Journal of
Ship&Ocean Technology

SOTECH Vol.2;No.1, pp. 1~12

The Transverse Dynamic Stability of Hard-chine Planing Craft

Edward M. Lewandowski

Davidson Laboratory, Stevens Institute of Technology Castle Point Station, Hoboken
New Jersey, USA

Abstract

A method to predict the dynamic roll stability of hard-chine planing craft is presented. Starting with the equation of motion, an equation governing small roll perturbations is developed. The roll restoring moment acting on the hull is evaluated by considering "static" and dynamic contributions. The contribution of rudders and skegs, which is significant for this type of craft, is also determined. A worked example is presented to show how the method can be used to find the maximum center of gravity height for transverse stability.

1. Introduction

Planing craft are probably the most widely used high-speed marine vehicles, with applications ranging from small pleasure boats to large military craft. Procedures for the hydrodynamic design of planing hulls are fairly well established at present; however most of the published information has been focused on "performance" (resistance, power and trim vs. speed) and seakeeping (wave-induced motions and bottom pressures). Little information is available on the dynamic transverse stability of these craft. As a results, designers must resort to procedures which have been developed for displacement craft, which may be satisfactory at low speed but which do not account for dynamic effects. With increasing maximum speeds and the corresponding low trim angles, potentially dangerous instabilities which are not apparent at low speeds are being observed more frequently[Savitsky, D., 1992] [Codega, L. & Lewis, J., 1987][Blount, D. L. & Codega, L., 1991]. As late as 1991 it was stated that "little known about the fundamental causes, and no guidelines presently exist to ensure adequate dynamic stability"[Blount, D. L. & Codega, L., 1991].

The present work provides a means to predict dynamic stability of hard- chine planing craft which can be used in preliminary design. The method is based on the equation of roll motion of the craft, together with a semiempirical method to predict the roll restoring moment as a function of speed, trim angle, and particulars of the vessel.

2. Equation for Roll Motion

The most convenient coordinate system for describing the motions of a surface craft has its origin at the center of gravity(CG) of the vessel, with the X-axis horizontal through the bow; the Y-axis is horizontal to starboard, and the Z-axis is vertical, positive downward. This coordinate system, referred to as the "waterplane coordinate system", yaws with the vessel, but the XY plane remains horizontal. This coordinate system is shown on Figure 1.

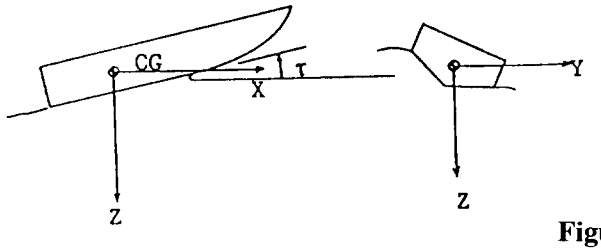

Figure 1. Waterplane coordinate system

Relative to this coordinate system, the roll moment equation can be written as follows[Lewandowski, E. M., 1994]:

$$K = \frac{d}{dt}\left(I_x p - I_{xy} q - I_{zx} r\right) - R\left(I_y q - I_{yz} r - I_{xy} p\right) \tag{1}$$

where K is the "applied" roll moment, I_{ij} are the moments and products of inertia relative to the waterplane axes; R is the yaw angular velocity about the Z-axis; and p, q and r are angular velocity components about body-fixed axes coincident with the waterplane axes at zero speed. They are related to the rates of change of the trim and roll angles as follows:

$$p = \dot{\phi} \tag{2}$$
$$q = \dot{\tau} \tag{3}$$
$$r = R \cos\tau \sec\phi - \dot{\tau} \tan\phi \tag{4}$$

where ϕ and τ are the roll and trim angles, respectively. It is noted that the moments and products of inertia relative to the waterplane axes vary in time as the vessel rolls and trims.

