Hydrodynamic Aspects of Ships Colliding with Fixed Structures

Aspects hydrodynamiques de la collision de navires avec des structures fixes

Hydrodynamische Aspekte bei Schiffskollisionen mit stationären Objekten

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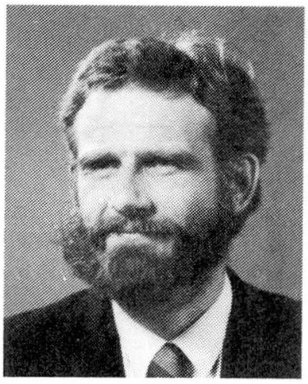

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SUMMARY

A ship impact against an elastic structure carries a certain added mass of water with it, the quantification of which is crucial in the design. The magnitude of this hydrodynamic added mass is influenced by: vessel size, shape, draught, underkeel clearance, spring characteristic of the obstacle and collision mode. The paper describes a series of experiments to study the added mass during impact. Results are of interest to ship-to-ship collisions and to fender design. The resulting added mass values appear to be higher than the expected and frequently used design values.

RÉSUMÉ

Le choc d'un navire contre une construction élastique s'accompagne d'une certaine masse d'eau, dont la quantification est capitale pour une étude. L'importance de cette masse hydrodynamique ajoutée est fonction de la taille du navire, de sa forme, du tirant d'eau, de la profondeur sous la quille, de l'élasticité de l'obstacle, et du mode de collision. Un programme d'essais a été entrepris pour étudier la masse ajoutée pendant l'impact. Les résultats sont intéressants pour les collisons entre navires et pour la conception des défenses. Les valeurs réelles de la masse ajoutée sont plus élevées que celles fréquemment utilisées dans les calculs.

ZUSAMMENFASSUNG

Ein Schiff, das auf ein elastisches Objekt prallt, nimmt eine gewisse zugefügte Wassermasse mit sich mit, deren Festlegung während des Entwurfes wichtig ist. Die Größe dieser hydrodynamischen zugefügten Masse wird beeinflußt von: Schiffsgröße, Form, Tiefgang, Bodenfreiheit, Federkonstante des Hindernisses und Kollisionsart. Experimente zur Untersuchung der zugefügten Masse während des Aufpralls werden beschrieben. Die Resultate sind in Hinsicht auf Schiff-gegen-Schiff-Kollisionen und Fenderentwürfe interessant. Die gefundenen Werte liegen höher als erwartet und sind auch größer als die gewöhnlich benutzten Entwurfswerte.

1. INTRODUCTION

When a ship approaches a jacket, a jetty or some other structure, the amount of kinetic energy present will in most cases be absorbed by elastic or plastic deformation of usually some fendering system. The most common way to obtain the design loads on the structures and fendering systems is to estimate this amount of kinetic energy of the ship. For this estimation some assumptions have to be made. The biggest ship approaching the structure is selected and the maximum speed at which and the mode under where contact is made, is usually taken. When a ship with a certain approaching speed will be decelerated, also a certain amount of water, which is moving along with the ship, has to be decelerated. This means that the mass to be taken into account for the kinetic energy is not only the mass of the ship; there is a certain additional mass, the so-called hydrodynamic added mass.

The most common practice is that a constant amount of added mass is applied to the ship's mass, independent of all factors affecting it; see Saurin [19] and Vasco Costa [21] and [22]. Thoresen [20] takes possible eccentricities into account during collision. Giraudet [11] accounts for underkeel clearance only. In recent years the possibility of an accidental collision between a ship and an offshore platform has drawn considerable attention ([6], [10], [16], [17] and [18]), in which basically the structural aspects were considered.

An advanced computer program, dealing with all mentioned parameters affecting the added mass, has been developed at N.S.M.B. This program was presented at the OTC by Van Oortmerssen in 1974, [15]. The data presented in this paper are a result of an extensive model test program for further investigation and validation of the computer program. Another goal of the experiments was to supply potential designers with more test results in order to support their designs.

Parts of the results of this large test series have already been presented in papers contributed to the 1979 and 1983 OTC Conferences, [1] and [2].

2. CURRENT DESIGN PRACTICE

2.1 Basic equations

The current practice in the design calculations and dimensioning of fendering systems and flexible structures is almost invariably based on energy considerations [8], [12], [13] and [14]. In these it is assumed that the loss of kinetic energy of the ship is transformed into an equal amount of energy absorbed by the fender or structure, i.e.:

$$W = \tfrac{1}{2} M_v . V_0^2 \quad \ldots \quad (1)$$

in which:
W = work done by the fender
M_v = virtual mass of the ship, which is the sum of the ship's mass and the hydrodynamic added mass
V_0 = initial velocity of the ship on hitting the fender.

