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Ship collision with offshore structures

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ABSTRACT: Static versus dynamic analysis of ship collision with offshore structures is discussed. Modelling of collision within the nonlinear space frame program USFOS is described. The significance of including dynamic effects is evaluated through numerical studies of collision with a jack-up and a jacket platform.

1 INTRODUCTION

Offshore platforms are exposed to many risks and one of the more severe is ship collision. The largest damage potential is associated with collision with large merchant vessels. Their kinetic energy is, however, so large that it is virtually impossible to design steel platforms such as jackets and jack-ups for this event. The risk should instead be controlled by keeping the probability of occurrence acceptably low. Encounters with attendant vessels, on the other and, have a rather high probability of occurrence, approximately 0.15 per platform year (Wicks et. al. 1992). To this end there has been no catastrophic failures, but rather severe accidents have taken place. In 1975 a jacket on the Auk field lost three braces and had a fourth one severely bent by a supply vessel impact. In 1988 the Oseberg B platform was hit by a German submarine 25 m below sea surface. A diagonal was dented and distorted and had to be repaired at substantial costs (Thuestad &Nielsen 1990). The impact energy was estimated to be in the range of 5 MJ.

The concern for ship collision is reflected in various design codes. Since 1980 the Norwegian Petroleum Directorate (NPD) requires that platforms normally be designed for impacts from supply vessels of 5000 tons displacement with a speed of 2 m/s yielding a kinetic energy of 14 MJ for beam impact and 11 MJ for bow or stern impact, taking into account specified values for hydrodynamic added mass (NPD 1984). The design is carried out in the limit state of progressive collapse (PLS), i.e. local failures in the form of denting, plasticity, buckling etc. are allowed but the total integrity should not be put in jeopardy. Furthermore, in damaged condition the platform shall be able to resist the design environmental forces, however, with all partial safety factors equal to unity.

Until quite recently the platforms in the British Sector were normally designed for an impact energy in the range of 0.5 MJ. With the 4th edition of the guidance notes issued by the Health and Safety Executive (HSE 1990) the recommended level of energy has become the same as the level of energy considered in the Norwegian Sector. The platform's contribution to energy dissipation should be minimum 4 MJ. This differs from the NPD requirement, where the share of energy depends upon the relative stiffness of the ship and platform. For conventional jackets, however, the energy dissipated will normally be larger than 4 MJ.

The present industry standard in Norway for designing against ship impacts is summarized in the design guidance manual for offshore steel structures exposed to accidental loads (Veritec 1988), developed jointly by Veritec and Sintef. According to this the following mechanisms contribute to the dissipation of

energy
- local denting of hit cross-section
- beam deformation of hit member
- frame deformation of platform
- deformation of ship

In all modes of deformation the response will be dominated by plastic straining, apart from frame deformation of the platform where the elastic straining can be significant. The relative contribution from each of the modes depends upon the impact scenario considered; ship deformation is normally neglected for bow and stern impact, whereas the energy dissipated in local denting is normally small for brace impact.

In the past, the design has typically been based upon application of simplified hand calculation methods in combination with linear elastic frame programs. Presently, non-linear finite element methods are coming more and more into use. Special purpose codes for accidental load effect analysis of offshore structures are also being developed .

Most often a static approach is adopted. However, this may be questioned: In a static case the collision force is transferred from the point of contact to the ground through shear action, predominantly by axial compression and tension of the braces in the structure. In the dynamic case shear forces have to be transferred initially through the members above the contact point in order to accelerate and displace the topside structure. In the late stages of impact or after the contact has ceased, the momentum of the topside may induce some sort of "whip-lash" effect.

The purpose of this work is to study the effect of dynamics on the platform response in terms of energy dissipation and load effect.

2. NONLINEAR FINITE ELEMENT PROGRAM - USFOS.

The computer program used in the numerical studies is USFOS (Søreide et. al. 1991). It has been developed especially for progressive collapse analysis of offshore steel platforms and features among others analysis of extreme waves (static pushover analysis), elasto-plastic, cyclic analysis of extreme storms (Stewart et. al. 1993), integrated analysis of fires (Eberg et. al. 1992), explosion response analysis etc. Below, a brief description of the theoretical basis is given, with particular emphasis on the modelling of collision.

