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On Eddy Making Component of Roll Damping
Force on Naked Hull *

by

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and Norio TANAKA^{****}

1. Seaking
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SUMMARY

From the results of the forced roll tests of two-dimensional cylinders having various cross sections, it is found that the eddy making component of roll damping force is proportional to the square of the frequency and the amplitude of roll motion. In other words, the damping coefficient C_R of this component for ship-like sections can be regarded to be independent of the period parameter for practical use.

An empirical formula for the eddy making component of the roll damping force for two-dimensional sections is deduced from the theoretical considerations on the basis of these experimental results. For three-dimensional ship forms without bilge keels at zero Froude number, the estimated results of the roll damping force which consist of the eddy making, the wave making and the frictional component, are in fairly good agreement with experimental results of ship models.

1. INTRODUCTION

The rolling motion of ships is one of those important problems which have not yet been solved because of the difficulties arising from the fluid viscosity and caused by forward speed. In this survey, the roll damping at zero forward speed is to be treated, on the assumption that the roll damping of naked hull can be divided into three components, i.e., the wave, the friction and the eddy components, and that the mutual interference among them can be neglected.

For the wave damping of ship, it has been known^{2),3)} that the radiation wave amplitudes¹⁾ of the oscillating two-dimensional cylinders show good agreements between the potential theory and the experiment.

For the friction damping, we can cite here Kato's formula⁴⁾ which was deduced from the analysis of the problem of the boundary layer on a circular cylinder. Kato's formula agrees well with the measurement for the ellipsoid models. In the case of an actual ship, however, it is difficult at present time to measure only the friction damping separately, so that we cannot know strictly the validity of Kato's formula. Nevertheless, considering the ratio of the friction damping to other ones is generally quite small, we can safely use Kato's formula.

On the other hand, the research of the eddy damping has rarely been carried out, except by Tanaka⁶⁾, one of the authors. Making the visualization of the eddy motion and measuring the roll damping for many kinds of models with various cross sections, Tanaka obtained interesting conclusions that the eddy damping increases rapidly as the bilge radius

decreases, and that this component also prevails at bow and stern of ships. However, many problems have been left for the quantitative analysis.

In this survey, assuming that the eddy damping can be obtained by subtracting the predicted values of the wave and the friction damping from the total damping force measured for the model without bilge keels at zero speed, the authors carried out an extensive forced roll tests for two-dimensional cylinders with various shiplike sections, and clarified experimentally the effects of roll amplitude and frequency on the roll damping. The results are that the roll damping is nearly proportional to the squares of frequency and amplitude for almost all cases.

The authors also propose an prediction formula for the eddy damping assuming the shape and the magnitude of the pressure distribution on the hull during its roll motion. Applying the formula to the ordinary three-dimensional ship hull forms, satisfactory values compared with the measured ones are obtained.

2. FORCED ROLL TESTS FOR TWO-DIMENSIONAL CYLINDERS

Recent researches for the oscillating bluff bodies^{7),8),9)} have clarified that the drag coefficient of the bluff body depends on the period parameter, $U_{\max}T/D$, where $U_{\max}$ represents the maximum velocity, T the period, D the representative dimension. This parameter is also called the relative displacement⁸⁾ or K-C number. The authors^{5),10)} also found that the drag of the bilge keels shows a strong dependency on the period parameter.

For the rolling of the actual naked ship hull, however, it has not yet been known whether and how the period parameter dependency exists. To investigate the problem, the forced roll tests are carried out for two-dimensional cylinders of ship-like sections, the particulars of which are shown in Table 1. A pair of end plates (0.9m breadth and 0.8m depth) is equipped at the both ends of the model to keep two-dimensionality during the test.

The method for extracting the eddy damping from the test data is the following. To begin with, we assume the roll damping moment M_R can be expressed in the form,

$$M_R = B_1 \dot{\theta} + B_2 \dot{\theta} |\dot{\theta}| \quad (1)$$

where B_1 and B_2 represent the damping coefficients and θ the roll angle. And we assume further that both the wave and friction dampings are included in the first term in Eq.(1) and can safely be obtained from the existing methods. In the analysis here, we use Ursell-Tasai method¹⁾ for the wave damping and Kato's formula⁴⁾ for the friction damping. Then subtracting these predicted values from the total damping measured, we can obtain the eddy damping.

In the analysis of the forced roll test, we define B_{44}^* as a kind of equivalent linear damping coefficient.

$$B_{44}^* = \frac{M_R}{\omega \theta_0} \quad (2)$$

Further, the nondimensional form can be written in the form,

$$\hat{B}_{44}^* = \frac{B_{44}^*}{\rho V B^2 \sqrt{2g}} \quad (3)$$

where ω , V , B and θ_0 stand for the circular frequency, the displacement volume, the breadth and the roll amplitude. In Eq.(2), M_R represents the value at the time when $\theta=0$, i.e., the roll velocity becomes maximum. Therefore, the value of B_{44}^* in Eq.

(2) is slightly different from the B_{44} value usually defined through equating the energy loss during a unit cycle of roll. For instance, in the case when only the nonlinear damping exists, the following relation holds.

$$B_{44}^* = \frac{3\pi}{8} B_{44} \quad (4)$$

The equivalent linear damping coefficient of the eddy damping B_E^* ($=B_2\omega\theta_0$) is obtained by subtracting both the linear wave damping coefficient B_W^* and the friction one B_F^* from the total value B_{44}^* measured.

