Report no. 218.

LABORATORIUM VOOR SCHEEPSBOUWKUNDE

TECHNISCHE HOGESCHOOL DELFT

DEVELOPMENT AND DESIGN OF PASSIVE ROLL STABILISERS

by

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Discussion by Ir. J.H. Vugts.

March 1968. Paper no. 6 R.I.N.A.-Meeting

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Discussion by Ir J.H. Vugts to:

"DEVELOPMENT AND DESIGN OF PASSIVE ROLL STABILISERS".

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(paper No. 6 of the R.I.N.A.-meeting on March 27, 1968).

The information given in the paper is an interesting illustration of the influence of variation of parameters in the equations (1) and (2). The whole paper is, in fact, based on these equations, which formulate the problem of a damped vibratory system with attached an also damped vibration absorber. However, this mathematical model is not, or anyhow only in a very restricted way, applicable to the rolling of a ship equipped with a roll damping tank of the free surface type. In our opinion this set of equations can present very misleading results. Therefore it looks of more value to state the reasons why the mathematical model is not correct and to demonstrate its effect on the predicted rolling of the ship than to dispute individual statements made in the paper.

Unfortunately the theoretical treatment of roll damping tanks is mostly completely borrowed from the double pendulum model or from the vibration absorber, which are fully equivalent in principle. Both cases represent mechanical systems with well-defined and unvariable quantities as masses and spring constants. By lack of any better approximation until now the ship in itself when rolling in beam seas must mostly be considered to be equivalent to such a mass-spring system with constant coefficients. And possibly for the U-tube tanksystem this concept is equally applicable as a fair approximation. Horn [1, disc] in 1911 and later Chadwick and Klotter [2,disc.] in 1954 have elaborated on these theories.

When the interest in passive tank systems revived and the free surface type was studied again, it was logical that investigators tried to place it within the scope of known theories. But experiments have shown that for a free surface type of tank the system-coefficients are largely dependent on frequency, and also on amplitude of motion, and thus are far from constants. Therefore the above-mentioned mathematical model fails completely to describe the problem of rolling of a ship with a free surface tank. Of course it is possible to reduce the actual mathematical model per frequency of motion to a set of equations of the form (1) and (2). But then for any other frequency different values of the tank parameters k and μ should be used.

When the moment exerted by the tank on the ship has been determined by experiments a tank equation is not necessary any more. Then only a ship roll-equation remains with an additional frequency dependent term for the tank moment.

Thus:

$$I_\phi \ddot{\phi} + N_\phi \dot{\phi} + R_\phi \phi = K_w + K_t,$$

where:

I_ϕ = virtual ship mass moment of inertia

N_ϕ = hull damping coefficient

R_ϕ = restoring coefficient

K_w = wave exciting moment

K_t = tank moment, which can always be written as $K_t = \Delta R_\phi \cdot \phi + \Delta N_\phi \cdot \dot{\phi}$, with ΔR_ϕ and ΔN_ϕ experimentally determined coefficients. Compare ref.[3,disc.], where all numerical information on the tank data has been published.

Rewriting this equation there appears

$$\ddot{\phi} + \left\{ \nu_\phi \omega_\phi + (\Delta \nu_\phi) \cdot \omega_\phi \right\} \dot{\phi} + \left\{ \omega_\phi^2 - \frac{\Delta R_\phi}{R_\phi} \omega_\phi^2 \right\} \phi = \alpha_w \omega_\phi^2 \cos \omega t.$$

where $\nu_\phi \omega_\phi \equiv K$ of the paper,

and $\Delta \nu_\phi \cdot \omega_\phi = -\frac{\Delta N_\phi}{\sqrt{I_\phi R_\phi}} \cdot \omega_\phi$, or equivalently $= \Delta K^*$

$\frac{\Delta R_\phi}{R_\phi}$ is equivalent with μ^*

The latter two quantities: ΔK^*, and μ^* are strongly dependent on frequency and no constants. Ultimately one obtains:

$$\ddot{\phi} + (K + \Delta K^*) \dot{\phi} + (1 - \mu^*) \omega_\phi^2 \phi = \alpha_w \omega_\phi^2 \cos \omega t.$$

According to this equation some of the calculations have been carried out again for the same ship and for the same tanks as used in the paper. The results have been compared to the given results and are shown by dotted lines in figures I, II and III, corresponding to figures 2,3 and 4 of the paper.