2.1 Fundamental concepts.

The basic idea behind USFOS is to represent each physical element with one finite element. This facilitates that the finite element mesh from conventional linear elastic analysis be used directly in the nonlinear progressive collapse analysis. This is enabled by means of two important concepts: First, the interpolation function used for the elastic displacements is the exact solution to the differential equation for a beam subjected to end forces. Secondly, nonlinear material is modelled by means of plastic hinges.

The elastic stiffness matrix, which contains the effect of large lateral deformations on element level (geometric effect), comes out to be the conventional stiffness matrix for a 3-D beam, multiplied with the so-called Livesly's stability functions. These are simple analytic functions of the axial force normalized with respect to the Euler buckling force. The stability functions contain all information needed to predict buckling for the elastic element.

Incremental relationships for the step-iterative procedure are obtained by taking the second variation of the strain energy. Again, closed-form solutions can be obtained for all nonlinear stiffness terms, so that no numerical integration is required.

The yield criterion represents full plastification of the cross-section and is expressed as

$$F = \left(\frac{N}{N_p}, \frac{Q_y}{Q_{yp}}, \frac{Q_z}{Q_{zp}}, \frac{M_x}{M_{xp}}, \frac{M_y}{M_{yp}}, \frac{M_z}{M_{zp}}\right) = 0$$

where N is axial force, Q_y and Q_z are shear forces in two orthogonal directions, M_x is torsional moment and M_y and M_z are the bending moments. For a tubular beam subjected to combined axial compression/tension and bending equation the criterion takes the simple form:

$$F = \cos(\frac{\pi}{2}\frac{N}{N_p}) - \frac{M}{M_p} = 0$$

The incremental, elasto-plastic stiffness matrix, $\mathbf{k}_i^{ep}$, is obtained by application of the consistency criterion and the normality criterion for the plastic displacement increment and can be written as:

$$k_i^{ep} = (k_i^{el} - k_i^{el}\phi^T(\phi^T k_i^{el}\phi)^{-1}\phi k_i^{el})$$

where $\mathbf{k}_i^{el}$ is the elastic incremental stiffness matrix and ϕ is a vector of containing the partial derivatives of F with respect to the cross-sectional force components.

2.2 Interaction with local denting

Depending upon the slenderness of the hit tube and the area of contact local denting may take place. The effect of the dent is two-fold: First, energy is dissipated in the denting process. Second, the dent reduces the effective bending capacity of the section and causes an additional bending moment from the axial force through the eccentricity created in the damaged section.

Resistance curves for tubes subjected to denting are given in Figure 1 (Veritec 1988), based upon plastic analysis of the yield line model indicated. The curves are developed on exact non-dimensional form and show good agreement with experimental data, except for large deformations where the resistance is overestimated.

Wierzbicki & Suh (1988) considered the effect of axial force on the denting resistance as well, and obtained the following relationship

$$P / (\sigma_y \frac{t^2}{4}) = 16\sqrt{\frac{2\pi}{3}\frac{D}{t}\frac{\delta}{D}(1-\frac{1}{4}[1-\frac{N}{N_p}]^3)}$$

where σ_y is yield stress, δ is dent depth, D and t is tube diameter and thickness, respectively.

For small axial force the equation shows somewhat poorer agreement with test data. Hence, the prediction of the resistance is based upon parameterization of the curves in Figure 1, modified for the effect of axial force according to the equation above. The resulting expression reads:

$$P / (\sigma_y \frac{t^2}{4}\sqrt{\frac{D}{t}}) = (22+1.2\frac{B}{D})\,(\frac{\delta}{D})^{\frac{1.925}{3.5+\frac{B}{d}}} \cdot \sqrt{\frac{4}{3}(1-\frac{1}{4}[1-\frac{N}{N_p}]^3)}$$

The dent causes a reduction of the capacity in bending and axial compression. It is assumed that the cross-section consists of a flattened part, which is only partly effective, and an undamaged part, which remains virtually unaffected.

