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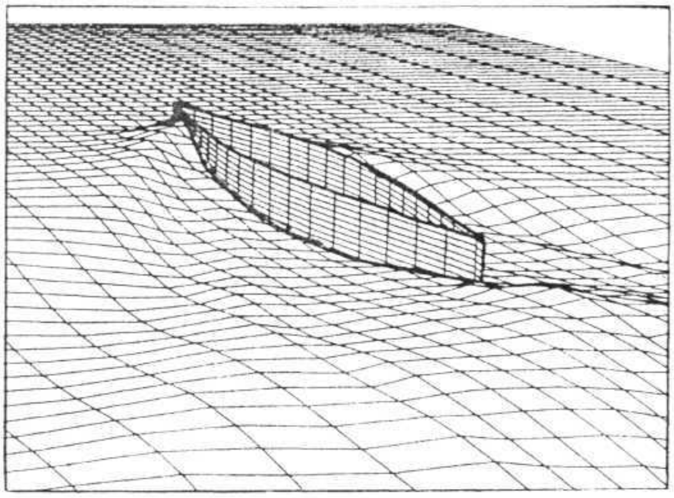

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VOL 4

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Water waves and floating bodies

FORCES AND MOMENTS IN THE RIGID CONNECTIONS BETWEEN A BARGE AND ITS TUG WITH FORWARD SPEED IN WAVE

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Abstract

We propose in this paper a 3D method to compute forces and moments in the rigid connections between the two parts of an ocean going tug-barge system.

As the relative movements of the tug and its barge are small as compared to the motions in the waves,this physically complex problem is reduced to the determination of the potential of the flow around the immersed part of the two bodies together.Boundary conditions are linearized and calculated at the mean boundary position of each hull.

The total velocity potential is obtained by summing incident,diffraction and radiation velocity potentials.Each simple problem is resolved by a 3D integral method using a distribution of sources on the bodies;

When the motions in wave of the tug and barge are fixed,we can calculate the forces and moments on the tug (or the barge)and introduce connections forces and moments as unknowns.The very classic equation which binds these forces and relative displacements owing to stiffness matrix gives us the elements of movements between the two bodies.

The last step consists in calculating first the displacement and then the forces on each connection.

We have compared our numerical results to experimental results of a tug-barge system,obtained in David W.Taylor Naval Ship Pesearch . and Development Center.

I .General problems raised by tug-barges

The use of barges and tug-boats is not a new idea.However,the innovation was to place the tup at the aft.Part of the barpe and to more or less connect it to the barge.Such idea enables to obtain an improved efficiency of the tug-barge assembly with respect to the other traditional towing solution using roges or cables.The two main advantages to be obtained by th« tuR-barge system are a decrease of the resistance in operation.a better running stability and manoeuvrability.The two advantages have direct financial incidents on exploitation : energy costs reduced barge rotation speed increased.

The differences between the various tug-barge systems depend on the mechanical particulars of the connection between the components of the assembly :

- the flexible system where the tup keeps all or part of its free motions with respect to the barge.
- the rigid system where the tug is embedded in a slot,the shape of which is adapted to the aft part of the barge.

The two systems present the same advantages and disadvantages as regards the agitation of the water in which the assembly is working :

- calm water (rivers,small lakes),simple pushing
- "sheltered" water (lakes,coastal waters) : flexible system where certain motions (lateral) are blocked, (soft lines)
- agitated waters (oceans,seas,wide lakesKntesrated system.

The main problems met with are in connection with :

- manoeuvring problems (for example : disenpageraent of a tup, "soft lines" to pass in simple towing)
- problems of structures : stresses due to swell on the connection system and on the structures of the barge and the tug.

The solution to this latter problem constitutes the obiect of present studies carried out at ACP.

In this paper we deal in particular with the rigid connection systems,in which the tug-barge assembly forms a conical system with respect to the forces due to swell.

The connection technique we wich to modelize is that Patented by ACB.

"This technique uses,for the connections between the tug and the barge,an assembly *of* hydraulically inflated rubber cushions connected to accumulators.enabling to absorb the pressure and contact surface variations during working operations.

The movements regarding this type of connection are of about a few millimetres.

Furthermore, the cushions have four advantages :

- a very easy fitting of the tug in the bar- - ge notch, as they can be deflated (resulting in the reduction of the manufacturing allowances)
- a quick setting (simple inflation and adjus tment by hydraulic circuits)
- a possibility of adjustment of pressure, therefore of the forces and surface contacts.
- a possibility or damping or regulation by hydraulic accumulators.

Thanks to the aforesaid,it becomes easy to obtain sea trial results,as the only thing to do is to record the pressures in each cushion.

The cushions have a parallelepipedic shape (1000x1000x120mm)and a rigid tug-barge connec- tion is comprised of about thirty cushions plus on longitudinal link (for example under the form or an articulated bar which will be prestressed by pressurization of the cushions).

The object of our study is therefore to calcu late,in particular,the efforts applied on each cushion when the assembly in running in swell. So, it was suitable to use a hydrodynamical theory taking into account the running speed pf the assembly in swell.

Besides, the dimensions of the barges used (low L/B ratio) as well the necessity to calcu- late the hydrodynamic efforts on the tug loca- ted at the aft part of the assembly forming one and only ship,bind us to use a three-dimen- sional hydrodynamic model (the end effects not being represented in the -strip theory in spite of certain attemps of corrections)

Therefore, we present hereunder a calculation program,DYNAPLOUS 81,developed by the ENSM,using a traditional method to solve the problem of diffraction-radiation with running speed.

The program does not calculate the resistan- ce to operation which will be taken into account only under the form of a constant torque aoplied by the tug propelling system on the connection.

II. General assumptions regarding the model and limits

The assumptions concern the mechanic and the hydrodynamic.

The mechanical basis of the DYNAPLOUS 81 program is that of the linear theory.The tug- barge assembly is regarded as a rigid package. The combined movements are very important with respect to the relative displacements.

The rigidity matrix of the connection is the- refore assumed as practically infinite in a first step. This permits to solve the hydrodyna- mical problem for a single body.

The therefore try to find out the effects due to a sinusoidal swell,said of low amplitude, single float.

The program calculates the forced reply of the same frequence as that of excitation. The movements are also assumed of low amplitude and what we only do is to set up the conditions at Che limits on the frame said as "medium" which is the frame corresponding to the ship at rest (hydrostatic equilibrium).We therefore obtain the linear equations of the movements.

Thereby;it is useless to try to find some- thing else than a linear representation of the connection.

By the way,we shall note the importance of the assumption of linearity on the hydrostatic matrix.

