The Effect of Forward Speed on Ship Roll Damping

by

Douglas Bruce Colbourne

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Signature of Author ______________________
Department of Ocean Engineering

Certified by ______________________
Prof. Martin A. Abkowitz

Accepted by ______________________
Prof. A.D. Carmichael
Department Graduate Committee

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Abstract

A series of trials was carried out in the M.I.T. Ship Model Towing Tank on a bare hull Mariner model to determine the influence of vessel forward speed on the roll damping coefficient. The model was tested at a number of roll frequencies and at Froude numbers varying from zero to 0.40.

The results showed good agreement with established theoretical calculations and indicated that within the range of roll frequencies tested, the roll damping coefficient increases with vessel forward speed.

Thesis Supervisor: Professor M.A. Abkowitz
Title: Professor of Ocean Engineering

Acknowledgement

I would like to thank Professor M.A. Abkowitz and "Midge" Mejia for their help with this work.

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Nomenclature

$\bar{A}$	rotating mass eccentricity
$\bar{B}$	rotating arm eccentricity
B	vessel beam
b b_{44}	roll damping coefficient
b_o	zero speed roll damping coefficient
b_1	wave damping coefficient
b_2	viscous damping coefficient
$\bar{C}$	rotating mass height
C	restoring moment function
F_n	Froude Number
GM	metacentric height
g	gravitational constant 32.2 ft/sec^2
I	roll moment of inertia
I_h	hydrodynamic (added) moment of inertia
L	vessel length
M	mass
M_o	peak exciting moment
T	roll period
t	time
Δ	vessel displacement
ε	phase angle
ε_r	phase angle at resonance
ω	frequency
ω_o	natural roll frequency

ω_d	damped natural roll frequency
ψ	roll angle
ψ_o	peak roll response
ψ_r	roll response at resonance

Numbers in parentheses indicate references at end of paper.

Introduction

Rolling is typically the most severe mode of motion encountered by seagoing vessels. The onset of heavy rolling due to resonant wave excitation can seriously impair vessel and crew performance and often necessitates voluntary speed reduction or more commonly a change in course. Both these actions are clearly detrimental to efficient vessel utilization.

In recent years, considerable effort has been devoted to the development of mathematical models for the prediction of roll motions in seaways. Under resonant excitation the roll response is limited only by the energy dissipation or damping due to wave generation and viscous drag on the hull and its appendages. In general, the wave damping is a linear function of roll rate and the viscous damping is a non-linear function (usually taken as a quadratic). For normal ship types the damping coefficients due to both these mechanisms are small and thus roll motions at resonance can be quite large.

The prediction of motion amplitudes is dependent on an accurate estimation of the damping coefficients. For full scale vessels without appendages such as bilge keels, viscous damping is frequently neglected and the linear wave making damping coefficient taken to be the dominant parameter. A commonly used method for predicting the damping coefficient is the two dimensional slender body strip theory. It is well known that roll damping increases with the forward speed of the vessel (1, 2, 3) and strip theory does make some correction for shape related speed effects. The theory can not however predict the effect of interaction between waves generated by the two modes of motion (i.e., roll and forward translation). Thus, predictions using strip theory are somewhat lacking for vessels underway.

Three-dimensional theories such as that developed by Newman (6) for a submerged oscillating ellipsoid indicate that the forward speed has a marked effect on the damping coefficients. For the case of roll, the coefficient is shown to increase with forward speed for low frequency numbers and to decrease with increasing speed for higher frequency numbers. For most vessels, resonant rolling occurs at lower frequency numbers and thus it is reasonable to expect significant increases in wave related roll damping with forward speed and a consequent reduction in maximum predicted roll amplitudes. Newman's results are, however, for a shape that is not a typical ship shape and, in addition, the ellipsoid does not penetrate the free surface.

In order to determine if these theoretical results were in fact qualitatively similar to those achieved with a surface vessel, a series of experiments were carried out at the MIT Ship Model Towing Tank on a bare hull merchant ship model. The purpose of these trials was to demonstrate the effect of forward speed on ship roll damping over a range of roll frequencies. This would indicate whether the three dimensional theoretical results were similar to actual results and allow the development of a correction to be applied to the two dimensional strip theory calculations.

Theory

The basic non-linear equation describing the forced roll motion of a vessel is as follows

$$(I+I_h)\ddot{\psi}+b_1\dot{\psi}+b_2\dot{\psi}|\dot{\psi}|+C(\psi) = M_o(\omega,t) \qquad (1)$$

where

I	Roll moment of Inertia
I_h	Added hydrodynamic moment of Inertia
b_1	linear damping coefficient
b_2	quadratic damping coefficient
$C(\psi)$	restoring moment function
$M_o(\omega,t)$	exciting function
ψ	roll angle

In order to simplify the equation, the restoring moment function was taken to be a linear one. This is a reasonable assumption for small to moderate roll angles, particularly for straight sided vessels where freeboard is not exceeded. In addition, the quadratic damping term was neglected. This term is largely due to viscous effects which are felt to be significant in small scale model tests (5). However, for Froude numbers greater than .15, the damping has been shown to be a linear function (2). Thus this neglect of viscosity may effect the trials run at low speeds but should become less significant as speeds increase. As a caution, viscous effects should be accounted for before the results presented here are scaled.

Finally, the model was supplied with constant frequency sinusoidal, roll excitation. The combined effect of these simplifications was to reduce Equation 1 to the more easily dealt with linear form given by,

$$(I+I_h)\ddot{\psi}+b\dot{\psi}+C\psi = M_o \sin \omega t \tag{2}$$

where $C = \Delta GM$ for small angles.

The steady-state response is given by

$$\psi = \psi_o \sin(\omega t-\varepsilon) \tag{3}$$

where

$$\psi_o = \frac{\dfrac{M_o}{(I+I_h)}}{\sqrt{(\omega_o^2-\omega^2)^2+\left(\dfrac{b\omega}{I+I_h}\right)^2}} \tag{4}$$

$$\omega_o^2 = \frac{\Delta GM}{(I+I_h)} \quad = \text{Natural Roll Frequency}$$

$$\varepsilon = \tan^{-1}\left[\frac{b\omega}{(I+I_h)(\omega_o^2-\omega^2)}\right] \tag{5}$$

ε being the phase angle between the excitation and the response.

The easiest point at which to measure the effects of the damping coefficient (b) is the point of resonance where the excitation frequency ω is equal to the natural frequency ω_o. At this point, the above expressions reduce to

$$\psi_r = \frac{M_o}{b\omega} \tag{6}$$

$$\varepsilon_r = 90° \tag{7}$$

Thus, with the exciting amplitude M_o and frequency ω known ψ_r can be measured and b calculated.

In practice, the natural frequency of the model can be determined to a suitable degree of accuracy by allowing the model to roll without restraint in calm water and timing the roll period. An alternate method is to excite the model over a range of frequencies and tune the excitation to the point where large amplitudes are evident and the near resonance "beating" phenomena ceases to occur. Either of these methods will allow one to determine the natural frequency but assuming that the excitation frequency can be accurately measured, the second method probably offers a better result. For these experiments, both methods showed reasonable agreement although the first was less consistent.

It should be noted that no discrimination is made between the natural roll frequency and the damped natural roll frequency given by

$$\omega_d = \sqrt{\omega_o^2 - \nu^2} \tag{8}$$

where

$$\nu = \frac{b}{2(I+I_h)}$$

Because roll is a very lightly damped phenomena, the difference between the two frequencies during these experiments was less than 1%. This difference was less than the measurement error and thus the frequency measured as the resonant roll frequency in the tank was taken to be the undamped resonant frequency.

Experimental Method

The roll damping experiments were carried out at the MIT Towing Tank using a 1:96 scale model of the familiar Mariner hull. The model was tested without appendages or propellers to isolate the hull damping characteristics. The addition of appendages, particularly bilge keels, has been shown to have a considerable effect on the damping at zero and non-zero forward speeds (1,4). It was felt that these effects were better dealt with on a separate basis from whence they could be added to the bare hull effects if desired.

