THE NUMERICAL ANALYSIS OF TRANSITORY DYNAMIC RESPONSE, BASED ON THE NON-LINEAR HYDROELASTICITY THEORY, FOR A BARGE TEST SHIP★

LEONARD DOMNIȘORU[1], DANIELA DOMNIȘORU[2]

[1] University "Dunărea de Jos" of Galați, Naval Architecture Faculty, Romania
[2] The Naval Transport High-School of Galați, Physics Department, Romania
Tel./Fax: +40 236 495400, email: leonard.domnisoru@ugal.ro

Received September 26, 2006

In this paper there is presented the study of the transitory vertical displacement response, oscillations and vibrations, of a barge model, under initial imposed fore-pick displacement conditions. The analysis is carried on with eigen program DYN-NL, based on the non-linear hydroelasticity theory. The numerical model includes time domain implicit integration procedure of the motion equations, at zero speed and still water conditions. For the numerical analyses validation, there are used experimental results provided by the Bureau Veritas Register Paris, in the frame of EU-FP6 Marstruct Project. The numerical results are in a good agreement with the experimental data. The study points out the best approach in order to calculate the hydrodynamic terms.

Key words: ships hydroelasticity, non-linear numerical analysis, transitory dynamic response.

1. INTRODUCTION

This study is focused on the time domain analysis of the barge transitory dynamic response, prismatic ship with small changes in the fore pick, proposed by the Bureau Veritas Register [5]. The numerical non-linear analysis is carried on with eigen program pack DYN-NL (module TRANZY).

2. THE THEORETICAL MODEL FOR TRANSITORY SHIP RESPONSE

2.1. THE HYPOTHESES

1) The ship hull girder is modelled using the finite element method (FEM), with Ne Timoshenko beam elements [2, 3], including bending and shearing deformations in vertical plane.

★ Paper presented at the National Conference on Applied Physics, June 9–10, 2006, Galați, Romania

2) The hydrodynamic forces are calculated according to the strip theory Gerritsma & Beukelman [3], with the inclusion of the non-linearities due to the time variation of the hydrodynamic coefficients, at the instantaneous ship-wave position, and the bottom-slamming component.

3) Based on the modal analysis technique, there is considered the ship dynamic response decomposed on the $r = 0, 1$ rigid modes and $r = 2, n$ the first n-2 eigen ship elastic girder modes.

4) The hydrodynamic terms on the eigen rigid modes $r = 0, 1$ are calculated at the ship vertical oscillation circular frequency ω_{osc}. The hydrodynamic terms on the eigen vibration modes $r = 2, n$ are calculated for ω_2, the circular frequency of the fundamental vibration mode.

5) The geometry of the ship hull transversal sections is modelled using the conformal multi-parametric transformation [2, 3], including also the possibility of total emersion from water.

6) There is no external excitation wave taken into account, so that no stabilized response occurs.

7) The speed of the ship is zero and the ship-girder initial conditions are non-zero displacements.

2.2. THE MOTION NON-LINEAR DIFFERENTIAL EQUATIONS

According the 3-rd and 6-th hypotheses, the ship transitory dynamic response is decomposed in: $x \in [0, L]$; $n = 4$

$$w_{nl}(x,t) = \sum_{r=0}^{n} w_r(x)\, p_{nlr}(t); \quad \theta_{nl}(x,t) = \sum_{r=0}^{n} \theta_r(x)\, p_{nlr}(t);$$
$$\gamma_{nl}(x,t) = \sum_{r=0}^{n} \gamma_r(x)\, p_{nlr}(t) \tag{1}$$

where: $w_r(x)$, $\theta_r(x)$, $\gamma_r(x)$ are the displacement, bending and shearing rotations eigen modes form functions (FEM calculated); $p_{nlr}(t)$ are the non-linear principal modal coordinates; L is the ship length.