The "applied" roll moment K consists of hydrodynamic and hydrostatic moments on the hull and appendages:

$$K = K_h + K_r \tag{5}$$

where K_h is the contribution of the hull and K_r is the contribution of the rudders and other appendages in the presence of the hull. It is conventional in ship maneuvering work[Mandel, P., 1967] to assume that hydrodynamic forces and moments can be expressed as functions of the orientation of the vessel and its linear and angular velocity components; thus for the hull moment,

$$K_h = a_1\phi + a_2\Psi + a_3 R + a_4\dot{\phi} + a_5\ddot{\phi} + \cdots \tag{6}$$

where Ψ is the hydrodynamic drift angle,

$$\Psi = -\tan^{-1} V/U$$

and U and V are the X and Y components of the vessel velocity U. The coefficients a_j must be determined from test data as no reliable means presently exists to compute these quantities.

The contribution of a rudder, on the other hand, can be reliably estimated based on the lift rate of the rudders and an effective lever arm:

$$K_r = -\frac{dF_r}{d\alpha_r}\alpha_r l_r \tag{7}$$

where $dF_r/d\alpha_r$ is the lift curve slope of the rudder, α_r is the hydrodynamic angle of the attack of the rudder and l_r is the lever arm of the rudder force(in the YZ plane) relative to the CG of the vessel. The negative sign in the equation above reflects the fact that a positive force on the rudder (to standard) induces a negative roll moment (counterclockwise looking forward) since the rudder is located below the origin (CG). The lift of the rudder can be determined from the following semiempirical expression from Reference[6]:

$$\frac{dF_r}{d\alpha_r} = \left(\frac{1}{2}\rho U_r^2 A_r\right)\frac{1.8\pi}{1+\frac{2.8}{AR_e}} \tag{8}$$

where U_r and A_r are the velocity of the flow at the rudder and rudder planform area, and AR_e is the effective aspect ratio of the rudder; for rudders mounted against the hull, this can be taken to be twice the geometric aspect ration to account for the "reflection plane" effect of the hull. The lever arm of the rudder force in a vertical plane is given by

$$l_r = y_r \sin\phi_r + (z_r \cos\tau - x_r \sin\tau)\cos\phi_r \tag{9}$$

where (x_r, y_r, z_r) are the coordinates of the center of rudder force in a body-fixed system with origin at the CG as shown on Figure 2, and ϕ_r is the roll orientation (cant angle) of the rudder.

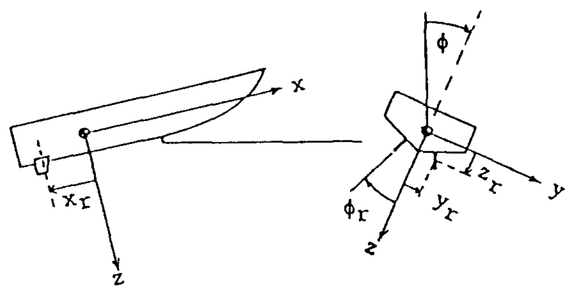

Figure 2. Body axes coordinates of appendages

2.1. Perturbation Equation

The behavior of the vessel subsequent to a small pure roll perturbation ϕ' to a steady state equilibrium condition denoted by $(U_s, V_s, W_s, p_s, q_s, r_s, \phi_s, \tau_s, \cdots)$ can be examined by substituting

$$\begin{aligned} \phi &= \phi_s + \phi' \\ \dot{\phi} &= \dot{\phi}_s + \dot{\phi}' \\ \ddot{\phi} &= \ddot{\phi}_s + \ddot{\phi}' \end{aligned}$$

in Equations (1)-(6) above. Further, since ϕ_s is itself a solution to these equations (provided that the other variables are equal to their steady-state equilibrium values), the "steady-state" equation can be subtracted from both sides of Equation (1) to yield an expression for the roll perturbation ϕ':