Under tow, the model was restrained in all degrees of freedom, excepting roll, for which a low friction bearing and angle measurement transducer were fitted. Roll excitation was provided by a variable speed motor driving a set of contrarotating eccentric masses. This provided an exciting moment that was sinusoidal and variable in both amplitude and frequency. Figure 1 shows the arrangement of the roll measurement and excitation equipment in the model and Figure 2 illustrates the exciting moment calculations.

The model was towed at speeds corresponding to Froude numbers ranging from zero to 0.40. For trials at zero speed, the model was positioned sideways in the middle of the tank to minimze the effects of wave reflection from the tank walls. The natural frequency of the model was varied by adjusting the height of fixed ballast. This enabled trials covering a number

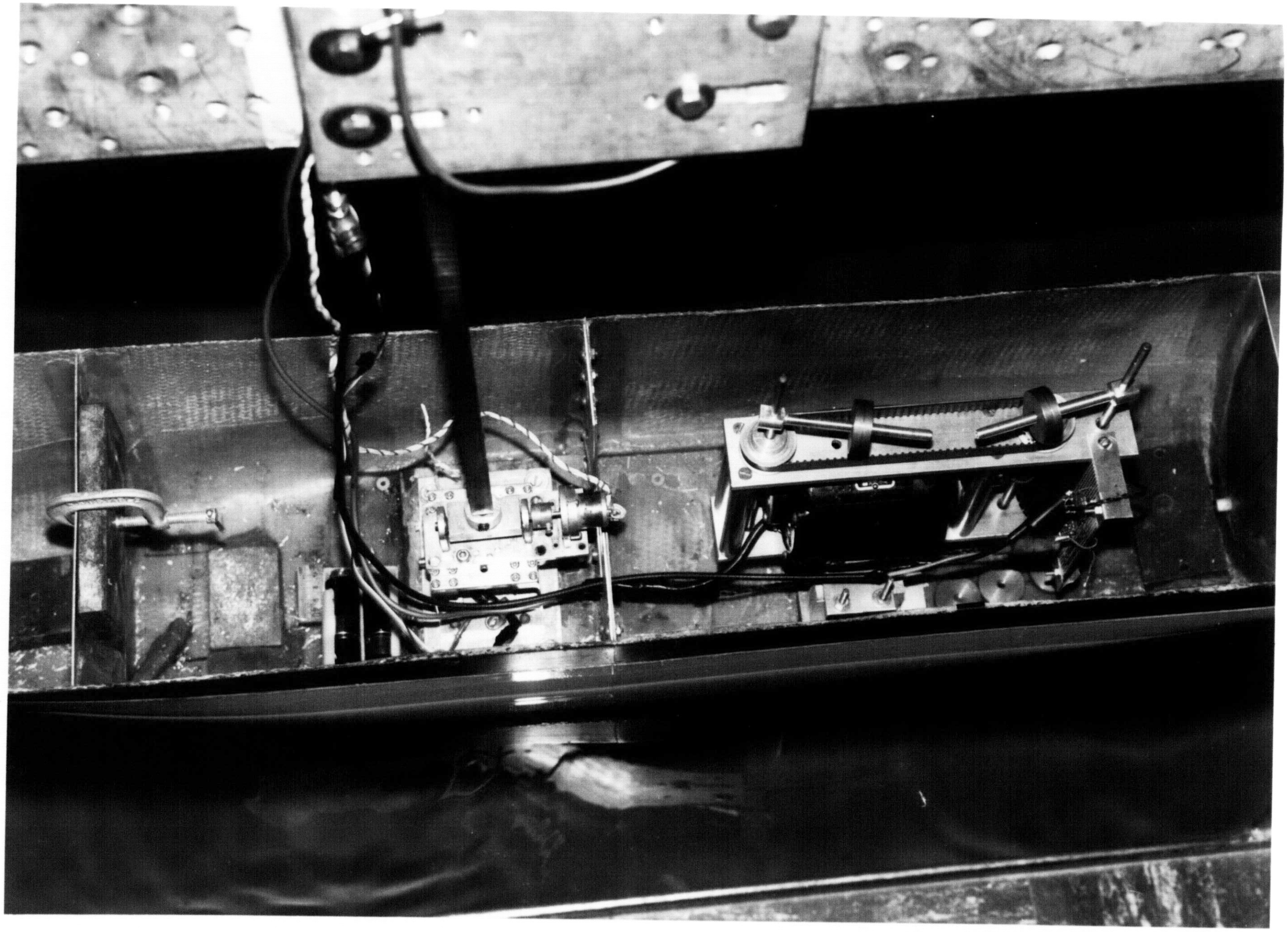
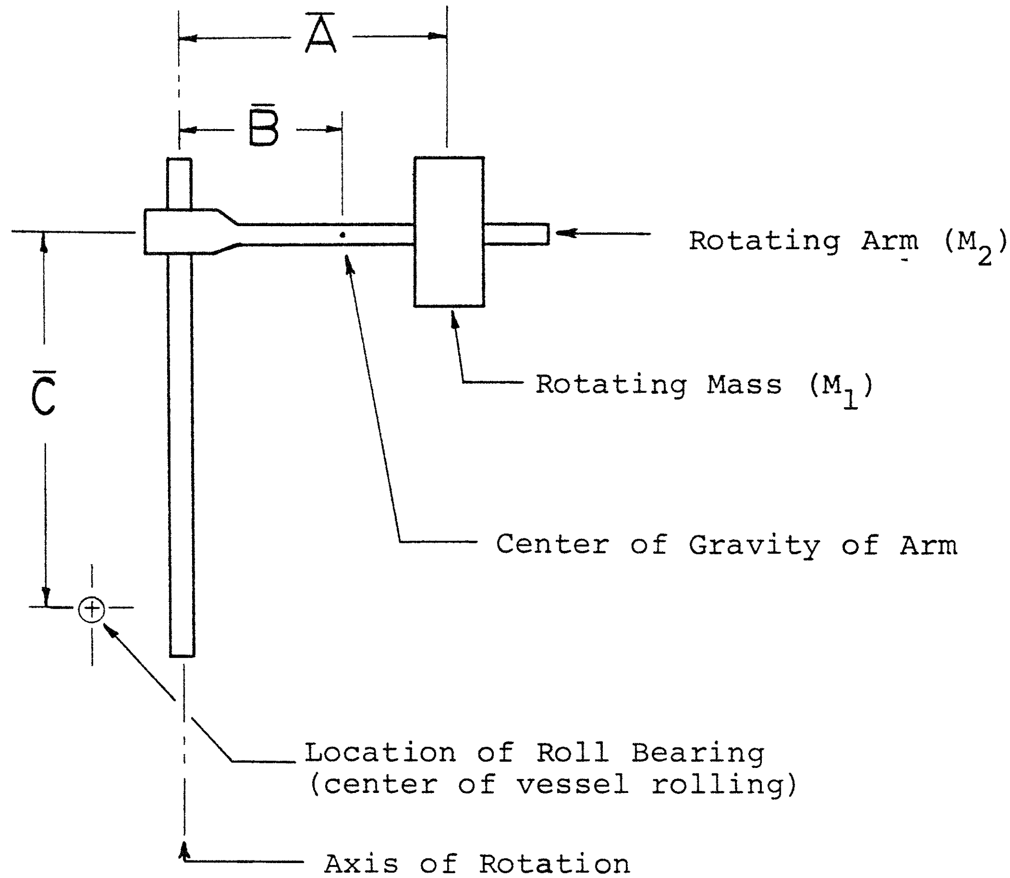


Figure 1 Instrumentation and Excitation Equipment

Figure 2 Roll Inducing Mechanism



The roll inducing mechanism consisted of two sets of rotating masses (one shown above) driven at constant angular velocity (ω) in opposite directions. Phasing of of the rotations was such that fore-aft dynamic forces cancelled leaving a dynamic side force which was absorbed by the model towing rig. Thus the mechanism generated a rolling moment with static and dynamic components calculated below.