Each bearing point (or cushion) of the con- nection, assumed as punctual, is modelized by a linearized displacement stiffness matrix gi- ven in a local frame.

No damping term relating to the connection is taken into account. This point could .make the object of further studies.

In the rigid connections, the relative tug- barge displacements being very weak due to its design, the problems relative to the friction on the bearing may be disregarded.

We therefore see that the assumptions made on the connection do not restrict the use of the program to only the calculations of the for- ces on hydraulic cushions. It is even possible to thus take into account the local stiffness of the structure but which also need to be li- nearized.

III. Hydrodynamic problem

III.1. Notations and hypotheses

Let S be the surface of the hull (tug and barge together) and $\vec{n}$ its external normal vec- tor.

SL is the free surface, C is the waterline and D is the whole fluid domain.

(0,x,y,z) is a moving system of axis binded to the mean position of the two bodies which translates with the uniform velocity $U.\vec{i}_x$ in a fixed frame. Oscillations are defined by the velocity $\vec{V}$ (t) and -the angular velocity vec- tor $\vec{\Omega}$ (t).

Incident wave is characterized by its inci- dence 8, its pulsation a and its amplitude a.

We shall assume the following hypotesises

(i) The fluid is almost perfect*, isovolume and its flow is irrotationnal ;
(ii)The incident wave has a small degree of steepness ;
(iii)The motions of the two bodies are small around their mean position.

* The general equation of the almost perfect fluid is $\frac{1}{\rho}$ grad $p = \vec{F} - \vec{\gamma} - 2\varepsilon \vec{V}$
where ε is a very small positive time constant.

III.2. Boundary problem

These hypothesises imply the existence of a velocity potential function Φ (M;t) in D. In the moving frame, the absolute potential is the solution of the following boundary problem (to collate 4)

$$\Delta\Phi(M;t)=0 \quad \forall M \in D \tag{1}$$

$$\frac{\partial^2}{\partial t^2}\Phi(M;t) - 2U\frac{\partial^2}{\partial t\partial x}\Phi(M;t) + U^2\frac{\partial^2}{\partial x^2}\Phi(M;t) + 2\varepsilon\frac{\partial}{\partial t}\Phi(M;t) - 2U\varepsilon\frac{\partial}{\partial x}\Phi(M;t) + g\frac{\partial}{\partial z}\Phi(M;t) = 0 \quad \forall M \in SL \tag{2}$$

$$\lim_{z\to-\infty}\Phi(M;t) = 0 \quad \forall M \in D \tag{3}$$

$$\frac{\partial}{\partial n}\Phi(M;t) = \left[\vec{U} + \vec{V}(t) + \vec{\Omega}(t)\wedge\vec{OM}\right].\vec{n} \quad \forall M \in D \tag{4}$$

$$\lim_{|\vec{OM}|\to\infty}\left[\Phi(M;t) - \Phi_I(M;t)\right] = 0 \quad \forall M \in D \tag{5}$$

$$\Phi_I(M;t) = -\frac{ag}{\sigma}e^{k_o z}\cos\left[k_o(x\cos\beta + y\sin\beta)-\sigma t\right] \quad \forall M \in D \tag{6}$$

$$\omega = \sigma - Uk_o\cos\beta \quad \text{with } k_o = \frac{\sigma^2}{g} \tag{7}$$

Let Φ (M;t) be the velocity potential function defined by

$$\Phi_p(M;t) = \Phi(M;t) - \Phi_I(M;t) \tag{8}$$

Φ_r (M;t) must satisfy equations (1),(2),(3) and (5), and the equation (9) can be substitued to (4)

$$\frac{\partial}{\partial n}\Phi_p(M;t) = \vec{U}.\vec{n} - \frac{\partial}{\partial n}\Phi_I(M;t) + \left[\vec{V}(t)+\vec{\Omega}(t)\wedge\vec{OM}\right].\vec{n}$$

The different parts of the right-hand side characterize respectively :
(i) Neumann kelvin problem, the solution Φ (M;t) of which is not dependent on the time in the re lative frame. Tus induces constant loads on the hull, the mean position of which is changed but this problem can be traited separately,
(ii) Diffraction problem, the solution of which is Φ_D(M;t). Φ_I(M;t) and Φ_D(M;t) excite the ships with a sinusoidale load or encounter pulsation w. Thus, the bodies perform sinusoidal oscillations of pulsation around their mean position,
(iii) Radiation problem in the six modes. Solutions Φ_{Ri} (M;t) of these are linear functions of twelve constants (six amplitudes and six phases) We can solve the radiation problems for only six movements without phases, and we can deduce from same the results for a phase of $\pi/2$.

Therefore, we now only have to solve seven simple boundary problems : one for the diffraction and six for the radiation (one in each mode).

III.3. Determination of the motions

After having computed Φ_D(M;t) and Φ_{Rj}(M;t) $j\in[1,6]$ we can compute the pressure on the hull with lagrange linearized formula. Thus we obtain hydrodynamic coefficients and forces.

The Newton equation gives us a linear system of twelve equations whereas the motions of the two bodies together are the twelve unknowns.

III.4. Integral equation for the distribution of sources

The application of the third Green formula and the introduction of the Green function G(M,M';t), give us an integral formulation of the potential function.

If we consider the particular case where the potential is continued and the normal velocity $\frac{\partial}{\partial n}\Phi$ (M;t) only is discontinued on the boundary, thus we obtain the following expressions.

$$-\frac{1}{4\pi}\iint_S\left[\sigma^*(M')G^*(M;M')-\sigma^{**}(M')G^{**}(M,M')\right]dS(M') - \frac{U^2}{4\pi g}\int_C\left[\sigma^*(M')G^*(M;M')-\sigma^{**}(M')G^{**}(M,M')\right](\vec{n}'.\vec{i}_x)dy' = \phi^*(M)$$

$$-\frac{1}{4\pi}\iint_S\left[\sigma^{**}(M')G^*(M,M')+\sigma^*(M')G^{**}(M,M')\right]dS(M') - \frac{U^2}{4\pi g}\int_C\left[\sigma^{**}(M')G^*(M,M')+\sigma^*(M')G^{**}(M,M')\right](\vec{n}'.\vec{i}_x)dy' = \phi^{**}(M)$$

formulae in which σ (M;t) is the discontinuity of the normal velocity with

$$\sigma(M;t) = \sigma^*(M)\cos\omega t + \sigma^{**}(M)\sin\omega t \tag{12}$$

$$\Phi(M;t) = \phi^*(M)\cos\omega t + \phi^{**}(M)\sin\omega t \tag{13}$$

$$G(M,M';t) = G^*(M,M')\cos\omega t + G^{**}(M,M')\sin\omega t \tag{14}$$

Writing the normal velocity in each point of the hull, we obtain an integral equation where the unknown is the discontinuity σ (M;t)