Whatever the design of the fender, the work done by the fender can be written as:

$$W = \int_{s(t_0)}^{s(t_1)} F.ds \quad \ldots \quad (2)$$

in which:
F = reaction force exerted by the fender
s = deflection of the fender
t_0 = instant of hitting the fender
t_1 = instant when the maximum deflection is attained.

Equation (1) is strictly valid for the situation where all the kinetic energy is absorbed, such that during the slowing down process no other modes of motion develop than the one present. That means that all kinetic energy is supposed to be absorbed when the fender deflection has reached its maximum.

2.2 Practical implementation

For the more complicated cases, in which yawing starts to develop during the slowing down process, the kinetic energy is never to reach zero at the point of maximum fender deflection; see Vasco Costa [21] and [22]. Equation (2) offers the possibility to determine the magnitude of the absorbed energy of the fender over the first part of the impact. This is independent of the shape of the fender characteristic. (Relation between load and deflection).

The most common formulation of the energy equation, (see [3], [19] and [21]), taking the various effects into account is:

$$W = \tfrac{1}{2}\, M.V_0^2.C_M.C_E.C_S \quad \ldots \quad (3)$$

in which:

M = mass of the ship

C_M = coefficient of added mass

C_E = coefficient of eccentricity, accounting for the position of the point of contact relative to the ship's centre of gravity. It boils down to a reduction of the energy to be absorbed, since:

$C_E = 1$ for impact aside of the centre of gravity and

$C_E \approx 0.5$ for impact at one quarter ship's length from the fore or aft perpendicular.

C_S = coefficient of deformation, taking the elastic hull deformation into account. If 10 per cent of the energy is absorbed by the hull the fender need only take 90 per cent, hence:

$C_S = 0.9$.

It will be shown in the sequel that these assumptions can be grossly in error. As already suggested by the above expression the added mass is associated with a sway motion even if the ship has a certain rate of yaw before and after hitting the fender. In so far as the authors are aware no information is known to exist in which the added mass is split up into a coefficient pertaining to sway and another to yaw in a hydrodynamically sound way.

In [1] a review has been given of most of the values for C_M in use so far. They do not take into account all effects influencing the C_M magnitude. One of the results of the test series, as presented herein, is that the accepted and frequently used C_M values appear to be too low; a finding which is of direct consequence to the designers.

3. EXPERIMENTAL WORK

The most important aim of the test program was to gather as much as possible test data on the subject for the benefit of the practising engineer. A secondary purpose was the availability of reliable test data to validate a sophisticated N.S.M.B. developed computer program on this subject.

In order to cover realistic cases the test parameters had to be selected carefully. Also, the test set-up wherein the tests were done had to be very accurate and able to reproduce the test conditions and results precisely.

3.1 The test set-up

The tests were performed at a model scale of 1 to 75. A sketch of the test set-up is shown in Fig. 1. The structure, to which the fenders were connected, was completely transparent in order to avoid reflection from some quay wall. This effect would produce lower values for the added mass coefficient, because a reflected wave would slow down the ship faster and as a result the measured peak load at the contact point would be lower.

The ship was connected to two endless wires by means of two electromagnetic couplings. By means of these wires the ship could be directed to the fender at a very precisely controlled heading and speed. Just before the moment of contact the ship was released of the wires, after which it was free to move in any mode of motion.

At the moment of impact, measurements were taken of: sway, surge, roll, yaw, roll rate, yaw rate, fender force and fender stroke.

3.2 The parameters

In order to gather data, corresponding to the most frequent situations in reality, the following selection of parameters was made:

- VLCC: 225,000 DWT, being one of the most popular sizes; see [7].
- Draught: full load, since the mass is then largest.
- Underkeel clearance: 20 per cent of the ship's draught.
- Fender characteristics: a linear relationship between deflection and reaction force, being 1600 kN/m, 4200 kN/m, 12000 kN/m and 20600 kN/m.
- Collision modes:
  Collision type 1: ship having a pure sway motion, hitting the fender at the midships.
  Collision type 2: ship having a pure sway motion, hitting the fender at 15, 20 and 30 per cent of its length forward of the midships.
  Collision type 3: ship's heading making an angle of 15 degrees with the final berthed position. Approach of the ship was a pure translation.
- Collision velocities: a range of velocities has been chosen, corresponding to 0.04 m/s up to 0.30 m/s for the full scale; see [3] and [4].