From Fig.1 to 4 showing variations of the eddy damping coefficient B_E due to the change of the circular frequency ω , we can find that B_E^* is proportional to the frequency which means the nonlinear coefficient B_2 is independent of the frequency. This tendency is similar as that of the drag of the bluff body and bilge keels^{7),5)}.

Figs.5 to 16 show the amplitude (or period parameters) dependencies for the various cylinders. From these figures, we can safely conclude that the eddy damping B_E^* is also almost proportional to the roll amplitude θ_0 , and therefore that the coefficient B_2 does not depend on the amplitude. This fact seems to be quite different from that of bilge keels. The reason is not completely clear here, and should be left for future study.

We can, anyway, define an another expression for the eddy damping as the following form, considering the facts above mentioned,

$$C_R = \frac{M_{RE}}{\frac{1}{2}\rho d^4 L \dot{\theta} |\dot{\theta}|} = \frac{2\sqrt{B^2}}{d^4 L} \hat{B}_2 \quad (5)$$

,where M_{RE} , d and L represent the eddy damping moment, the draft and the length of the cylinder respectively. And the non-dimensional coefficient $\hat{B}_2$ is defined as $B_2/(\rho V B^2)$. The relation between B_E and C_R is obtained from the assumption of the equivalent energy loss in a unit roll cycle.

$$\hat{B}_E = \frac{8}{3\pi} \hat{B}_2^* = \frac{4Ld^4\dot{\theta}\theta_0}{3\pi\sqrt{B^2}} C_R \quad (6)$$

When we express the roll damping as Eq.(5), the coefficient C_R depends only on the shape of the cylinder.

3. PREDICTION METHOD FOR EDDY ROLL DAMPING OF NAKED HULL

In this chapter the theoretical and empirical prediction of the eddy damping by the use of the experimental results above mentioned will be described.

We must consider at first the location of the eddy arising. As shown in Fig.17 schematically, we can consider that the one or two eddies take place near the hull and that the condition for the appearance of the eddies seems to depend on the two parameters related to the hull shape, i.e., the half breadth-draft ratio $H_0 (=b/2d)$ and the area coefficient $\sigma (=S/Bd, S$ the area of the section under waterline). Moreover, the boundary between the cases of one-and two-points separations need not be prescribed correctly, because in those cases the eddy damping can be negligibly small.

The procedure for deducing the prediction formula for C_R is at first to assume the location of the separation of the flow, secondly to presume the shape and magnitude of the eddy pressure distribution on the hull and lastly to integrate the pressure

over the hull, similarly as Watanabe-Inoue's formula, to obtain the damping moment of the cylinder.

For the location of the eddy, we assume here that the position does not vary during half roll cycle, and that it is at the downstream edge of bilge circle for the two-points separation and at the bottom center for the one-point separation.

From Fig.18 showing the pressure distribution on the hull, we can find that the pressure decreases monotoneously as the distance from the edge. In Fig.18³⁾ the pressure coefficient C_{p1} is defined as $P^*/(0.5\rho r^2 \theta_0^2 \omega^2)$, where P^* is the pressure on the hull at the time $\theta=0$, r is the distance from the roll axis to the edge. Then we can assume a linear variation of the difference of the pressure distribution between the right and the left sides of the hull as shown in Fig.19, separately to the cases of one- and two-points separation.

The eddy damping moment M_{RE} is obtained by integrating the value of the pressure times a suitable rolling lever all over the surface. For the case of two-points separation, it becomes,

$$M_{RE} = Ld^2 \left\{ \left(1 - \frac{R}{d}\right) \left(1 - \frac{OG}{d} - \frac{R}{d}\right) + \left(H_0 - \frac{R}{d}\right)^2 \right\} \frac{P_m}{3} \quad (7)$$

where $OG/d \ll 1$, and R represents the bilge radius and OG the distance from the still water level to the roll axis, positive when taken downward.

On the other hand, for the case of the one point separation it is not easy to obtain in a similar manner. Instead, we can deduce M_{RE} at first for the cylinder of a triangular section ($\sigma=0.5$) in the form.

$$M_{RE} = Ld^2 \left(1 - \frac{OG}{d} - \frac{1}{2} H_0^2\right) \frac{P_m}{3} \quad (8)$$

And then introducing a modification factor f_2 into the third term in the bracket of Eq.(8), we can express M_{RE} for the arbitrary case of one-point separation as follows,

$$M_{RE} = Ld^2 \left(1 - \frac{OG}{d} + f_2 H_0^2\right) \frac{P_m}{3} \quad (9)$$

Eq.(9) corresponds to the case where the ship section is replaced by a triangular section with the beam $\sqrt{-2f_2}B/2$. The value f_2 should be determined from the experimental data, and will be mentioned later.

Combine Eq.(7) and (9), we can obtain an general expression for the eddy damping coefficient C_R including the both cases of one- and two-points separation,

$$C_R = \frac{2}{\rho d^2 \theta |\dot{\theta}|} \left\{ \left(1 - f_1 \frac{R}{d}\right) \left(1 - \frac{OG}{d} - f_1 \frac{R}{d}\right) + f_2 \left(H_0 - f_1 \frac{R}{d}\right)^2 \right\} \frac{P_m}{3} \quad (10)$$

,where the bilge radius R in Eq.(10) can be written in the form.

$$R = \begin{cases} 2d \sqrt{\frac{H_0(\sigma-1)}{\pi-4}} & (R < d, R < B/2) \\ d & (H_0 \geq 1, R/d > 1) \\ \frac{B}{2} & (H_0 \leq 1, R/d > H_0) \end{cases} \quad (11)$$