Static Moment Component $M_s = 2[M_1 g\bar{A} + M_2 g\bar{B}]$
$= 2g[M_1\bar{A} + M_2\bar{B}2$

Dynamic Moment Component $M_o = 2\bar{c}\omega^2[M_1\bar{A} + M_2\bar{B}]$

Total Exciting Moment = $(M_s + M_o)\sin\omega t$

of roll frequencies extending from 2.67 rad./sec. to 6.83 rad./sec., which adequately covers the range of resonant roll frequencies for the full scale vessel. Model displacement was maintained at the design condition of 46.25 + 1.0 lbs F.W. for all trials except the high frequency case where it was necessary to add considerable ballast in order to achieve a suitably low roll period. Table 1 gives the model conditions during the trials.

During the performance of the trial runs the excitation was started with the model stationary. The roll motion were allowed to settle down to steady-state resonant conditions and the model was then towed the full length of the tank. Motion response was recorded during the run on strip chart. To allow for slight variations in excitation frequency, each run was performed three times to insure agreement. Figure 3 shows the model under tow during a low speed trial at moderate frequency.

Table 1 Model Condition During Trials

Roll Period T. (sec)	2.25	1.80	1.44	1.25	1.15	0.92
Frequency (sec^{-1})	2.79	3.49	4.36	5.03	5.46	6.83
Displacement (lbs.)	47.23	47.23 (46.75)	47.23 (46.75)	45.22 (46.75)	46.23	66.20
GM (Ft)	.030	.047	.073	.097	.114	.178
Peak Exciting Moment (M_o) (ft-lbs)	.124	.266 (.164)	.368 (.227)	.366 (.235)	.351	.644

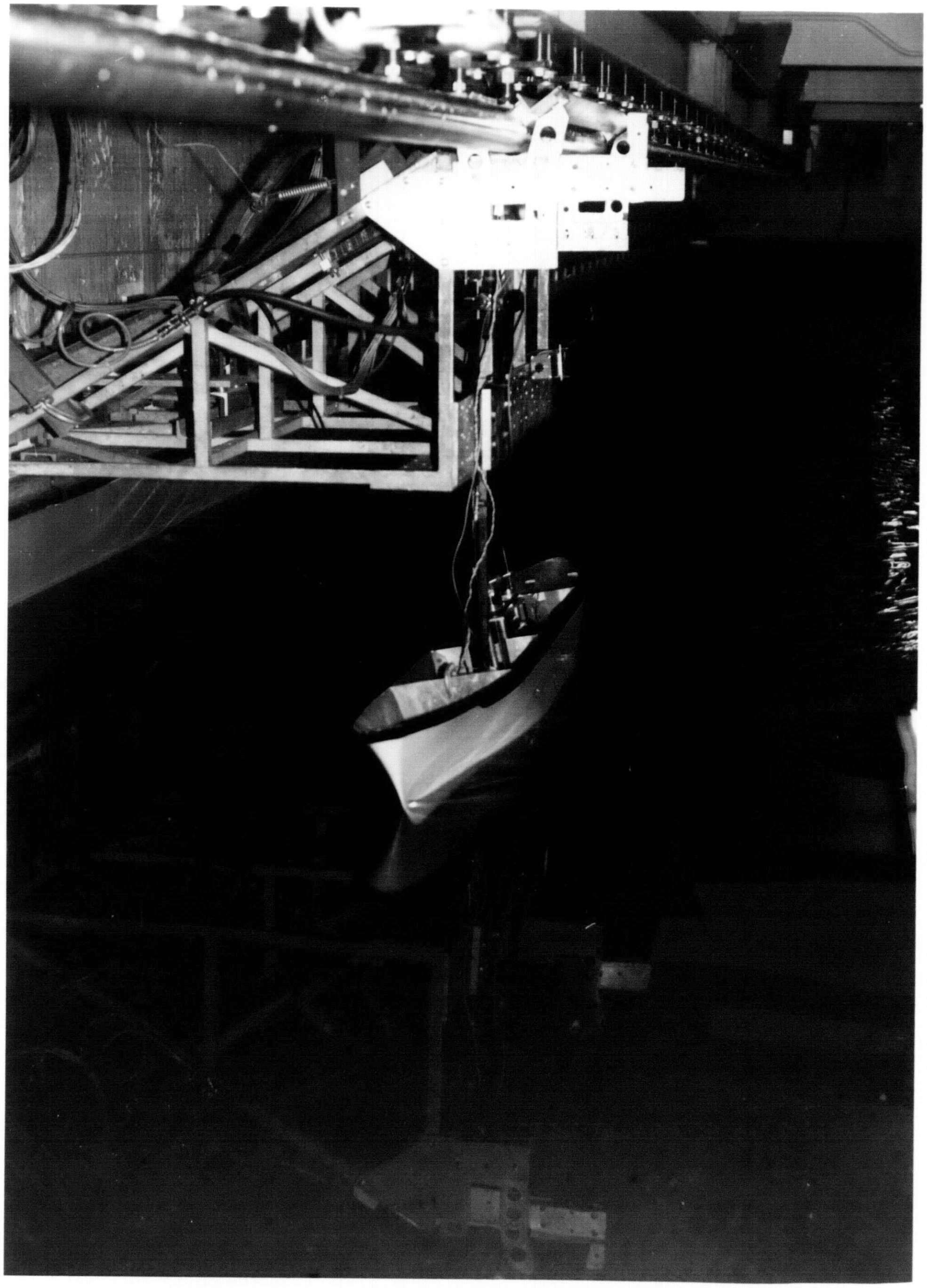
Numbers in parentheses indicate displacements and exciting moments used during repeat trials

Table 1A

The model was run at the following speeds in each condition:

Speed kts.	fps	Froude Number
0.0	0.0	0.0
0.759	1.28	.10
1.543	2.60	.20
1.985	3.35	.25
2.316	3.91	.30
2.781	4.69	.35
3.062	5.17	.40

Figure 3 Model Under Tow



Results and Discussion

Recorded and calculated results are presented in the following tables and figures. Table 2 gives peak roll amplitudes and corresponding model displacements recorded during the trials. These figures give the largest amplitude recorded over several trials. In some cases, slightly different displacements were used on repeated runs and these differences are noted. As stated previously, the high frequency (ω=6.83 rad./sec.) trials were performed using a much higher displacement., Consequently, the underwater volume and effective hull shape were substantially altered. Despite the fact that the damping coefficients are non-dimensionalized on displacement, it is felt that the high frequency results are probably not directly comparable with the remaining figures. The primary object of performing the high frequency trials was to illustrate that the relative increase in damping due to forward motion is much less for that particular frequency.

Figure 4 shows examples of trial data as it was recorded. These two recordings represent resonant rolling at the same frequency but at different forward speeds. It can be seen that the increase in speed results in a significant decrease in roll amplitude. The indicated roll period of 1.8 seconds is slightly longer than the scaled value of the normal Mariner roll period. Thus, it would seem likely that this sort of reduction in amplitude could be realized for the full scale vessel. It should be noted that the larger amplitudes recorded at one end of the higher speed results, occurred while the model was accelerating from zero up to trial speed.

Non-dimensional damping coefficients are presented in Table 3. The coefficients have been non-dimensionalized based on vessel displacement and beam as these are the relevant parameters for wave related roll damping. The roll frequency is non-dimensionalized to a frequency number based on the half length rather than the length of the vessel. This was done to facilitate comparison of the results with the theoretical work done by Newman (6).

The tabulated results are plotted in Figures 5 through 10. These plots clearly illustrate the effects of increasing forward speed on the roll damping coefficient. Figures 6 through 9 best represent the normal range of resonant roll frequencies for the full scale vessel and these curves show similar characteristics. It is felt, based on these curves, that it would be reasonable to develop an empirically based formula of the form

$$b = b_o(1 + d(F_n)^k)$$

(where b and b_o are non-dimensional damping coefficients). to give an estimate of the roll damping coefficient at a given Froude number based on that determined or calculated for zero speed (b_o). It is, however, felt that further trials on different vessel types should be done to verify this form.

Based on the results of these trials, fitted using the least squares method;

$$d = 0.86$$

$$k = 2.95$$

These values are valid for the range $F_n = 0.0$ to 0.35 and for the Mariner in the design condition. For higher Froude numbers the curves start to flatten out and thus a higher order expression would be required to adequately predict the damping coefficient.