The consistency criterion must now take into account the effect of the dent, yielding

$$\Delta F = \frac{\partial F}{\partial S}\Delta S + \frac{\partial F}{\partial \delta}(\frac{\partial \delta}{\partial P}\Delta P + \frac{\partial \delta}{\partial N}\Delta N)$$

The first term in the parenthesis is interpreted as a consistent load increment while the second term contributes to the gradient used to obtain the elasto-plastic stiffness matrix, i.e.

$$\frac{\partial F}{\partial N} \equiv \frac{\partial F}{\partial N} + \frac{\partial F}{\partial \delta}\frac{\partial \delta}{\partial N}$$

2.3 Numerical modelling.

The nonlinear equations are integrated in the time domain by means of the HHT-alpha algorithm (Hilber et. al. 1976). It is based upon the Newark-β scheme, but introduces numerical damping by means of time averaging. The equilibrium equation reads:

$$M\ddot{r}_{n+1} + (1+\alpha)C(\dot{r})\dot{r}_{n+1} - \alpha C(\dot{r})\dot{r}_n + (1+\alpha)K(r)r_{n+1} - \alpha K(r)r_n = (1+\alpha)Q_{n+1} - \alpha Q_n$$

where n and n+1 denote two consecutive time states. The effect of the α parameter is to damp out higher order frequencies.

The mass matrix, consisting of structure inertia and hydrodynamic added mass, is considered to be constant. Several physical phenomena contribute to damping, for example structural damping, hydrodynamic drag, wave generation etc. The maximum response will occur during impact or in the first period of free vibration when the effect of damping is small. Damping is therefor neglected.

The numerical integration can be performed implicitely in the conventional way or by means of the predictor-corrector scheme. The latter allows for automatic time step scaling in the predictor phase in order to bring the force point close to the yield surface once the occurrence of a yield hinge is detected. Due to the nonlinearity of the incremental equations the scaling is not exact. However, minor deviations from the yield surface are removed by equilibrium iterations.

2.4 Impact modelling

The ship is modelled as a mass point connected to the platform through a nonlinear spring. The spring can be given arbitrary properties. Here, the load-deformation characteristics shown in Figure 2 is used., being recommended for supply vessel beam impact (DnV 1981). A minor change is made in the sense that a finite stiffness is considered in the first phase of deformation rather than the instantaneous jump to 7 MN. During unloading the force state follows a curve parallel to the initial stiffness.

The mass representing the ship is given an initial velocity corresponding to the impact speed and the analysis is carried out as a free vibration problem. The ship force unload once the spring starts to elongate, i.e. the ship and the vessel go away from each other. When the contact force has vanished the ship mass is disconnected from the model.

3. CASE STUDIES

Collision response analysis are carried out for to platforms; a jack-up and a four-legged jacket. In both cases supply vessel beam impact against a leg chord is considered.

In the jack-up case the location of impact is assumed to be on a main joint. This is a "hard point" in the leg. Alternatively, a potential location of contact is halfway between two joints. This would yield a "softer" platform response. However, it should be taken into consideration that the ship deformation curve is developed for equal penetration over the entire ship side. The depth of the design vessel is 7 m and is identical to the vertical distance between two joints on the jack-up leg. Thus, it is natural to assume that the contact force be distributed to one joint, as in the present case, or to two joints.

In the case of stern impact contact may take place halfway between the joints, because the stern measures only 3.5 m. On the other hand, the maximum design force created by the stern is 16 MN. This is to be compared with the three hinge mechanism load for the leg chord, being in the range of 15-19 MN, depending on the direction of deformation. These values refer to a concentrated load. Taking into account the height of the stern it will be higher. Thus, it is concluded that interjoint stern impact probably is of less concern than beam impact

As to the jacket, the vertical distance between each horizontal frame is in the range of 20-25 m. Hence, inter-joint impact is more likely. In the present study, the contact force is (conservatively) applied as a concentrated force. In the calculation of local denting the width of the contact area is accounted for and is assumed to be twice the leg diameter, or 3 m.

