Engineering Notes

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Wedge Effect on Planing Hulls

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Introduction

IN 1971 Brown[1] reported a series of test results on planing surfaces with trim flaps. For the full-span flap series tested, flap angle δ varied between 0 and 15 deg, but flap length L_F was kept at 20% of the beam b of the planing hull for most of the tests conducted. The effect of some flap parameters, as determined experimentally on planing hull performance, were reported, and these were later incorporated into the widely used Savitsky equations[2] to study the resistance D and running trim τ of a planing hull with trim flaps or wedges.[3] The relations developed were:

1) Flap lift coefficient:

$$\Delta C_{LF} = \frac{\Delta_F}{0.5\rho V^2 b_F{}^2} = 0.046\lambda_F \cdot \delta \qquad (1)$$

2) Distance from transom to point of resultant force due to flap lift:

$$LOF = 0.6\, b_F \qquad (2)$$

3) Added resistance coefficient:

$$\Delta C_D = \frac{\Delta D}{0.5\rho V^2 b_F{}^2} = 0.0052 \Delta C_{LF}(\tau + \delta) \qquad (3)$$

where b_F is the flap width, $\lambda_F = L_F/b_F$ the flap length-beam ratio, V the speed of the planing hull, ρ the fluid density, and Δ_F and ΔD the added lift and resistance due to flap, respectively.

For flap lengths beyond those tested by Brown, Millward[4] found that the experimentally determined parameters are not applicable. This might be attributed to the limited range of validity of the parameters determined from Brown's restricted range of flap length. This Note presents a method based on the use of Savitsky's equations in an effort to obtain more rational parameters which may give a better understanding of the effect of trim flaps or wedges on planing hulls.

Analysis

When a trim flap or wedge is attached to the rear end of a planing hull, the mean wetted length $\lambda\, b$ of a planing hull is generally much larger than the flap length under most planing conditions. The fluid under the planing hull flows parallel to the bottom of the planing hull before it reaches the trim flap. It is assumed that the added lift of a trim flap can be approximated by imagining the trim flap acts like another planing surface with a mean wetted length equal to the flap length at a trim angle δ. The effective mean wetted length-beam ratio of the trim flap is taken to be $\lambda_F + 0.3$, and it is further assumed that Savitsky equations[2] can be applied to evaluate the added lift due to the trim flap as provided by

$$\Delta C_{LF} = C_{LOF} - 0.0065\beta C_{LOF}^{\,0.6} \qquad (4)$$

where β is the deadrise angle of the planing surface, and C_{LOF} is the flap lift coefficient at zero deadrise angle as given by

$$C_{LOF} = \delta^{1.1}\left(0.012(\lambda_F + 0.3)^{0.5} + 0.0055(\lambda_F + 0.3)^{2.5}/C_V^2\right) \qquad (5)$$

where $C_v = V/(gb)^{0.5}$ is the speed coefficient, and g is the gravity acceleration. Figure 1 shows the relation between ΔC_{LF} and δ for different λ_F values computed from Eqs. (1) and (4). Also shown in the figure are the test data reported by Brown[1] for $\lambda_F = 0.2$ and 0.1, where C_v^2 varied between 10 and 50, and λ/λ_F varied between 11 and 21. The test data are comparable with results computed from both Eqs. (1) and (4) within the test range covered by Brown. But as the flap length increases, the difference between ΔC_{LF} values computed from Eqs. (1) and (4) becomes larger. It is observed from Fig. 1 that Eq. (1) tends to overestimate the added lift due to trim flap at larger flap lengths, and may not be applicable beyond the data range where it was derived.