$$\begin{aligned} K(\phi_s + \phi') - K(\phi_s) &= \frac{d}{dt}\left[I_x\left(\dot{\phi}_s + \dot{\phi}'\right) - I_{xy}q_s - I_{zx}r_s\left(\phi_s + \phi'\right)\right] \\ &- R_s\left[I_y q_s - I_{yz}r_s\left(\phi_s + \phi'\right) - I_{xy}\left(\dot{\phi}_s + \dot{\phi}'\right)\right] \\ &- \frac{d}{dt}\left[I_x\left(\dot{\phi}_s\right) - I_{xy}q_s - I_{zx}r_s\left(\phi_s\right)\right] \\ &- R_s\left[I_y q_s - I_{yz}r_s\left(\phi_s\right) - I_{xy}\left(\dot{\phi}_s\right)\right] \end{aligned} \quad (10)$$

In the equation above, $r(\phi)$ indicates that r is a function of ϕ, as shown in Equation (4), and K is given by Equations (5)-(7). Furthermore , in steady-state the roll and pitch angular velocities and accelerations must be zero. Thus equation (10) reduces to the following form:

$$\Delta K = \dot{I}_x\dot{\phi}' - \dot{I}_{zx}\Delta r_s + I_x\ddot{\phi}' - I_{zx}\Delta\dot{r}_s + R_s I_{yz}\Delta r_s + R_s I_{xy}\dot{\phi}' \quad (11)$$

where the notation

$$\Delta f = f(\phi_s + \phi') - f(\phi_s)$$

has been introduced. The left-hand side of this equation, from Equations (5)- (7), has the following form:

$$\Delta K = a_1\phi' + a_4\dot{\phi}' + a_5\ddot{\phi}' + \cdots - \frac{dF_r}{d\alpha_r}\Delta\alpha_r l_r \quad (12)$$

Equations (11) and (12) constitute a second-order ordinary differential equation for a roll perturbation about any equilibrium value. If the initial condition is chosen to be straight-ahead motion, as is conventional in stability studies, then $R_s = 0, \phi_s = 0, \dot{r}_s = r_s = 0$ and in this case, neglecting the higher order terms in Equation (12) is justified. Thus the equation for a small roll perturbation about straight-ahead motion becomes

$$(I_x - a_5)\ddot{\phi}' - a_4\dot{\phi}' - a_1\phi' + \frac{dF_r}{d\alpha_r}\Delta\alpha_r l_r = 0 \quad (13)$$

where the (small) product of the rate of change of I_x and the roll velocity perturbation (the first term in Equation (11)) has been neglected. The change of angle of attack of the rudder induced by a small roll perturbation about straight-ahead motion is given by

$$\Delta\alpha_r = -\phi'\sin\tau\cos\phi_r - \dot{\phi}'(x_r\sin\tau\cos\phi_r - z_r)/U \quad (14)$$

where (as before) x_r and z_r are the body-fixed coordinates of the effective center of the rudder, and ϕ_r is the roll orientation (cant angle) of the rudder. Here it has been assumed that the trim angle τ is sufficiently small that its cosine can be replaced by 1. Thus using Equation (14), Equation (13) becomes

$$(I_x - a_5)\ddot{\phi}' - [a_4 + \Lambda(x_r\sin\tau\cos\phi_r - z_r)/U]\dot{\phi}' - (a_1 + \Lambda\sin\tau\cos\phi_r)\phi' = 0 \quad (15)$$

where

$$\Lambda \equiv \frac{dF_r}{d\alpha_r}l_r$$

When more than one rudder is used, or skegs or other appendages are present, Equation (15) should be replaced by

$$(I_x - a_5)\ddot{\phi}' - \left[a_4 + \sum_i \Lambda_i(x_{ri}\sin\tau\cos\phi_{ri} - z_{ri})/U\right]\dot{\phi}' - \left(a_1 + \sum_i \Lambda_i \sin\tau\cos\phi_{ri}\right)\phi' = 0 \quad (15a)$$

where the summations are taken over the number of appendages.