3.1 Jack-up

Figure 3 shows the finite element model of a jack-up with three legs. It is situated at a water depth of 105 m. The location of impact is indicated by the mass and the spring representing the non-linear deformation characteristics of the ship. The ship is assumed to move in the negative global X-direction hitting the platform in its axis of symmetry. Due to the low diameter to thickness ratio of the chord and the strengthening effect of the racks, local indentation will be small and is therefor neglected.

For reference a static analysis is first carried out. The collision energy considered is 14 MJ. It is found that 5.4 MJ is dissipated by ship deformation and 8.6 MJ by platform deformation. The maximum collision force amounts to 17.4 MN yielding a maximum displacement at deck level of 0.9 m.

Forty-six per cent of the collision force, or 8 MN is transferred to the sea floor directly through the hit leg, the remainder, 9.4 MN, is transferred via the deck through the two other legs.

Dynamic analysis is carried out both for the conventional 2 m/s (14 MJ) impact as well as for an impact speed of 3 m/s corresponding to a kinetic energy of 31.5 MJ. Except for some limited yielding in a member connected to the contact node in the 3 m/s impact the structure behaves fully elastic during the whole response simulation. Figure 3 shows screen plots of the jack-up at maximum collision force and at maximum lateral displacement for the 2 m/s impact. The displacements are magnified by a factor of 10. It is easily observed that the two deformation modes are quite different.

Figure 4 shows the displacement history at deck level and the evolution of the contact force. The highest contact force is obtained in the 3 m/s impact, but the force level drops dramatically once rupture takes place in the ship side. In the 2 m/s impact rupture does not take place so that the maximum force is smaller, but the intensity is high for a longer period so that the total impulse to the platform for the two cases are relatively comparable.

The difference in the contact force histories for the two cases demonstrates the importance of the interaction with the platform response. As Figure 5 displays the nonlinear spring reproduces very well the ship deformation curve given in Figure 2.

Maximum displacement at deck level is 0.77 m for the 2 m/s impact and is almost twice the value at the end of the contact period. For the 3 m/s impact the maximum displacement is 0.89 m and occurs immediately after the end of the contact period. Both values are close to the 0.9 m obtained in the static case. However, the maximum collision force, which in the static case is 17.4 MN, now amounts to 20.9 MN for the 2 m/s impact and 27 MJ for the 3 m/s impact, the latter value being equal to the maximum possible collision force.

The deviation between the static and dynamic collision force is due to the large force needed to accelerate the deck. This is further evidenced in the axial forces in the cross-braces just above the contact point. In the static case the maximum force is 5.2 MN. In the dynamic case it becomes 9.2 MN and 10.0 MN for 2 m/s and 3 m/s, respectively. The increase is more than proportional with the increase in the maximum contact force, especially for the 2 m/s impact.

Also the share of the energy dissipation differs very much from the static case. The ship absorbs 7.2 MJ as strain energy and is given a rebound of 0.5 m/s corresponding to an energy of 0.9 MJ for the 2 m/s impact. The remainder, 5.9 MJ is absorbed by the platform. For 3 m/s there is no rebound; 23.6 MJ is absorbed by ship deformation and 7.9 MJ by the platform. It should be noted that the amount of rebound is dependent upon the steepness of the ship deformation curve during unloading, which is assumed parallel to the initial stiffness.

In order to get information of the significance of the ship deformation properties, an analysis is carried out where the ship strength is increased with 50%. For the 2 m/s impact this has the effect of reducing the collision duration from 1.2 seconds to 1.0 seconds. The maximum collision force and maximum brace force increase to 28 MN and 11 MN, respectively, but still only limited yielding take place. Thus, it is concluded that the platform response is not very sensitive to the uncertainty of the ship deformation characteristics.