Fig.I shows clearly that the fully drawn lines are in error, and that they are misleading even for a qualitive approach. The character of the dotted lines is completely different.

From fig.II it appears that the open rectangular tank without any restriction for this case already corresponds to a k -factor of about 1.5. And with a smaller waterdepth curve 4 with a k-factor of 2 can be approached; compare figI with h/b between the dotted curves 2 and 3. The mathematical model of the paper suggests wrongly that the response curves have two distinct peaks and that curves for different k's have two fixed crossing points. When a restriction would be brought in the dotted line would be changed, but not according to curve 3 passing into curve 4. Something like curve 4 can be obtained with restrictions as well at the expense of much more water than with the unrestricted rectangular tank. It is moreover questionable whether curve 4 is the best to be selected. The athwartship's accelerations cause at least as much inconvenience as the roll amplitudes. The reproduced line denotes a constant acceleration and from this point of view the dotted line is better than all full curves shown. Considering irregular seas instead of regular waves the shape of the wave slope spectrum is of importance as well.

To figure III the same comments as to figure II could be made.

A proof that the equation used in this discussion is a very realistic representation of the case of pure rolling can be found in ref. [4,disc.] fig.8, which is represented here as fig. IV. When the ship-data are determined experimentally as depending on frequency of motion as well, the calculated and measured roll responses fully coincide, both for amplitude and phase. When the ship-data are taken as approximate constants, as generally done, the agreement is still very reasonable and certainly a good basis for comparison of the ship without and with tank in operation.

Summarizing I state that the mathematical model of the paper is definitely insufficient and leads to erroneous results, even in a qualitative sense. The effect of the variation of internal tank damping can not be judged from figures 3 and 4. From previous results published by the Shipbuilding Laboratory At Delft in [12] and [3 disc.] it is evident once more that it is a rule rather than an exception that the plain rectangular tank is best, or at least as good as one with any type of restriction. This fact was already established by R.E. Froude, in the discussion to Watt's paper of 1881. Froude had a very clear physical insight in the problem. I cite his remarks from the Transactions:

"It is clear that the work which the water takes out of the ship must be absorbed in the friction of the water - the action of the water slushing about. Since all the work is to be absorbed, it might seem at first sight the greater amount of obstruction there is to the motion of the water, the more effective the operation of the water in absorbing the work would be. But the only way in which the water influences the rolling of the ship is by its presence alternately on the two sides of the ship; and the friction of the water must operate to resist the passage of the water from side to side, and so do harm."

Only in the range of the very low roll resonance frequencies requiring waterdepht- ratios h/b of, say, 0.03 or less, restrictions cause a favourable shift in the field where the tank is particularly active. Therefore only in these cases restrictions need to be considered. But it is quite disputable if there is much sense in equiping these tender vessels, which are generally easy rollers, with a roll damping tank.

Finally I like to comment on some other points of the paper. Under case 4 it is stated that the magnification factor at $\omega = o$ has a different value for different tank sizes, due to the loss in roll stiffness. This is only true in the mathematical model, both that in the paper and the one advocated in this discussion. But it isworth while to remember that actually at sea a ship with any $\overline{GM}$, with or without any size of tank, will have a magnification factor of 1 in very long waves, as the angle of roll will always be equal to the wave slope. This is due to the combined effect of rolling and swaying, which is left out of consideration in the mathematical model. The ship is subjected to the

velocity distribution of the orbital motion of the wave particles and will like a raft adjust itself to the apparent vertical, perpendicular to the water surface.
In the introduction is said that information on ships fitted with the system of Van den Bosch and Vugts is awaited with interest. I like to remark that any tank of the free surface type, with or without some type of restrictions and wether it is called a N.P.L.-system, a Flume-system or otherwise, belongs to, what is called, "their" system, but in fact is Watts' system. The shipbuilding Laboratory in Delft has been involved in the design of several unrestricted tanks, but only little experience is fed back to it because The Laboratory has no commercial interest in the applications. It is known that several bulk-carriers in ballast condition fill a hold to a certain level to act as roll-damping tank and that its effect is"dramatic".(to use the sea officers' words); the ship will seemingly not roll at all. A test on a naval ship at sea has been reported in ref.[5 disc.] Model tests in comparison with a U-tube tank can be found in ref. [4, disc.].
In the end it is emphasized that the criticism expressed in this discussion only concerns the theoretical background and the design philosophy of the paper, and certainly not the tanks installed on recommendations of N.P.L.