Figure 11 is plotted for comparison with the theoretical results derived by Newman (6) and shown graphically in Figure 12. Newman's results are potential theory calculations of the wave damping coefficient in rolling for an ellipsoidal shape oscillating below a free surface. These theoretical results cover a much broader range of frequencies and Froude numbers than could be achieved with the model trials. However, within the range the model trials did cover, comparison can be made. Again, it should be noted that the high frequency trials ($\omega=6.83$ $\omega^2L/2g=3.98$) may not be directly comparable to the lower frequency result but should indicate a general trend.

The theoretical results show the damping at low Froude numbers and low frequencies tending to zero whereas the experimental results show significant values in this area. This discrepancy is credited largely to viscous damping which is not accounted for in the theoretical calculations. Viscous effects would dominate at the lower frequencies and speeds, particularly in view of the relatively large roll amplitudes recorded during these trials. Thus, it is felt that the low frequency end of the curve shown for Froude numbers equal to zero and 0.10 is probably more indicative of viscous damping trends.

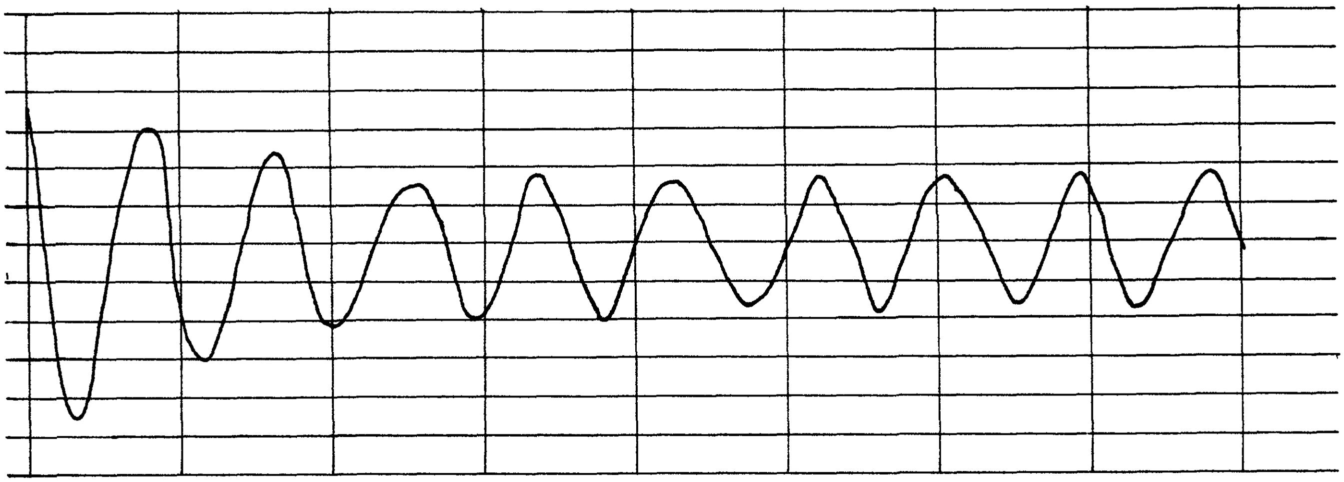
The remaining curves follow the trend of the theoretical results including the convergence around the value $\frac{\omega^2 L}{2g} = 4.0$. There is, however, a pronounded "bump" in the experimental results for the higher Froude numbers in the area $\frac{\omega^2 L}{2g} =$ 2.0-2.5. In this region, both the waves generated by the forward motion and those generated by the rolling motion are approaching a wave length equal to the vessel length. This condition seems to be analagous to the "humps" observed in resistance curves. In this case, the superposition of similar wavelength wave systems increases the energy absorbed by the waves and thus the damping effect. This phenomena is not evident in the theoretical results and it would seem that this is due to the submergence of the ellipsoid. In addition, the calculated results are based on radiated wave energy at an infinite radius from the body. This may not be able to account for effects due to local wave interference at the free surface.

Table 2 TRIAL RESULTS

Peak Recorded Roll Amplitude (Deg)
[Model Displacement (lbs)]

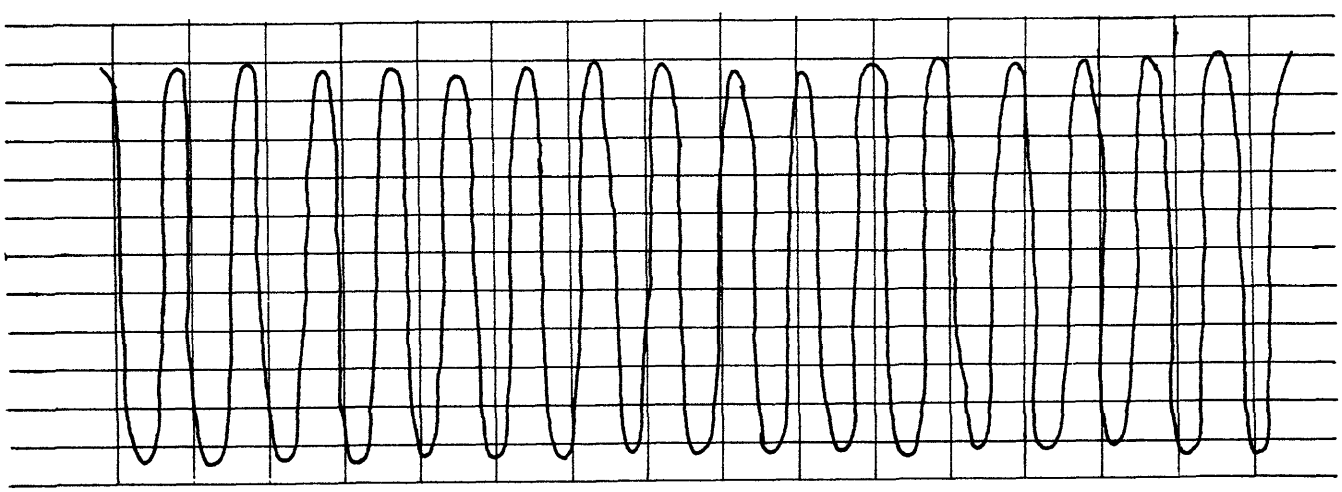
Speed (kts) \ Roll Period (sec)	2.25	1.80	1.44	1.25	1.15	0.92
0.0	15.4 [47.23]	13.5 [47.23]	12.5 [47.23]	12.3 [45.22]	10.8 [46.23]	5.7 [66.20]
0.759	15.8 [47.23]	13.3 [47.23]	12.5 [47.23]	11.4 [45.22]	11.1 [46.23]	5.3 [66.20]
1.543	10.3 [47.23]	10.8 [47.23]	9.1 [47.23]	7.1 [45.22]	6.8 [46.23]	4.6 [66.20]
1.985	7.5 [47.23]	5.7 [46.75]	8.7 [47.23]	6.2 [45.22]	5.3 [46.23]	4.8 [66.20]
2.316	6.5 [47.23]	7.0 [47.23]	6.5 [47.23]	4.9 [45.22]	4.3 [46.23]	4.6 [66.20]
2.781	3.8 [47.23]	5.5 [47.23]	2.9 [46.75]	2.8 [46.75]	4.0 [46.23]	4.3 [66.20]
3.062	3.5 [47.23]	2.9 [46.75]	2.7 [46.75]	3.7 [45.22]	3.4 [46.23]	4.0 [66.20]

Figure 4 Recorded Trial Results



Run No. 16-C
Model Speed 2.781 kts.
Roll Period 1.8 sec.

Chart Scale .2 in = 2.5deg
Chart Speed 10 mm/sec



Run No. 13-C
Model Speed 0.759 kts.
Roll Period 1.8 sec.

Chart Scale .2 in = 2.5deg
Chart Speed 5 mm/sec.