Thus the superficial distribution of singularities, kinematically equivalent to the hull, is σ (M;t). This distribution of sources satisfies the following integral equations.

$$-\frac{1}{4\pi}\iint_S\left[\sigma^*(M')\frac{\partial}{\partial n}G^*(M,M')-\sigma^{**}(M')\frac{\partial}{\partial n}G^{**}(M,M')\right]dS(M') - \frac{U^2}{4\pi g}\int_C\left[\sigma^*(M')\frac{\partial}{\partial n}G^*(M,M')-\sigma^{**}(M')\frac{\partial}{\partial n}G^{**}(M,M')\right](\vec{n}'.\vec{i}_x)dy' + \frac{1}{2}\sigma^*(M) = \frac{\partial}{\partial n}\phi^*(M) \tag{15}$$

$$-\frac{1}{4\pi}\iint_S\left[\sigma^{**}(M')\frac{\partial}{\partial n}G^*(M,M')+\sigma^*(M')\frac{\partial}{\partial n}G^{**}(M,M')\right]dS(M') - \frac{U^2}{4\pi g}\int_C\left[\sigma^{**}(M')\frac{\partial}{\partial n}G^*(M,M')+\sigma^*(M')\frac{\partial}{\partial n}G^{**}(M,M')\right](\vec{n}'.\vec{i}_x)dy' + \frac{1}{2}\sigma^{**}(M) = \frac{\partial}{\partial n}\phi^{**}(M) \tag{16}$$

The right-hand sides of these equations are given by the following formulae

$$\frac{\partial}{\partial n}\,\phi_D^{*}(M) = -\frac{\partial}{\partial n}\phi_I^{*}(M);\quad \frac{\partial}{\partial n}\,\phi_D^{**}(M) = -\frac{\partial}{\partial n}\,\phi_I^{**}(M) \quad (17)$$

$$\frac{\partial}{\partial n}\,\phi_{Rj}^{*}(M) = \vec{V}_{Ej}^{*}.\vec{n} \quad ; \quad \frac{\partial}{\partial n}\,\phi_{Rj}^{**}(M) = \vec{V}_{Ej}^{**}.\vec{n} \quad (18)$$

The integral equations will now be solved with a method of discretization. Let the hull be discretized with N panels. We shall assume that on each panel the density of singularity is constant. Thus we have a linear system of 2N equations with 2N unknowns (N' σ^{*} and N σ^{**}).

III.5.Green function

The Green function of the diffraction-radiation whit forward speed is given by the following integral expression (to collate 4 and 13)

$$G(M,M';t) = \left[\frac{1}{|MM'|} - \frac{1}{|MN'|}\right]\cos\omega t \quad (19)$$
$$-\lim_{\varepsilon\to 0^+}\frac{1}{\pi l}\,\mathrm{Re}\left\{e^{i\omega t}\int_{-\pi/2}^{\pi/2}d\theta\int_0^{\infty}\frac{e^{K(Z+Z'+i\Omega)}\,K\,dK}{(\tilde{\omega}-FK\cos\theta)^2-K-2i\tilde{\varepsilon}(\tilde{\omega}-FK\cos\theta)}\right\}$$
$$-\lim_{\varepsilon\to 0^+}\frac{1}{\pi l}\,\mathrm{Re}\left\{e^{-i\omega t}\int_{-\pi/2}^{\pi/2}d\theta\int_0^{\infty}\frac{e^{K(Z+Z'+i\Omega)}\,K\,dK}{(\tilde{\omega}+FK\cos\theta)^2-K+2i\tilde{\varepsilon}(\tilde{\omega}+FK\cos\theta)}\right\}$$

expression in which

$$X = \frac{x}{l}\quad ;\quad Y = \frac{y}{l}\quad ;\quad Z = \frac{z}{l}$$
$$\tilde{\omega} = \omega\sqrt{\frac{l}{g}}\quad ;\quad \tilde{\varepsilon} = \varepsilon\sqrt{\frac{l}{g}}\quad ;\quad F = \frac{U}{\sqrt{gl}}$$

and $\Omega = (X-X')\cos\theta + (Y-Y')\sin\theta$

N' is the point symmetrical to M' in relation to the free surface.

This Green function satisfy the equations (1) (except in M'), (2), (3) and (5)

IV.EQUATIONS OF THE MECHANICS

IV.1.Combined movements

The movements of the tug and barge assembly are very wide with respect to the relative displacements. The regidity matrix of the connection is assumed as infinite for the resolution of the equations of the movement- Therefore

$$\eta_T - \eta_B = E \ll \eta_T, \eta_B \quad (20)$$

p_T . Tub displacement vector

p_B • Barge displacement vector

E • Relative displacement vector

The equation of the mechanics at 6 degrees of freedom is written for the assembly. All the torques are reduced to *A* same point 0 (Whatever a priori) :

$$\left[M+MA\right]_{T+B}\ddot{\eta} + TA_{T+B}\,\dot{\eta} + TH_{T+B}\,\eta = \left[F_I + F_D\right]_{T+B}$$

The matricial system represents 12 equations with 12 unknown quantities (6 amplitudes and 6 phases)

$\eta = \eta_1,\ldots\eta_6$ Vector of the combined movements

$\eta_i = \eta_i^{*}\cos\omega t + \eta_i^{**}\sin\omega t$

The η_i^{*} and η_i^{**} are the unknown quantities

M inert mass matrix
MA added water mass matrix
TA Damping terms matrix
TH Hydrostatic matrix

F_i column vector of the Froude-Krilov excitation forces

F_D column vector of the diffraction excitation forces

The resolution of the equations (21) enable to fix vector η after having solved the hydrodynamical problems.

IV.2.Calculation of the connection efforts torque

Need only be to fix the connection torque between the barge and its tug, write down the equation of the mechanics for one of the two floats.

For the tug, we thus have :

$$\left[M+MA\right]_T\ddot{\eta} + TA_T\,\dot{\eta} + TH_T\,\eta - \left[F_I + F_D\right]_T = F_{B\to T} \quad (22)$$

$F_{B\to T}$: Torque of the connection efforts exerted by the barge on the tug.

The equation (22) permits to calculate $F_{B\to T}$ as we know the displacement vector η

IV.3.Calculation of the relative movements

By virtue of relation (20) we now have a simple problem of decomposition of a torque on a elastic bearing system.

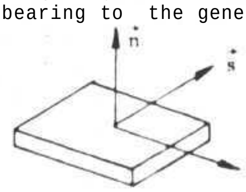
$$F_{B\to T} = -K_{eq}\,E \quad (23)$$

K_{eq} : Equivalent stiffness matrix of the flexible connection of the assembly reduced to point 0.