3.3 Analysis of the test results

For the analysis of the test results various principles from classic mechanics were invoked. These comprised the energy preservation law and the law of change of momentum.

Most concisely we can express the equations in the following form:

$$\int_{t_a}^{t_b} F(t).\cos\alpha.dt = C_{My}.M\{\dot{y}(t_b) - \dot{y}(t_a)\} \quad \ldots\ldots\ldots\ldots\ldots (4)$$

momentum equation for sway

$$\int_{s(t_a)}^{s(t_b)} F(t).ds = \tfrac{1}{2}\, C_{My}.M\{\dot{y}^2(t_b) - \dot{y}^2(t_a)\} \quad \ldots\ldots\ldots\ldots\ldots (5)$$

energy equation for sway

$$\int_{t_a}^{t_b} F(t).\overline{QG}.\cos\alpha.dt = C_{M\psi}.I_G\{\dot{\psi}(t_b) - \dot{\psi}(t_a)\} \quad \ldots\ldots\ldots\ldots\ldots (6)$$

momentum equation for yaw

For collision type 1, where before and after the impact solely a sideways sway motion existed, the first two equations were used. Both of them would yield a added mass coefficient for sway motion, which would not be exactly the same, see Section 5.1. For collision type 2 and 3 equation (5) could not be used anymore, since the energy would be distributed over all modes of motion, in particular, sway and yaw. The energy approach would give us more unknowns than equations. So for these cases the momentum approach remained and equations (4) and (6) were used. For collision type 2 the angle between ship and jetty was zero, so that cos α reduced to unity.

Each equation would give an added mass coefficient for any one test. This coefficient could be based on various time periods, depending on the choice of t_a and t_b. In the analysis three distinct time instants were defined, t_0 being the moment of impact, t_1 being the instant of maximum spring deflection and force and t_2 the instant of spring force being back to zero. These three points in time define three time periods which were all taken for the analysis, $t_0 - t_1$ being the most important as the added mass thus derived has a direct engineering application, while the others are of lesser importance.

4. ANALYTICAL WORK

4.1 Simple mass-spring system model

When a ship has a purely sideways motion and runs into a fender or other elastic structure exerting a force at the midships, the hydrodynamics associated with the arrested motion of the ship and the decelerated flow of the water around the ship are rather complicated. It would be interesting to see to what extent the motion of the ship, most dominantly sideways, can be predicted by a simple mathematical model and if the deflection of the fender and the exerted force can be predicted as well.

Suppose a ship with rigid body mass M and added mass coefficient C_{My} strikes an elastic structure with spring rate C at the origin of time t_0. We may illustrate this as done in Fig. 2. This single degree of freedom model can easily be solved. We may also suppose the ship to hit the fender at a point well forward of the midships as illustrated in Fig. 3. Solving the equations of motion we can evaluate the following:

maximum spring deflection: $$\hat{s} = V_0\sqrt{\frac{C_{My}.M.k^2}{C(a^2 + k^2)}} \qquad \text{(7)}$$

maximum spring force : $$\hat{F} = V_0\sqrt{\frac{C_{My}.M.C.k^2}{(a^2 + k^2)}} \qquad \text{(8)}$$

time to reach maximum : $$T_{01} = \frac{\pi}{2}\sqrt{\frac{C_{My}.M.k^2}{C(a^2 + k^2)}} \qquad \text{(9)}$$

in which "a" represents the eccentricity distance and "k" stands for the radius of gyration in yaw. When we set "a" to zero the centric impact is obtained. It can be shown that the test results agree with these simple relations, see Section 5.

4.2 Linear mathematical model with fluid memory effect

The usual approach in ship motion theory is to use an equation of motion in the frequency domain, that takes the following form:

$$\{M + a_{yy}(\omega)\}\ddot{y} + b_{yy}(\omega)\dot{y} + C_{yy}.y = F_y(\omega) \qquad \text{(10)}$$

This equation, in which all coefficients are frequency dependent, can only be solved for discrete frequencies, hence harmonic oscillations. This implies that, contrary to the looks of the equation, it is an algebraic equation rather than a differential equation.

