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Development of a Simplified Dynamic Analysis Procedure for Offshore Collisions

by

Sang-Rai Cho*

해양구조물 충돌의 간이 동적해석법 개발

조 상 래*

Abstract

A simple numerical analysis procedure has been proposed to trace the response of unstiffened offshore tubular members subjected to lateral impacts and eventually to estimate the consequential extent of damage. In the procedure a tubular member is reduced to a spring-mass system having two degrees-of-freedom, one for local denting deformation and the other for that of overall bending. Results of impact tests have been correlated with those of numerical analysis in order to achieve an empirical representation of the strain-rate sensitivity and other dynamic effects upon the spring coefficient for bending deformation. The theoretical estimates of extents of damage correlate reasonably well with those obtained in experiments.

요 약

해양구조물의 비보강원통 부재가 충돌로 인해 횡 충격하중을 받는 경우 비보강원통의 동적거동을 추적하고 그 결과로 발생되는 손상의 정도를 예측할 수 있는 간이 수치해석 방법을 제안하고자 한다. 이 방법에서는 국부적인 변형과 전체굽힘 변형을 별도의 자유도로 분리하여 2 자유도의 스프링-질량계로 치환하여 해석하게 된다. 변형속도를 비롯한 기타의 동적효과가 굽힘변형을 나타내는 스프링의 상수값에 미치는 영향은 실험자료로부터 얻어진 수정계수를 도입하여 고려하였다. 제안된 해석방법을 사용하여 얻어진 손상정도의 값들은 실험결과와 비교적 잘 일치하고 있다.

1. Introduction

For more efficient design of offshore structures considering offshore collisions it is necessary to be able to predict the probability of occurrence of minor collisions, the consequential extent of damage and the residual strengths of the damaged structures. In this paper, however, only the extent of damage of offshore tubulars due to minor collisions will be considered. A question may be raised whether offshore minor collisions can be considered as quasi-static phenomena or need to be treated as dynamic ones.

An experimental study was carried out by Arochiasamy et al.[1] on the response of a hydro-elastic

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* Member, Dept. of Naval Architecture, Shipbuilding and Ocean Engineering, University of Ulsan

semi-submersible to bergy-bit impacts. It was observed from the experiments that the rebound velocity of the bergy-bits after impact were approximately 70 to 75% of the impact velocity. In other words, about half of the initial kinetic energy of the bergy-bit was transferred to the semi-submersible while the other half was retained by the rebounding ice flow. A series of lateral impact tests on tubulars having simply supported roller end conditions were reported in ref. 2. The results of the tests showed that the rebound velocities of the striker were about 40 to 90% of those of initial impact. Nataraja and Pemsing [3] evaluated the collision energy distribution of an offshore fixed platform based on the measured extent of damage and the estimated impact velocity. It was shown that the elastic strain energy stored by the whole platform is greater than that absorbed by the impacted structural elements. From the aforementioned findings it can be suggested that dynamic elastic-plastic analyses must be employed in order to avoid excessive conservatism in predicting the consequences of offshore collisions.

The ductile response of unstiffened tubular members under lateral impacts can be divided into local denting of the cylinder wall and overall bending of the member as a beam. Some combination of these two modes is the most likely outcome for offshore tubulars. The dynamic behaviour of beams, particularly of rectangular solid sections, under transverse impulsive loadings or impacts has been extensively examined. The local effects, i.e. the local deformation surrounding the region where a striker impinges on a structure, are neglected and consequently the initial sectional configuration is assumed to be unchanged. As far as offshore tubulars are concerned it is, however, unlikely that the local deformation can be neglected in the analysis not only because roughly 10 to 15% of the total available energy would be locally dissipated[4] but also because the dent depth of a damaged tubular is one of the most influential factors upon its ultimate strength[5, 6, 7].

In order to fully take into account the local denting of cylinder wall, the influence of elastic vibrations and material strain-rate sensitivity on the permanent plastic deformations, it seems inevitable to solve the problem using a dynamic elastic-viscoplastic numerical shell analysis with the aid of the finite element method or finite difference technique. These numerical procedures are, however, expensive to operate particularly for preliminary design studies and even for parametric studies to derive any simple design equations. In the present study, therefore, an attempt has been made to develop a simple numerical procedure for solving the dynamic response of the tubular member in which the tubular member is reduced to a spring-mass system with two degrees-of-freedom. The results of impact tests have been correlated with numerical analysis to empirically represent the strain-rate sensitivity and other dynamic effects upon the spring coefficient for bending deformation. Material strain hardening and the influence of transverse shear force and rotary inertia are not considered.

2. Spring-Mass Model with Two Degrees-of-Freedom

Provided that the transverse sectional shape of a beam does not change throughout the procedure the dynamic flexural response of a tubular member under lateral impact can be investigated approximately by reducing a given problem to a spring-mass system with one degree-of-freedom. In this simplification the fundamental mode of vibration of the system under consideration needs to be estimated. The effects of higher modes of vibration which may contribute to the response will not be contained in the simplified model. When considering the local denting deformation of the cylinder wall the problem, however, becomes more complicated. In order to overcome this difficulty the local denting and the overall bending deformations are uncoupled. Adopting a spring-mass model with two degrees-of-freedom, one for overall bending and the other for local denting, the problem becomes tractable.