The connection is modelized by punctual flexible bearings points each one of the n bearings, an (Fig. I) is defined by :

- $\vec{OA}_n$: position vector in the general frame
- k_n : Local displacement stiffness matrix
- P_n : Matrix of passage from the local frame of the bearing to the general frame.

Figure I : Bearing (cushion)



$$K_n = \begin{bmatrix} k_{xx} & k_{xy} & k_{xz} \\ k_{yx} & k_{yy} & k_{yz} \\ k_{zx} & k_{zy} & k_{zz} \end{bmatrix} \qquad P_n = \begin{bmatrix} l_x & s_x & n_x \\ l_y & s_y & n_y \\ l_z & s_z & n_z \end{bmatrix}$$

From these data, it is possible to calculate the stiffness matrix equivalent to point 0 :

$$K_{eq} = \begin{bmatrix} \sum_n K_n & \sum_n K_n B_n^T \\ \sum_n B_n K_n & \sum_n B_n K_n B_n^T \end{bmatrix}$$

with $K_n = P_n \; k_n \; P_n^{-1}$

and B_n : Vectorial product matrix of $\vec{OA}_n$ such as : $\vec{OA}_n \wedge \vec{E}_n = B_n \; \vec{E}_n$

It is therefore possible to calculate $\vec{E}$, relative displacement vector, with equation (23).

IV.4. Calculations of the efforts on each bearing

Knowing the relative displacement $\vec{E}$ in the general frame, it is possible to calculate the displacements on each bearing, and then the local efforts.

We have : $\vec{d}_n = P_n^{-1} \; \vec{E}_n$

$\vec{d}_n$: local displacement vector on bearing n

$\vec{F}_n$: local force vector on bearing n

Vectors $\vec{F}_n$ and $\vec{d}_n$ have three components as we simulate only punctual bearings.

V. RESULTS AND COMPARISON WITH MODEL TESTS

V. 1 .Choice of the tests

Our comparison between calculations and basin tests is based on the report intented for Maritime Administration, U.S. Department of Commerce. "Experimental research relative to improving the hydrodynamic performance of ocean going tub/barge systems".

This report is composed of four chapters and we mainly used Part B "Seaworthiness experiments" by Grant A. Rossignol.

In this report, experiments have been carried out on two pushed barge concepts
- first concept : rigidly connection
- second concept : articuled connection

In this paper we are only dealing with the first concept.

The tug and barge together shape a 60.000 T ship with a block coefficient of C_B - 0.75. The main characteristics of the pusher, of the barge and of the tug-barge system are summed up in table I.

In the model tests the tug and the barge was joined together at three points (S* 1,2 and 3 in Fig. 2). The centerline block gage arrangement

Particular	Symbol	Tug	Barge	T/B
Length P.P	LPP (m)	56,4	–	232
Length waterline	LWL (m)	53.83	204	235,9
Beam max.	B (m)	23,5	32	32
Draft mean	T (m)	10,67	10,67	10,67
Displacement	Δ (Tf)	5892	55068	60960

Figure 2 shows the tug-barge system.

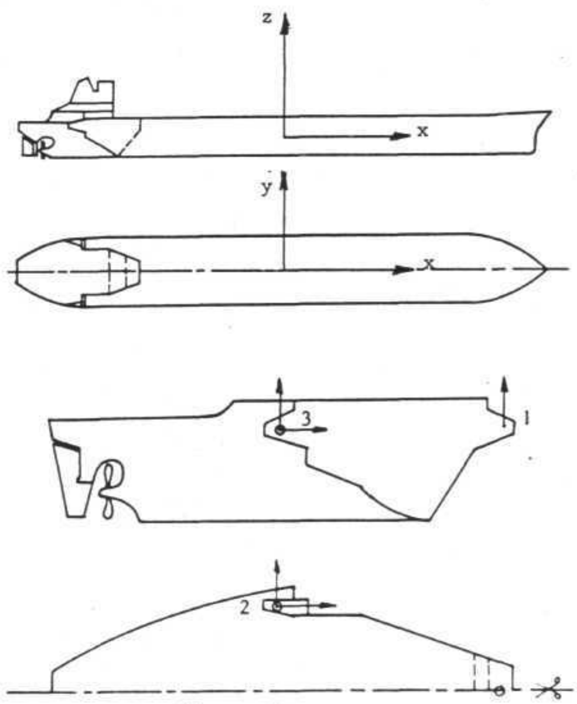


Figure 2

(N° 1) provides only vertical restraint. The other connections points are port (N°2) and starbvard (N°3) pins which provide total translatory restraint.

The locations of the three connection points and the accelerations measurement point are given in table 2 •

The deatiled informations about model, connection, instrumentation are given in the mentionned report of the David W. Taylor Naval Ship Research and Development center.

The measured variables necessary to our comparisons are :
- Heave of the overall concept LCC ; Z (fig 3)
- Pitch of the overall concept ; (fig 4)
- Roll of the overall concept ; (fig 5)
- Vertical acceleration of the tug pilot house; ZPH

- longitudinal acceleration of the tug pilot house ; XPH
- Vertical force between the tug and barge at the centerline (N°l) ; FVCL
- Vertical force between the tug and barge at the starboard pin (N°3) ; FZS
- Vertical force between the tug and barge at the port pin (N°2) ; FZP
- Longitudinal force between the tug and barge at the starboard pin (N°3) ; FXS
- longitudinal force between the tug and barge at the port pin (N°2) ; FXP
- Transverse force between the tug and barge at the starboard pin (N°3) ; FYSC
- Transverse force between the tug and barge at the port pin (N°2) ; FYP

From this measurements, the following results was computed by NSRDC :
- total vertical pin force between the tug and barge : FVPN : FZS + FZP (Fig. 9)
- total longitudinal pin force between the tug and barge : FLPN : FXS + FXP (Fig. 10)
- total transverse pin force between the tug and barge : FTPN : FYS + FYP (Fig. 11)
- Axial moment about the longitudinal axis between the tug and the barge (Roll moment) : ML = d/2 x (FZS - FZP) (Fig. 12)
- Axial moment about the vertical axis between the tug and barge (Yaw moment) : MV = d/2 (FXS - FXP) (Fig. 13)

Where d is the transverse distance between pin locations (d - 22,42 m)

	Connection points			A celeration
	1 (m)	2 (m)	3 (m)	tug pilot house
X to A. P.	54.2	26.6	26.6	46.4
X to C.L.	0.	+ 11.2	- 11.2	0.
Z to B.L.	16.05	12.6	12.6	25.