3. Stabilty Analysis

The solution of Equation (15) or (15a) is well known:

$$\phi' = \phi_1 e^{\sigma_1 t} + \phi_2 e^{\sigma_2 t} \quad (16)$$

The stability of the vessel when subjected to a small pure roll perturbation is determined by the sign of real parts of the quantities σ_1 and σ_2 , which (following conventional stability and control terminology) will be referred to as the stability indices for rolling motion. Substitution of Equation (16) in Equation (15) ultimately yields the following expression for the stability indices in terms of the coefficients of Equation (15):

$$\sigma_{1,2} = \frac{-B \pm \sqrt{B^2 - 4AC}}{2A} \quad (17)$$

where

$$\begin{aligned} A &= I_x - a_5 \\ B &= -[a_4 + \Lambda(x_r \sin\tau\cos\phi_r - z_r)/U] \\ C &= -(a_1 + \Lambda\sin\tau\cos\phi_r) \end{aligned}$$

Stability is ensured if C is greater than zero [Mandel, P., 1967]. The coefficient a_1 is the roll restoring moment rate, which is a negative quantity for a proper design (i.e., a positive roll angle induces a negative roll moment). The second term in braces in the expression for C represents the effects of the rudder on the roll resorting momemt; it is positive for positive trim angles. Thus the presence of a rudder reduces the roll stability of the craft at positive trim angles.

The sign of the quantity $(B^2 - 4AC)$ determines whether the vessel undergoes oscillatory motion: If this quantity is negative, the stability indices are complex and the motion is a damped oscillation. If this quantity is positive, the roll disturbance decays (for $C > 0$) or grows (if $C < 0$) exponentially.

3.1. Evaluation of Coefficients

The roll stability of a vessel depends to a large degree on the magnitude and sign of the roll restoring moment coefficient a_1; it is of interest to the designer to determine this quantity at an early stage of the design process. The most reliable way to do this is by analysis of test data (i.e., roll restoring moment at a range of speeds, roll angles, and trim angles) for the particular hull under consideration; however, this is not always practical. To provide some guidance for cases in which no data are available, the following formulation is offered.

The roll restoring coefficient is assumed to be expressible as the sum of static and dynamic contributions:

$$a_1 = a_{1s} + a_{1d} \tag{18}$$

For displacement hulls, the restoring moment is purely static; the roll restoring moment rate is given by the product of the displacement and the transverse metacentric height (GM). A more general expression, which is more applicable to the present situation, is

$$a_{1s} = -\rho g I_{WP} + BG \cdot \Delta_s \tag{19}$$

Where I_{WP} is the waterplane area moment of inertia, BG is the distance from the center of buoyancy of the displaced fluid to the VCG, and Δ_s is the "static" lift, or the buoyancy due to the fluid displaced by the hull. The waterplane area moment of inertia for a planing hull can be approximated as follows:

$$I_{WP} = \int_{-b/2}^{b/e} y^2 dA_{WP} = \int_{-b/2}^{b/2} y^2 l(y) dy \approx \frac{b_{avg}^3}{48}(L_k + 3L_c) \tag{20}$$

where $l(y)$ is the local wetted length, and L_k and L_c are the wetted keel and chine lengths, respectively; b is the average wetted chine beam. The weight of the displaced fluid can be approximated as

$$\Delta_s \approx \rho g \frac{b^2 \tan\beta}{12}(2L_c + L_k) \tag{21}$$

and the height of the center of buoyancy is

$$KB \approx \frac{b}{3}\tan\beta \tag{22}$$

Combining with Equation (19), we have

$$a_{1s} \approx 0.624\left\{-\rho g \frac{b^3}{48}(L_k + 3L_c) + (KG - \frac{b}{3}\tan\beta) \cdot \Delta_s\right\} \tag{23}$$

where the factor of 0.624 reflects the reduction of static lift due to displacement of the free surface around the hull[Brown, P. W., 1971]. Since the wetted chine and keel lengths are functions of speed, it can be seen that the "static" restoring moment rate is actually speed dependent, and in fact generally decreases with increasing speed because of its dependence on L_k and L_c.