3.2 Jacket

Figure 6 shows the finite element model of the jacket, which is located at 95 m water depth. The contact point is indicated by a nonlinear spring representing the ship deformation properties and the mass point.

In all analyses considerable yielding and plastic deformation take place in the hit beam and adjacent members as shown in Figure 6. Apart from that the platform behaviour is essentially elastic.

In the static case the maximum collision force attains 10.5 MN with a corresponding displacement of 0.88 m at the contact point first period of free vibration when the effect of damping is small. Damping is therefor neglected.

The numerical integration can be performed implicitely in the conventional way or by means of the predictor-corrector scheme. The latter allows for automatic time step scaling in the predictor phase in order to bring the force point close to the yield surface once the occurrence of a yield hinge is detected. Due to the nonlinearity of the incremental equations the scaling is not exact. However, minor deviations from the yield surface are removed by equilibrium iterations.

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In all analyses considerable yielding and plastic deformation take place in the hit beam and adjacent members as shown in Figure 6. Apart from that the platform behaviour is essentially elastic.

In the static case the maximum collision force attains 10.5 MN with a corresponding displacement of 0.88 m at the contact point and 0.06 m at deck level. The ship dissipates 1.1 MJ, the dent 2.7 MJ whereas 10.2 MJ is absorbed as elastic and plastic strain energy in the platform, predominantly by beam deformation of the hit member. The final dent depth in the member with diameter 1.5 m and thickness 38 mm, becomes 0.36 m.

Figures 7 and 8 depict results from dynamic simulations with and without local denting taken into account. It is seen that the dent softens the response; the maximum force is reduced and the duration is increased. In both cases the maximum displacement exceeds the value obtained in the static calculation and is in the range of 1 m. Whether this can be achieved without rupture due to excessive straining of the chord wall has not been examined.

The displacement at deck level is much smaller. Due to the permanent displacements induced by the collision the average final position of the deck is opposite of the impact direction.

The rebound of the ship is significant when denting is disregarded. The rebound velocity is 0.9 m/s corresponding to a kinetic energy of 2.8 MJ. The ship dissipates 1.7 MJ. With denting the rebound speed is only 0.4 m/s with a kinetic energy of 0.6 MJ. The dent dissipates 3.6 MJ and the ship 1.0 MJ.

During deformation the leg subjected to impact is allowed to unload most of its share of the topside weight. This load redistribution is particularly enabled through action of the diagonals supporting the deck structure. It is therefor very essential that they remain intact. In all cases the maximum force in the diagonals is found to lie considerably below the capacity, and the dynamic analyses do not yield higher values than the static analysis.

It is noteworthy that the results from the dynamic analyses do not differ very much from the static analysis. This is due to the fact that the collision force is limited the local mechanism developing in the hit member and that the duration of the impact is quite long relative to the natural period for the governing motion which being 1.4 sec.This means that global frame energy is small. This picture may change, however, if collision against a main node is considered.

4. CONCLUSIONS

The present study is very limited and one should be cautious in drawing general conclusion with respect to the effect of dynamics on ship-platform impacts. Three factors seem to be important:

- the local strength of the platform and the strength of the ship relative to the overall strength of the platform
- the duration of the collision relative to the fundamental period of the governing motion
- the strength of the members transmitting forces needed to accelerate the deck

Both platform studied survive the selected collision events. In particular, the jack-up behaves elastically for the design ship beam impact. Considerable dynamic magnification is found in the diagonals above the collision zone, but the load level is tolerable. The jack-up shows also little sensitivity to uncertainty in ship deformation characteristics and impact speed (energy).

The jacket response for the impact scenario considered can be reasonably well predicted by a static approach, because the impact duration is relatively long compared to the fundamental period of the governing motion and contact takes place at a "soft" point. If the platform is struck at a joint it is expected that dynamics have a larger effect. This event deserves further investigation.