The comparisons of the previous results and the computer program DYNAPLOUS 81 are only based on the regular wave experiments.

The tug barge system has been tested for a forward speed of 16 knots (the Froude number is equal to 0.171) at headings of 0, 45, 90, 180 and 225 degrees.

In the computer program the immersed part of the pusher has been discretized with 18 panels on the half hull and the immersed part of the barge has been discretized with 44 panels on the half hull. The part of the two hulls between the bodies has not been taken into account to compute the hydrodynamic coefficients,
The hydrostatic particulars were calculated for the tug and barge separately and together equally for the wetted part of the tug.
The three connection joints of the tug/barge are modelised by elastic if bearing strings in the same directions as those of the block gage arrangement* (i.e chapter 4). The stiffness characteristics are not important as the whole connection is isostatic. The program has run at the ACB computer center on a UNIVAC 1100/10.

V:2. Motions of the two bodies together

We present here only three movements (heave pitch and roll) because the experimental results was given only for these modes. Nevertheless the computer program gives the results for all the six modes in the same theoretical conditions.

The results of the movements are nondimensional. The amplitude of the heave response is divided by the wave amplitude a, and the amplitudes of the pitch and roll responses are divided by the product of the amplitude and the wave number k_o.

Figure 3 shows the tug-barge heave transfer function at various headings. This results gives the proof of a good agreement between the experiments and our numerical method for all the angles of incidence. Perhaps the difference between the results when β is equal to 225° and little λ/L , comes from natural frequency response which is always more stiff for the nymerical results than for the experimental results.

Figure A shows the tug-barge pitch transfer function at various headings. This results show a fairly good accuracy in our results with experimental mesurements except for the angle of incidence β equal to 45°. The same remark, than the aforesaid, can be made here because the difference is localized around the natural frequency.

Figure 5 shows the tug-barge roll transfer function at various headings. Naturally, the difference between the motions calculated and mesured is sensible for two reasons. Firsthy the non - linear effects are not negligeable in this case, because the hemichrone roll appears for λ' equal to $(3/2)^2$ λ (where λ is the wave-length of the natural frequency). Secondly the viscous effect are very important for this mode ; the computer program can take into account damping coefficient arising from the logarithmic decrement calculated from model tests result. Nevertheless we have not use this possibility for the present calculations.

V.3. Accelerations of the tug pilot house

We present in this chapter the vertical acceleration and the longitudinal acceleration of the tug pilot house.

The results of the accelerations are nondimensional. The amplitude of these accelerations are divided by the produit of the amplitude of the wave and the square of the encounter angular frequency.

Figure 6 shows the tug pilot house vertical a eleration transfer functions at various headings. This results proves a good correlation between the experiments and the numerical method except for the angle β equal to 45°.that is a consequence of the results of the pitch motion and the same natural frequency remarks mentioned above is valid.

Figure 7 shows the tug pilot house longitudinal acceleration transfer function at various headings. This results do not permit to conclude

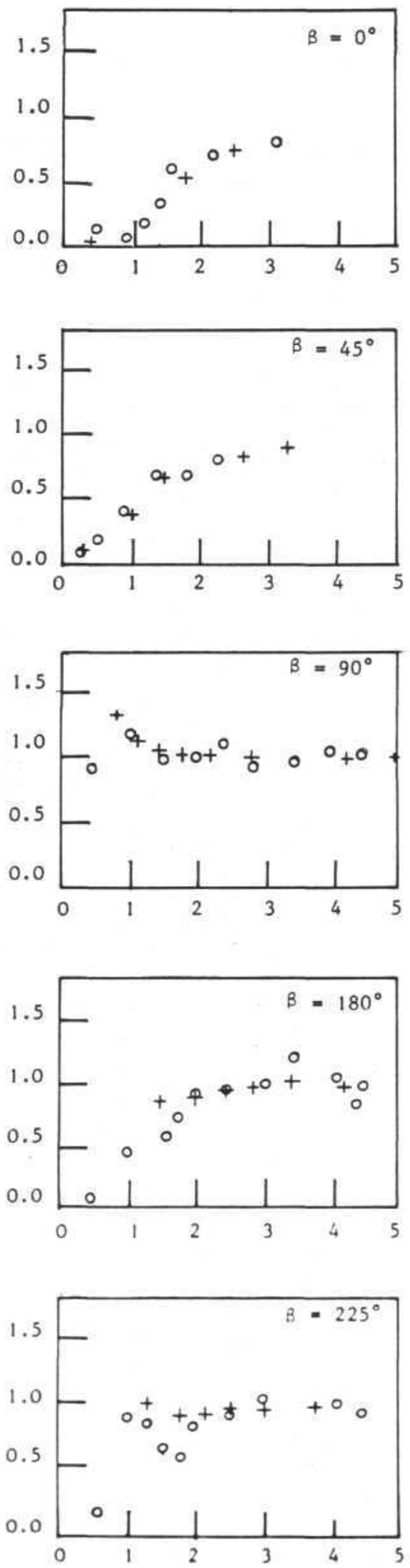


Figure 3 - Nondimensional Tug/Barge Heave Transfer Function versus Wavelength/Barge Length for the Tug/Barge at Various Headings.
o experimental results
+ numerical results

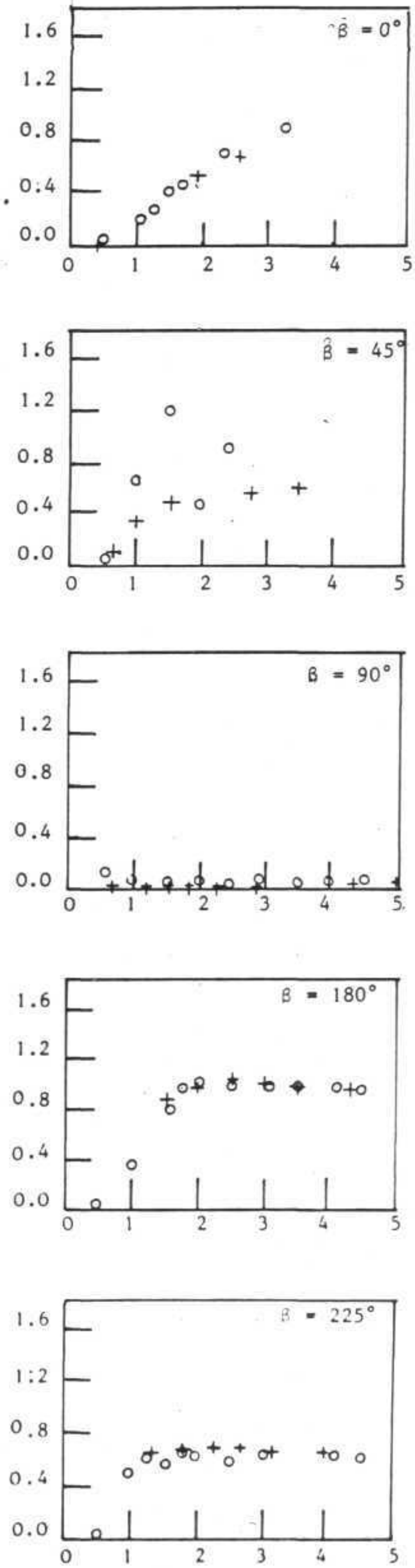


Figure A - Nondimensional Tug/Barge Pitch Transfer Function versus Wavelength/Barge Length for the Tug/Barge at various Headings
o experimental results
+ numerical results