In order to simulate time processes, like in the subject case, one needs a true differential equation in which the coefficients are constants and the time constitutes the sole independent parameter. Such an equation can be obtained by taking the Fourier transform of the above equation which leads to the following expression:

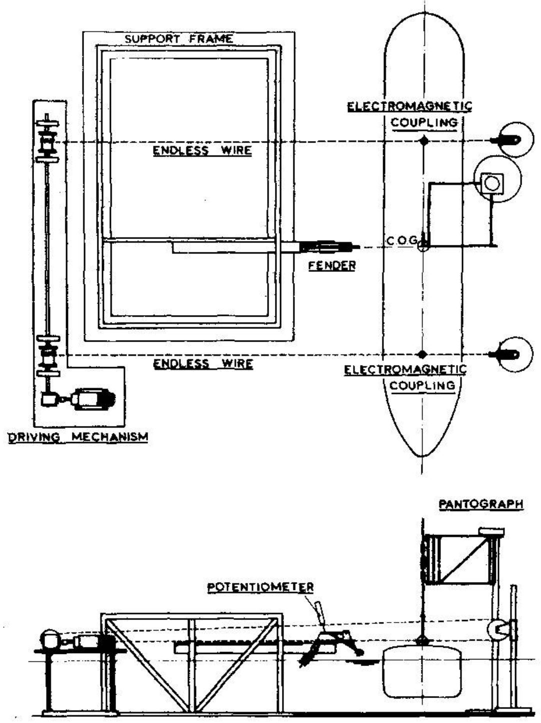

Fig. 1 Test set-up

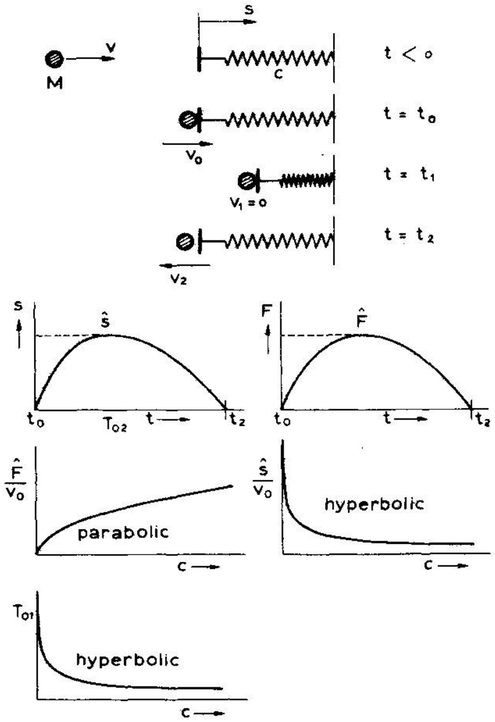

Fig. 2 Scheme and time histories for simple mass-spring system model for central impact

$$(M + M_{yy}).\ddot{y}(t) + \int_{-\infty}^{t} k_{yy}(t-\tau).\dot{y}(\tau).d\tau = F_y(t) \quad \ldots\ldots\ldots\ldots\ldots (11)$$

This one degree of freedom model can be extended to more degrees of freedom as given by Cummins [5], Van Oortmerssen [15] and Fontijn [9]. This time domain model has been employed in the present study to check its applicability to the results of the model experiments and it is shown in the sequel that the results from experiments and computations agree fairly well, see Section 5.3.

5. DISCUSSION OF THE RESULTS

5.1 Some notes on the added mass coefficients

For the centric impact case there are basically two ways to obtain the added mass coefficient, namely through the use of the momentum equation or the energy equation. It has been observed that the C_{My} coefficients derived from both techniques do not correspond. As shown in the following expressions the C_{My} coefficients cannot be equal.

The loss of kinetic energy in the fluid can be written as:

$$\tfrac{1}{2}\{\rho \iiint \left(\frac{V_1(x,y,z)}{\dot{y}_0}\right)^2 dx\,dy\,dz - \rho \iiint \left(\frac{V_0(x,y,z)}{\dot{y}_0}\right)^2 dx\,dy\,dz\}\dot{y}_0^{\,2} \quad \ldots\ldots (12)$$

Likewise the loss of fluid momentum in y-direction can be written as:

$$\{\rho \iiint \left(\frac{V_{y1}(x,y,z)}{\dot{y}_0}\right) dx\,dy\,dz - \rho \iiint \left(\frac{V_{y0}(x,y,z)}{\dot{y}_0}\right) dx\,dy\,dz\}\dot{y}_0 \quad \ldots\ldots (13)$$

The terms within brackets represent a kind of lumped added mass, associated either with energy or with momentum; and it is clear that the two expressions are not the same.