In Eq.(10), the factor f_1 represents the differences of the flow such that the f_1 value takes unity when the flow is the case of one-point separation and zero for the two-point separation. From Fig.17 we can express f_1 in the form.

$$f_1 = \frac{1}{2} [1 + \tanh \{20(\sigma - 0.7)\}] \quad (12)$$

Then the pressure difference P_m in Eq.(10) accompanied by the constant $1/3$ depending on the form of the pressure distribution, can be expressed by the following nondimensional coefficient

ent C_p ,

$$\frac{P_m}{3} = \frac{1}{2} \rho r_{\max}^2 \dot{\theta} |\dot{\theta}| C_p \quad (13)$$

where the term $r_{\max}$ is the maximum distance from the roll axis to the hull surface which can be approximately expressed in Appendix. The pressure difference coefficient C_p depends largely on the strength of the eddies, which may be effected by the magnitude of the local velocity and the pressure gradient. Here we assume that the value of C_p solely depends on a ratio γ of the maximum velocity $V_{\max}$ to the average velocity V_{mean} ($\gamma = V_{\max}/V_{\text{mean}}$). The value of γ for a cylinder with an arbitrary shape can be determined approximately by a potential flow calculation as will be shown in Appendix. Using the experimental values in the preceding chapter, together with Eq.(10) (12) and (13), we can find a relation between C_p and γ as shown in Fig.20 and in Eq.(14)

$$C_p = \frac{1}{2} (0.87e^{-\gamma} - 4e^{-0.1877\gamma} + 3) \quad (14)$$

The value of C_p becomes zero for the circular cylinder ($\gamma=1$), and approximately 1.5 for the flat plate ($\gamma=\infty$).

Lastly, the value of the modification factor f_2 can be determined in such a way that the values of C_p for the cylinders of the stern sections (Figs.14 to 16) should agree with the formula Eq.(14). Then we obtain the expression for the factor f_2 in the form,

$$f_2 = \frac{1}{2} (1 - \cos \pi \sigma) - 1.5 (1 - e^{-8(1-\sigma)}) \sin^2 \pi \sigma \quad (15)$$

and also shown in Fig.21.

Through the above statements, the prediction formula is settled completely. On the calculation of the eddy damping B_E for an arbitrary cylinder, we can start to obtain γ from the parameter H_0 and σ of the cylinder section, then to determine C_p from Eq.(14), $P_m/3$ from Eq.(13), f_1 and f_2 from Eqs.(12) and (15), and to substitute all these values into Eq.(10), and last to obtain B_E from Eq.(6).

The longitudinal distribution of the eddy damping coefficient can be obtained for actual ship forms as shown in Figs.22 to 23. The magnitude of C_R becomes large at the bow and stern, and moreover at the midship part where the C_R value depends largely on the midship area coefficient C_M . In Fig.23 the variation of the wave damping is also shown, from which it should be noted that there is a clear difference between the distributions of the wave and the eddy dampings.

4. FORCED ROLL TEST FOR ORDINARY SHIP HULL FORM

In this chapter, comparison of the roll damping between the formula in the preceding chapter and the experimental value measured by the forced roll test for the ordinary ship hull form. Models used are the SRL08 container ship with single screw and Todd Series 60 parent forms ($C_B=0.6, 0.7, 0.8$) the particulars of which are shown in Table 1. ^BThe method of analysing the test data is the same as mentioned previously. For the prediction of the eddy damping for three-dimensional ship forms, the values of the two-dimensional ship sections are integrated over the ship length, in which the values at FP and AP are replaced by the flat plate value $C_R=1.5(1-OG/d)$ and the so called end effect is not considered here. Furthermore, the wave damping is calculated by the "Ordinary Strip

Method" and the friction damping is obtained using Kato's formula for three-dimensional ship form.

Figs.24 to 26 shows the results plotted against the roll amplitude. The agreements between the experiment and the theory are quite good, which shows that the assumption for C_R to be independent of the period parameter is also valid for the ordinary three-dimensional ship hull form.

Figs.27 to 29 show an dependency on the roll frequency. Since for the predicted values the eddy damping $\hat{B}_E^*$ is proportional to $\hat{\omega}$ and $\hat{B}_F^*$ to $\sqrt{\hat{\omega}}$, only the wave damping shows somewhat complex variation with ω . From these figures the prediction formula is successful in the whole range of ω . In the vicinity of the natural period for the ordinary ship forms ($\hat{\omega}=0.4$ to 0.5), the eddy damping plays the greater role than the wave damping.

Fig.30 shows the comparison with the measured values for the 3m length cargo ship model carried out by Tasai and Takaki¹²⁾. The effect of bilge keels are estimated by the authors' previous formula¹⁰⁾. In this case the agreement is also well.

From these comparisons, we can safely conclude that the roll damping of the ordinary ship hull form at $F_n=0$ can be obtained by summing up the predicted values for four component dampings, that is, due to wave, friction, eddy and bilge keels.

Concerning the scale effect of roll damping, the eddy and the bilge keel dampings have little effect of Reynolds number considering the experimental results for the bluff bodies. So we need consider only of the friction damping, which is very small in magnitude.

In the presence of a forward velocity, there will arise another component damping, that is, due to hydrodynamic lift on the hull and bilge keels. And moreover wave damping shows the hump-hollow undulation. These problems at forward speed are left for the future study.

5. CONCLUSION

The eddy damping of the naked ship hull at zero forward speed is treated through experiments and some theoretical considerations. The following conclusion can be obtained.