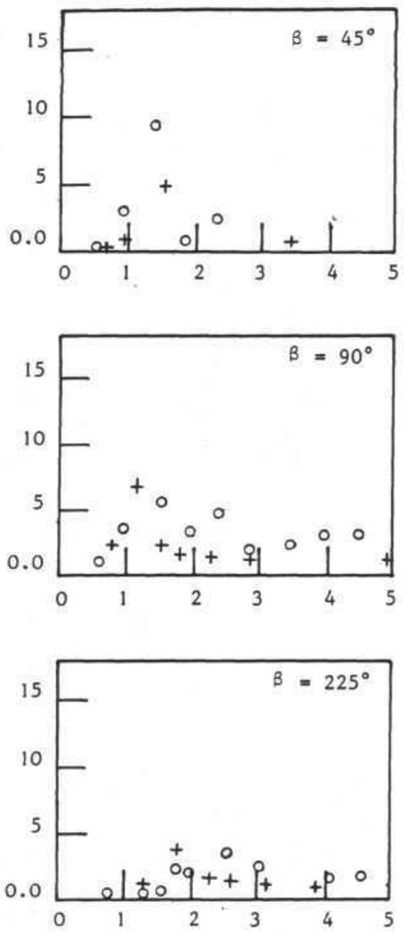


Figure 5 - Nondimensional Tug/Barge Roll transfer Function versus Wavelength/Barge Length for the Tug/Barge at Various Headings.

o experimental results
+ numerical results

because the values are very small and the precision of the plots is not fairly good.

V.4. Forces between the tug and barge

We present here the forces between the tug and barge for all the modes except the transversal moment.

The results of the forces and moments are nondimensional. The amplitude of the forces are multiplied by the barge lenght and divided by the product of the weight of the barge and the wave amplitude. The amplitude of the moments are divided by the product of the weight of the barge and the amplitude of wave.

Figure 8 shows the vertical centerline force transfer function at various headings. The comparison between the numerical results and the experimental results shows a fairly good relationship for this forces.

Figure 9 shows the vertical pin face transfer function at various headings. These results give the proof of a very good agreement between the experiments and calculations.

Figure 10 shows the longitudinal pin force transfer function at various headings. At first sight, the results seem to be very bad.. Nevertheless the experimental tests for this component are very difficult because the longitudinal force is in the same direction as that of the forward speed and the problems of speed stability and added wave resistance are very important in this case as the model is self propelled.

Figure 11 shows the transverse pin force transfer function at various headings the comparison between the experiments and calculations is very good except for the natural frequency of rool at 90° of heading. In quartering sea, the difference between the test and the numerical points is probabily due to the steering control. In fact it is impossible to take into account, in the computer program, the directional instability of a real ship running in following seas.

Figure 12 shows longitudinal axial moment transfer function at various headings. Comparaison between experiments and calculations is very good for 90° and 225° of heading. A great difference appears for a angle of incidence of 45° in the range of λ/L_B around 1, as the for the previous results at same heading.

Figure 13 shows vertical axial moment transfer function at various headings. This results do not permit to conclude because the values are very small and the precision of the plots is not fairly good.

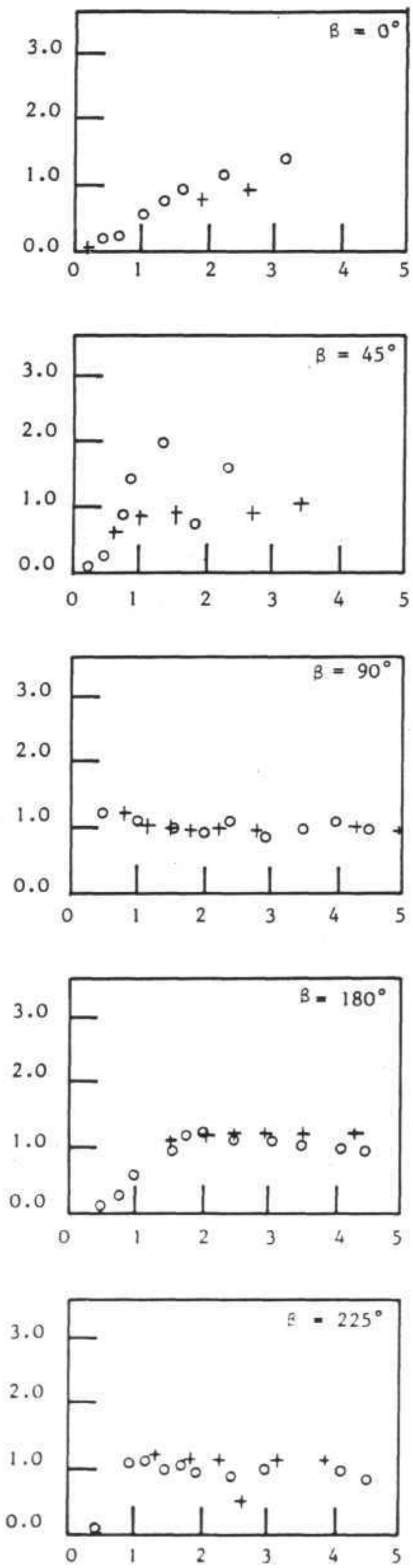
VI - ANALYSIS OF THE RESULTS

The previous comparisons between experimental and numerical results show a good relationship. The sensible differences occur for longitudinal pin force transfer function (fig. 10) and for the various peak responses throughout the experimental results.

We present hereunder some comments to explain these differences :
The real tug is fit into the barge afterbody with a 6 inch gap full scale. But the forebody shapes of the tug model are made to wedge tightly into the notch of the barge model (see. afore mentionned report phase 11 ; part A page 2)

This remark is very important for two reasons :
- firstly we must take the correct part of the hydrostatic force on the tug model.
We have made different calculations with the hydrostatics of the entire tug or only the after body wetted surface of the tug.
We have taken into account the problem of the hydrostatic couplings between two join bodies
Some differences may occur because we do not know the exact configuration of seakeeping tests.