Table 3   Non Dimensional Damping Coefficients

$$\frac{100 b_{44}}{\Delta B \sqrt{L/g}}$$

b_{44} = roll damping coefficient

Δ = vessel displacement

B = vessel beam

L = vessel length

Frequency $\left(\frac{\omega^2 L}{2g}\right)$ / F_n (speed, kts)	2.74 (.67)	3.49 (1.04)	4.36 (1.62)	5.03 (2.16)	5.46 (2.55)	6.83 (3.98)
0.0 (0.0)	1.067	2.096	2.503	2.290	2.254	4.375
0.10 (.759)	1.041	2.122	2.503	2.473	2.194	4.703
0.20 (1.543)	1.598	2.613	3.318	3.966	3.582	5.018
0.25 (1.985)	2.199	3.091	3.596	4.540	4.593	5.196
0.30 (2.316)	2.536	4.036	4.812	5.749	5.663	5.418
0.35 (2.781)	4.334	5.136	6.718	6.215	6.086	5.796
0.40 (3.062)	4.709	6.077	7.215	7.614	7.157	6.235

Figure 5 Roll Damping Coefficient Versus
Froude Number

Roll Frequency = 2.79 rad./sec.

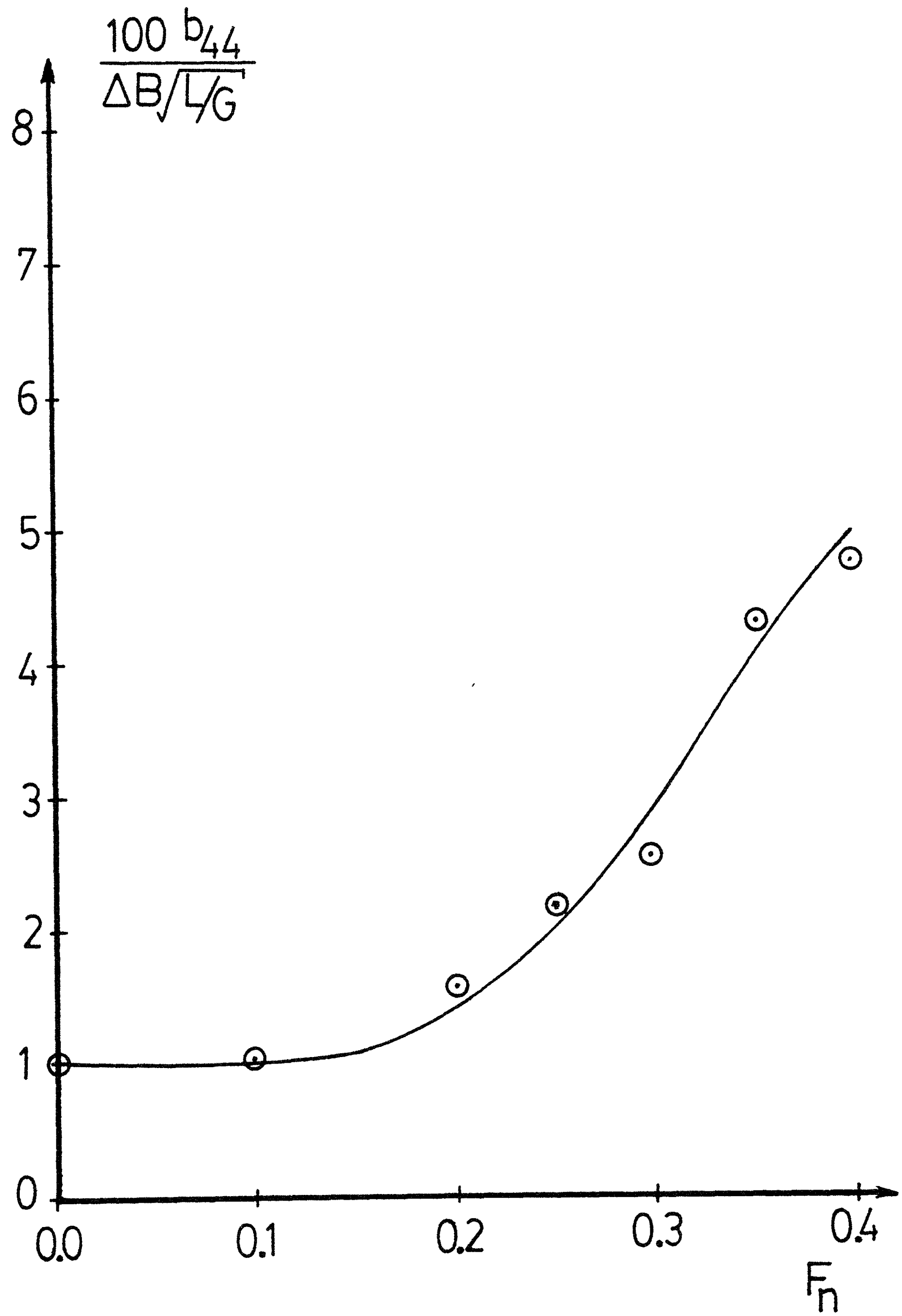


Figure 6 Roll Damping Coefficient Versus Froude Number

Roll Frequency = 3.49 rad./sed

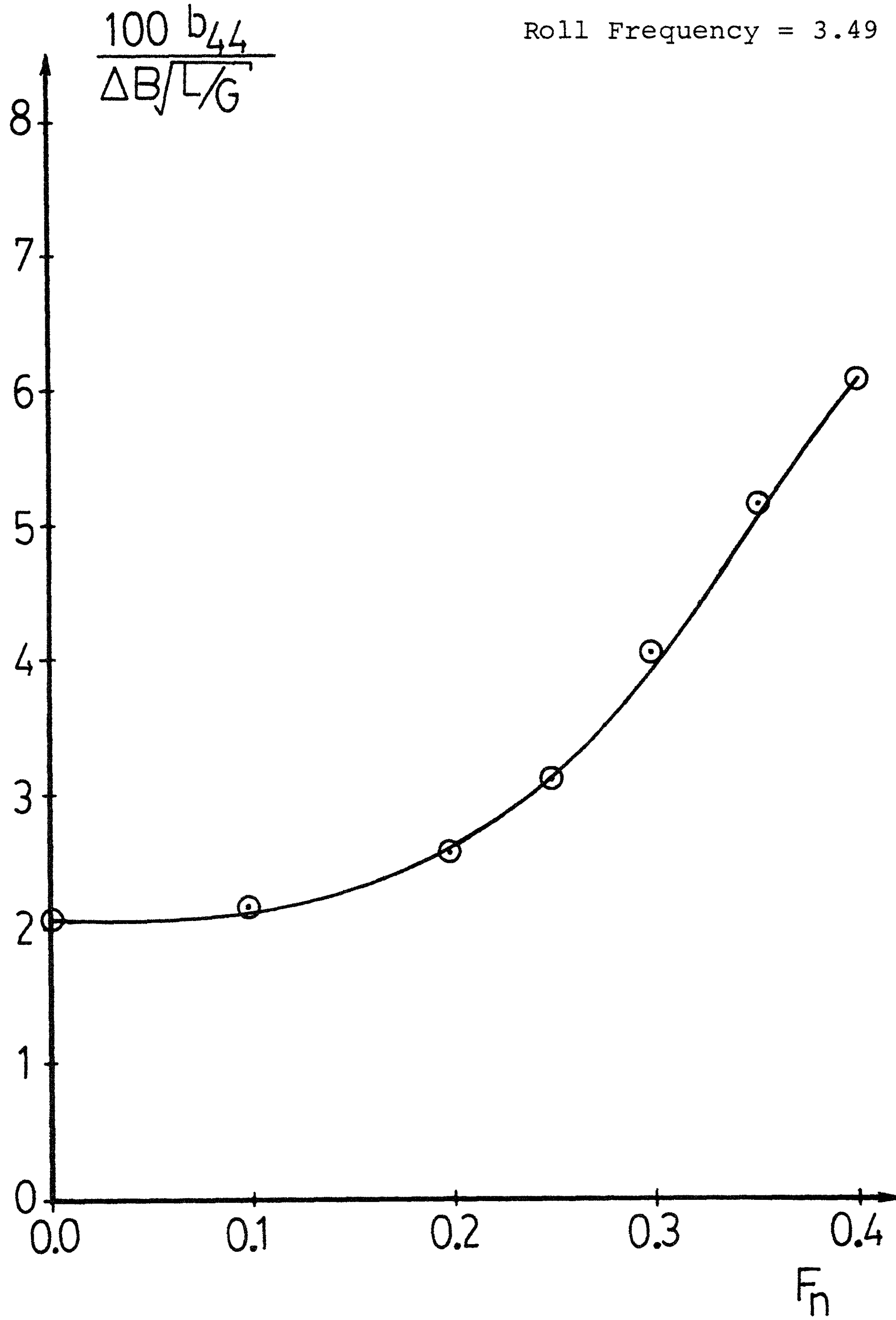


Figure 7 Roll Damping Coefficient Versus Froude Number

Roll Frequency = 4.36 rad./sec.