- 1) The eddy damping of the naked hull is proportional to both the frequency and the amplitude of roll. This means that the viscous drag coefficient of the hull in its roll motion is little affected by the period parameter.
- 2) The prediction formula for the eddy damping of the two-dimensional cylinders with ship-like sections are established. According to the formula, the eddy damping becomes large at the bow and stern portion for fine ship forms, while it shows greater effect at the midship portion for blunter ship forms.
- 3) The predicted values for the ordinary three-dimensional ship form are in good agreement with the experiments.
- 4) The total roll damping of ships at zero Froude number can be predicted by summing up the contributions of the component dampings, that is, the wave damping by the "Ordinary Strip Method", the friction one by Kato's formula, the eddy one by the present formula and the bilge keel damping by the authors' formula previously reported.

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APPENDIX VELOCITY INCREMENT RATIO γ

The parameter $\gamma (=V_{\max}/V_{\text{mean}})$ could be obtained by the potential flow theory which takes exactly into account of the free surface condition and even the effect of sway motion of the cylinder. However, the rôle of the parameter γ considered here is restricted to relate the experimental value of the pressure difference coefficient C_p between the right and the left sides of the hull. Therefore we only need to obtain the γ value approximately, as follows.

Let us consider only the lower portion under the roll axis of the cylinder, in which the half beam-draft ratio H'_0 and the area coefficient σ' can be related to those parameter H_0 and σ of the original section of the cylinder under the water line.

$$\sigma' = \frac{\sigma - OG/d}{1 - OG/d} \quad H'_0 = \frac{H_0}{1 - OG/d} \quad (A-1)$$

Define the mean velocity on hull V_{mean} as the following form, using the area S' of the modified cylinder or the radius r_{mean} of the circle with the same area S' .

$$\left. \begin{aligned} V_{\text{mean}} &= r_{\text{mean}} \dot{\theta} \\ r_{\text{mean}} &= \sqrt{\frac{2S'}{\pi}} = 2d \left(1 - \frac{OG}{d} \right) \sqrt{\frac{H'_0 \sigma'}{\pi}} \end{aligned} \right\} \quad (A-2)$$

The maximum velocity $V_{\max}$ can be obtained approximately by

considering that the flow at the bilge circle corresponds nearly to that of the modified cylinder under the roll axis rotating in a infinite fluid, and that the position of the maximum velocity is assumed to be the point where the distance from the roll axis to the hull becomes a maximum $r_{\max}$. Then the expression for $V_{\max}$ can be written in the form,

$$V_{\max} = \left(r_{\max} + \frac{2M}{H} \sqrt{A^2 + B^2} \right) \dot{\theta} \quad (A-3)$$

and

$$\left. \begin{aligned} M &= \frac{B}{2(1+a_1+a_3)} \\ H &= 1+a_1^2+9a_3^2+2a_1(1-3a_3)\cos 2\psi - 6a_3\cos 4\psi \\ A &= -2a_3\cos 5\psi + a_1(1-a_3)\cos 3\psi + \{(6-3a_1)a_3^2 + (a_1^2-3a_1)a_3 + a_1^2\}\cos \psi \\ B &= -2a_3\sin 5\psi + a_1(1-a_3)\sin 3\psi + \{(6+3a_1)a_3^2 + (3a_1+a_1^2)a_3 + a_1^2\}\sin \psi \end{aligned} \right\} \quad (A-4)$$

$$r_{\max} = M \sqrt{\{(1+a_1)\sin \psi - a_3\sin 3\psi\}^2 + \{(1-a_1)\cos \psi + a_3\cos 3\psi\}^2} \quad (A-5)$$

where the constants a_1 and a_3 are the Lewis form parameters corresponding to the shape of the modified cylinder below the roll axis. And in Eq.(A-4), the term ψ represents the Lewis argument on the transformed unit circle. When $r=r_{\max}$, the corresponding ψ value may be the following,

$$\psi = \begin{cases} 0 & = \psi_1 \\ \frac{1}{2} \cos^{-1} \frac{a_1(1+a_3)}{4a_3} & = \psi_2 \end{cases} \quad (A-6)$$

where ψ_1 and ψ_2 correspond one- and two-points separations respectively. In the actual calculation, however, we need not distinguish the type of the separation, and we only need to select the ψ value in which the $r_{\max}$ is large.

$$\psi = \begin{cases} \psi_1 & \text{for } \{r_{\max}(\psi_1) \geq r_{\max}(\psi_2)\} \\ \psi_2 & \text{for } \{r_{\max}(\psi_1) < r_{\max}(\psi_2)\} \end{cases} \quad (A-7)$$

In eq.(A-3), there would be a considerable error for the cylinders with small bilge radius because of the inappropriateness of the Lewis form fitting. Herein we can introduce an modification factor f_3 for compensating these situations as the following form,

$$V_{\max} = f_3 \left(r_{\max} + \frac{2M}{H} \sqrt{A^2 + B^2} \right) \dot{\theta} \quad (A-8)$$

,where the f_3 value can be determined experimentally in such a way that Eq.(14) holds in the case of small bilge radius.

$$f_3 = 1 + 4 \exp \{-1.65 \times 10^6 (1-\sigma)^2\} \quad (A-9)$$

Finally, we can obtain an expression for the velocity increment ratio γ in the form.

$$\gamma = \frac{\sqrt{\pi} f_3}{2d \left(1 - \frac{OG}{d}\right) \sqrt{H_0' \sigma'}} \left(r_{\max} + \frac{2M}{H} \sqrt{A^2 + B^2} \right) \quad (A-10)$$

When the shape of a cylinder is known, the parameter H_0 and σ can be firstly calculated, then H_0' and σ' from Eq.(A-1), the Lewis form parameter a_1 and a_3 ¹⁾, the argument ψ_1 and ψ_2 from Eq.(A-6). Taking the greater $r_{\max}$ value among those corresponding ψ_1 and ψ_2 values, and determining the ψ and $r_{\max}$ from Eq.(A-7), we can evaluate the value of f_3 and γ .