2.1 Equations of Motion

In Fig. 1 the analytical system model is illustrated. It is assumed in the system that damping is negli-

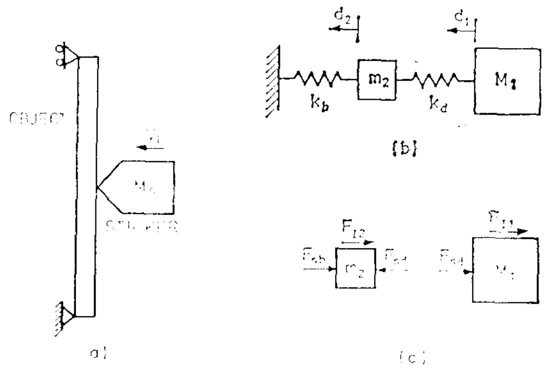

Fig. 1 (a) Pyhsical system (b) Mathematical model (c) Free body diagram

gible. Thus the dynamic equilibrium in the system is established by equating to zero the sum of the inertial forces and the spring forces. At time t_i the equilibrium of these forces can be written as follows:

for *mass* M_1 ; $F_{I1}(t_i)+F_{sd}(t_i)=0$ (1.a)

for *mass* M_2 ; $F_{I2}(t_i)+F_{sb}(t_i)-F_{sd}(t_i)=0$ (1.b)

where

F_{I1}, F_{I2} ; inertia forces of the masses M_1 and m_2 respectively

F_{sd}, F_{sb} ; spring forces for local denting deformation and overall bending deformation respectively

M_1 ; m_1+M_s, during impact
m_1, after separation

m_1 ; equivalent mass of the tube wall for local denting mode

m_2 ; equivalent mass of the tube for overall bending mode

M_s ; mass of the striker

The dynamic equilibrium at a short time Δt later can be expressed as

$$F_{I1}(t_i+\Delta t)+F_{sd}(t_i+\Delta t)=0 \qquad (2.a)$$

$$F_{I2}(t_i+\Delta t)+F_{sb}(t_i+\Delta t)-F_{sd}(t_i+\Delta t)=0 \qquad (2.b)$$

Subtracting eqns. (1.a) and (1.b) from eqns. (2.a) and (2.b) respectively results in the differential equations of motion in terms of increments, namely

$$\Delta F_{I1}+\Delta F_{sd}=0 \qquad (3.a)$$

$$F_{I2}+\Delta F_{sb}-\Delta F_{sd}=0 \qquad (3.b)$$

It is assumed here that the masses M_1 and m_2 and the spring coefficients k_d and k_b remain constant during the interval Δt. The incremental forces in eqns. (3.a) and (3.b) can be expressed as

$$\Delta F_{I1}=M_1(t_i)\Delta\ddot{d}_{1i} \qquad (4.a)$$

$$\Delta F_{I2}=m_2(t_i)\Delta\ddot{d}_{2i} \qquad (4.b)$$

$$\Delta F_{sd}=k_{di}(\Delta d_{1i}-\Delta d_{2i}) \qquad (4.c)$$

$$\Delta F_{sb}=k_{bi}\Delta d_{2i} \qquad (4.d)$$

where dots denote differentiation with respect to time.

The spring coefficients k_{di} in eqn. (4.c) and k_{bi} in eqn. (4.d) are defined as the current evaluation for the derivatives of the spring forces with respect to the corresponding displacements, namely,

$$k_{di}=\left\{\frac{d(F_{sd})}{d(d_d)}\right\}_{d_d=d_{di}} \qquad (5.a)$$

$$k_{bi}=\left\{\frac{d(F_{sb})}{d(d_2)}\right\}_{d_2=d_{2i}} \qquad (5.b)$$

where $d_d=d_1-d_2$.

Substituting eqns. (4.a-d) into eqns. (3.a) and (3.b) convenient forms for the incremental equations of motion can be obtained as follows:

$$M_{1i}\Delta\ddot{d}_{1i}+k_{di}(\Delta d_{1i}-\Delta d_{2i})=0 \qquad (6.a)$$

$$m_{2i}\Delta\ddot{d}_{2i}+k_{bi}\Delta d_{2i}-k_{di}(\Delta d_{1i}-\Delta d_{2i})=0 \qquad (6.b)$$

