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Energy Absorption in Ship-Platform Impacts

Absorption de l'énergie dans les chocs navire – plate-forme

Energieabsorption bei Zusammenstößen zwischen Schiffen und Plattformen

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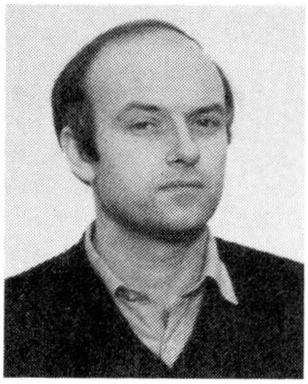

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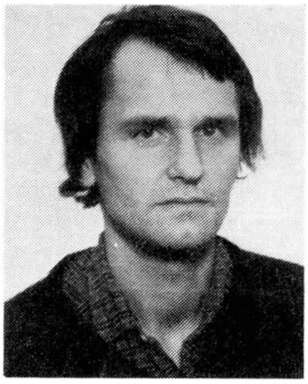

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SUMMARY

The paper deals with theoretical and experimental studies on ship-platform collision. General impact mechanics is presented together with a discussion on present design methods. A series of tests on models of platform bracing and bulbous bows is described with emphasis on load-deformation behaviour and the influence from impact velocity on the energy absorption capability.

RÉSUMÉ

L'article aborde certaines études théoriques et expérimentales portant sur les collisions navire – plate-forme. La mécanique générale des chocs est décrite et des commentaires sont faits sur les méthodes actuellement employées pour le projet. Une série d'essais a été pratiquée sur des modèles d'armature de plate-forme et de bouées en forme de bulbe. Une attention particulière est accordée au comportement de déformation en charge et à l'influence exercée par la vitesse du choc sur les capacités de l'absorption d'énergie.

ZUSAMMENFASSUNG

Der Artikel behandelt theoretische und experimentelle Untersuchungen von Zusammenstößen zwischen Schiffen und Plattformen. Allgemeine Aufprallmechanismen werden gemeinsam mit einer Erörterung über gegenwärtige Konstruktionsverfahren dargestellt. Eine Reihe von Tests mit Modellen von Plattformverstrebungen und Wulstbugen wird beschrieben, wobei der Schwerpunkt auf dem Lastverformungsverhalten und dem Einfluß der Aufprallgeschwindigkeit auf die Energieabsorptionseigenschaften liegt.

1. INTRODUCTION

Ship collisions with platforms have been identified as one of the possibly major hazards in connection with offshore oil activity. Statistics for world wide operation platforms during 1/1-70 to 31/12-80 indicate 9 cases with total or severe damage of the platform among 114 incidents with similar result [1]. At the same time many impact situations with minor consequences have been reported. As the number and size of vessels used in offshore operations (especially in the North Sea) increase, collision risk should be seriously considered in the design of platforms.

The event collision is characterized by the probability of occurrence and the inherent consequences. The two factors must be related to each other in the sense that major collisions endangering human lives, structures and environment must have low probability of occurrence while minor impacts occurring frequently must have small consequences. The probability as well as the consequences of collisions are affected by several factors such as traffic monitoring, navigational aids, operational limits, size of vessels and platform design and fendering. Consequently, a design procedure including a full evaluation of probability and consequences would be complex and not very feasible for practical use. Instead, socalled design accidents are evaluated based on judgement of probability of occurrence and consequences.

2. PRESENT DESIGN METHODS

The most frequent impacts against offshore platforms come from authorized vessels operating close to the platform. The consequencies of such impacts are normally small like local deformation of tube wall in bracing elements. However, with increasing size of supply vessels collision from such ships evidently defines a design limit state for the platform. The DnV rules for mobile offshore units [2] specify a ship of 5000 tons displacement with impact speed 2 $^{m}/sec$ as a design limit state.

A new design philosophy is related to impact analysis in the sense that structural capacity is given as energy absorbing capability rather than as ultimate load. The DnV rules [2] specify 14 MJ (Mega Joule) as impact energy for sideway collision (40 percent added mass included) and 11 MJ for bow or stern collision (10 percent added mass included).

The present methods for design of offshore platforms against collision are conservative in the sense that the striking ship is normally considered as undeformable so that the platform is designed to absorb all impact energy.