Table 1 Particulars of models

	No.	Model	L (m)	B (m)	d (m)	V (m ³)	H ₀ (=B/2d)	σ	Roll axis
2-dim. cylinder	A	bilge radius 0cm	0.8	0.28	0.112	0.02509	1.25	1.0	O
	B	bilge radius 1cm	0.8	0.28	0.112	0.02501	1.25	0.997	O
	C	bilge radius 2cm	0.8	0.28	0.112	0.02495	1.25	0.995	O
	D	bilge radius 3cm	0.8	0.28	0.112	0.02475	1.25	0.988	O
	E	flat plate	0.8	0.024	0.07	0.00116	0.171	0.866	OG/d=-1.753
	F	flat plate	0.8	0.064	0.15	0.00763	0.21	0.992	CG/d=-0.80
	G	Series60, C _B 0.6 SS 8 1/2	0.8	0.185	0.192	0.02321	0.485	0.799	O
	H	SS 7	0.8	0.398	0.193	0.0549	1.036	0.893	O
	I	SS 5	0.8	0.237	0.096	0.01775	1.232	0.977	O
	J	SS 1 1/2	0.8	0.343	0.192	0.03163	0.8895	0.593	O
	K	SS 1 1/4	1.0	0.193	0.125	0.01228	0.777	0.541	O
	L	SS 3/4	1.0	0.151	0.144	0.00933	0.5235	0.430	O
3-dim. model	M	SR108 (Single screw)	1.75	0.254	0.095	0.0241			O
	N	Series60, C _B 0.6	1.8	0.2365	0.096	0.0247			O
	O	Series60, C _B 0.7	1.8	0.2570	0.1028	0.0331			O
	P	Series60, C _B 0.8	1.8	0.2769	0.1108	0.0439			O

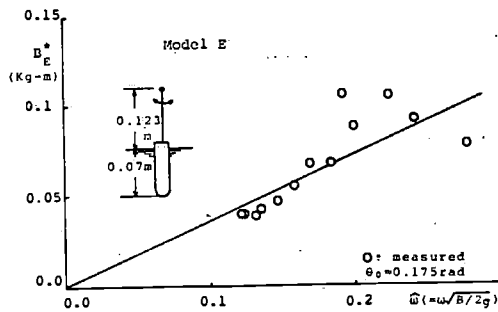


Fig. 1 Eddy making component of roll damping coefficient (Model E)

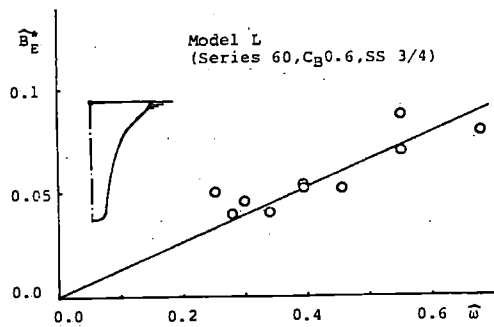


Fig. 2 Eddy making component of roll damping coefficient (Model L)

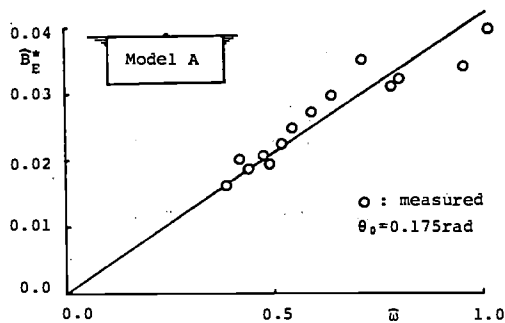


Fig. 3 Eddy making component of roll damping coefficient (Model A)

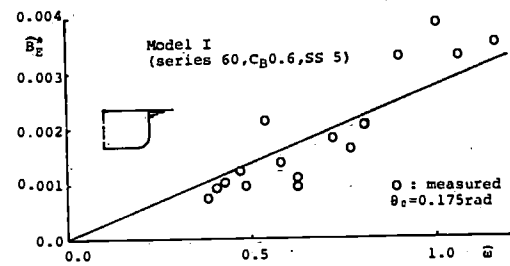
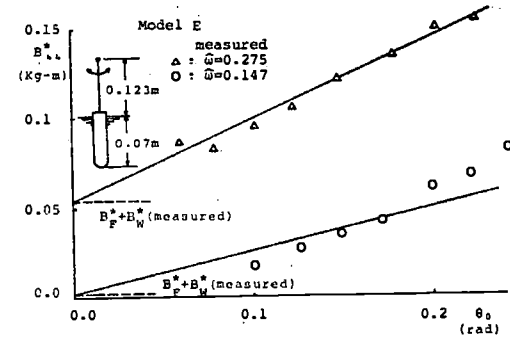
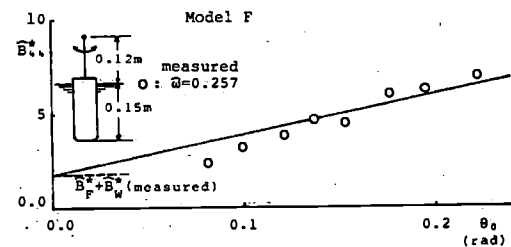