When a trim flap is attached to the stern of a planing hull, and added drag is anticipated, the added drag due to the trim flap can be approximated by[5]

$$\Delta C_D = \lambda_F \tan\delta \cdot f(\delta) \qquad (6)$$

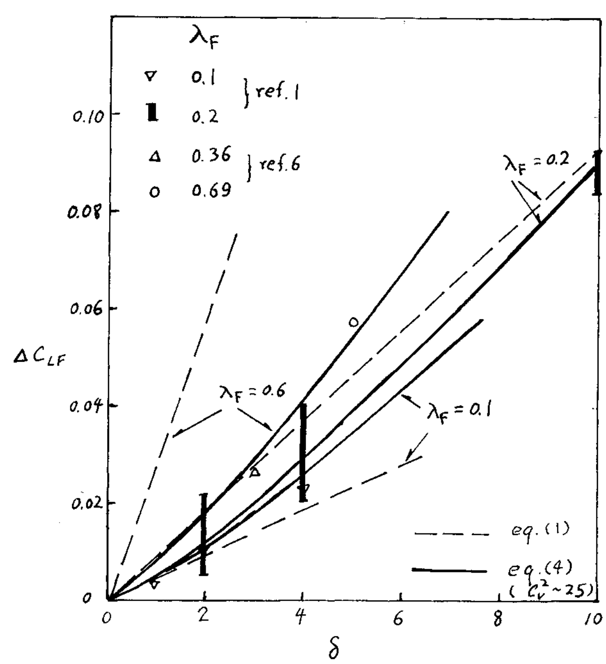

Fig. 1 Relation between ΔC_{LF} and δ.

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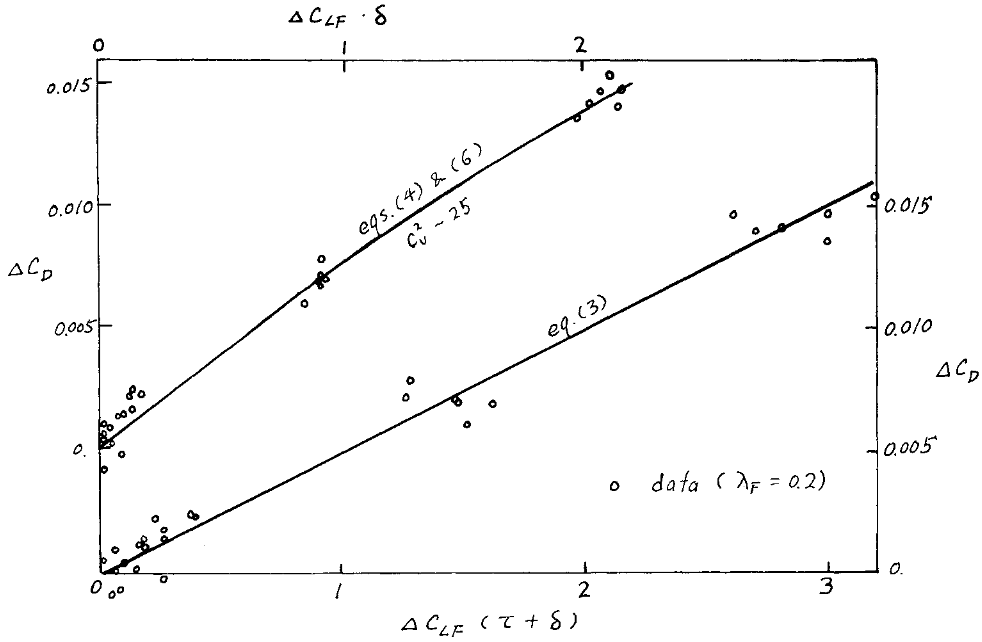

Fig. 2 Relation between ΔC_D and ΔC_{LF}.