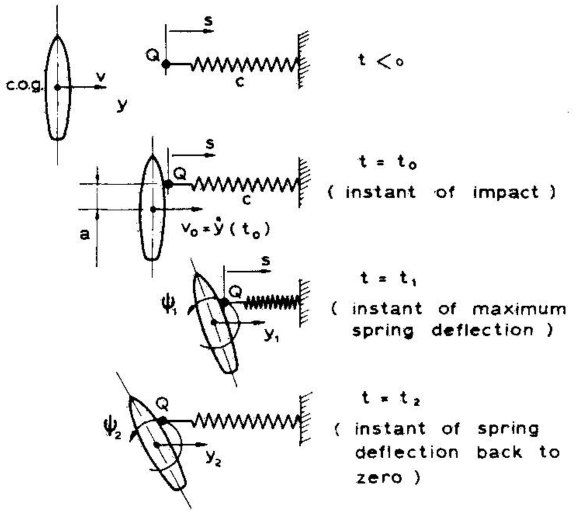

Fig. 3 Scheme for simple mass-spring system model for eccentric impact

FENDER CONTACT AMIDSHIPS

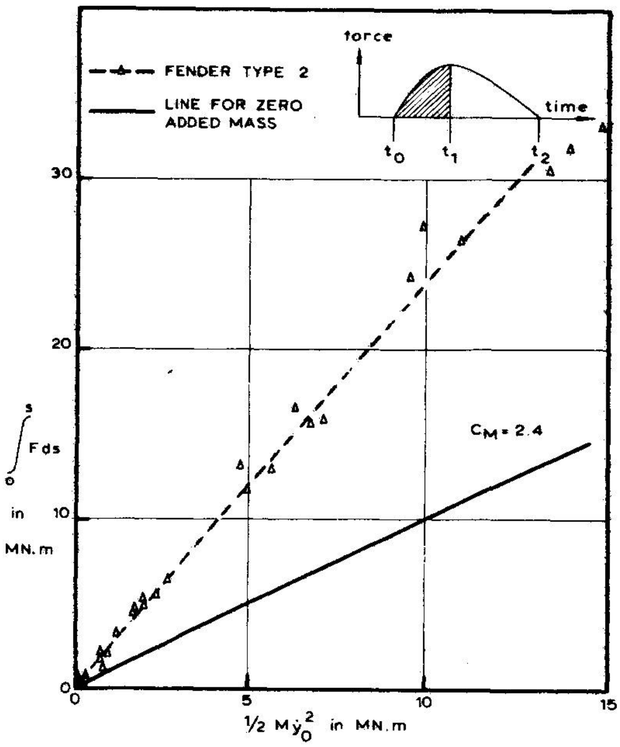

Fig. 4 Energy balance during compression phase

FENDER CONTACT AMIDSHIPS

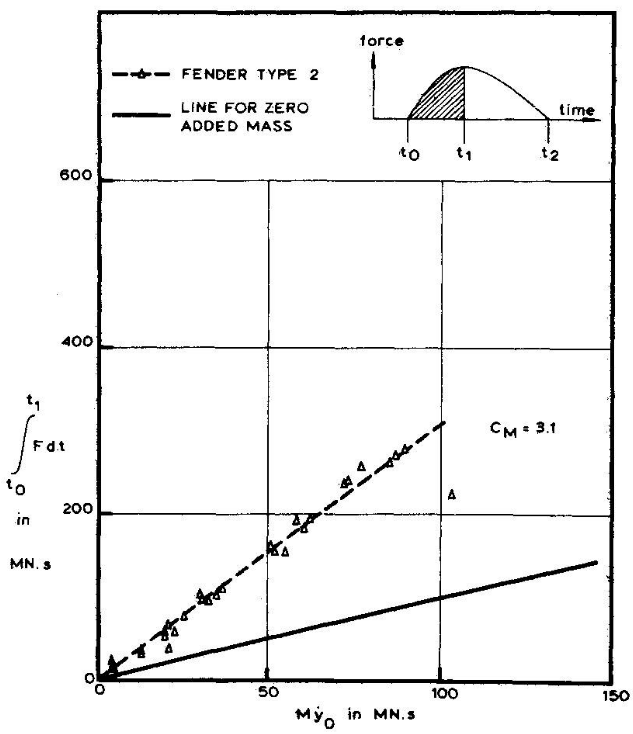

Fig. 5 Impulse versus change of momentum during compression phase

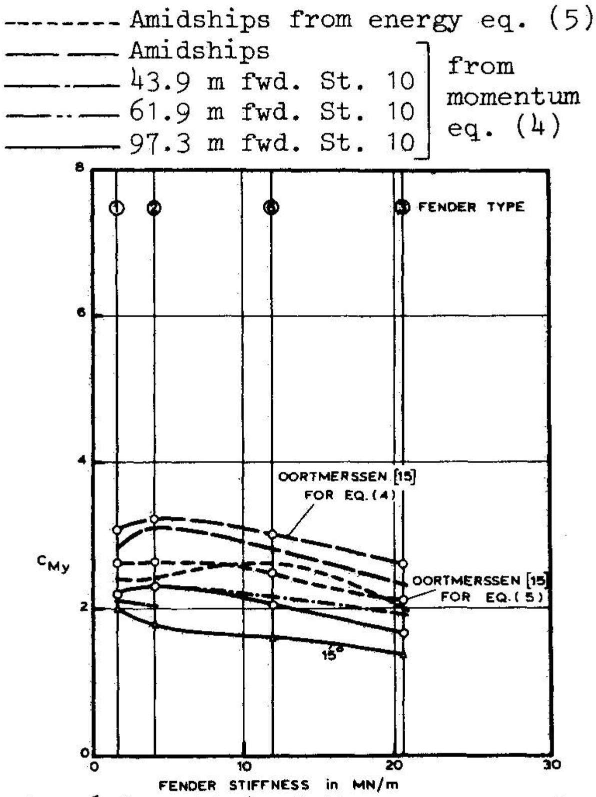

Fig. 6 C_{My} obtained from energy and momentum balance (eq. (5) and (4)) over compression phase