Fig. 4 Eddy making component of roll damping coefficient (Model I)

Fig. 5 Roll damping coefficient B_{44}^* (Model E)Fig. 6 Roll damping coefficient B_{44}^* (Model F)

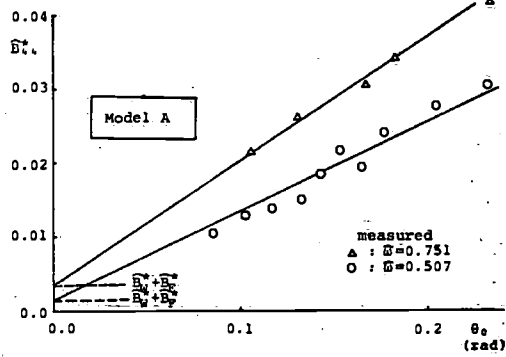


Fig. 7 Roll damping coefficient $\bar{B}_{44}^*$ (Model A)

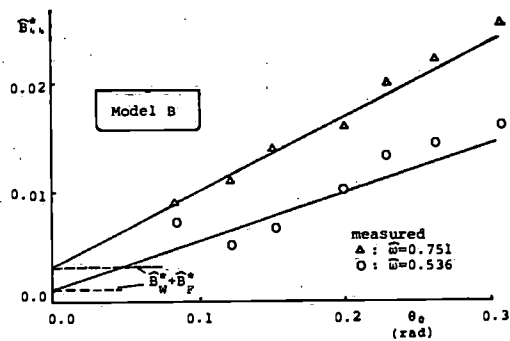


Fig. 8 Roll damping coefficient $\bar{B}_{44}^*$ (Model B)

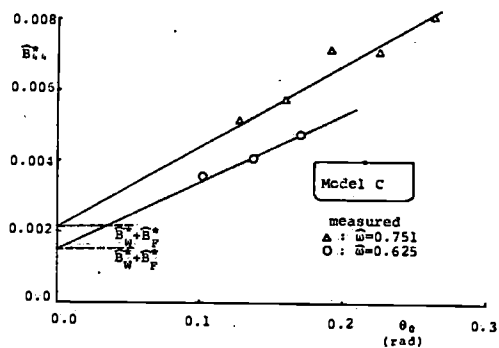


Fig. 9 Roll damping coefficient $\bar{B}_{44}^*$ (Model C)

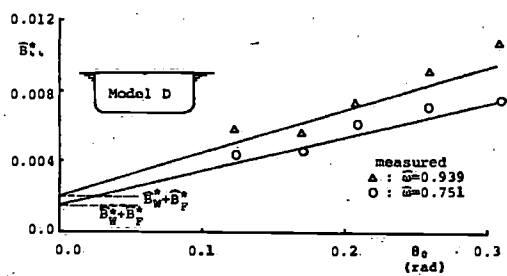


Fig. 10 Roll damping coefficient $\bar{B}_{44}^*$ (Model D)

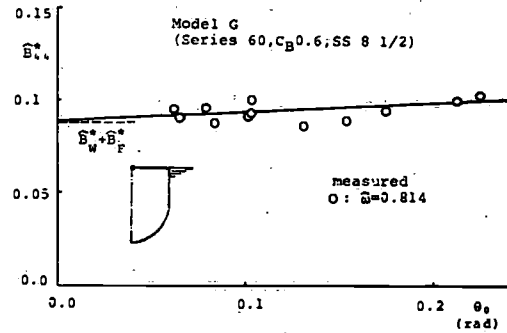


Fig. 11 Roll damping coefficient $\bar{B}_{44}^*$ (Model G)

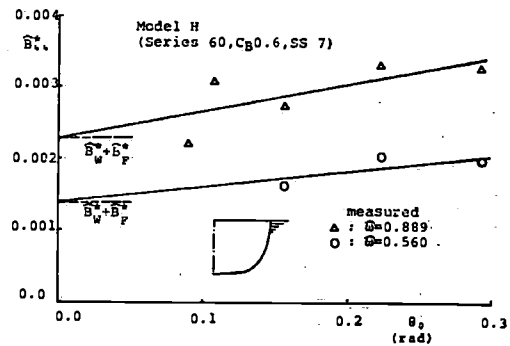


Fig. 12 Roll damping coefficient $\bar{B}_{44}^*$ (Model H)

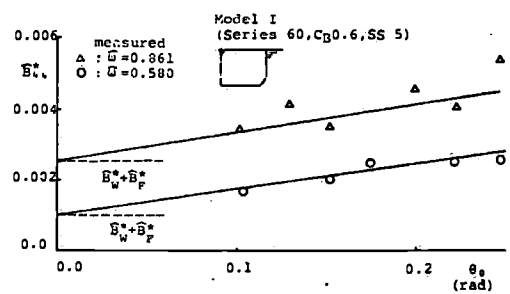


Fig. 13 Roll damping coefficient $\bar{B}_{44}^*$ (Model I)

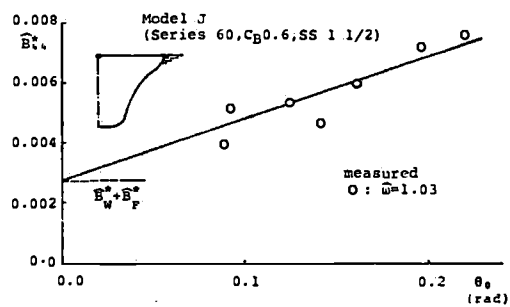


Fig. 14 Roll damping coefficient $\bar{B}_{44}^*$ (Model J)