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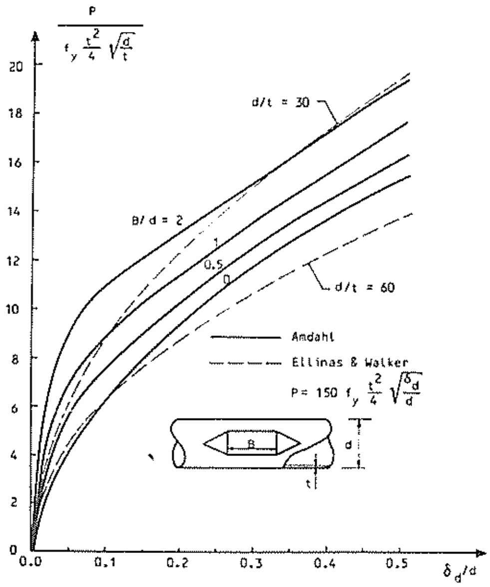


Figure 1. Load-local denting relationship for tube.

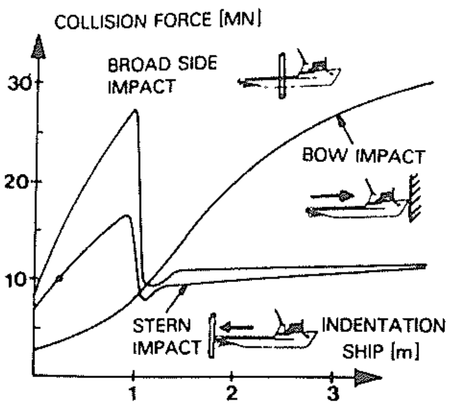
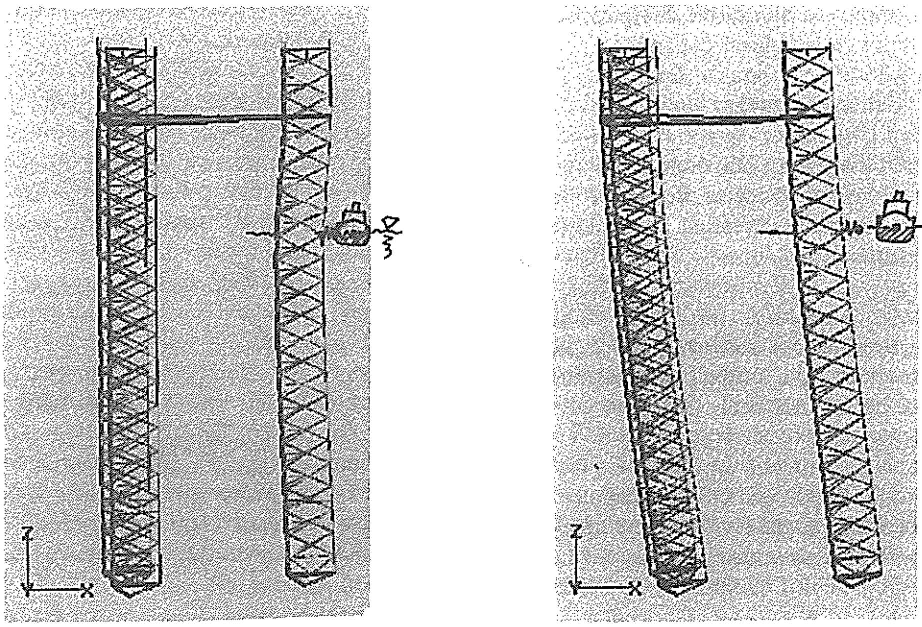


Figure 2. Load-indentation relationship for supply vessel.