2.2 Integration of Equation of Motion

Among the many methods available for the solution of the non-linear equations of motion, probably one of the most effective is the step-by-step integration method. In this method the response is evaluated at successive increments Δt of time, usually taken of equal length of time for computational convenience. The non-linear characteristics of the masses M_1 and m_2 and the spring coefficients k_d and k_b are considered in the analysis by reevaluating them at the beginning of each time increment. In this study the linear acceleration method[8] is adopted in performing the step-by-step integration of eqns. (6.a) and (6.q). Using the incremental displacements Δd_1 and Δd_2 as the basic variables in the analysis the incremental velocities and accelerations can be obtained as follows

$$\Delta\ddot{d}_{1i}=\frac{6}{\Delta t^2}\Delta d_{1i}-\frac{6}{\Delta t}\dot{d}_{1i}-3\ddot{d}_{1i} \qquad (7.a)$$

$$\Delta\ddot{d}_{2i}=\frac{6}{\Delta t^2}\Delta d_{2i}-\frac{6}{\Delta t}\dot{d}_{2i}-3\ddot{d}_{2i} \qquad (7.b)$$

$$\Delta\dot{d}_{1i}=\frac{3}{\Delta t}\Delta d_{1i}-3\dot{d}_{1i}-\frac{\Delta t}{2}\ddot{d}_{1i} \qquad (7.c)$$

$$\Delta\dot{d}_{2i}=\frac{3}{\Delta t}\Delta d_{2i}-3\dot{d}_{2i}-\frac{\Delta t}{2}\ddot{d}_{2i} \tag{7.d}$$

The substitution of eqns. (7.a) and (7.b) into eqns. (6.a) and (6.b) respectively leads to the following simultaneous equations for Δd_{1i} and Δd_{2i}:

$$A_{1i}\Delta d_{1i}-k_{di}\Delta d_{2i}-B_{1i}=0 \tag{8.a}$$

$$-k_{di}\Delta d_{1i}+A_{2i}\Delta d_{2i}-B_{2i}=0 \tag{8.b}$$

where

$$A_{1i}=\frac{6M_{1i}}{\Delta t^2}+k_{di} \tag{9.a}$$

$$A_{2i}=\frac{6M_{2i}}{\Delta t^2}+k_{di}+k_{bi} \tag{9.b}$$

$$B_{1i}=3M_{1i}\left[\frac{2}{\Delta t}\dot{d}_{1i}+\ddot{d}_{1i}\right] \tag{9.c}$$

$$B_{2i}=3M_{2i}\left[\frac{2}{\Delta t}\dot{d}_{2i}+\ddot{d}_{2i}\right] \tag{9.d}$$

Eqns. (8.a) and (8.b) may be solved for the incremental displacements Δd_{1i} and Δd_{2i}:

$$\Delta d_{1i}=\frac{k_{di}B_{2i}+A_{2i}B_{1i}}{A_{1i}A_{2i}-k_{di}^2} \tag{10.a}$$

$$\Delta d_{2i}=\frac{k_{di}B_{1i}+A_{1i}B_{2i}}{A_{1i}A_{2i}-k_{di}^2} \tag{10.b}$$

The displacements and velocities at time $t=t_i+\Delta t$ can then easily be obtained using eqns. (10.a) and (10.b). Finally the accelerations $\ddot{d}_{1i+1}$ and $\ddot{d}_{2i+1}$ at the end of time step are directly obtained from eqns. (6.a) and (6.b). It is noteworthy here that in order to minimise accumulated errors the accelerations are calculated from the dynamic equlibrium equations rather than using the equations for the incremental accelerations, eqns. (7.a) and (7.b). Having determined the displacements, velocities and accelerations at time t_{i+1} the outlined procedure is repeated to calculate these quantities at the following time step $t=t_{i+1}+\Delta t$ and the process is continued to any desired final value of time.

2.3 Equivalent Masses and Spring Coefficients

2.3.1 Equivalent Masses

In the step-by-step integration of the non-linear equations of motion described in the previous section the equivalent masses and stiffness properties of the system need to be evaluated at the initiation of each time increment. In the analysis the mass M_1 is assumed during impact to be the sum of the striker's mass M_s and the equivalent mass of the locally deformed tube wall, m_1, during impact and to be the mass m_1 alone after the separation of the striker from the struck model. For overall bending deformation Cox[9] obtained an equivalent mass, m_2, equal to 17/35 of the beam's mass under the assumption that the deflection curve of the beam during impact does not differ much from the elastic curve produced by a static concentrated load at its mid-length and the velocity distribution along the length has the same form as that of deflection. When considering the fundamental mode shape of elastic vibration of the beam an equivalent mass equal to 1/2 of that of the beam can be obtained [10].

For most of practical cases the mass of the striker M_s can be much greater than the mass m_1. Thus the influence of m_1 on the mass M_1, and consequently on the extent of damage seems negligible. Notwithstanding the accuracy in determining m_1 can be transferred in the prediction of the elastic local shell vibration after the separation of the striker. which is not of importance from a practical design viewpoint. In the present study, however, the equivalent mass for local denting deformation, m_1, is approximated to be a half of the mass of the locally deformed tube wall, while the equivalent mass for overall bending deformation, m_2, is assumed to be a half of the mass of the tube.

2.3.2 Equivalent Spring Coefficients

In order to evaluate the stiffness properties of the system the force-displacement curves for overall bending deformation under static lateral loads have been derived numerically, while for local denting deformation an empirical representation of the relationship has been attempted. The spring coefficients are then obtained from the slope of these curves. It seems highly likely that strain-rate and higher mode effects are attributable to the difference in the structural behaviour of beam-like structures under moderate dynamic loads from those under static actions. To consider these dynamic effects in the analysis, a modification factor to the spring coefficient for overall bending deformation was introduced. The value of the factor was obtained from an empirical correlation with the experimental data.