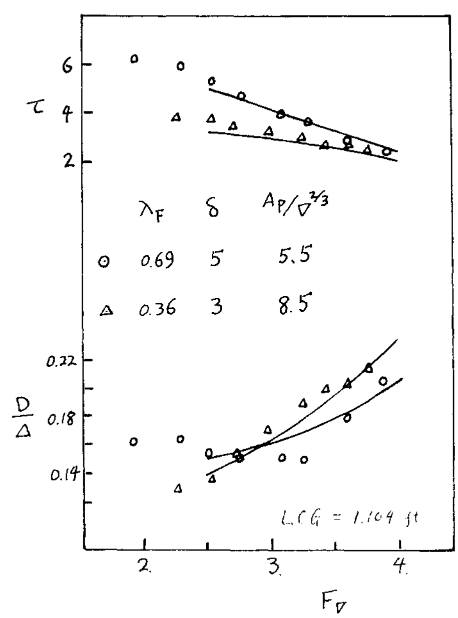

Fig. 3 Comparison with experimental results.

where

$$f(\delta) = 2\delta/90 \qquad \text{for } \delta < 5 \text{ deg}$$

$$= 0.111 + 0.0169(\delta - 5) \qquad 5 \text{ deg} \leq \delta \leq 15 \text{ deg}$$

Figure 2 shows two ways of replotting the data reported by Brown.[1] The lower curve corresponds to the relation represented by Eq. (3), and thus test data plotted have the same degree of scattering as that shown in Fig. 16 of Ref. 1. The upper curve corresponds to a relation between ΔC_D and $\Delta C_{LF} \cdot \delta$ computed from Eqs. (4) and (6) for the test condition conducted by Brown. It is apparent that better correlation between the theoretical curve and experimental data appears in the upper curve, suggesting that Eq. (6) may be a better representation than Eq. (4) to give the added drag due to a trim flap or wedge.

Millward[4] pointed out in his paper that it is difficult to believe that the additional lift due to the flap acted at a fixed geometric point on the hull which does not change as the flap length increases. A re-examination of the data on added moment due to trim flap as shown in Fig. 17 of Brown's report tends to show that LOF is between 0.5 and 0.6 b_F, which is approximately equal to the effective flap mean wetted length 0.5 b_F corresponding to a flap length of 0.2 b_F used by Brown. Therefore, it seems reasonable to assume at this stage that the distance forward of the transom to the point of resultant force on the hull due to flap lift is approximately equal to the effective mean wetted length of the trim flap, i.e.,

$$LOF = (\lambda_F + 0.3) b_F \tag{7}$$

The drag and running trim of a planing hull with trim flap or wedge can now be evaluated by substituting Eqs. (4), (6), and (7) into the following governing equations.[3] They are:

1) The effective lift coefficient of the planing hull

$$C_{L\beta} = \frac{\Delta - \Delta_F}{0.5\rho V^2 b^2} = C_{L0} - 0.0065\beta C_{L0}^{0.6} \tag{8}$$

where Δ is the load of the planing hull.

2) The effective lift coefficient at zero deadrise angle

$$C_{L0} = \tau^{1.1} (0.012\lambda^{0.5} + 0.0055\lambda^{2.5}/C_V^2) \tag{9}$$

where τ is the trim angle and λ the mean wetted length-beam ratio.

3) The equation relating the effective longitudinal center of gravity

$$\frac{\Delta LCG - \Delta_F LOF}{(\Delta - \Delta_F)\lambda b} = 0.75 - (5.21 C_V^2/\lambda^2 + 2.39)^{-1} \tag{10}$$

where LCG is the longitudinal distance of gravity from transom, and

4) The total horizontal hydrodynamic drag component

$$D = \Delta \tan\tau + \frac{c_f \lambda \rho V^2 b^2}{2\cos\tau\cos\beta} + \Delta D \tag{11}$$

where c_f is the friction drag coefficient.

Most of the test results of model length 3 to 4 ft reported by Millward[4] were in the preplaning range and were likely subject to tank (1.4 m wide × 0.84 m deep) wall effect when wedges were attached to the planing hulls. They are not suitable for the present analysis. Instead, models of Series 62 4667-1 planing hulls of 3 ft length with larger wedge lengths were tested at the Ship Model Basin (4 m wide × 2.5 m deep tank) of the National Taiwan University.[6] Figure 3 shows the test data as well as the analytical solutions obtained by solving Eqs. (8-11). In the planing range, comparisons between analytical and experimental results are reasonably good to suggest that Eqs. (4), (6), and (7) introduced in this Note could replace Eqs. (1-3) originally proposed by Brown[1] to study the effects of trim flaps or wedges on planing hulls, since Eqs. (4), (6), and (7) are applicable for both small and large flap or wedge lengths.