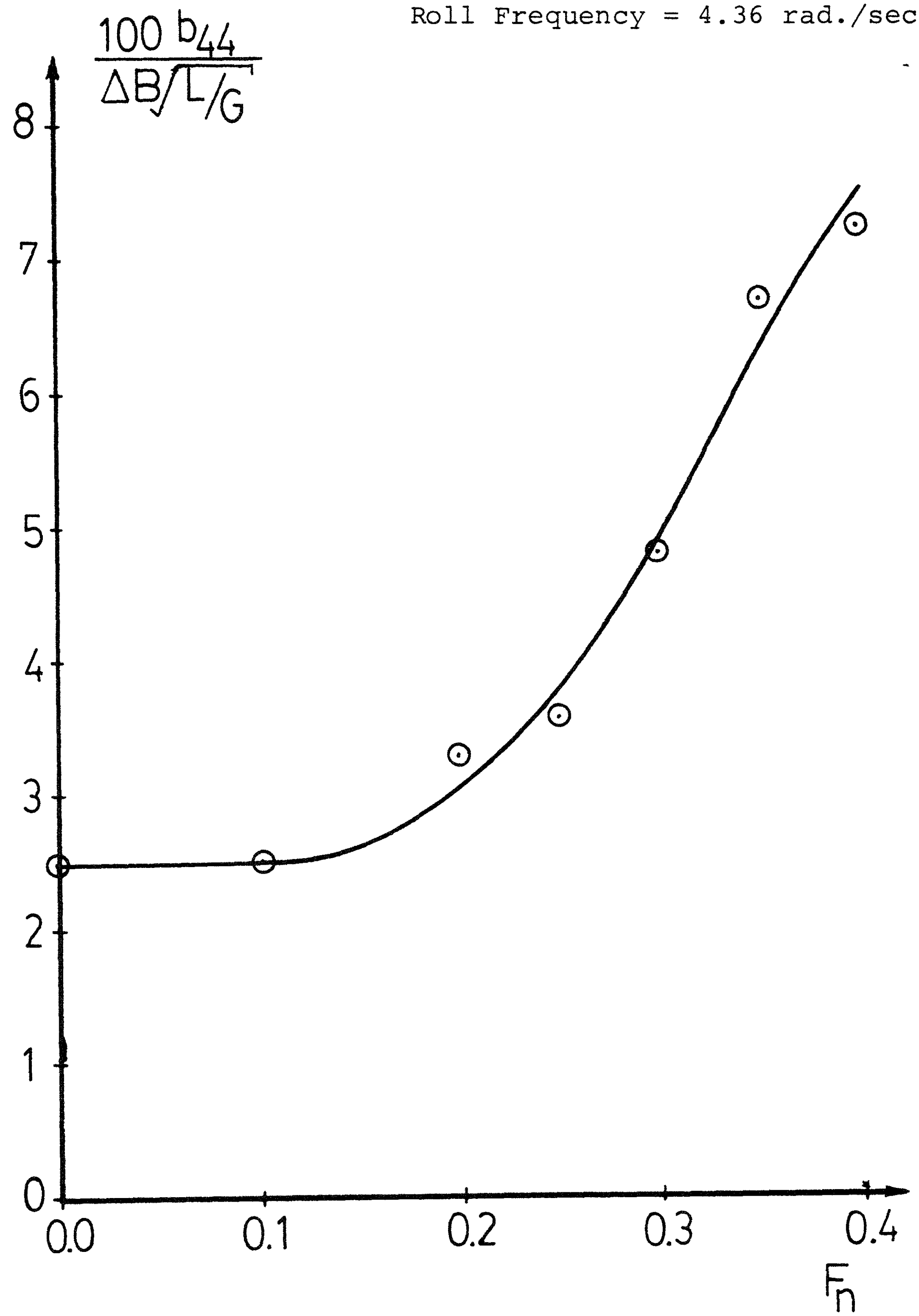


Figure 8 Roll Damping Coefficient Versus Froude Number

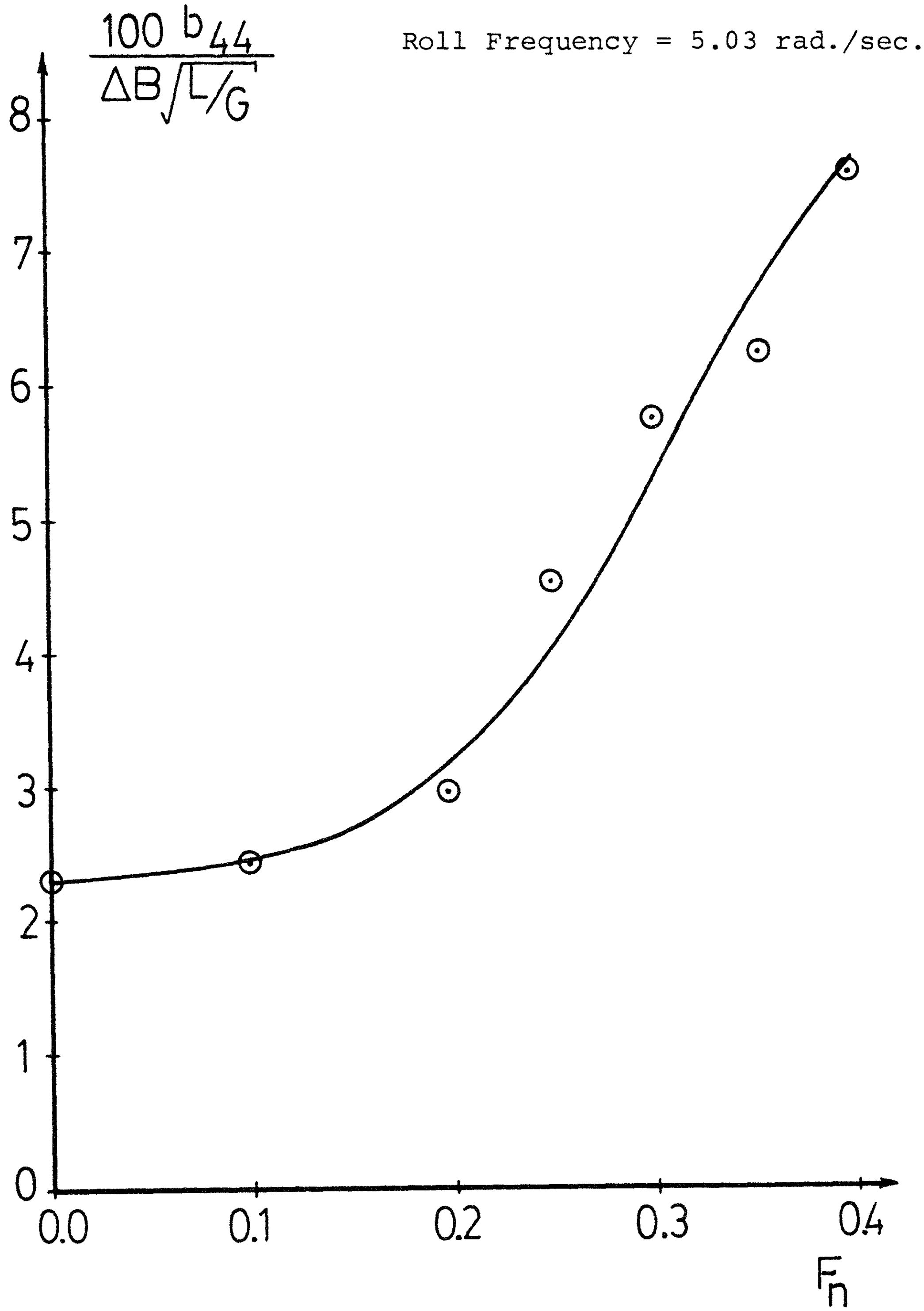


Figure 9 Roll Damping Coefficient Versus Froude Number

Roll Frequency = 5.46 rad./sec.