Additional references.

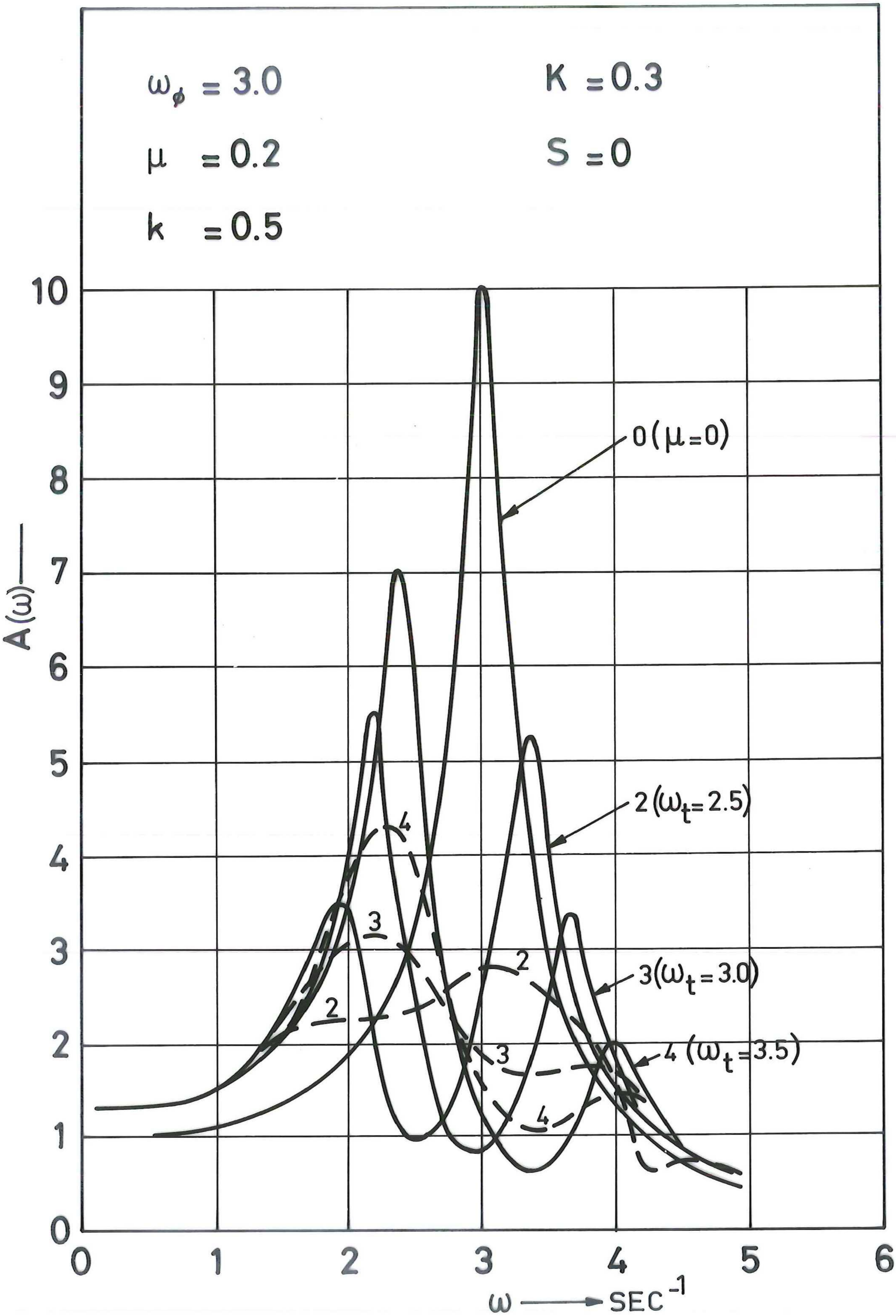
[1,disc.] Horn, F., "Zur Theorie der Frahmschen Schlingerdämpfungstanks", Jahrbuch der Schiffbautechnischen Gesellschaft 12, 1911; p.453.