∘ Local Denting : There have been five load-indentation cnrves reported so far in the literature on this matter. Three curves were presented by Smith[5, 11] and the other two by Ueda and Rashed [12]. In all the tests a quasi-static lateral load was applied at midspan through a solid knife-edge with a tip of small radius. The back of the tube at midspan was supported in a soft cylindrical cradle except for specimen P1A in ref. 5 where two cradles, located equidistant from midspan, were employed.

The force-indentation relation for loading has been obtained using the least-square method to provide a best fit to experimental data. Then the equation for the spring coefficient is obtained by differentiating the force-indentation equation with respect to displacement. However, for unloading the equation for the spring coefficient is directly derived using the test results. The equation of the spring coefficient for local denting are given as follows:

$$k_{ds}=1.25\frac{m_p}{D}(D/t)^{0.2}(E/\sigma_Y)^{0.5}\delta_\alpha^{-0.5}\ ; \quad \text{for } \textit{loading} \qquad (11.a)$$

$$=5.0\frac{m_p}{D}(E/\sigma_Y)\left\{3\left(\frac{\delta_d-\delta_{do}}{\delta_{dp}-\delta_{do}}\right)^4+\frac{2}{5}\right\}\ ; \quad \text{for } \textit{unloading} \qquad (11.b)$$

where

k_{ds} ; static spring coefficient for local denting

m_p ; plastic moment resultant of the tube wall, $1/4\sigma_Y t^2$

D ; diameter to mid-thickness of the tube

t ; thickness of the tube

σ_Y ; static yield stress

δ_d ; non-dimensionalised depth of dent, d_d/D or $(d_1-d_2)/D$

δ_{dp} ; non-dimensionalised depth of dent at which unloading starts

δ_{do} ; non-dimensionalised depth of dent when completely unloaded, $\delta_{dp}-1/2(D/t)^{0.2}(E/\sigma_Y)^{-0.5}\delta_{dp}^{0.5}$

∘ Overall Bending : Provided that the depth of dent of the tube does not increase during overall bending deformation the force and midspan lateral deflection relation of a simply supported beam under concentrated load at midspan can be computed using the Newmark integration method[12]. The bending moment at any section along the beam can easily be determined from static equilibrium conditions and the curvature along the beam can then be calculated using moment-curvature equations. The deflection at midspan for a given lateral load can be obtained by integrating twice the curvature along the beam length. By increasing the lateral force incrementally up to ultimate the non-linear force-deflection relation can be established. Using a computer program based upon the procedure described above an extensive parametric study has been carried out. The force-deflection curves resulting from the parametric study were then approximately represented by a linear equation up to the elastic limit and by an exponential equation for the elastic plastic regime. The approximate equations for the relation were obtained by regression. The spring coefficient for overall bending deformation was then obtained from the slopes of the relations as follows:

$$k_{bs}=4\frac{M_p}{L^2}a \quad ; \text{for } \textit{elastic regime} \qquad (12.a)$$

$$=4\frac{m_p}{L^2}c_1c_2(\delta_o-\delta_{o1})^{c_2-1}(f_1-f_{max})\exp\left\{c_1(\delta_o-\delta_{o1})^{c_2}\right\} \quad ; \text{for } \textit{elastic plastic regime} \qquad (12.b)$$

where

k_{bs} ; static spring coefficient for overall bending

M_p ; fully plastic moment of the intact tubular section, $tD^2\sigma_Y$

δ_o ; non-dimensionalised overall bending deformation, d_o/L or d_1/L

δ_{o1} ; non-dimensionalised elastic limit bending deformation, f_1/a

Details of f_{max}, f_1, a, c_1 and c_2 are given in ref. 13. For unloading and reloading up to the deflection at which the unloading started the spring coefficient is assumed to be the same as that for the elastic regime.

2.4 Modification Factor for Dynamic Effects

It is well known that the strain-rate sensitivity of the material can significantly increase the bending stiffness of beam-like structures subjected to severe dynamic loadings. On top of that the higher flexural vibration mode can also raise the spring coefficient for bending deformation based on the force-deflection relation under static load. The higher modes can affect flexural behaviour of the beam especially in the early stage of the impact.

In this study, however, the dynamic effects, especially the strain-rate sensitivity and higher mode effects, are roughly accounted for by an empirically derived modification factor, f_D, to the spring coefficient for bending deformation obtained from the static load-deflection relationship. Generally, the response of a tubular under lateral impact may consist of elastic-plastic deformation stage ($0<t<T_1$), elastic spring-back stage ($T_1<t<T_D$) and free elastic vibration stage ($t<T_D$). Having considered the nature of each stage it was decided to adopt different values of f_D for the first and second stages but no modification factor for the elastic vibration stage and to assume the form of the equation for f_D to be

$$f_D=1+f_{D1}\left\{\frac{d_1(t)}{V_i}\right\}^{1/2} \quad ;\ 0<t<T_1 \qquad (13.a)$$

$$=1+f_{D2}\left\{\frac{d_1(t)}{V_i}\right\}^{1/2} \quad ;\ T_1<t\leq T_D \qquad (13.b)$$

$$=1 \quad ;\ t>T_D \qquad (13.c)$$

The analysis procedure described in the following section was correlated with the experimental data of impact tests varing modification factors in an attempt to find values which would give a satisfactory estimate of extent of damage. It was found in this correlation work that the local denting damage can be determined by the value of f_{D1} independent of f_{D2}. Thus for each test case a value of f_{D1} was first identified for which theoretical and experimental denting damage were equal. Parameters which might influence f_{D1} were judged to be