43.9 m forward of St. 10
61.9 m forward of St. 10
97.3 m forward of St. 10

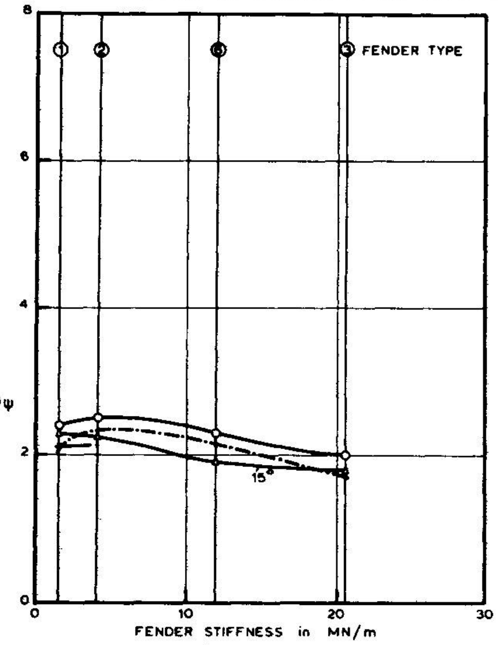

Fig. 7 $c_{M\psi}$ obtained from momentum balance (eq. (6)) over compression phase

Amidships
43.9 m forward of St. 10
61.9 m forward of St. 10
97.3 m forward of St. 10

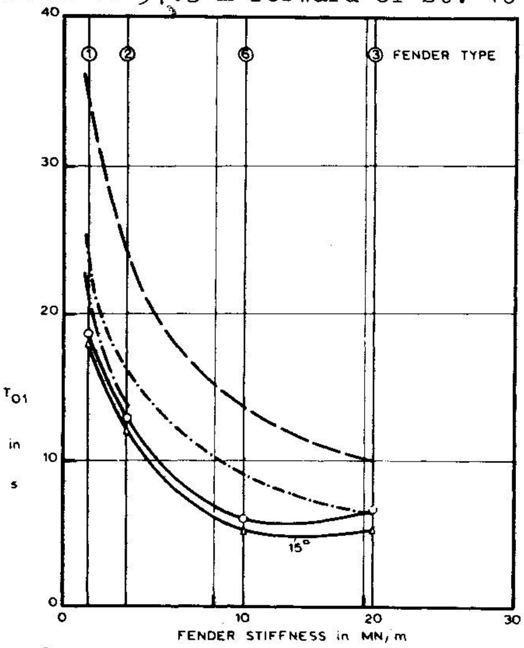

Fig. 8 Duration of ship-fender contact in compression phase

Amidships
43.9 m forward of St. 10
61.9 m forward of St. 10
97.3 m forward of St. 10