[2,disc.] Chadwick, J.H. and Klotter, K., "On the dynamics of anti-rolling tanks", Schiffstechnik 8, Febr. 1954; p. 85.

[3,disc,] Van den Bosch, J.J. and Vugts, J.H., "Roll damping by free surface tanks", Report no 83 S of the Netherlands' Ship Research Centre TNO, April 1966.

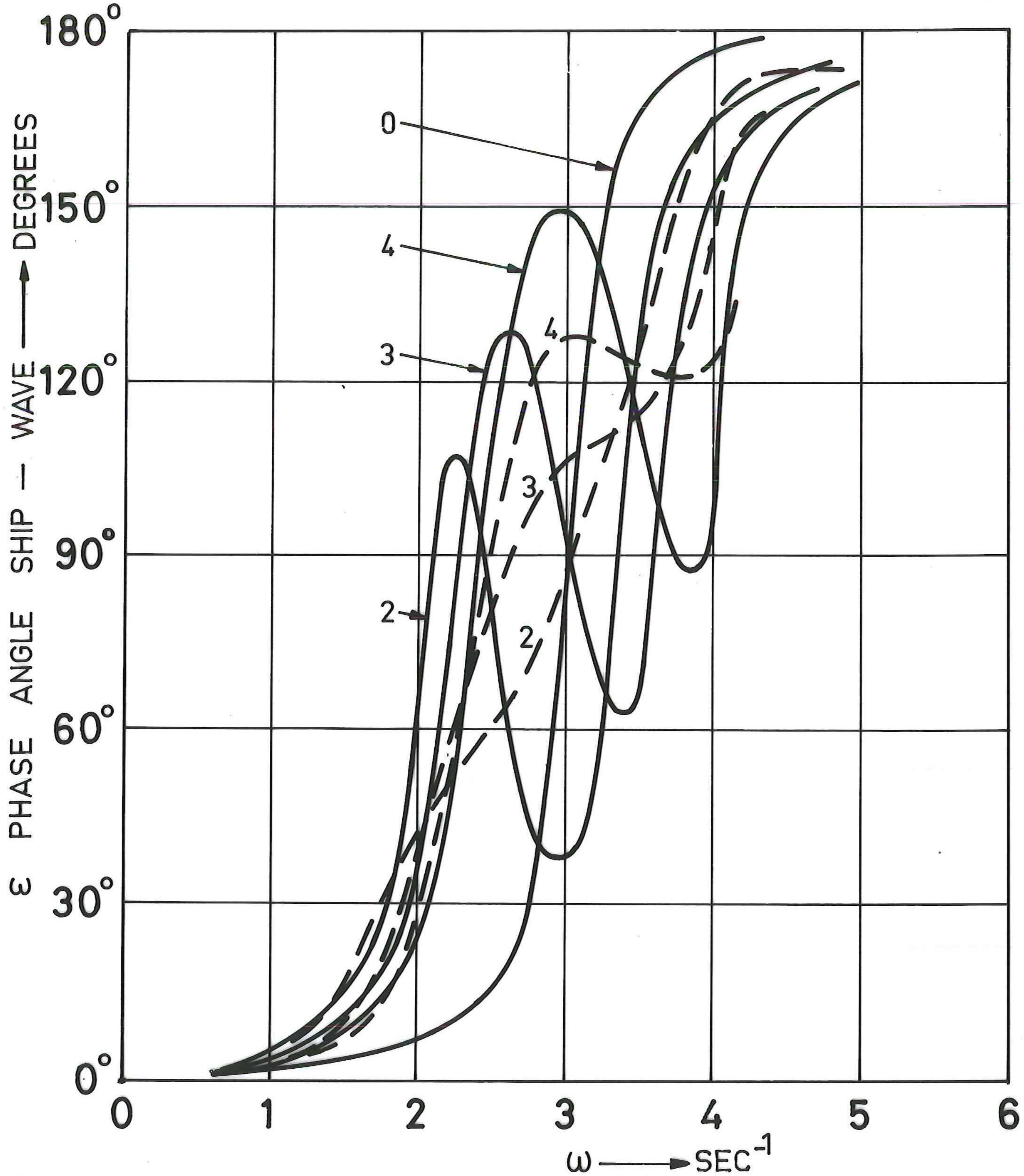
[4,disc.] Bootsma, J. and Van den Bosch, J.J., "On the efficacy of two different roll-damping tanks", Report no. 97S of the Netherlands' Ship Research Centre TNO, July 1967.