R_k : initial static stiffness ratio, $\frac{(k_{ds})_{\delta_d=0.001}}{(k_{bs})_{\delta_o=0}}$

R_m : initial mass ratio, $\frac{(M_1)_{t=0}}{(m_2)_{t=0}}$ or $\frac{M_s}{1/2\rho\pi DtL}$

R_v : non-dimensionalised impact velocity, $V_i/(L/T_b)$

and

R_E : energy ratio, $(1/2M_sV_i^2)/(M_p^2L/2EI)$

where the initial static stiffness for local denting is taken for $\delta_d=0.001$ rather than for $\delta_d=0.0$ due to mathematical difficulties as $\delta_d\to 0$ and T_b is the natural period of the flexural beam vibration of the inact tubular, $2\pi\sqrt{(m_2)_{t=0}/(k_{bs})_{\delta_o=0}}$.

A non-linear regression equation of the form

$$f_{D1}=\alpha_0 R_k^{\alpha_1} R_E^{\alpha_2} R_m^{\alpha_3} R_v^{\alpha_4} \qquad (14)$$

was assumed and the values for α_0, α_1, α_2, α_3 and α_4 were found to provide a best fit to the identified values for f_{D1}. The effect of parameters R_m and R_v was found to be negligible and the equation finally obtained was

$$f_{D1}=0.08R_kR_E \qquad (15)$$

Using the experimental values for overall bending damage together with eqn. (15) a value of f_{D2} for each test was identified and then following a similar procedure to that for f_{D1} the equation of f_{D2} was found as

$$f_{D2}=f_{D1}\exp(0.07R_vR_m^2) \qquad (16)$$

2.5 Step by Step Solution

The algorithm for a step-by-step solution of a non-linear spring-mass model with two degrees-of-freedom involves initial calculations and calculations for each time step. Initially, geometric and material parameters (L/D, D/t, E/σ_Y, M_p and m_p) and basic system parameters(R_k, R_E, R_m, and R_v) need to be

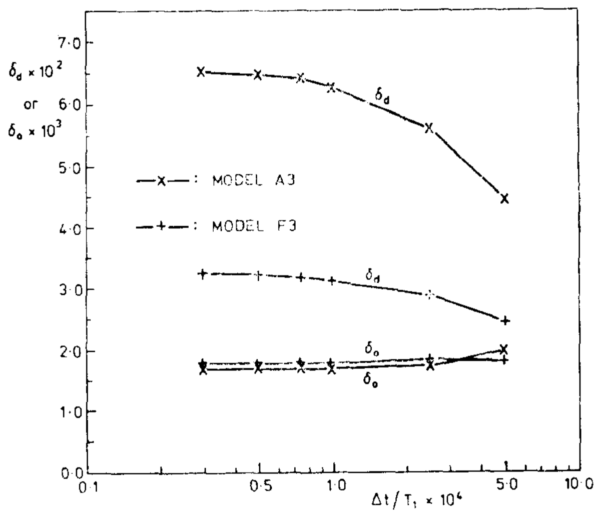

Fig. 2 Sensitivity of predicted extent of damage to time increment step

calculated together with initial conditions and the time step Δt. For each time step evaluations of the equivalent masses and spring coefficients are necessary to calculate incremental displacements. Using the obtained incremental displacements, the displacements, velocities, spring forces, accelerations and strain energies can be calculated.

As in any numerical method the accuracy of the step-by-step integration method depends upon the magnitude of the time increment selected. Generally, the natural period of the structure, the rate of the variation of the loading function and the complexity of the stiffness and damping functions can be considered in the selection of time step Δt. In this study the sensitivity of the predicted extent of damage to Δt was investigated for the cases of models A3 and F3 in ref. 13. In Fig. 2 plots are presented of the predicted extent of damage against non-dimensionalised Δt divided by T_1 which is the natural period for local denting vibration, i.e.

$$T_1 = 2\pi\sqrt{\frac{M_s}{(k_{sd})_{\delta_d=0.001}}} \tag{17}$$

For both the models, as the incremental time step Δt decreases the local denting damage increases while the overall bending damage decreases but in each case the response is asymptoted. However, the predicted overall bending damage is relatively insensitive to the time step for the range $0.00003T_1-0.0005T_1$. Compromising on the accuracy of prediction and the computing efficiency, the incremental time step finally selected to carry out the correlation and parametric studies was $0.0001T_1$.