The "dynamic" roll restoring rate can be estimated using the well-established planing equation. The expression for planing lift which is most useful in the present application is that given by Brown 1971, in which the dynamic and static contributions are distinct. The dynamic contribution to the planing lift coefficient is:

$$C_{L\beta d} = \frac{L_d}{\frac{1}{2}\rho U^2 b^2} = \sin 2\tau \left[\frac{\pi}{4}(1 - \sin\beta)\cos\tau \frac{\lambda}{1+\lambda} + \frac{1.33}{4}\lambda \cos\tau \sin 2\tau \cos\beta\right] \tag{24}$$

where β is the deadrise angle, b is the beam at the chine, and λ is the mean wetted length to beam ratio,

$$\lambda \equiv \frac{L_k + L_c}{2b}$$

When the hull is at a roll angle, Equation (24) can be used to compute the contributions of the port and starboard sides, using effective deadrise angles shown on Figure 3:

$$\beta_{e,stbd} = \beta - \phi; \beta_{e,port} = \beta + \phi \tag{25}$$

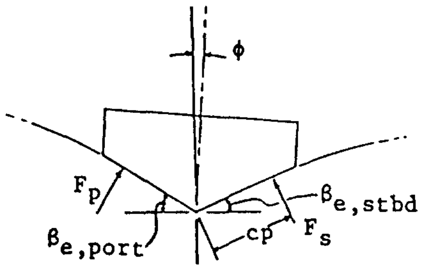

Figure 3. Effective deadrise angles for planing hull at a roll angle

For a prismatic plaining hull the difference between the keel and chine wetted lengths is also a function of deadrise angles[Savitsky, D., 1964]:

$$L_k - L_c = \frac{b \tan\beta}{\pi \tan\tau} \tag{26}$$

thus values of L_c and λ can be computed for the port and starboard sides if it is assumed that L_k is constant for small roll angles(this is, in fact, supported by experimental evidence[9]); for example:

$$\lambda_{port} = \frac{L_k}{b} - \frac{\tan\beta_{e,port}}{2\pi\tan\tau} \tag{26a}$$

Thus the dynamic force action normal to the port deadrise surface, for example, is

$$\begin{aligned} F_{d,port} &= \frac{1}{2}\rho U^2 b^2 \left\{ \frac{\sin 2\tau}{2\cos\beta_{e,port}} \left[\frac{\pi}{4}(1 - \sin\beta_{e,port})\cos\tau \frac{\lambda_{port}}{1+\lambda_{port}} \right.\right. \\ &+ \left.\left. \frac{1.33}{4}\lambda_{port}\cos\tau\sin 2\tau\cos\beta_{e,port} \right] \right\} \end{aligned} \tag{27}$$

with a similar expression for the starboard side. An approximate expression for the location of the center of pressure is given in Reference [10]:

$$cp = 0.8 \cdot \frac{\pi}{4} \cdot s$$

where s is the width of the deadrise surface,$s = b/(2\cos\beta)$,and cp is measured from the keel (Figure 3). Thus the lever arm for the dynamic moment about the CG is

$$arm = 0.8\frac{\pi b}{8\cos\beta} - KG \cdot \sin\beta \tag{28}$$

The dynamic roll moment is then given by

$$K_d = (F_{d,port} - F_{d,stbd}) \cdot \left(0.8\frac{\pi b}{8\cos\beta} - KG \cdot \sin\beta\right) \tag{29}$$

and so the restoring moment rate is

$$a_{1d} = \left[\frac{d}{d\phi}K_d\right]_{\phi=0} (F_{d,port} - F_{d,stbd})_{\phi=0} \cdot \left(0.8\frac{\pi b}{8\cos\beta} - KG\sin\beta\right) \tag{30}$$

It is possible to obtain an analytical expression for a_{1d} in terms of τ, β , and , L_k but the expression is quite lengthy; alternatively the approximation

$$\left[\frac{dK_d}{d\phi}\right]_{\phi=0} \approx \frac{K_d(\phi^*) - K_d(0)}{\phi^*} = \frac{K_d(\phi^*)}{\phi^*} \tag{31}$$

can be used, where ϕ^* is a small roll angle.