In order to get a representative model of a ship/platform collision, deformation and energy absorbing characteristics of the two colliding bodies must be known. Pioneering work on the energy absorbing capability of ships has been carried out by Minorsky [3] relating the amount of energy absorbed to the volume of damaged material. Most of the research on ships has been directed towards the protection of the reactor in nuclear powered ships [4] and few attempts have been made to develop general analytical models for the deformation process.

3. IMPACT MECHANICS

The derivation of a mathematical model of ship/platform impact is based upon two criteria:

a. Conservation of momentum
b. Conservation of energy

Assuming that the impact duration is short compared with the natural periods of motion for platform and ship the subsequent energy expression emerges

$$E_s + E_p = \frac{1}{2} m_1 v_1{}^2 \frac{(1 - {}^{v}2/v_1)^2}{1 + \frac{m_1}{m_2}} \tag{1}$$

where

m_1 = mass of striking ship including added mass (40 percent of vessel displacement for sideway collision and 10 percent for bow or stern collision [2])
m_2 = mass of semisubmersible platform including added mass
v_1 = velocity of striking ship immediately before collision
v_2 = velocity of semisubmersible platform immediately before collision
E_s = energy absorbed by ship
E_p = energy absorbed by platform

From Eq. (1) it is seen that in case ship and platform have opposite directions of velocity prior to impact the amount of plastic energy to absorb may exceed the kinetic energy of the ship.

For collision against a fixed jacket type of structure the corresponding energy expressions are obtained by introducing $m_2 = \infty$, $v_2 = 0$ in Eq. (1).

In the lack of reliable data for energy absorption in ships E_s is usually neglected, leading to a conservative design of the platform structure.

4. LOAD-DEFORMATION CHARACTERISTICS OF PLATFORM BRACING

For moderate impacts the energy absorption associated with global deformation of the platform is of minor importance and is usually neglected. The main contribution comes from deformation of the stroken bracing element either in the form of local deformation of tube wall for high D/t ratios, beam deformation or a combination of both modes.

4.1 Theoretical Models

4.1.1 Local Deformation of Tube Wall

It is not possible to present one single analytical model for local energy absorption. Several types of models must be considered related to various impact situations. A head-on collision gives a more concentrated force than a sideway impact (Fig. 1) and results in a larger amount of local energy absorption.

A simple yield line model for the case of sideway impact has been presented by Furnes and Amdahl [5]. The theoretical model gives good agreement with test results for moderate indentations.

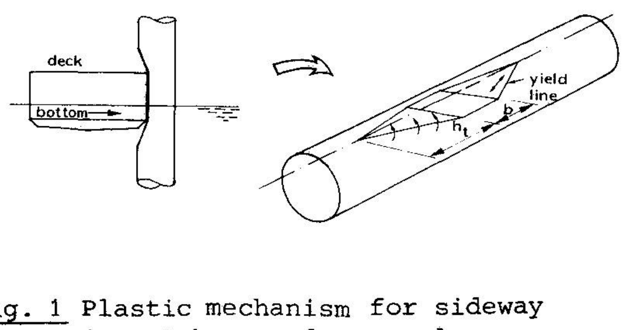

Fig. 1 Plastic mechanism for sideway impact by supply vessel

4.1.2 Analytical Model for Beam Deformation

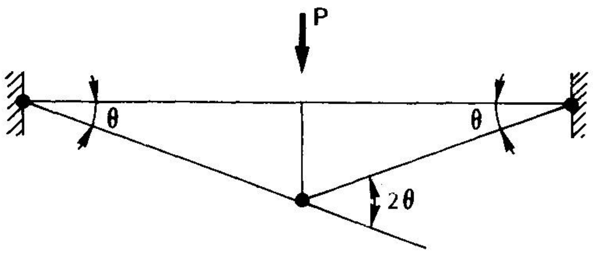

Fig. 2 Collapse mechanism for bracing element

Fig. 2 shows a collapse model of a bracing element where the ends are assumed restrained against axial movement and rotation. In this case the load carrying capacity increases considerable as the beam undergoes finite deflections due to the development of membrane tension forces. For a centrally loaded tubular beam the load-deflection expression reads [6]

$$\frac{P}{P_0} = \sqrt{1 - \left(\frac{w}{D}\right)^2} + \frac{w}{D} \text{ arc sin } \frac{w}{D} \; ; \; \frac{w}{D} \leq 1 \tag{2}$$

$$\frac{P}{P_0} = \frac{\pi}{2}\frac{w}{D} \; ; \; \frac{w}{D} > 1 \tag{3}$$

where w is the central deflection at the point of impact and D is the tube diameter. P_0 is the plastic collapse load of a circular tube in pure bending:

$$P_0 = \frac{8M_p}{\ell} = \frac{8\sigma_Y D^2 t}{\ell} \tag{4}$$

In a real frame structure the bracing elements sustain a certain degree of elastic support from the joints. Such elastic restrictions can be included in the above model [6], the major problem being to obtain realistic estimates on tubular joint flexibility.