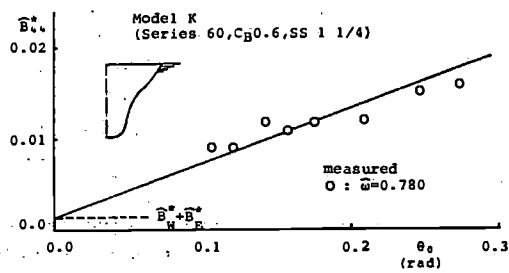


Fig. 15 Roll damping coefficient $\hat{B}_{44}^*$ (Model K)

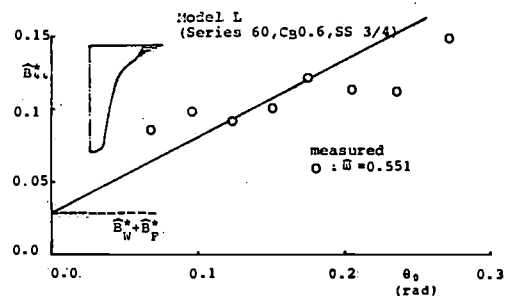


Fig. 16 Roll damping coefficient $\hat{B}_{44}^*$ (Model L)

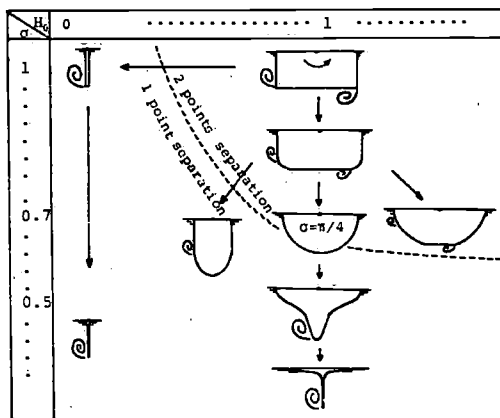


Fig. 17 Eddies near hull

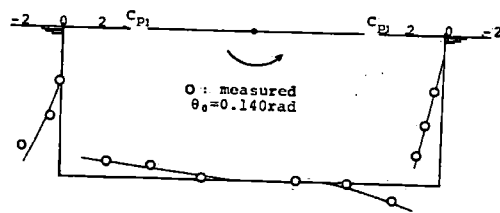


Fig. 18 Pressure distribution on hull

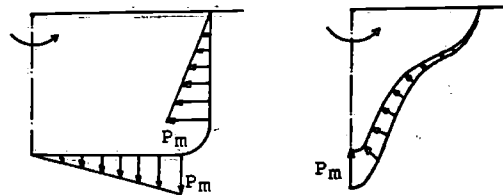


Fig. 19 Assumed profile of pressure distribution

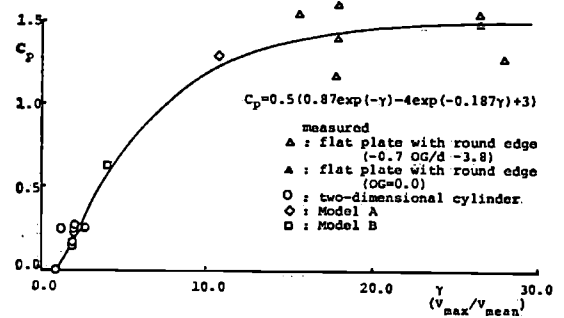


Fig. 20 Pressure coefficient C_P

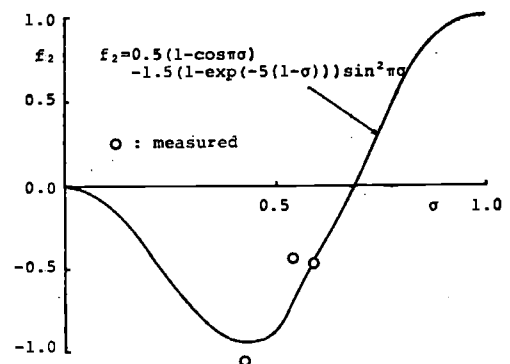


Fig. 21 Coefficient f_2 for after sections of ship

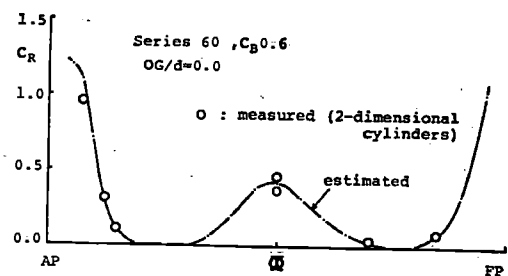


Fig. 22 Damping coefficient C_R for eddy making component

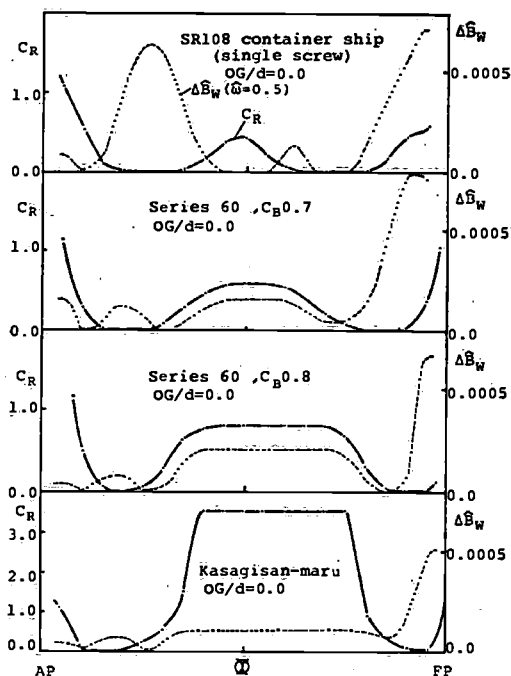


Fig. 23 C_R and ΔB_W distribution for ship forms

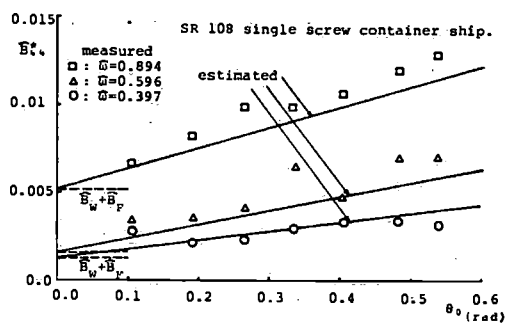