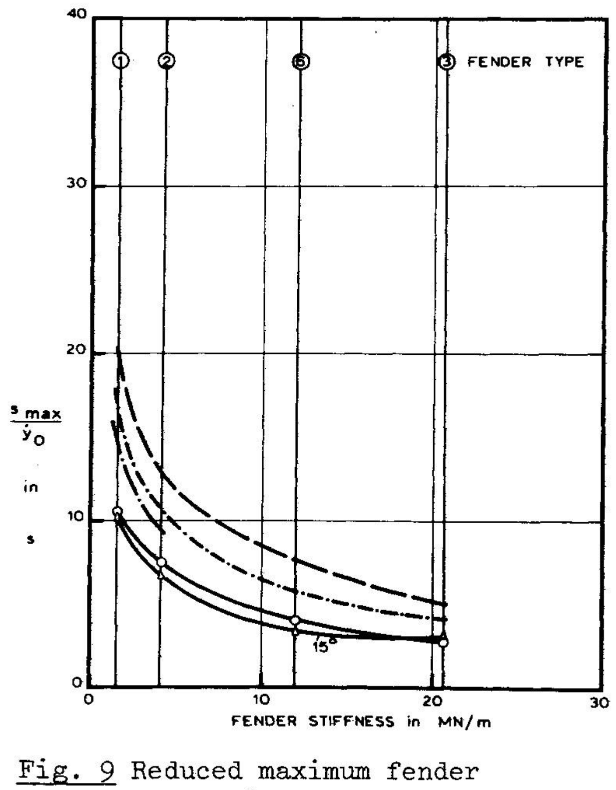

Fig. 9 Reduced maximum fender deflection

Amidships
43.9 m forward of St. 10
61.9 m forward of St. 10
97.3 m forward of St. 10

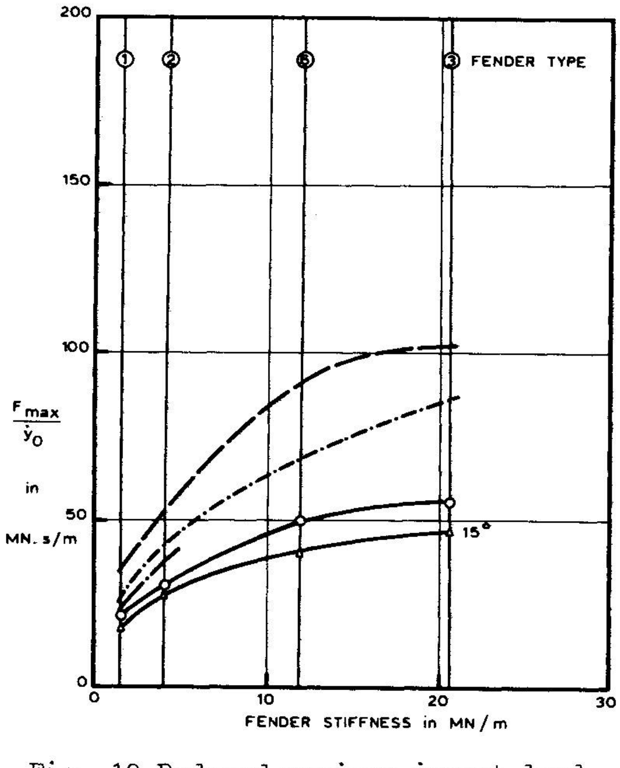

Fig. 10 Reduced maximum impact load

5.2 Discussion of the measured data

As outlined in Section 3.3, the derivations of the added mass coefficient can be done using the energy equation or the momentum equation. The results shown in Fig. 4 and 5 illustrate this point. The ensuing diagrams, Fig. 5 to 10 give in very concise form the total results of the whole test program, and they lead to the following observations. It is shown in Fig. 6 and 7 that as the fender point of contact moves further forward the added mass coefficients decrease. Theoretically there is no ground for this, but we can imagine that as the ship takes on an appreciable yaw angle the flow of water will be such that the sway and yaw added mass changes. Both the sway and yaw added mass coefficients are a function of the fender stiffness. In Fig. 8, 9 and 10 one can see that the shape of the curves corresponds fairly well to relations derived in Section 4.1.