[5,disc.] Van den Bosch, J.J., "Meting van de Slingerbeweging a/b HR.MS. "Wolf", report no.184 of the Shipbuilding Laboratory, Technological University, Delft, July 1967.



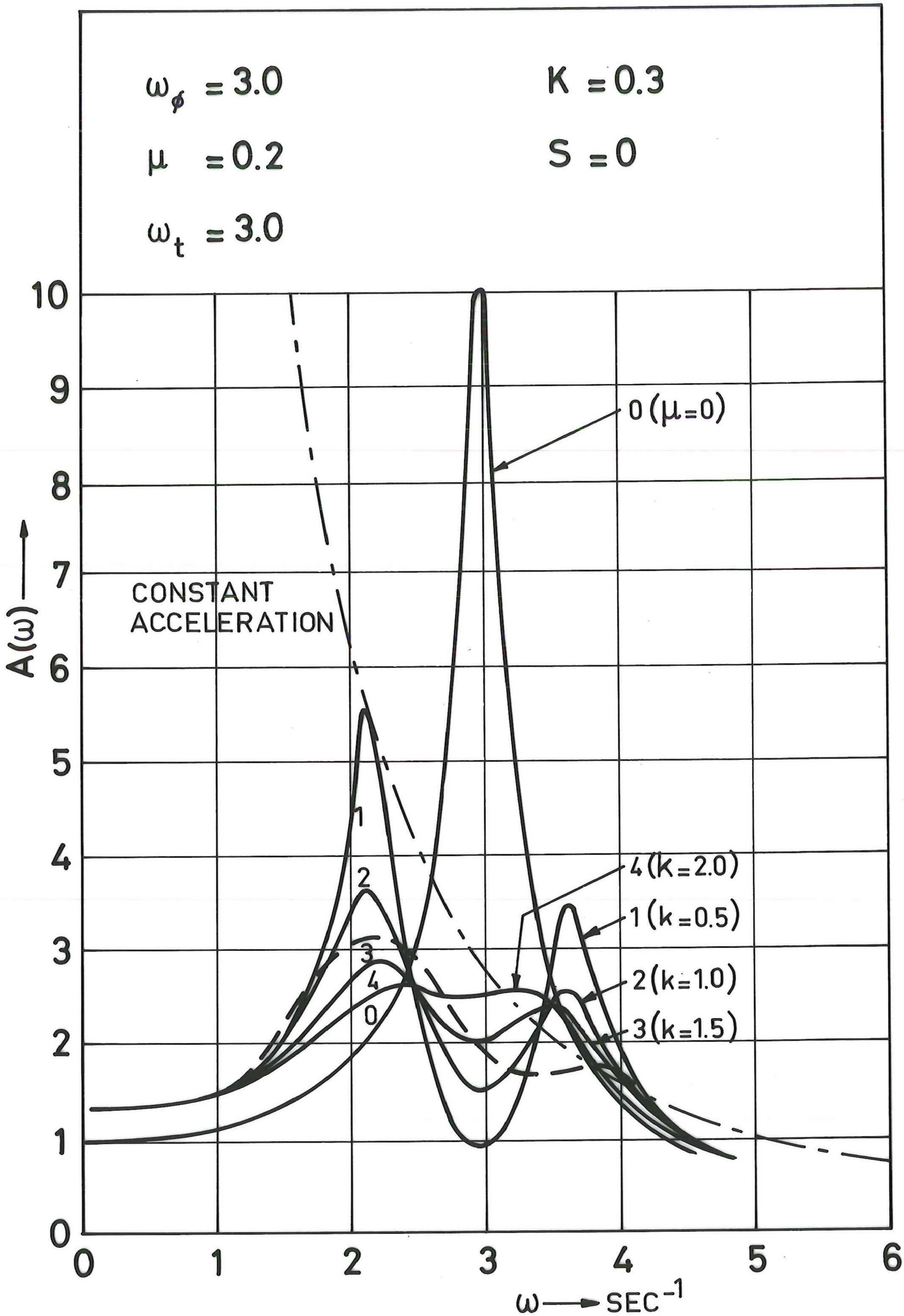
(A) AMPLIFICATION FACTOR

IA



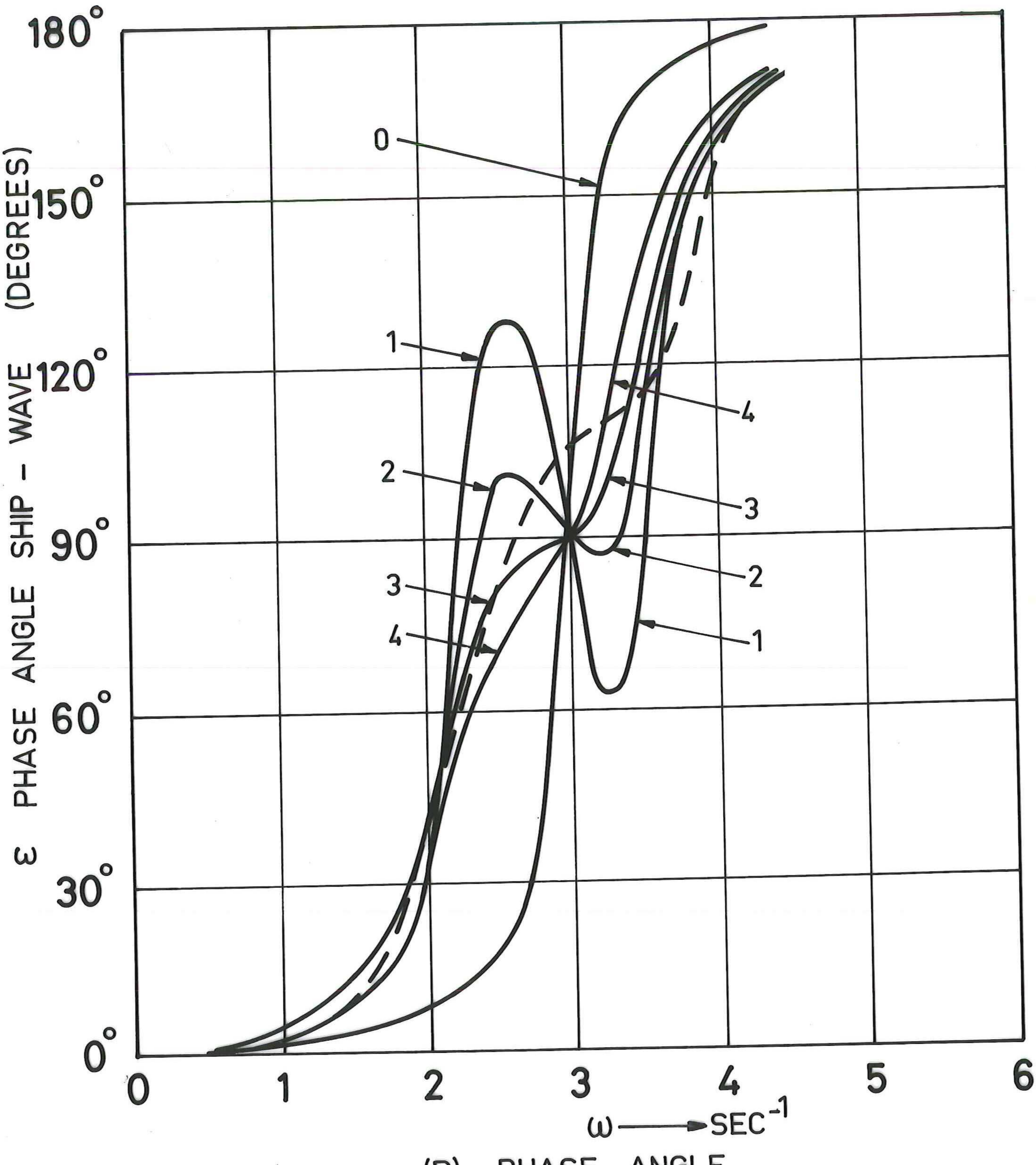
(B) PHASE ANGLE

I.B



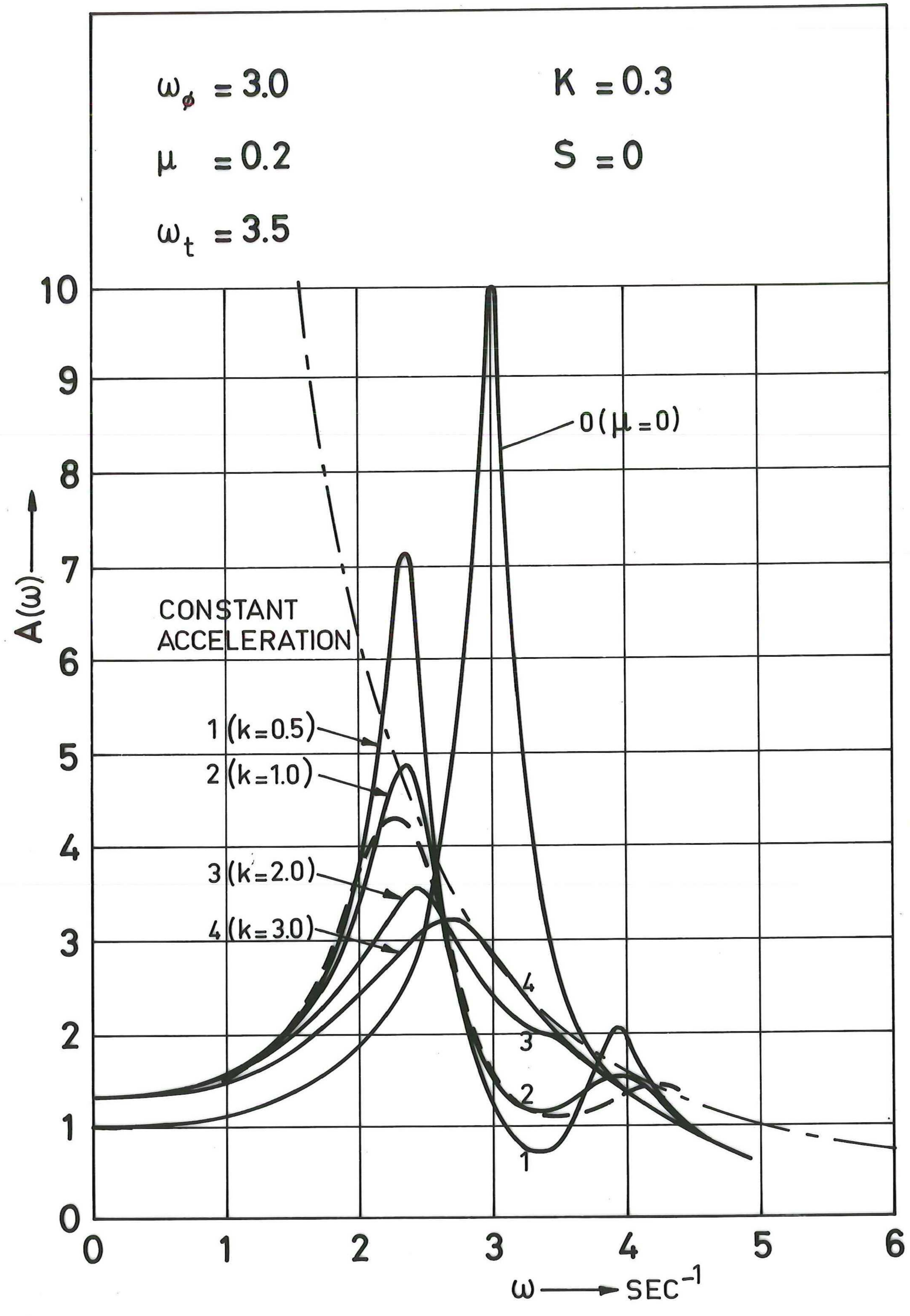
(A) AMPLIFICATION FACTOR

II A



(B) PHASE ANGLE

II B



$\omega_\phi = 3.0$
$\mu = 0.2$
$\omega_t = 3.5$
K = 0.3
S = 0
$A(\omega)$
CONSTANT ACCELERATION
0 (μ = 0)
1 (k = 0.5)
2 (k = 1.0)
3 (k = 2.0)
4 (k = 3.0)
ω → SEC^{-1}