3. Results and Discussion

3.1. History of Impact

Following the solution procedures described above the analysis has been carried out for the twenty-four test cases[2]. In Figs. 3.a and 3.b the history of displacements d_1, d_2, d_d, velocities and accelerations for masses M_1 and m_2 are illustrated for models A3 and F3 respectively. The history is also presented in the figures of the non-dimensionalised spring forces divided by maximum static lateral load, 4

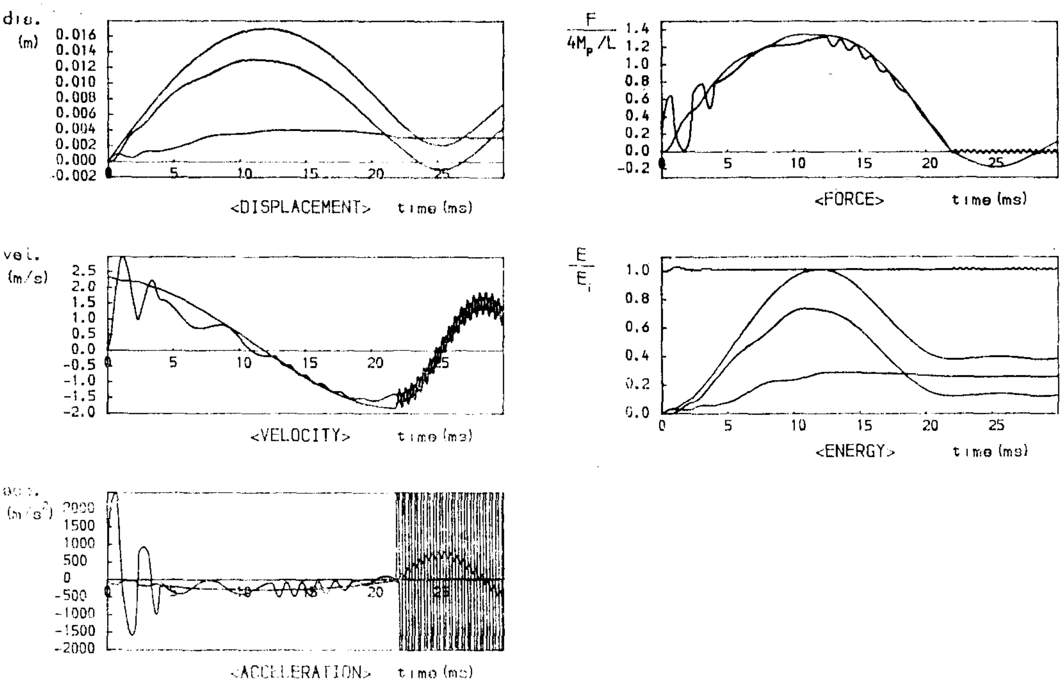

Fig. 3.a Theoretical impact history: model A3

M_p/L, and of the non-dimensionalised energies divided by impact energy, $E_k(=1/2M_sV_i^2)$. The characteristics of the impact history curves shown in figures can be specified as follows:

∘ purely local denting deformation is followed by overall bending together with additional local denting;

∘ bending deformation dominates in the elastic vibration stage and a smooth curve has been demonstrated by the total displacement d_1;

∘ in the purely local denting phase very high acceleration due to high local denting stiffness is imposed on mass m_2, which consequently develops the velocity of m_2 greater than the initial impact velocity;

∘ a high frequency local shell vibration is apparent in the elastic vibration stage, i.e. after separation of the striker from the struck model;

∘ maximum spring force can far surpass the maximum static lateral load, $4M_p/L$, and the spring force F_{sb} is the basis of the oscillation of the spring force F_{ds};

∘ the change of the strain energy of denting deformation in the elastic spring back and elastic vibration stages is negligible; and

∘ despite the fact that dynamic force equilibrium only is retained in the formulation, energy conservation has been achieved throughout the procedure with a negligible violation in the purely local denting phase.

3.2. Correlation with Test Results

A summary of the theoretical estimates is made in Table 1, which includes the extent of damage, δ_d and δ_o, peak bending deformation, δ_{opk}, impact duration, T_D, rebound velocity, V_r, energy absorbed plastically in the struck model, E_D, and maximum spring force for all of the test cases together with their parameters and experimental results.

In Fig. 4 the predictions for the extent of damage are compared with the test results. A reasonably good correlation can be seen in the figure except for two of the most severely damaged cases, i.e. for models C2 and D4. For these two cases the analysis method provides underestimated values. Another