Restrictions must be set on the maximum $^D/t$-ratio for which this rigid-plastic theory can be used so that the full plastic capacity is retained during deformation. Sherman [7] predicts from tests $^D/t < 35$ while the API rules [8] prescribe $^D/t < 9000/\sigma_Y$ (σ_Y is yield stress in $^N/mm^2$).

4.1.3 Tubular Joint Capacity

A major requirement for the yield hinge model in Fig. 2 to be valid is that the tubular joints can sustain bending moment and membrane force at the beam ends. Thus, failure criteria for tubular joints must be checked against plastic capacities of the bracing elements in order to ensure full energy absorption capability. Valuable information on capacity of unstiffened tubular joints has been presented by Yura et al. [9] and in design codes [8,10].

4.2 Tests on Bracing Elements

For jackets and semisubmersible platforms characteristic dimensions of bracing elements in water plane are:

$1.0 < D < 2.0$ m
$20 < {}^D/t < 100$
$10 < {}^L/D < 30$

A series of model tests on energy absorption in bracing elements has been performed with the following range of variation:

$63 < D < 125$ mm
$22 < D/t < 61$
$10 < L/D < 20$
$204 < \sigma_y < 328$ N/mm^2

The effect of membrane forces on the energy absorption capability of bracing elements is illustrated by Fig. 3 in which load-displacement curves are shown for static testing of two similar models with different end conditions. The lower curve relates to horizontally free end conditions and the upper curve relates to horizontally fixed ends. In both cases the ends are clamped against rotation.

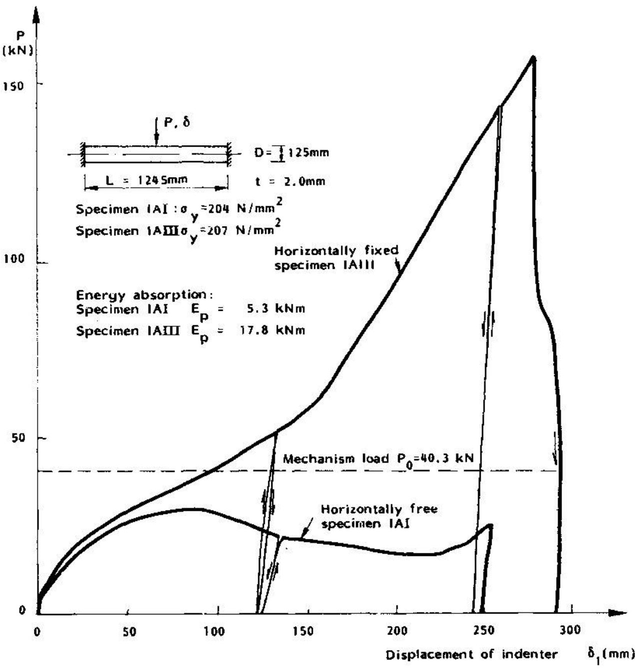

Fig. 3 Load-displacement curves for indenter.
Horizontally free and horizontally fixed ends

The energy absorbed is 5.3 kNm and 17.8 kNm, respectively. The two models showed two different collapse modes. For the horizontally free case local wall crippling occurred on the compression side of the end, while the membrane tension caused a rupture type of failure for the horizontally fixed specimen.

The effect of dynamic loading on the load-displacement curve is demonstrated by Fig. 4. Solid line represents static load and dotted line dynamic load corresponding to a real velocity of 2.0 m per second. It is seen that the energy absorption capability is raised by approximately 10 percent. Inertia forces are negligible for the actual range of velocity and the main increase in stiffness comes from the strain-rate effects.