- Secondly in full scale the laws of the pressure variations in the clearance between tug and barge are very difficult to modelise but perhaps these effects are of second order in the



Figurt 6 - Nondimensional Tug Pilot House Vertical acceleration Transfer Function versus wavelength/Barge Length for the Tug/Barge at Various Headings.
o experimental results
+ numerical results

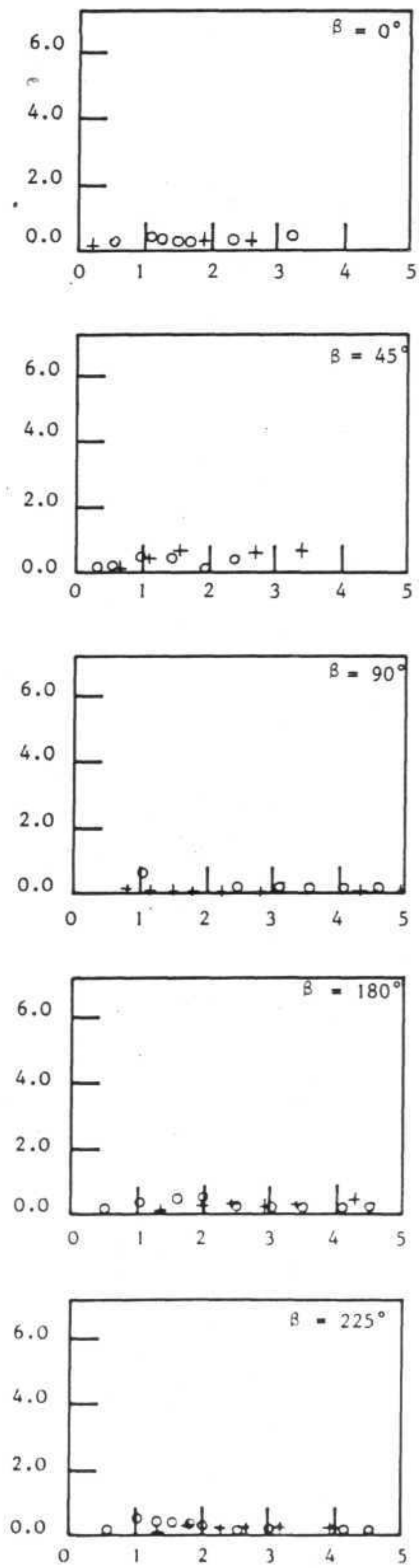


Figure 7 - Nondimensional Tug Pilot House longitudinal Acceleration Transfer function versus Wavelength/barge Length for the Tug/Barge at Various Headings.
o experimental results
+ numerical results

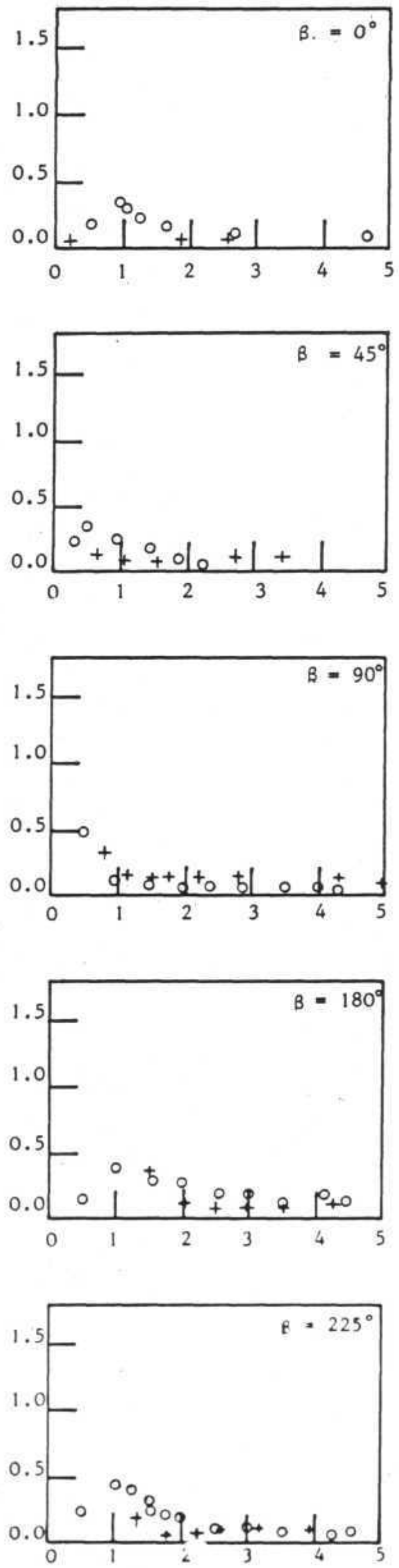


Figure 8 - Nondimensional Vertical Centerline Force Transfer Function Versus Wavelength/Barge Length for the Tug/barge at Various Headings.

o experimental results
+ numerical results

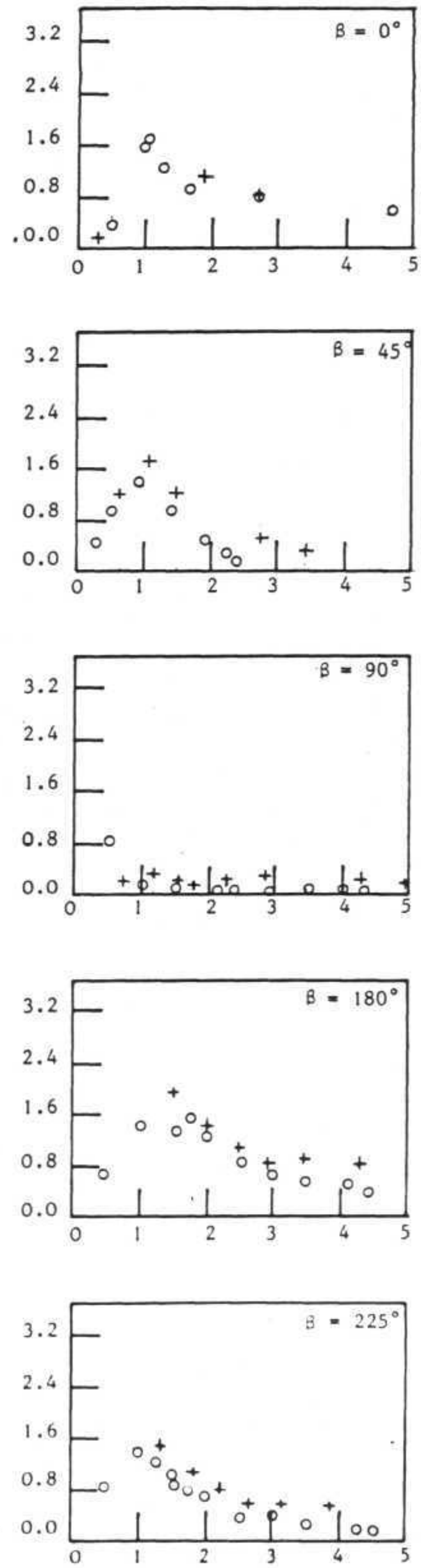


Figure 9 - Nondimensional Vertical Pin force Transfer Function versus Wavelength/Barge Length for the Tug/Barge at Various Headings.