Based on the 1-st and 3-rd hypotheses, the motion equations system is:

$$[a]\{\ddot{p}_{nl}(t)\} + [b]\{\dot{p}_{nl}(t)\} + [c]\{p_{nl}(t)\} = \{F_h(t)\}$$
$$[a] = diag\{a_{ss}\}\big|_{s=0,n}; \quad [b] = \{b_{rs}\}\big|_{r,s=0,n}; \quad [c] = diag\{c_{ss}\}\big|_{s=0,n}$$
$$a_{ss} = \int_0^L \left[\mu(x) w_s^2(x) + j_y(x)\theta_s^2(x)\right] dx; \quad \{p_{nl}(t)\} = \{p_{nl0}(t), \ldots, p_{n\ln}(t)\}^T$$

$$b_{rs}=\int_0^L\left[\alpha_r(x)GA_{fz}(x)\gamma_r(x)\gamma_s(x)+\beta_r(x)EI_y(x)\theta'_r(x)\theta'_s(x)\right]dx$$
$$c_{ss}=\int_0^L\left[EI_y(x)\left(\theta'_s(x)\right)^2+GA_{fz}(x)\gamma_s^2(x)\right]dx;\quad F_{hs}(t)=\int_0^L F_h(x,t)w_s(x)dx \tag{2}$$

where: $\mu(x)$, $j_y(x)$ are the ship mass and inertial mass moment per unit length; $EI_y(x)$, $GA_{fz}(x)$ are the bending and shearing ship rigidity; $\alpha(x)$, $\beta(x)$ are the structural damping coefficients.

According to the 2-nd hypothesis, the hydrodynamic terms and the wet sectional areas are decomposed in still and instantaneous ship-water position components:

$$m_{33}(x,t)=m_{330}(x)+m_{33nl}\big|_{w_{nl}}(x,t);\quad N_{33}(x,t)=N_{330}(x)+N_{33nl}\big|_{w_{nl}}(x,t);$$
$$A(x,t)=A_0(x)-b_0(x)w_{nl}(x,t)+A_{nl}\big|_{w_{nl}}(x,t) \tag{3}$$

Based on the 2–7 hypotheses, the generalized hydrodynamic force has the expression:

$$\{F_h(t)\}=\{F_{h0}(t)\}+\{F_{h1}(t)\}$$
$$\{F_{h0}(t)\}=-[A_h]\big|_{\omega2}^{\omega osc}\{\ddot{p}_{nl}(t)\}-[B_h]\big|_{\omega2}^{\omega osc}\{\dot{p}_{nl}(t)\}-[C_h]\big|_{\omega2}^{\omega osc}\{p_{nl}(t)\}$$
$$A_{hrs}=\int_0^L m_{330}(x)w_r(x)w_s(x)dx;\quad B_{hrs}=\int_0^L N_{330}(x)w_r(x)w_s(x)dx$$
$$C_{hrs}=\int_0^L \rho g b_0(x)w_r(x)w_s(x)dx;\quad F_{h1s}(t)=\int_0^L F_{h1}(x,t)w_s(x)dx \tag{4}$$
$$F_{h1}(x,t)=-m_{33nl}\big|_{w_{nl}}\frac{\partial^2 w_{nl}}{\partial t^2}-\left[N_{33nl}\big|_{w_{nl}}+\frac{\partial m_{33nl}}{\partial t}\bigg|_{w_{nl}}\right]\frac{\partial w_{nl}}{\partial t}+\rho g A_{nl}\big|_{w_{nl}}+$$
$$+K_{imp}\big|_{w_{nl}}\left[\frac{\partial w_{nl}}{\partial t}\right]^2$$

where $K_{imp}\big|_{w_{nl}}$ is the impact bottom-slamming coefficient.