Since the dynamic lift generally decreases with increasing deadrise(other factors being equal), the first term in Equation (30) is generally negative; the sign of the dynamic roll restoring moment rate thus depends on the sign of the lever arm term. The maximum KG for the dynamic moment to be stabilizing is then

$$\left(\frac{KG}{b}\right)_{MAX} = \frac{0.8\pi}{8\sin\beta\cos\beta}$$

which is illustrated in Figure 4 below.

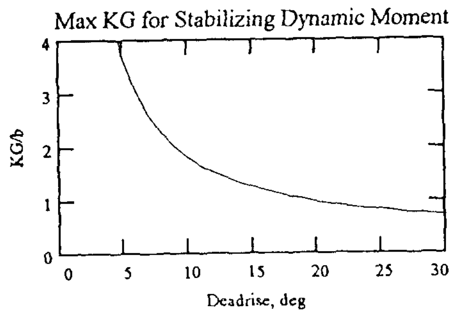

Figure 4. Maximum KG for dynamic moment to be stabilizing

Using Equation (17) the stability criterion is given by

$$-\left(a_{1s} + a_{1d} + \sum_i \Lambda_i \sin\tau\cos\phi_{ri}\right) > 0 \tag{32}$$

where as before the summation is taken over the number of rudders and skegs.

4. Application : Prediction of Dynamic Roll Stability

To illustrate the application of Equation (32) in a practical case, the example of a 64 ft planing craft cited by Wellicome and Campbell[1984] will be examined. Particulars are given in Table 1.

This craft resportedly suffered an apparent loss of stability at 35 knots during trials, where a "tendency to loll" on straight course and a "sharp inward list" during a turn were reported. The running trim of the craft is estimated to be 5.4 degrees at 35 knots based on the method[Savitsky,

Table 1. Characteristics of 64 ft Planing Craft[Wellicome, J. F., & Campbell, I. M. C., 1984]

LOA	64 ft
LBP	56.7 ft
Deadrise	15.8°
Chine Beam	
Amidships	15.44 ft
Transom	11.41 ft
Mean	13.43 ft
Displacement	37 ton
LCG	20.77 ft fwd transom
VCG(KG)	5.125 ft

Appendages(assumed to be normal to hull)		
	Rudders(2)	"P-brackets"(2)
Span	3.67 ft	3.67 ft
Mean Chord	1.36 ft	1.36 ft
Assumed Center of Force of Stbd Appendage(x,y,z),ft	(-20.0,2.85,6.226)	(-18.0,2.85,6.226)

D., 1964] and information[Wellicome, J. F., 1984]. Wetted keel and chine lengths were determined as 40 ft ([Wellicome, J. F., 1984]) and 27ft (Equation 26) respectively.

The limit of roll stability occurs when the quantity C is just equal to zero. Equation (32) can thus be used to determine the maximum value of KG for dynamic transverse stability. The calculation procedure is outlined in Table 2. The dynamic roll restoring moment rate is computed using Equation (31), with a value of ϕ^* of 1° ; thus the "effective" deadrise angles are 16.8° and 14.8° on the port and starboard sides, respectively.

The flow velocity at the rudder is required for computation of the rudder force rate as shown in Equation (8). The flow velocity at the rudders is increased by the propeller wash; a simple approximation for the increase is available from momentum theory[Glauert, H., 1959]:

$$\left(\frac{U_r}{U}\right)^2 = 1 + \frac{8}{\pi}\frac{K_T}{J^2} \tag{33}$$

where K_T and J are the propeller thrust coefficient and advance ratio, respectively. If these are not known (as in the present example), a representative value[Blount, D. L., 1975] of

$$K_T/J^2 \approx 0.2$$

can be used, giving

$$U_r^2 = 1.5U^2$$

Table 2 indicates that the maximum height of the CG above the keel for this vessel is 7.7 ft. The maximum KG for stability is shown as a function of Figure 5 below.