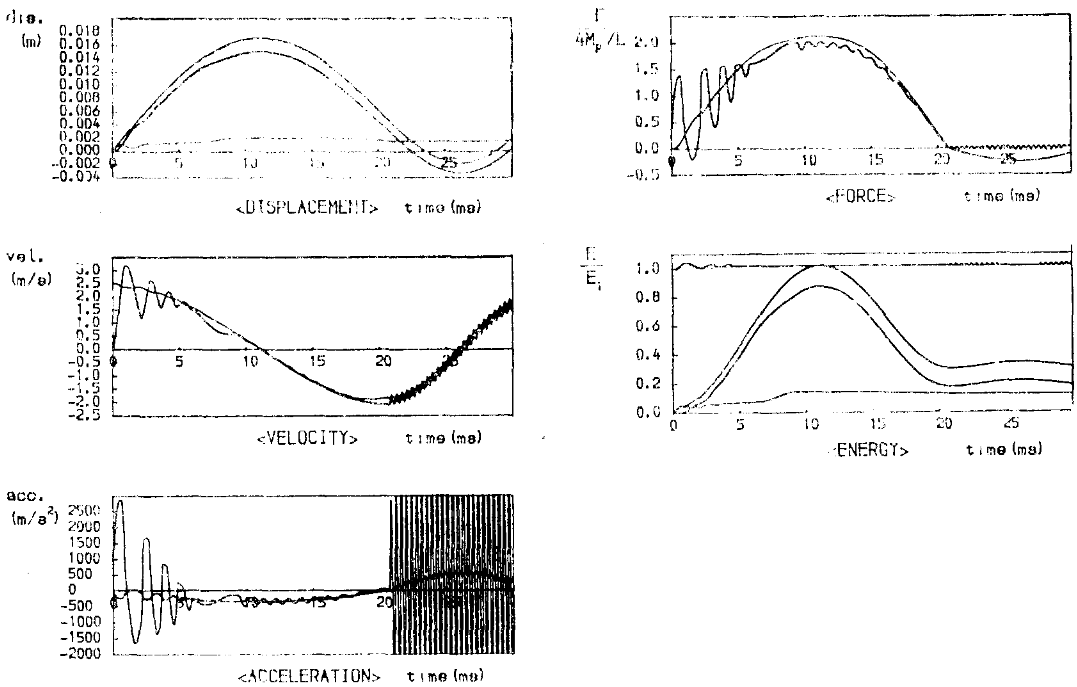

Fig. 3.b Theoretical impact history: model F3

Table 1 Comparison between experimental and theoretical results

Model	Test Parameters				Results												
					Experimental						Theoretical						
	R_k	R_E	R_m	$R_v\times 10^2$	$\delta_d\times 10^2$	$\delta_o\times 10^3$	$\delta_{opk}\times 10^3$	T_D(ms)	V_r/V_i	E_{Du}/E_k	$\delta_d\times 10^2$	$\delta_o\times 10^3$	$\delta_{opk}\times 10^3$	T_D(ms)	V_r/V_i	E_D/E_k	$F_{sm}/4M_p/L_i$
A3	25.2	0.47	19.1	2.22	7.1	2.9	14.7	33.6	0.40	0.84	6.26	1.71	9.7	21.9	0.781	0.380	1.347
A4	8.8	0.73	27.1	1.62	9.3	3.5	13.1	25.0	0.48	0.77	8.90	2.91	8.8	16.8	0.599	0.641	1.080
B1	25.9	0.64	23.8	2.39	6.2	2.8	14.4	37.8	0.58	0.66	9.40	3.53	10.2	22.3	0.664	0.558	1.556
B3	8.9	0.44	40.8	1.05	5.6	1.7	6.8	28.0	0.57	0.68	6.30	1.84	7.1	20.1	0.681	0.535	0.937
B4	25.7	0.64	28.7	2.16	4.4	1.7	—	—	—	—	9.80	4.43	9.9	23.7	0.606	0.633	1.593
C1	8.6	0.42	58.6	0.79	4.0	1.1	9.4	31.3	0.45	0.80	5.87	2.82	6.4	22.9	0.600	0.641	0.917
C2	8.7	1.63	58.2	1.55	20.9	15.0	25.6	51.4	0.36	0.87	14.10	11.59	12.3	20.3	0.225	0.953	1.386
C3	24.9	0.18	41.0	0.88	1.0	0.1	8.6	45.6	0.45	0.80	1.57	1.88	7.2	35.1	0.813	0.334	0.774
C4	24.9	0.90	41.0	1.96	13.7	8.6	18.0	61.7	0.51	0.75	12.34	7.40	9.6	24.1	0.392	0.850	1.844
D1	25.5	0.17	28.7	1.10	0.4	0.4	7.5	37.8	0.90	0.20	1.50	1.13	7.6	30.6	0.870	0.233	0.747
D2	8.9	1.12	40.4	1.68	12.5	5.9	—	30.5	0.44	0.80	12.52	7.62	10.7	18.7	0.400	0.843	1.261
D3	25.8	0.80	28.5	2.42	10.7	5.6	18.5	41.9	0.49	0.76	11.22	5.72	10.4	22.4	0.546	0.704	1.699
D4	25.9	1.19	41.3	2.46	18.3	14.8	27.0	69.7	0.42	0.82	15.42	10.19	11.6	22.7	0.293	0.917	1.987
E3	41.3	0.49	17.1	2.40	0.8	0.4	12.1	30.5	0.86	0.26	3.44	1.87	9.1	17.8	0.809	0.332	1.669
F1	39.0	0.05	30.5	0.53	0.0	0.0	3.7	39.8	0.56	0.68	0.10	0.38	4.3	33.8	0.953	0.079	0.480
F1p	39.0	0.51	25.0	1.84	1.6	0.8	13.4	36.1	0.59	0.65	3.61	2.84	8.4	20.5	0.738	0.450	1.670
F2	13.6	0.64	35.5	1.22	4.3	2.0	9.4	23.4	0.81	0.35	4.62	2.90	7.6	15.9	0.651	0.575	1.314
F3	84.9	0.48	13.4	3.17	2.5	1.5	25.4	42.2	0.67	0.55	3.12	1.77	8.6	20.8	0.822	0.304	2.119
G1	13.7	0.67	24.4	1.52	3.5	1.7	10.9	21.4	0.60	0.64	4.79	1.92	7.8	13.8	0.704	0.501	1.327
G2	39.5	0.63	17.1	2.50	3.7	2.4	16.4	32.0	0.71	0.50	4.39	2.03	8.7	16.9	0.773	0.392	1.913
G3	85.6	0.30	19.2	2.11	0.4	0.3	13.5	52.2	0.83	0.30	1.33	2.27	8.3	27.7	0.820	0.313	1.491
H1	39.5	0.52	11.4	2.80	0.6	0.3	13.0	25.3	—	—	3.57	0.77	8.7	15.1	0.849	0.253	1.721
H2	38.6	0.93	25.1	2.46	6.5	4.3	17.3	39.1	0.55	0.70	8.22	4.00	8.4	17.3	0.598	0.641	2.390
H3	13.5	0.24	35.5	0.73	0.1	0.0	6.2	23.1	0.59	0.65	1.52	0.64	5.6	18.2	0.852	0.267	0.822