From relations (2), (4) the motion equations system at non-linear dynamic analysis becomes:

$$[A]\{\ddot{p}_{nl}(t)\}+[B]\{\dot{p}_{nl}(t)\}+[C]\{p_{nl}(t)\}=\{F_{h1}(t,\{p_{nl}\},\{\dot{p}_{nl}\},\{\ddot{p}_{nl}\})\}$$
$$[A]=[a]+[A_h]\big|_{\omega2}^{\omega osc};\quad [B]=[b]+[B_h]\big|_{\omega2}^{\omega osc};\quad [C]=[c]+[C_h]\big|_{\omega2}^{\omega osc} \tag{5}$$

2.3. THE TRANSITORY SHIP GIRDER DYNAMIC RESPONSE

Because $\{F_{h1}(t)\}$ is function of the dynamic response $\{p_{nl}(t)\}$, it is necessary to use a integration in time domain method (6), based on the β-Newmark ($\beta = 1/2$) algorithm, in order to solve system (5).

The simulation time is $T_s = 8$ s with a time step $\delta t = 0.001$ s and the triggering frequency $f_{es} = 1000$ Hz. There are obtained 8001 values into a time record file at each transversal section.

Obs. Applying the spectral analysis with the Fast Fourier Transformation (FFT) [3] to the calculated time records, there are obtained the amplitude spectral functions of the dynamic response.

step $t=0$: $\{p_{nl}(0)\} \neq 0 ; \{\dot{p}_{nl}(0)\} = 0 \Rightarrow$

$$\Rightarrow \{\ddot{p}_{nl}(0)\} = \left[A^{-1}\right]\left(\{F_{h1}(0)\} - [C]\{p_{nl}(0)\}\right)$$

step t: $\{p_{nl}(t)\} ; \{\dot{p}_{nl}(t)\} ; \{\ddot{p}_{nl}(t)\}$

step $t+\delta t$: It will be solved the linear equation system in $\{\ddot{p}_{nl}(t+\delta t)\}$:

$$\left\{[A] + [B]\frac{\delta t}{2} + [C]\left(\frac{\delta t}{2}\right)^2\right\}\{\ddot{p}_{nl}(t+\delta t)\} = \{F_{h1}(t+\delta t)\} - \{F_{h1}(t)\} +$$

$$+\left\{[A] - [B]\frac{\delta t}{2} - [C]\left(\frac{\delta t}{2}\right)^2\right\}\{\ddot{p}_{nl}(t)\} - \{[C]\delta t\}\{\dot{p}_{nl}(t)\}$$

$$\{\dot{p}_{nl}(t+\delta t)\} = \{\dot{p}_{nl}(t)\} + \left[\{\ddot{p}_{nl}(t)\} + \{\ddot{p}_{nl}(t+\delta t)\}\right]\frac{\delta t}{2} ;$$

$$\{p_{nl}(t+\delta t)\} = \{p_{nl}(t)\} + \{\dot{p}_{nl}(t)\}\delta t + \left[\{\ddot{p}_{nl}(t)\} + \{\ddot{p}_{nl}(t+\delta t)\}\right]\left(\frac{\delta t}{2}\right)^2$$

..... iteration $t = Ts$

3. THE BARGE TEST SHIP MODEL

There is considered the model of the barge, as it is defined in the report [5] (see Fig. 1). The idealization of the input data for the barge model is presented in the Table 1 [4].

The structural damping coefficients, according to Johnson & Tamita [1], are:

$$\alpha_r(x) \approx \beta_r(x) = 0.001 \cdot \Gamma_r ; \quad \Gamma_0 = \Gamma_1 = 0; \quad \Gamma_2 = 1; \quad \Gamma_3 = 0.95; \quad \Gamma_4 = 0.9 \qquad (7)$$

Based on the method of Vugts [1, 3], there are obtained the hydrodynamic mass coefficients c_{33} and the hydrodynamic damping coefficients λ_{33}, at oscillations and vibrations frequencies domains.