Table 2. Sample Calculation

Given: β = 15.8 deg, b_{ave} = 13.43 ft, U = 35 knots, b = 15.44 ft, Δ = 82880 lb, L_k = 40 ft

	Quantity	**Value**	**Source**
1	τ	5.4 deg.	Ref[8]
2	L_c	27 ft	Eqn (26)
3	Δ_s	25,600 lb	Eqn (21)
4	KB	1.267 ft	Eqn (22)
5	a_{1s}	-264,098+15,974 KG	Eqn (23)
6	$\lambda_{port}(\phi^* = 1\ deg)$	2.470 ft	Eqn (26a)
7	$\lambda_{stbd}(\phi^* = 1\ deg)$	2.533 ft	Eqn (26a)
8	$F_{d,port}(\phi^* = 1\ deg)$	33,905 lb	Eqn (27)
9	$F_{d,stbd}(\phi^* = 1\ deg)$	35,242 lb	Eqn (27)
10	arm	4.385-0.2723 KG ft	Eqn (28)
11	$K_d(\phi^* = 1\ deg)$	-5863+364 KG	((8)-(9))X(10)
12	a_{1d}	-335,925+20,856 KG	Eqn (31)
13	$dF_r/d\alpha_r$	$27.75U^2 = 96,975\ lb/rad$	Eqns (8),(33); $U_r^2 = 1.5U^2$
14	$dF_b/d\alpha_r$ ("p-brackets")	$18.50U^2 = 64,650\ lb/rad$	Eqn (8)
15	l_r	3.475+0.959 KG ft	Eqn (9), Table 1
16	l_b	3.311+0.959 KG ft	Eqn (9), Table 1
17	Λ_r	336,930+92,983 KG	(13)X(15)
18	Λ_b	214,019+61,989 KG	(14)X(16)
19	Φ_r	15.8 deg (assumed normal to hull)	

Inserting the results of Steps 5, 12, 17 and 18 in Equation (32) yields the stability criterion in terms of KG:

-(-264,098+15,794KG - 335,925+20,856KG
+2[336,930+92,983KG+214,019 + 61,989KG]$\sin(5.4°)\cos(15.8°)) > 0$

$$KG < 7.7ft$$

where the factor of 2 preceding the square brackets accounts for two rudders and brackets.

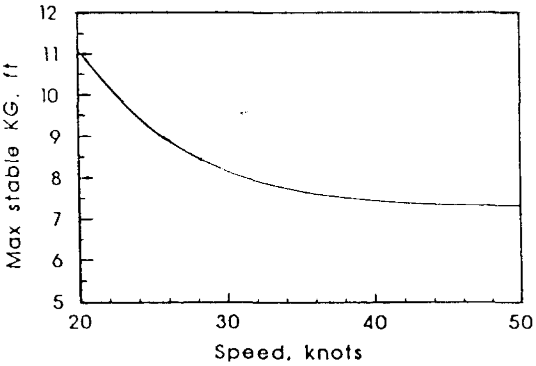

Figure 5. Behavior of maximum stable CG height with speed

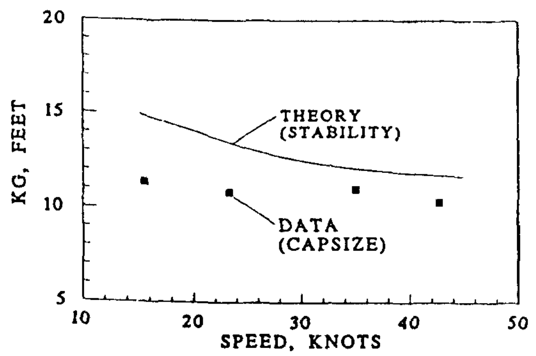

Figure 6. Comparison of stability prediction with data of Reference[11]