(a) Maximum collision force (b) Maximum lateral displacement

Figure 3. Screen plot of deformed jack-up

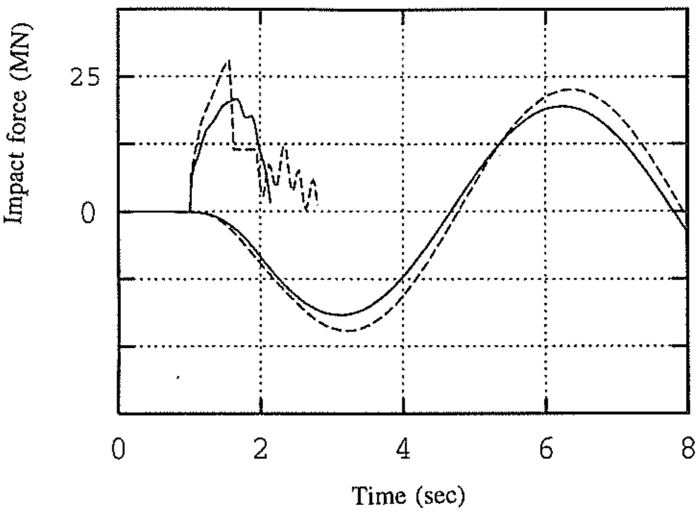


Figure 4. Contact force and deck displacement histories for jack-up

960

840 · 6.5

54 6.2

4.6

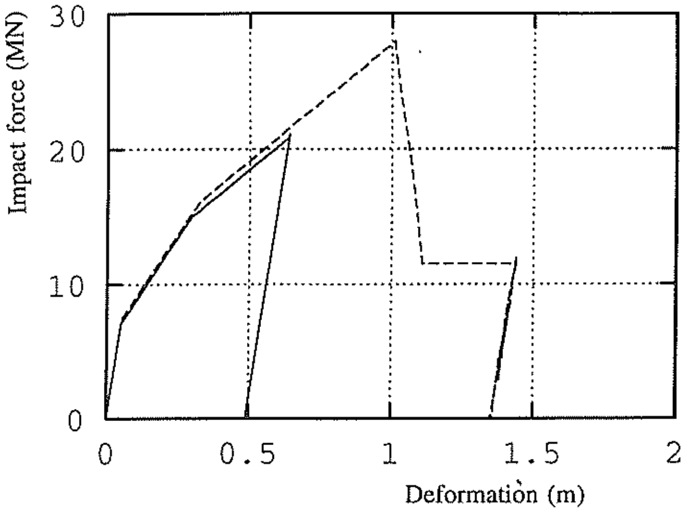


Figure 5. Simulated ship deformation relationships