Table 1

Barge model characteristics

Ne FEM beam elements	38	ρ [kg/m^3] water density	1000
D.O.F.degrees of freedom	78	Δ [kg] ship displacement	172.53
u_s [m/s] ship speed	0	A_{fz} [m^2] shear area	5.000E-04
Wave excitation	no	I_y [m^4] bending inertial moment	1.800E-09
L [m] ship length	2.445	μ [kg/m] mass per unit length	70.564
B [m] ship breadth	0.600	j_y [kgm^2/m] inertial mass / L	1.39E-05
D [m] ship depth	0.250	E [N/m^2] Young module	2.06E+11
d [m] ship draft	0.12000	G [N/m^2] Transversal module	7.92E+10
d_{aft} [m] aft draft	0.11316	g [m/s^2] gravity acceleration	9.81
d_{fore} [m] fore draft	0.12691	ρ_m [kg/m^3] material density	7.70E+03
c_B ship block coefficient	0.98	dx [m] FEM element length	0.0815 / 0.0545 / 0.019

$$m_{33} = c_{33} J_n \cdot \rho \pi b^2 / 8 \qquad N_{33} = \lambda_{33} \cdot \rho \omega b^2 / 4$$
$$c_{33}, \lambda_{33} = f\left(c_T, H, \delta\right) \qquad H = b/2d \qquad \delta = \omega^2 b / 2g \tag{8}$$

where: c_T, b, d are the section data, ω is the circular frequency, J_n is the Townsin coefficient [1].

In Fig. 2a, b there are presented the hydrodynamic coefficients c_{33}, λ_{33}, for a transversal section at the prismatic zone, function to the z/d, $z \in [0, d]$.

In Table 2 there are presented the eigen circular frequencies and in Fig. 3 the eigen oscillations and vibration modes.

Table 2

Eigen circular frequencies [rad/s]

Mode	$r = 0$ heave	$r = 1$ pitch	$r = 2$ flex1	$r = 3$ flex2	$r = 4$ flex3
ω osc./vib.	5.617	5.617	5.771	17.070	34.810

For this barge test $\omega_{osc} = 5.617$ rad/s $\approx \omega_{vib2} = 5.771$ rad/s and $\delta \approx 1$, so that the hydrodynamic damping is non-zero (with high values) and the hydrodynamic mass is between oscillations and vibrations values.

4. THE TIME DOMAIN ANALYSIS OF SHIPS TRANSITORY RESPONSE

According to report [5], the initial displacement conditions are obtained by pulling vertically the barge fore-pick to a prescribed level and then releasing the model by cutting the attached rope. In Table 3 there are presented the initial displacements from the experimental extinction test [5] and in table 4 there are the calculated initial conditions in terms of modal principal coordinates.

Table 3

Initial vertical displacements

Nr section S	x [m]	$w(x,0)$ [mm]
1	2.445	101.90
3	2.035	51.08
5	1.625	14.23
7	1.215	–1.19
9	0.805	–6.27
11	0.395	–6.74

Table 4

Initial modal principal coordinates

Mode r	$p_{nlr}(0)$
0	0.0160677
1	–0.0459936
2	0.0305438
3	–0.0072541
4	0.0020409

In Fig. 4 there is presented the initial experimental deformation of the barge girder. In Fig. 5 there is presented the deformations of the barge girder at some time values t.

In Fig. 6.1–2a, b there are presented the time records for transitory vertical displacement response, based on the numerical analysis. There are considered two cases for the hydrodynamic terms calculated on modes $r = 2, 3, 4$: $\omega = \omega_{vib2}$ or $\omega \to \infty$ [4]. In Fig. 7a, b there are presented the FFT amplitude spectrums for the time records at section 1.

In Fig. 8.1–2 there are presented the time records for transitory vertical displacement response, based on the experimental analyses presented in the Bureau Veritas Register report [5].

The time records in Fig. 6, Fig. 8 are non-dimensional, using the maximal value $A = 101.9$ mm.

5. CONCLUSIONS

The numerical time records for transitory response of vertical displacements [4] are in good agreement with the experimental data [5]. The differences that occur have the following main sources: the precision of the input data idealization used in the tests; the structural damping coefficients are based on empiric values; the eigen induced waves are neglected in the theoretical model and at the numerical analyses; method induced differences, because this study it is based on the 2D flow approach (strip theory).