It is noted that the stability limit is substantially lower at planing speeds than at low speeds; in fact the maximum KG for transverse stability at zero speed, determined by "traditional" static stability calculation, is 10.2 ft [Wellicome, J. F. & Campbell, I. M. C., 1984], 32% higher than the calculated value at 35 kt. In the present example, this is totally due to the effect of the appendages: The loss of "static" roll restoring moment, due to loss of waterplane area, is more than made up for by the dynamic roll reatoring moment , which is stabilizing in this case. Neglecting the influence of appendages, the maximum KG for dynamic transverse stability is 16.3 ft at 35 knots, much larger than the zero speed value. Thus consideration of appendage effects is critical in dynamic transverse stability analysis. These results show that addition of large skegs, or increasing the rudder size, to improve stability will have the opposite effect on dynamic roll stability.

Wellicome, J. F. & Campbell, I. M. C.[1984] also contains the results of transverse stability tests conducted on a 1/16 scale free-running model of a 22.5 meter patrol boat. The KG of model was remotely adjustable by means of a sliding weight. Test were conducted at a range of speeds to determine the minimum center of gravity height which resulted in loss of transverse stability (resulting in capsize). It was also found that the craft became uncontrollable for a range of KG values below that resulting in capsize. A comparison of these results with the maximum stable KG predicted by the present theory is shown on Figure 6. It can be seen that the predicted maximum KG exceeds the observed value for capsize, but that the values are fairly close at the higher speeds.

These comparisons show that the results of the present method can be a useful indicator of potential stability problems. It is up to the designer to ensure that an adequate safety margin exists relative to the prediction, and the examples cited may provide some guidance in this respect. It is emphasized that the most reliable means to assess stability, short of full-scale trials, is through a towing tank test. One possible test technique would involve towing the properly ballasted model free to roll (as well as free to trim and heave) and observing its behavior in the presence of small roll perturbations. The model must be equipped with all appendages for these tests, as shown by the results of the worked example above.

References

1. Savitsky, D., 1992, Overview of Planing Hull Developments, Proc. Intersociety High Performance Marine Vehicles Conference
2. Codega, L. and Lewis, J., 1987, A Case Study of Dynamic Instability in a Planing Hull, Marine

Technology, Vol.24, No.2

3. Blount, D.L. and Codega, L., 1991, Dynamic Stability of Planing Hulls, Proc. Fourth Biennial Power Boat Symposium, SNAME
4. Lewandowski, E.M., 1994, Trajectory Predictions for High Speed Planing Craft, International Shipbuilding Progress, Vol.41, No.426
5. Mandel, P., 1967, Ship Maneuvering and Control, Chapter VIII in Principles of Naval Architecture, SNAME
6. Lewandowski, E.M., 1989, The Effects of Reynolds Number, Section Shape, and Turbulence Stimulation on the Lift of a Series of Model Control Surfaces, Proc. 22nd American Towing Tank Conference
7. Brown, P.W., 1971,An Experimental and Theoretical Study of planing surfaces with trim of flaps, Davidson Laboratory Report SIT-DL-71-1463
8. Savitsky, D., 1964, Hydrodynamic Design of Planing Hulls, Marine Technology, Vol.1, No.1
9. Brown, P.W. and Klosinski, W.E., 1990, Directional Stability Tests of Two Prismatic Planing Hull, Davidson Laboratory Report SIT-DL-90-9-2614
10. Smiley, R.F., 1952, A Theoretical and Experimental Investigation of Effect of Yaw on Pressures, Forces, and Moments during Seaplane Landing and Planing, NACA Technical Note 2817
11. Wellicome, J.F. and Campbell, I.M.C., 1984, The Transverse Dynamic Stability of Planing Craft, University of Southampton, Department of Ship Science, REport No. 12
12. Glauert, H., 1956, The Elements of Airfoil and Airscrew Theory, Cambridge University Press
13. Blount, D.L. and Fox,D.L., 1975, Small Craft Power Prediction, presented at the Western Gulf Section, SNAME