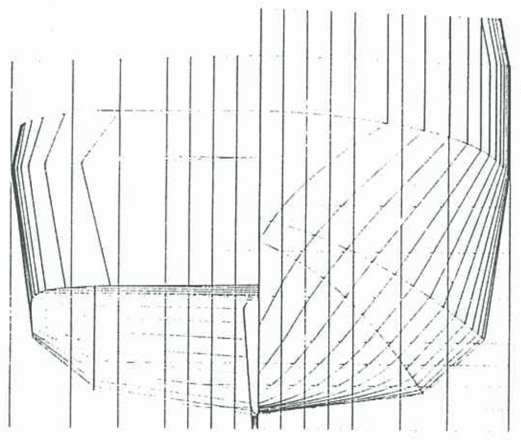
APPENDIX-A TUG PARTICULARS

The body plan and particulars of the tug on which the study has been based are as follows:

length overall	33.00 m
beam overall	11.65 m
length on waterline	30.50 m
beam on waterline	10.25 m
draft to baseline	4.25 m
propeller diameter	2.20 m
engine power x 2	2650 KW
bollard pull	50 tonnes
designer	Barnes and Fleck, Newcastle, Australia



Body Plan