Because $w_{\max} = 101.90$ mm $< d_{fore} = 126.91$ mm no bottom slamming occurs.

It results that in the case of hydrodynamic terms calculated for $\omega \to \infty$ (Fig. 6b), with zero hydrodynamic damping, the transitory response dose not corresponds to the experimental data.

In conclusion, because the vibration fundamental mode is dominant at the transitory response (see Fig. 7), the hydrodynamic terms must be calculated for $\omega \to \omega_2$ (Fig. 6.a), with non-zero hydrodynamic damping ($\omega_{\mathrm{osc}} \approx \omega_{vib2}$).

Also further investigations have to be carried on for the structural damping terms.

REFERENCES

1. R. E. D. Bishop, W. G. Price, *Hydroelasticity of Ships*, Cambridge University Press, 1979.
2. L. Domnisoru, D. Domnisoru, The Unified Analysis of Springing and Whipping Phenomena, The Royal Institution of Naval Architects, London, Part A,Volume 140, pp. 19–36, 1998.
3. L. Domnisoru, L. Stoicescu, D. Domnisoru, *The Analysis of Ships Dynamic Response Linear and Non-linear in Head Waves Based on the Hydroelasticity Theory*, (eBook) The University „Dunarea de Jos" Foundation Publishing House, Galati, 2006.
4. L. Domnisoru *et. al.*, The Numerical Analysis of Time Domain Transitory Ships Dynamic Response based on the Non-linear Hydroelasticity Theory. Barge Model Test Ship, WP.1 Methods and Tools for Loads and Load Effects, Report Grant EU Marstruct-FP6, 2006.
5. S. Malenica, G. Parmentier, Barge models. Specifications for comparisons, WP.1 Methods and Tools for Loads and Load Effects, Report Grant EU Marstruct-FP6, Bureau Veritas Paris, 2005.

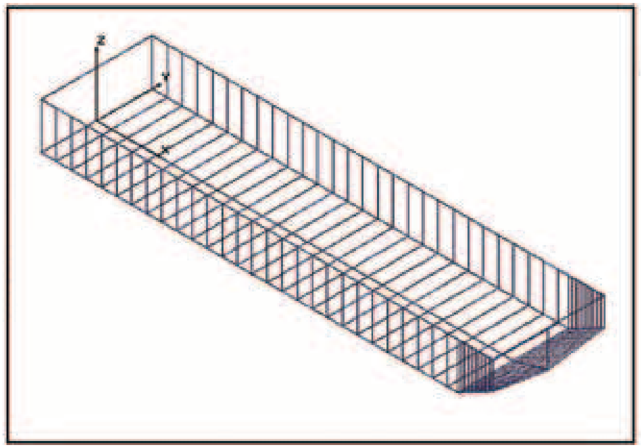


Fig. 1 – The barge model offset at DYN-NLN analysis.

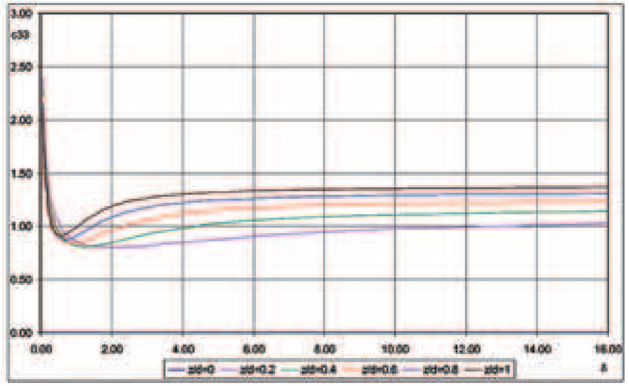


Fig. 2.a – The hydrodynamic mass coefficients c_{33}.