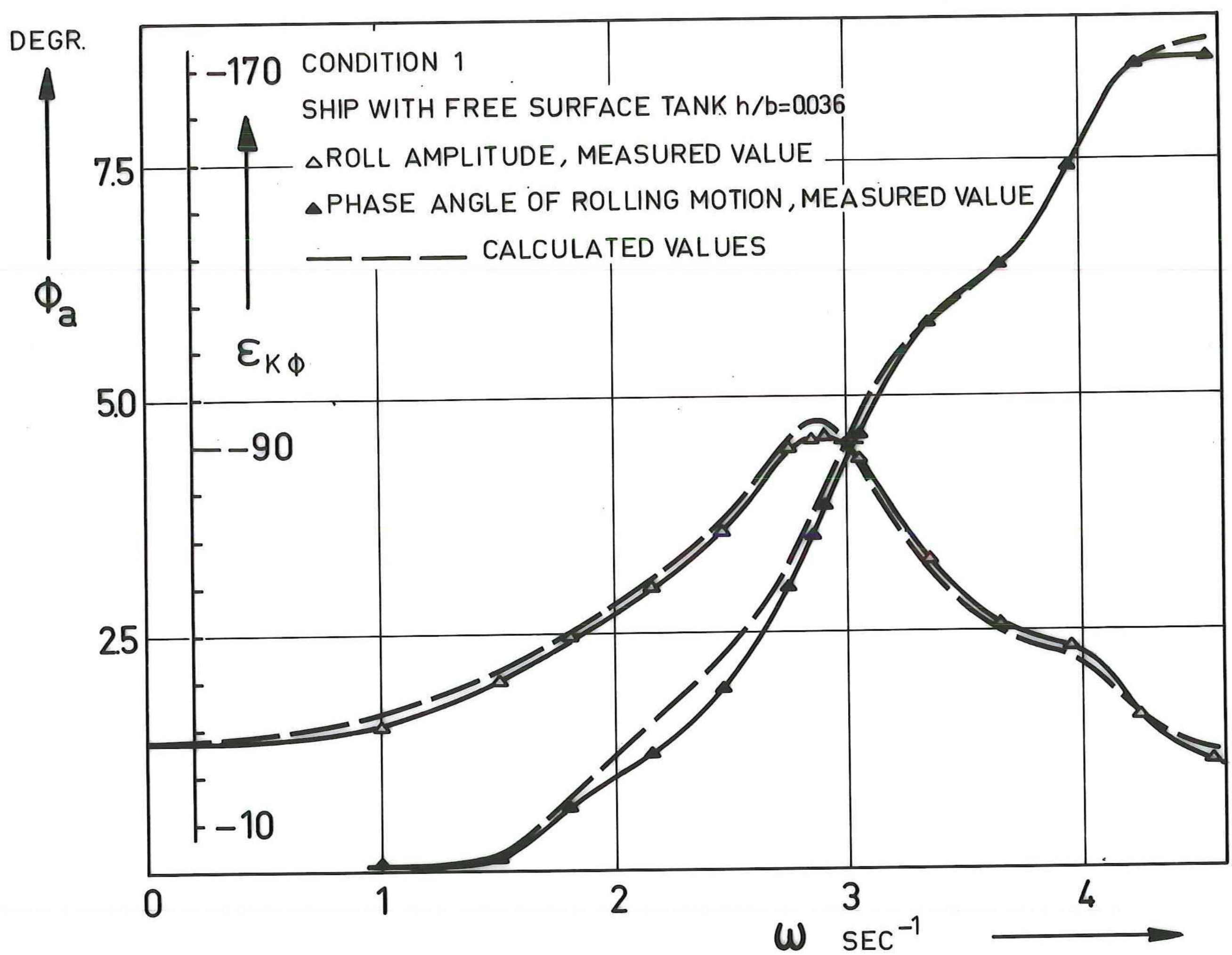


Fig.8 Rolling motion of the model with F.S.tank
Forced oscillation test
Comparison between experiment and calculation