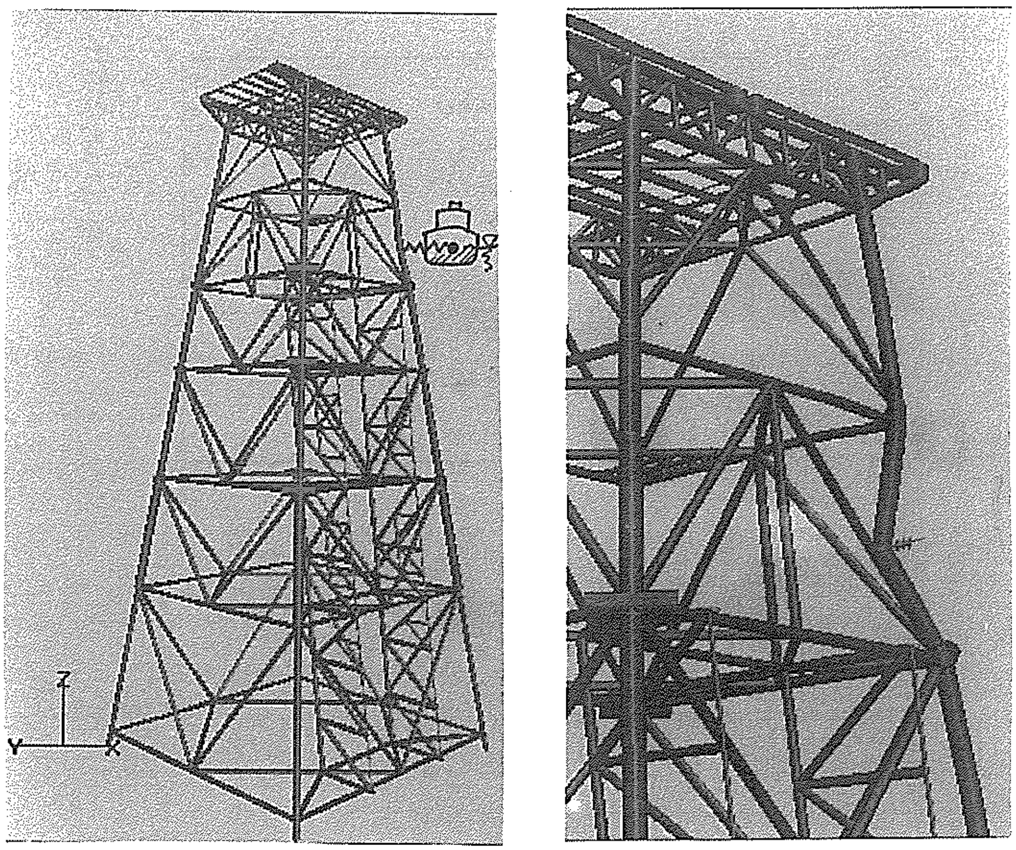


Figure 6. Screen plots of deformed jacket

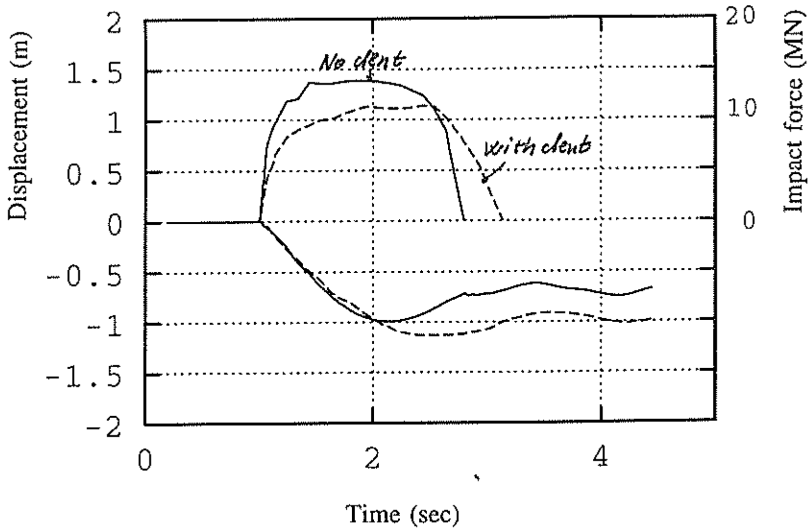


Figure 7. Contact force - and contact point displacement history for jacket.

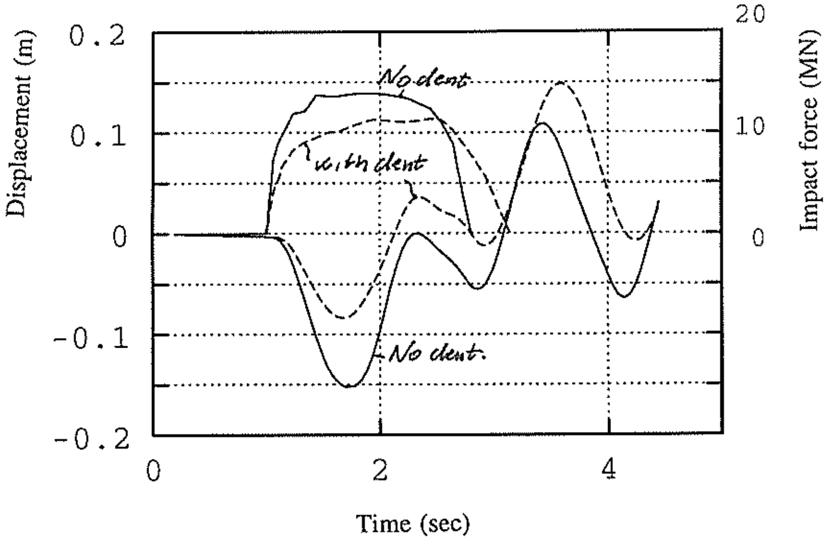


Figure 8. Deck displacement history for jacket