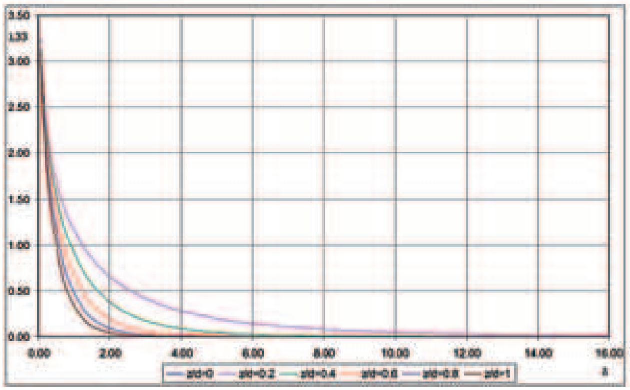


Fig. 2.b – The hydrodynamic damping coefficient λ_{33}.

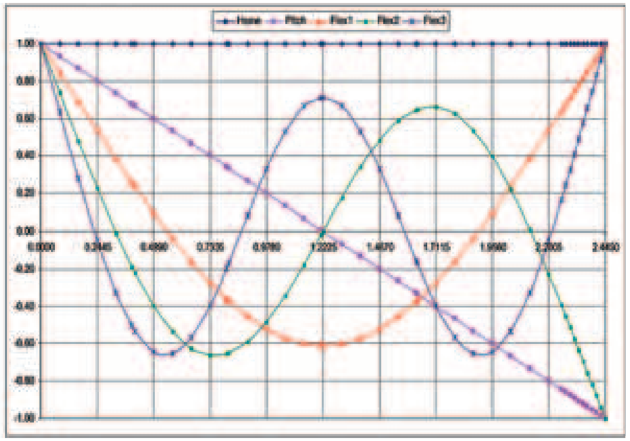


Fig. 3 – Eigen oscillations and vibration modes.

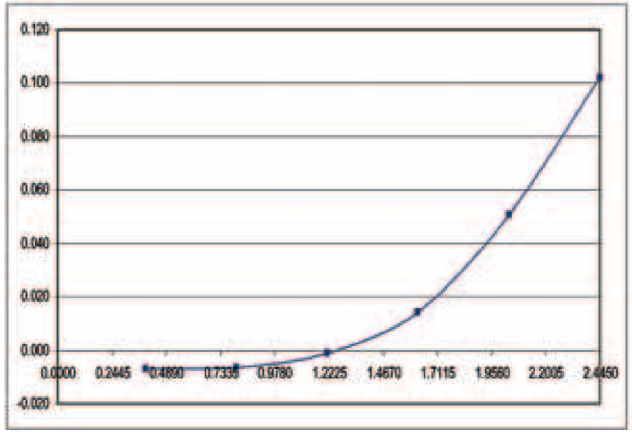


Fig. 4 – Initial experimental deformation [5].