Fig. 24 Roll damping coefficient B_{44}^* for container ship model

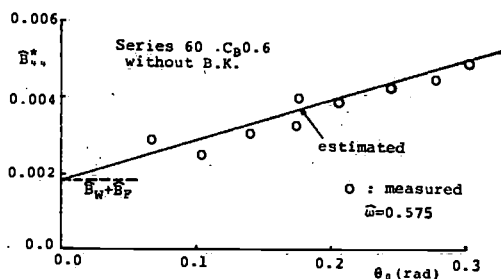


Fig. 25 Roll damping coefficient B_{44}^* for ship hull (Series 60 $C_B=0.6$)

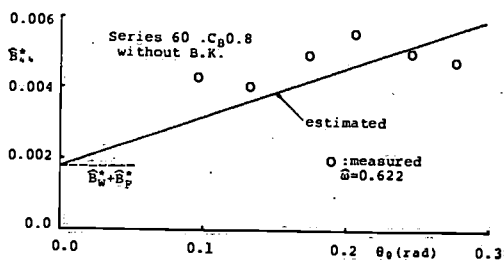


Fig. 26 Roll damping coefficient B_{44}^* for ship hull (Series 60 $C_B=0.8$)

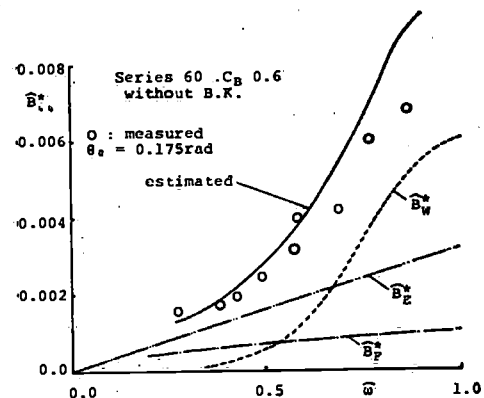


Fig. 27 Components of roll damping coefficient B_{44}^* for ship hull (Series 60 $C_B=0.6$)

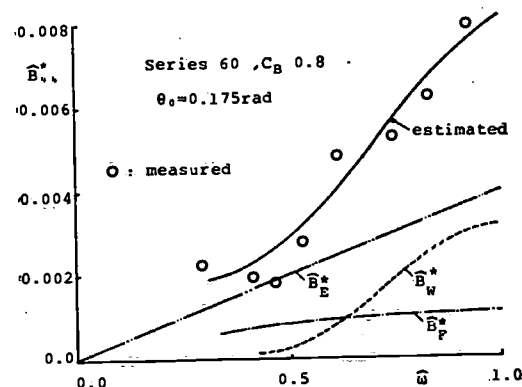


Fig. 28 Components of roll damping coefficient B_{44}^* for ship hull (Series 60 $C_B=0.7$)

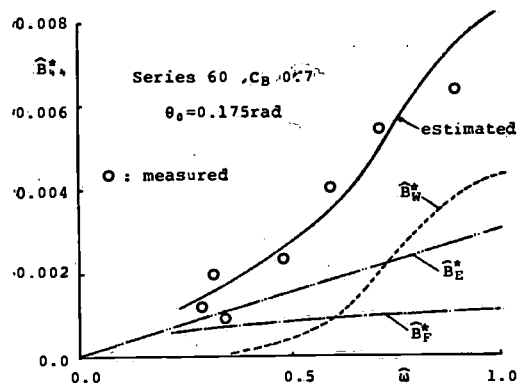


Fig. 29 Components of roll damping coefficient B_{44}^* for ship hull (Series 60 $C_B=0.8$)

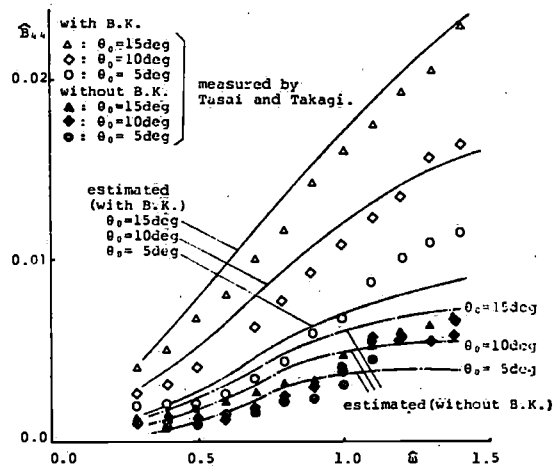


Fig. 30 Roll damping coefficient B_{44}^* for cargo ship model ($C_B=0.7119$)