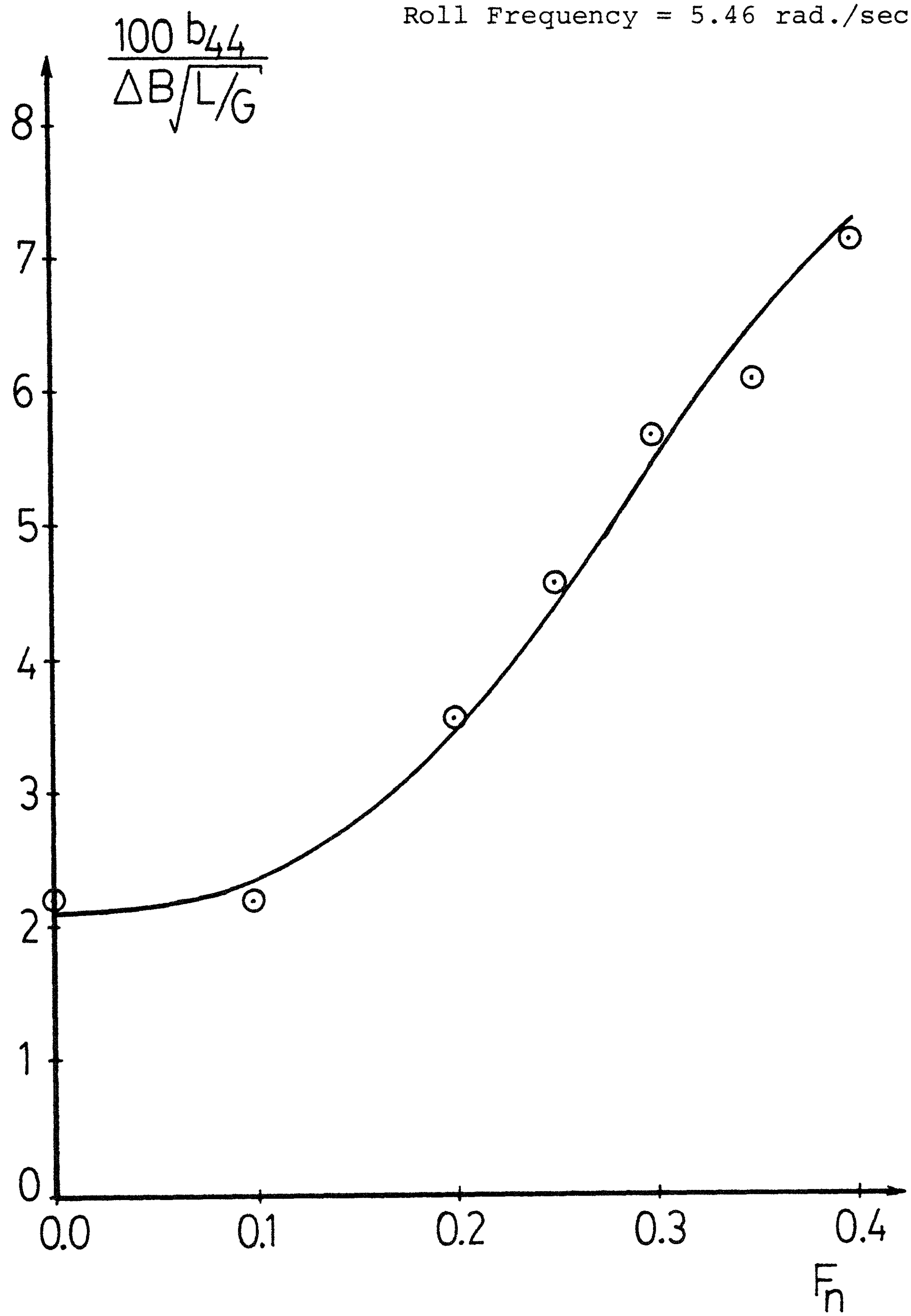


Figure 10 Roll Damping Coefficient Versus Froude Number

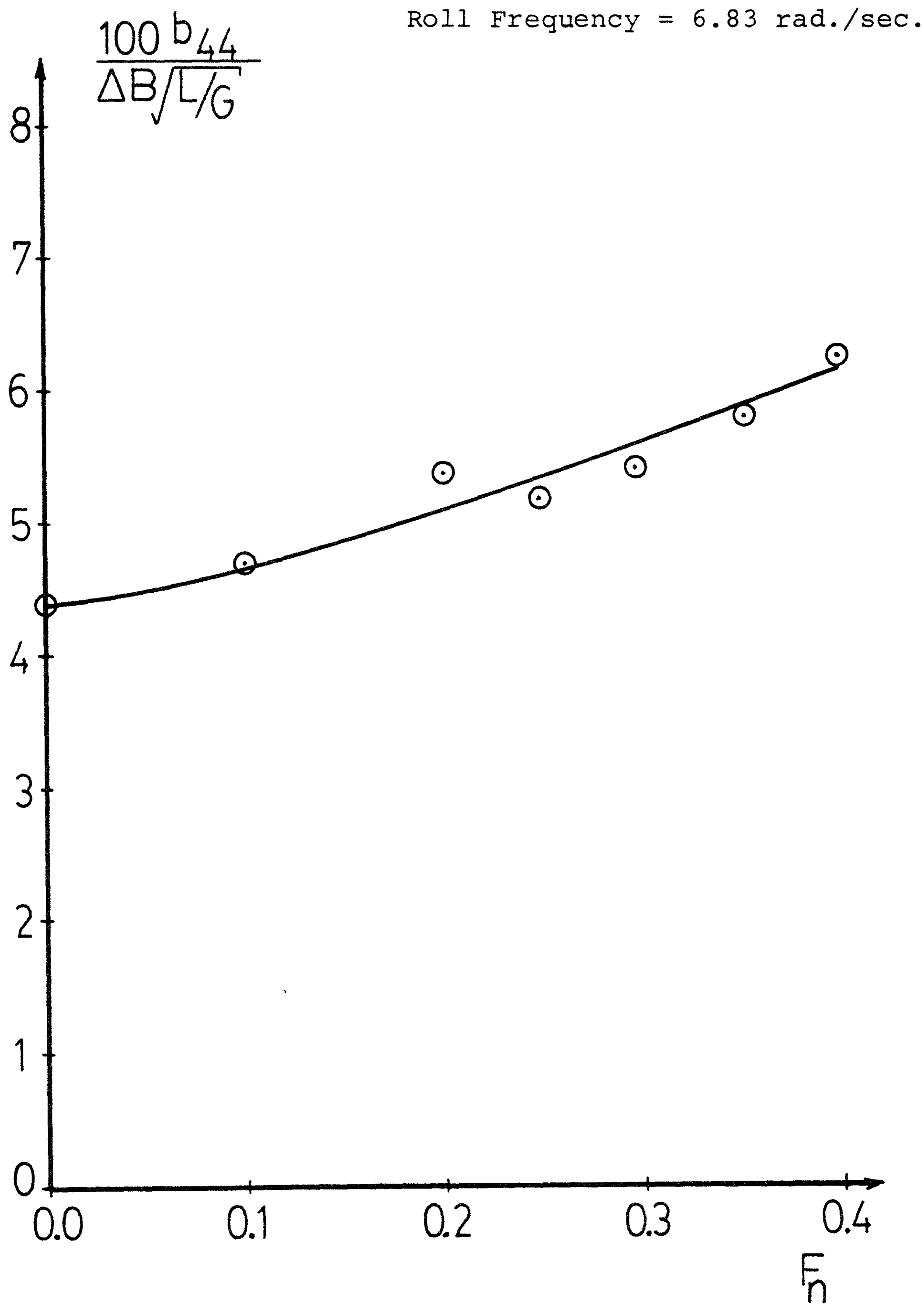


Figure 11 Roll Damping Coefficient Versus Frequency Number for Mariner Type Hull Model