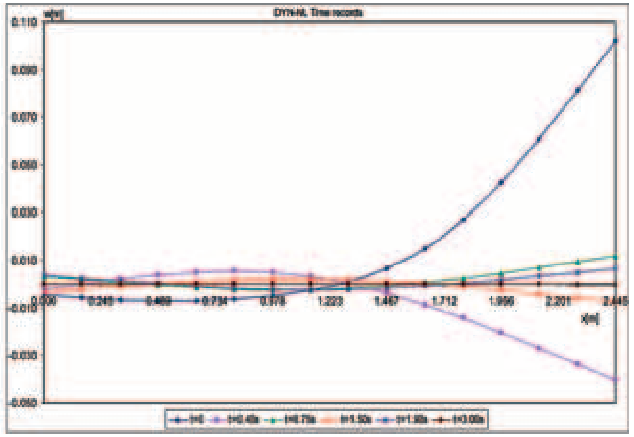
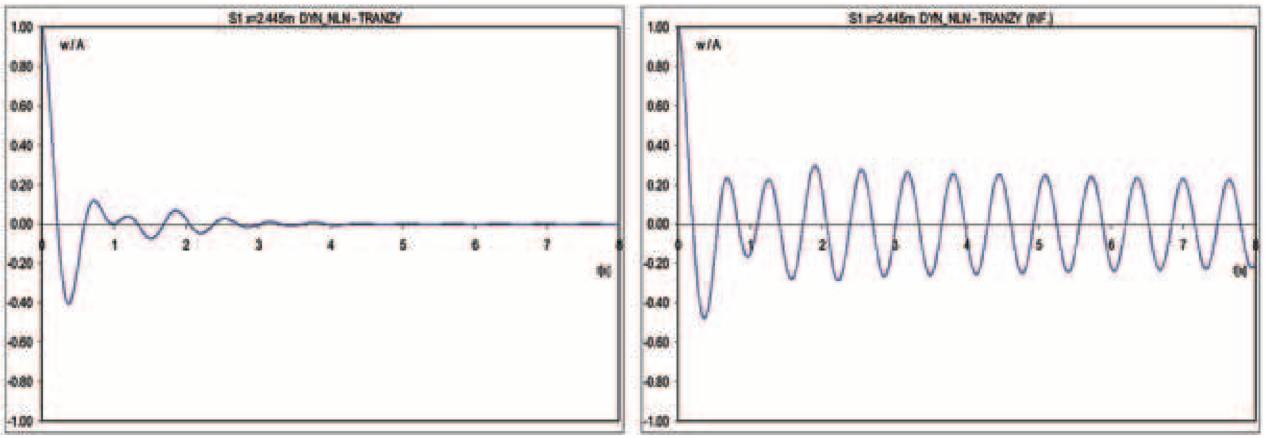
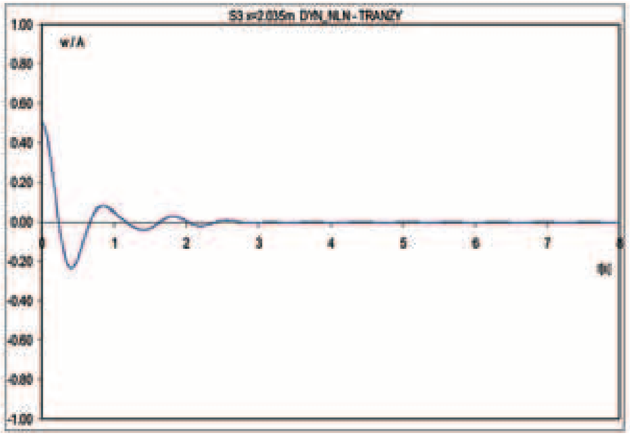


Fig. 5 – The deformations [4] at $t = 0; 0.4; 0.75; 1.5; 1.9; 3$ s.

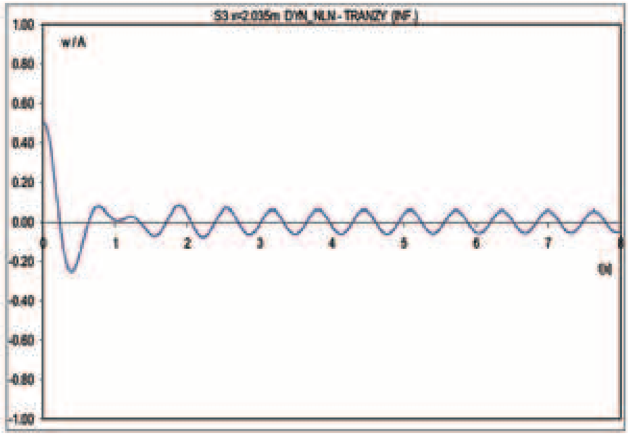


a) b)

Fig. 6.1 – (a) w/A time record, section S1, $\omega = \omega_{vib2}$, (b) w/A time record, section S1, $\omega \to \infty$.