5.3 Discussion on the comparison between test results and computations

The simple mathematical fitting model described in Section 4.1 agrees generally fairly well with the data, and is very instructive to gather some insight in the total impact event. However, this fit cannot predict the added mass value, so therefore a different model should be employed, as described in Section 4.2. That model gives a true time domain description of the total event, for instance the fender force as a function of time, which can then be analyzed in the same fashion as if it were a measurement result. This was done with regard to the sway added mass coefficient as shown in Fig. 6. The calculated added mass coefficient is indicated and it is clear that a close correspondence to the measured data exists.

It will be clear that a mathematical model of this kind can be used to advantage, for instance in the design stage of a jetty when due to the prevailing conditions there is little empirical data or experience to rely on.

6. EVALUATION

The results of the experiments presented herein and in [1] show the following:

- The added mass coefficients for sway and yaw are dependent upon the spring rate of the fender and on the characteristic of the fender's force-deflection curve, as is shown in [1].
- The maximum deflection of the fender decreases for increasing fender stiffness and for increasing eccentricity of the point of contact. The same applies to the characteristic time periods associated with the impact.
- The maximum force felt by the fender increases for increasing fender stiffness and decreases for greater eccentricity of the point of contact.
- The trend of these results can quite well be predicted by a simple mathematical model. However, the added mass coefficient can only be predicted by a sophisticated model in which due account is given to the fluid memory effect.

7. RECOMMENDED DESIGN PROCEDURE

It was stated in Section 3 that following the current design practice an added mass coefficient is to be selected applicable to the case. Usually, however, the designer takes a coefficient which has little if nothing to do with the subject case, and any dependency of the added mass coefficient on vessel size and shape, underkeel clearance, fender stiffness and characteristic and location of the point of contact on the ship's hull are eithr not rightly understood or considered of only marginal importance.

However, the present paper shows that these effects should not be left out of the design considerations. A simple mathematical model like the mass-spring system detailed in Section 4.1 can go a good way to explain the trends, but it takes a sophisticated mathematical model involving all the fluid memory effects in order to accurately predict the added mass and the time history of the impact. In the design of jetties and berths for large ships, under conditions for which one has little recourse to existing data, such computations will yield the right data to be used in the design.

NOMENCLATURE

a	$= \overline{QG}$ = distance between C.O.G. and point of contact	s	= fender stroke
$a_{yy}(\omega)$	= added mass in sway	$\hat{s}$	= maximum fender deflection
$b_{yy}(\omega)$	= damping in sway	s_0	$= s(t_0)$
C	= spring rate	s_1	$= s(t_1)$
C_E	= coefficient of eccentricity	s_2	$= s(t_2)$
C_M	= coefficient of added mass	t	= time
C_{My}	= added mass coefficient in sway	t_a, t_b	= time instants
$C_{M\psi}$	= added mass coefficient in yaw	t_0	= time instant when the ship hits the fender
C_S	= coefficient of deformation	t_1	= time instant when fender deflection is maximum
C_{yy}	= spring rate in sway	t_2	= time instant when contact between ship and fender is lost
dx dy dz	= infinitesimal fluid element	T_{01}	$= t_1 - t_0$
$\hat{F}$	= maximum fender force	V	= speed
$F(s)$	= fender force as a function of deflection	V_0	$= V(t_0)$
$F(t)$	= fender force as a function of time	$\overline{V}(x,y.z)$	= local fluid velocity vector
$F_y(\omega)$	= external exciting force	$V_y(x,y,z)$	= local fluid velocity vector y-component
$F_y(t)$	= external driving force	$V_0(x,y,z)$	$= V(x,y,z)$ at $t = t_0$
G	= index to centre of gravity C.O.G.	$V_1(x,y,z)$	$= V(x,y,z)$ at $t = t_1$
I_G	= mass moment of inertia in yaw relative to C.O.G.	$V_{y0}(x,y,z)$	$= V_y(x,y,z)$ at $t = t_0$
$k_{yy}(t)$	= retardation function in sway	$V_{y1}(x,y,z)$	$= V_y(x,y,z)$ at $t = t_1$
k	= radius of gyration in yaw	W	= work
M	= ship's mass	y	= sway motion
M_a	= added mass	α	= angle of incidence between ship and fender
M_v	= virtual mass = $= M + M_a = C_{My}.M$	ρ	= fluid density
M_{yy}	= added mass in sway	ψ	= yaw motion
		ω	= circular frequency
$\overline{QG}$	= a = distance between C.O.G. and point of contact	τ	= dummy time variable

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