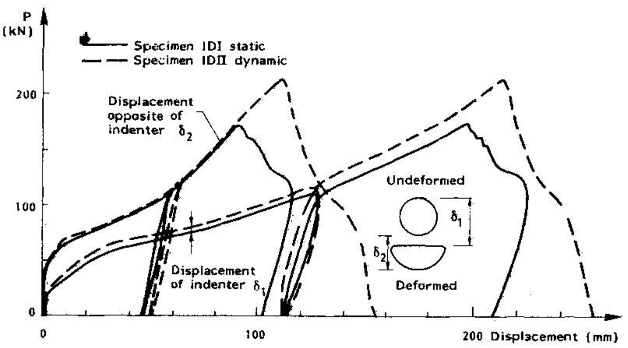

Fig. 4 Static and dynamic load-displacement curves

5. ENERGY ABSORPTION IN BULBOUS BOWS

The structural configuration of a ship bow is complex, being comprised of an outer shell stiffened by a grid of frames and stringers. This complexity makes it difficult to apply conventional collapse models for panels for estimating energy absorption capability.

5.1 Plastic Collapse Models for Bulbous Bows

Bulbous bows bear some resemblance to cylindrical shells. The simplest configuration as far as structural modelling concerns is a bulb in the form of a ringstiffened circular cylinder. Plastic collapse models for this case have been derived by Alexander [11] and by Johnson et al. [12].

After initial buckling [13] the ringstiffened cylinder continues to deform into the plastic region. Depending on the shell geometry the post-buckling behaviour may take on two forms, either with the shell in axisymmetric convolutions or with asymmetric folds.

For axisymmetric collapse the average load during plastic deformation is found to be [11]

$$P_{av} = \frac{\pi \sigma_y t^2}{\sqrt{3}} \left(\frac{\pi}{2}\frac{D}{h} + 1 + \sqrt{3}\frac{h}{t}\right) \tag{5}$$

where

σ_y = yield stress
t = shell thickness
D = shell diameter
h = stiffener spacing

Static considerations of impact may underestimate the energy absorption capability. A simple method of including strain rate effects, has been suggested by Cowper and Symonds [14]

$$\frac{\sigma_y'}{\sigma_y} = 1 + \left(\frac{\dot{\varepsilon}}{\dot{\varepsilon}_0}\right)^{\frac{1}{m}} \tag{6}$$

where $\dot{\varepsilon}_0$ and m are material constants determined from experiments. Recommended values for steel are m = 5, $\dot{\varepsilon}_0$ = 40 sec^{-1}.

The average strain rate in the cylinder wall may be approximated by [15, 16]

$$\dot{\varepsilon} = \frac{\dot{u}}{4h} \tag{7}$$

5.2 Tests on Bulbous Bows

A series of collision tests on bulbous bow models has been carried out. The first ten models comprised ringstiffened cylindrical shells. All models have diameter 400 mm and shell thickness and stiffener spacing are given in Table 1. The specimens MA1-MA4 are machined cylinders that are almost ideal in the sense that residual stresses and shape imperfections are small. FA1-FA6 are fabricated models rolled from 2 mm plate, and then closed by a longitudinal butt weld.

Table 1 gives comparison between average loads from experiments and according to the above mechanism models. The rate of displacement is $\dot{u}$ = 0.125 mm/sec. for all models and the theoretical solutions are based upon the dynamic yield stress according to Eqs. (6-7).

Test specimen	Wall thickness mm	Stiffener spacing mm	Static yield stress MPa	$P_{av,e}$ Exper. MN	$P_{av,th}$ Theory MN	$\frac{P_{av,th}}{P_{av,e}}$
MA1	0.97	23.0	267	68.1	63.6	0.93
MA2	1.22	37.0	267	103.2	75.3	0.73
MA3	0.99	33.5	267	80.5	55.5	0.69
MA4	0.98	33.5	267	62.5	55.6	0.89
FA1	2.03	47.0	236	158.2	126.6	0.80
FA2	2.06	57.0	236	147.8	122.7	0.83
FA3	2.06	67.0	236	143.6	120.6	0.84
FA4	2.04	97.0	228	139.9	125.9	0.90
FA5	2.05	117.0	228	147.4	141.5	0.96
FA6	2.05	-	228	101.4	82.1	0.81

Table 1 Experimental and theoretical average loads for bulb models

Table 1 indicates some discrepancy between analytic predictions and test results. For all models the mechanism calculations underestimate the average load. Several factors can explain this discrepancy. The most important effects are:

a. Inaccurate representation of deformation mode in mechanism models
b. Movable plastic hinges are not included in theoretical models
c. Inaccurate representation of material data

The load-displacement curve for specimen MA4 is shown in Fig. 5. The post-buckling behaviour is explained by the successive formation of plastic mechanisms between ring stiffeners.