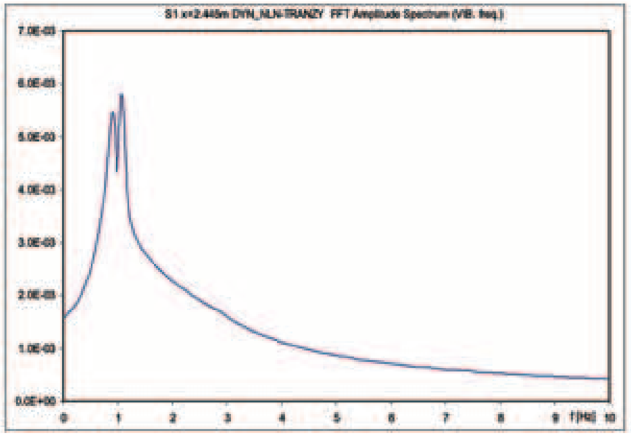


a)

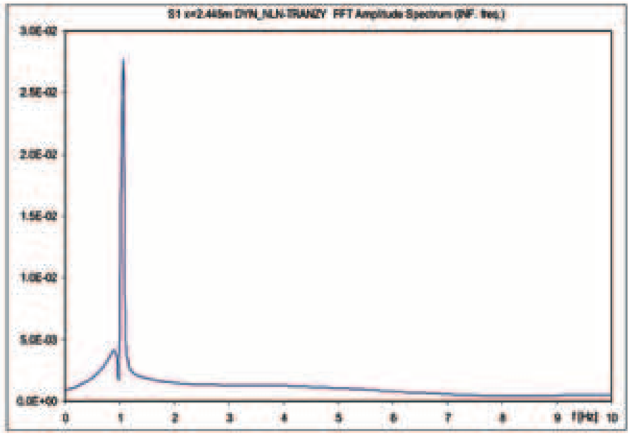


b)

Fig. 6.2 (a) *w*/*A* time record, section S3, $\omega = \omega_{vib2}$, (b) *w*/*A* time record, section S3, $\omega \to \infty$.



a)



b)

Fig. 7 – (a) FFT amplitude spectrum, section S1, $\omega = \omega_{vib2}$, (b) FFT amplitude spectrum, section S1, $\omega \to \infty$.

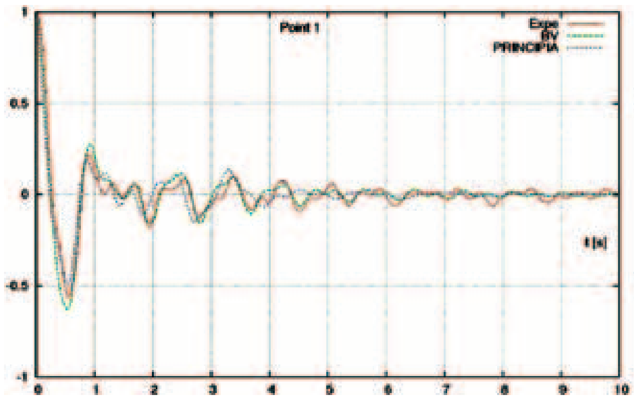


Fig. 8.1 *w*/*A* time record, section S1, experimental [5].

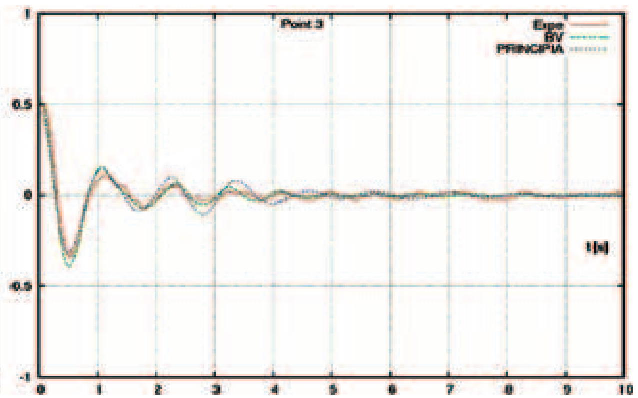


Fig. 8.2 *w*/*A* time record, section S3, experimental [5].