References

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Spread of Oil Slicks on a Natural Body of Water

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Introduction

THE accurate prediction of the rate of spread of oil accidentally spilled onto a natural body of water is important for initiating appropriate response measures. A number of theories exist in the literature for predicting the spread and movement of oil slicks under a variety of conditions. In particular Fay,[1] Fannelop and Waldman,[2] and Buckmaster[3] have, among others, given theoretical analyses for the spread of oil slicks on a quiescent body of water. The aforementioned analyses are based upon two restrictive assumptions. First, it is assumed that all of the oil is spilled "instantaneously," so that the total volume of the slick is conserved during the spread. Second, it is assumed that the viscous retarding force exerted on the underside of the slick by the underlying water layers can be predicted based upon concepts of laminar boundary-layer theory.

In many practical cases, such as in accidental spills at offshore drilling operations, the oil is released continuously rather than instantaneously. Moreover, flow conditions in the near-surface layers of a natural body of water are almost never laminar, so that the assumption of laminar flow to calculate the viscous force on the slick is unrealistic even when the calming effect of the slick on surface waves is taken into account. In this Note, we relax both of the foregoing restrictions and give power laws for the spread of continuous spills into a turbulent body of water. The analyses given herein are based upon simple phenomenological reasoning and on order-of-magnitude estimates of the various forces involved.

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Index categories: Hydrodynamics; Sea Pollution and Containment Control.

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Hydrodynamic Forces on the Slick

Using order-of-magnitude analyses of the forces on the slick, Fay[1] has shown that three separate regimes of spread can be identified, namely, gravity-inertial, gravity-viscous, and surface tension-viscous. In each regime the forces indicated in the pairing are assumed to be in balance, with the remaining (third) force being negligible. The points of transition from one regime of behavior to another is obtained by assuming that at three points the nonrepeating forces in two adjacent pairings are equal; that is, for example, the point of transition between the gravity-inertia and gravity-viscous regimes is found by assuming that at this point the inertia and viscous forces are equal.

Before proceeding with a description of the forces on the slick, one point of clarification regarding the nature of viscous forces on the slick is in order. The analyses given by Fannelop and Waldman[2] and by Buckmaster[3] are based upon the so-called slug-flow assumption. That is, the viscosity of the oil, relative to that of the underlying water, is assumed to be sufficiently high to insure that "the slick tends to move locally as a homogeneous slab relative to the water."[2] Consequently, the vertical gradients in the axial velocity within the slick are small, and the equations of motion can be integrated across the slick. The viscous retarding force on the slick is calculated by considering the developing *water* boundary layer below the slick, so that the force is independent of the viscosity of the *oil* itself.

For the sake of brevity, only spills into two-dimensional channels (one-dimensional spread) will be considered herein; results for spills onto open water (radial spread) follow readily and are not considered specifically. If at any time t, the length of the slick is l, then a measure of the velocity is (l/t). The different forces acting on the slick can then be estimated using order-of-magnitude analyses. If h is a measure of the thickness of the slick at time t, then we have the following estimates for the various forces (per unit width) on the slick:

Buoyancy force B

$$B \sim \rho G h^2$$

Inertia force I

$$I \sim \rho \frac{\partial u}{\partial t} \cdot lh \sim \rho \frac{l^2 h}{t^2}$$

Viscous drag force D (see Ref. 4, p. 108)

$$D \sim \rho \nu_w^{1/2} l^2 t^{-3/2}$$

Surface-tension force

$$\Sigma \sim \sigma$$

In the foregoing expressions G is the effective gravity (given by $G = g(\rho - \rho_0)/\rho$, where g is the acceleration due to gravity and ρ_0 and ρ are the densities of the oil and water, respectively), ν_w is the kinematic viscosity of water, and σ is the surface-tension spreading coefficient.[2] As in the existing