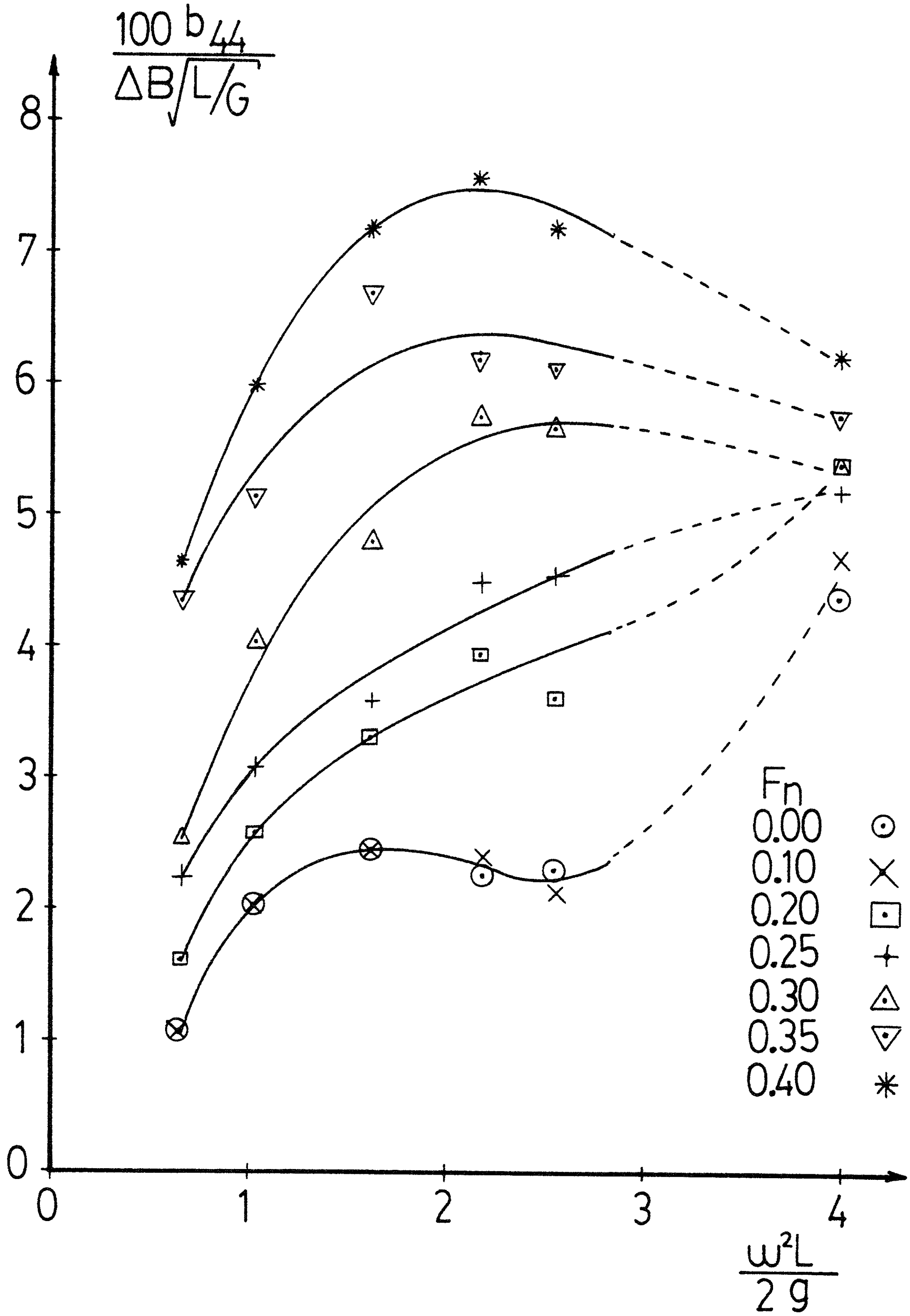
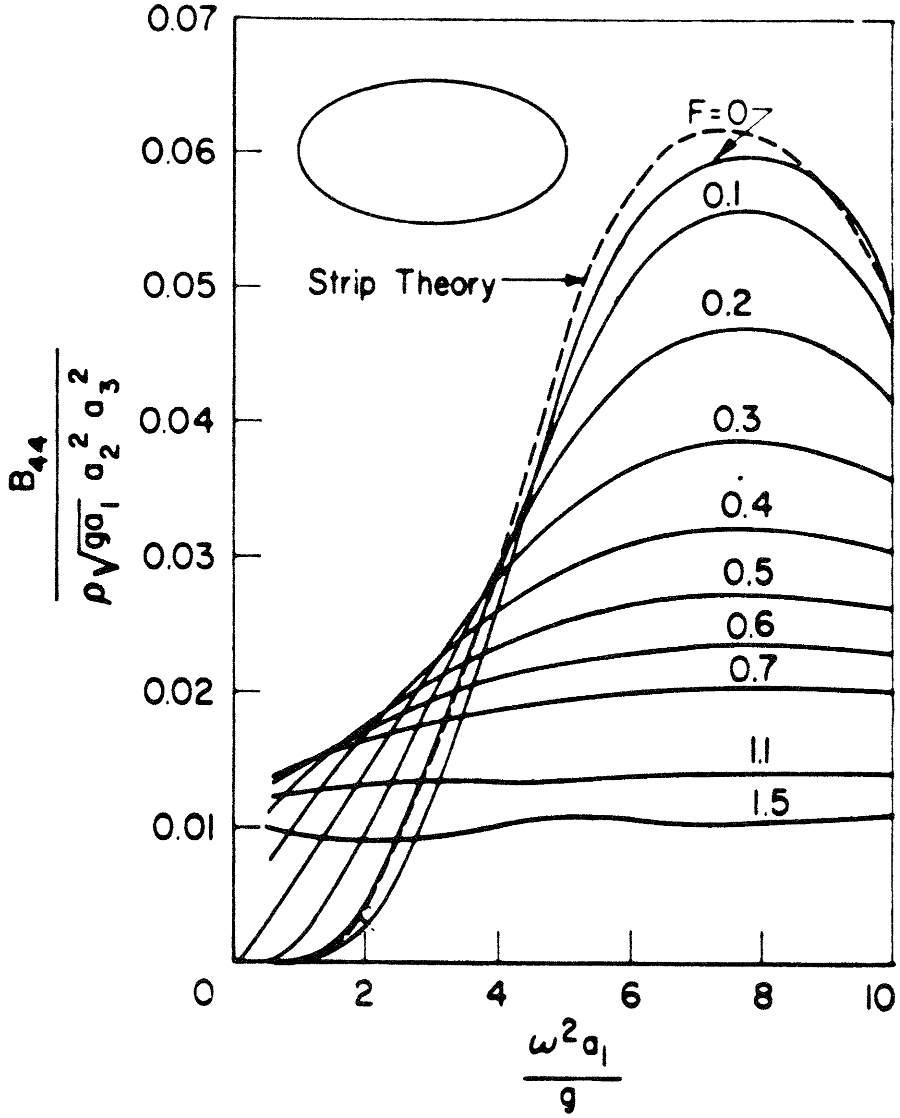


Figure 12 Roll Damping Coefficient Versus Frequency Number for Submerged Ellipsoid



Taken from: J.N. Newman
The Damping of an Oscillating Ellipsoid Near a Free Surface
Journal of Ship Research, Dec. 1961

Conclusion

In conclusion, it can be clearly seen that vessel forward speed has a significant effect on roll damping. For the vessel tested, wave related damping increases by a factor of 2 to 3 between the zero speed condition and normal operating speed. These results show good agreement with experiments done by Thews (1) in 1938 and demonstrate that the three-dimensional theory presented by Newman (6) is qualitatively valid for vessels operating on a free surface. This should allow reasonable predictions of the increase in roll damping for vessels underway, thus improving the accuracy of motion predictions.

On the practical side, it is interesting to note that although most vessels are unlikely to be able to increase speed sufficiently, under heavy rolling conditions to take advantage of the increase in damping, the standard procedure of reducing speed in oblique seas may result in increased roll amplitudes.

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