shortcoming of the method can be found in the skewness of the predicted impact duration, T_D, and peak bending deformations, δ_{opk}, which is illustrated in Fig. 5. The underestimation both of T_D and δ_{opk} is becoming more apparent as the value of $R_kR_ER_vR_m$ increases. Among other factors the consideration of

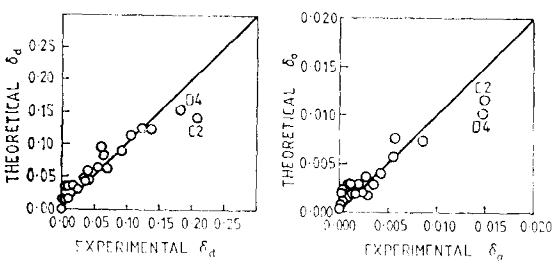

Fig. 4 Comparison between theoretical predictions and test results for extent of damage

overall bending damage in the derivation of the spring coefficient for denting deformation seems to improve these shortcomings. As described earlier the derived force-indentation relationship is based on the results of tests conducted with supports at the back of the dent centre which minimises the overall bending deformation. Therefore when the bending deformation is large an overestimated spring coefficient is obtained from the relationship.

The predictions for fourteen cases, whose extents of damage exceed the tolerance specifications given in ref. 14, provide a 20.9% COV with a mean of 1.080 and a 25.3% COV with a mean of 0.993 for local denting and overall bending damage respectively. It seems that these COVs are somewhat higher than those of static structural problems. However,

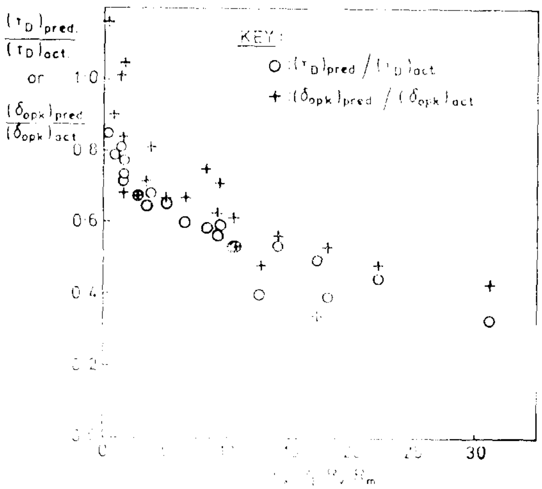

Fig. 5 Skewness of predicted impact duration(T_D) and peak bending deformation (δ_{opk})

considering the complexity of the dynamic problem and the computing efficiency the usefulness of the proposed method can be justified.

4. Conclusions

A simple analysis procedure has been developed to simulate the dynamic response of a tubular member having simply supported roller end conditions. In the procedure the tubular member is reduced to a spring-mass system with two degrees-of-freedom, one for overall bending deformation and the other for local denting deformation.

a) Correlation of the predictions following the proposed procedure with the available test results provides a 20.9% COV with a mean of 1.080 and a 25.3% COV with a mean of 0.993 for local denting and overall bending damage respectively.

b) A shortcoming of the proposed procedure is the underestimation of both impact duration, T_D, and peak bending deformations, d_{opk}, for the higher values of $R_k R_E R_v R_m$, which seems to possibly be improved by consideration of overall bending damage in the derivation of spring coefficient for denting deformation.

c) In order to provide a directly available analysis procedure for pedicting the extent of damage of tubular members of offshore platforms the simulating seems necessary of the actual boundary conditions of impacted members and realistic behaviour of the ship structure in the analysis. This can be achieved by the inclusion of more degrees-of-freedom in the proposed procedure.

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