The above ringstiffened cylindrical shells represent idealized models of bulbous bows. In addition to transverse stiffening the bulbs normally also contain longitudinal stiffening system consisting of stringers and centerline bulkhead. Further, the cross sections of bulbs are more elliptic in shape.

Test on a more realistic model is illustrated in Figs. 6-8.

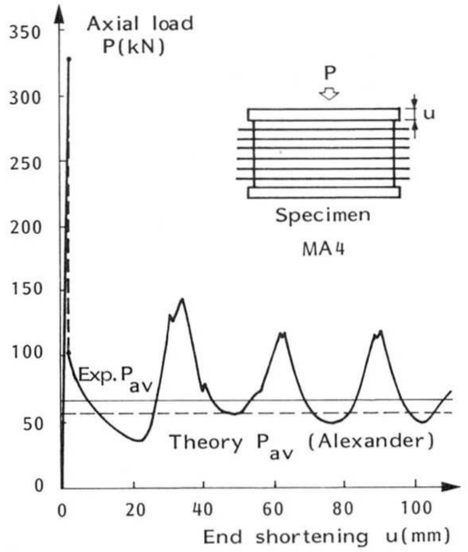

Fig. 5 Load-displacement curve for ringstiffened cylinder MA4

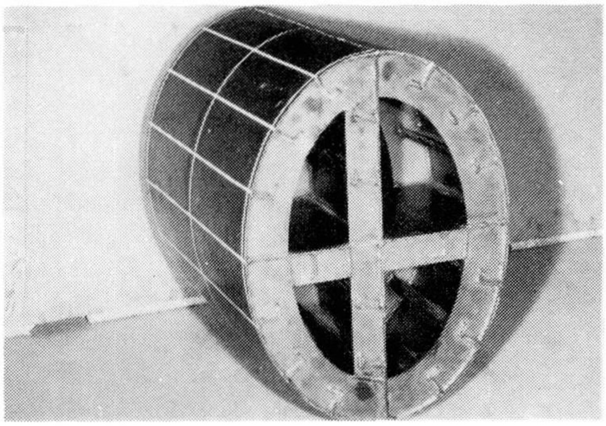

Fig. 6 Bulb model with combined stiffening

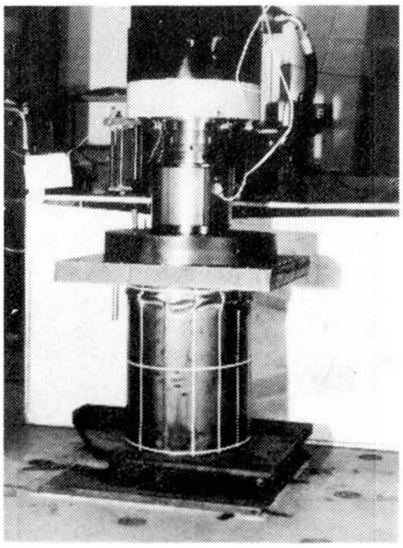

Fig. 7 Initial buckling of model in Fig. 6

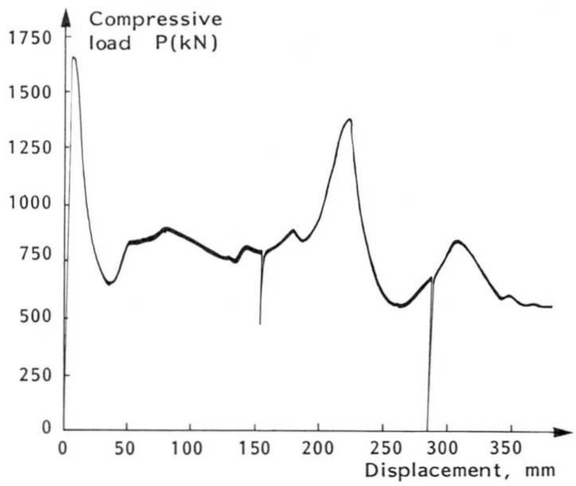

Fig. 8 Load-displacement curve for model in Fig. 6

6. CONCLUSIONS

The present work clearly demonstrates the difficulties in developing theoretical models for estimating energy absorption in ship/platform collisions. For representing collapse behaviour of platform bracing a simple beam mechanism model can be applied, combined with a yield line model for local wall indentation. However, the variation in structural configuration of ship bows makes it difficult to come up with a general design formula for energy absorption. The paper proves the need for combined experimental and theoretical work within this field.

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