o experimental results
+ numerical results

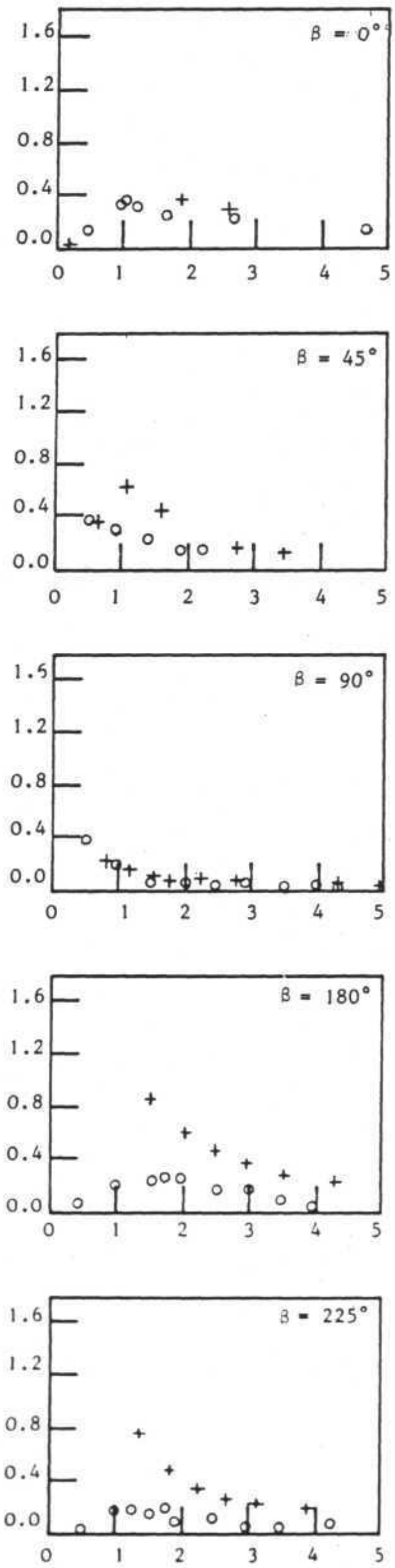


Figure 10 - Nondimensional Longitudinal Pin Force Transfer Function versus Wavelength/Barge Length for the Tug/Barge at Various Headings

o experimental results
+ numerical results

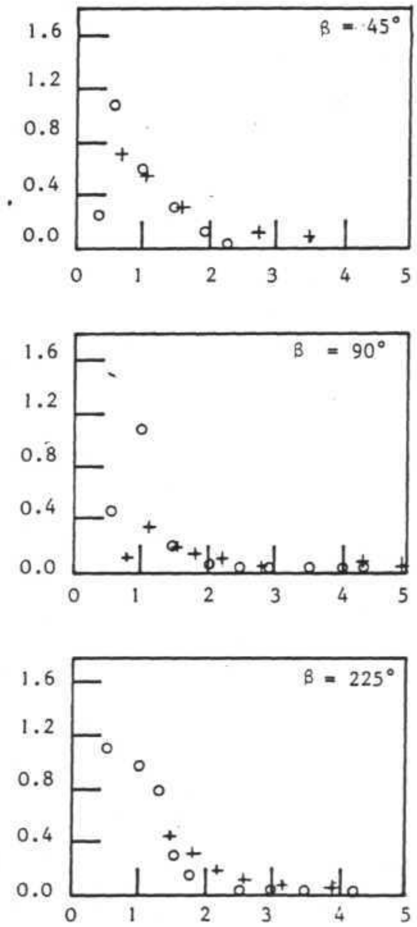


Figure 11 - Nondimensional Transverse Pin Force Transfer Function versus Wavelength/ Barge Length for the Tug/Barge at Various Headings

o experimental results
+ numerical results

calculation fo the forces.

An important difference between model tests and calculations are the different non-linearity effects. These are evident for the hydrostatic in particular for those on the tug only because its aft location.
The non-linear damping effects (for example due to viscosity) are not corrected in the program. This explains the difference in peak responses on roll.

The model was self-propelled and steered. This introcudes important differences between the conditions of the tests and calculations.

- firstly we have *no* taken in account the whole resistance at the forward speed. That could be made with a constant torque simulating the propulsation forces.

- Secondly we do not calculate the added resistance of ship with forward speed in waves. Studies about this 3 D problem have already starded at E.N.S.M.

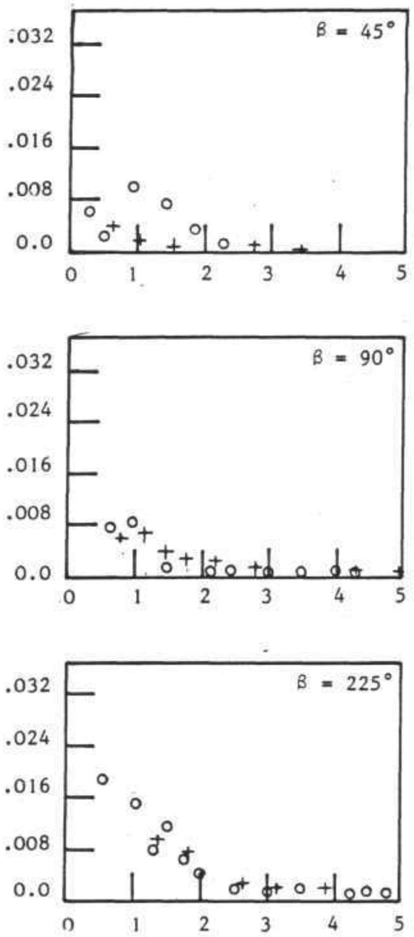


Figure 12 - Nondimensional Longitudinal Axial (Roll) Moment Transfer Function versus Wavelength/Barge Length for the Tug/Barge at Various Headings

o experimental results
+ numerical results

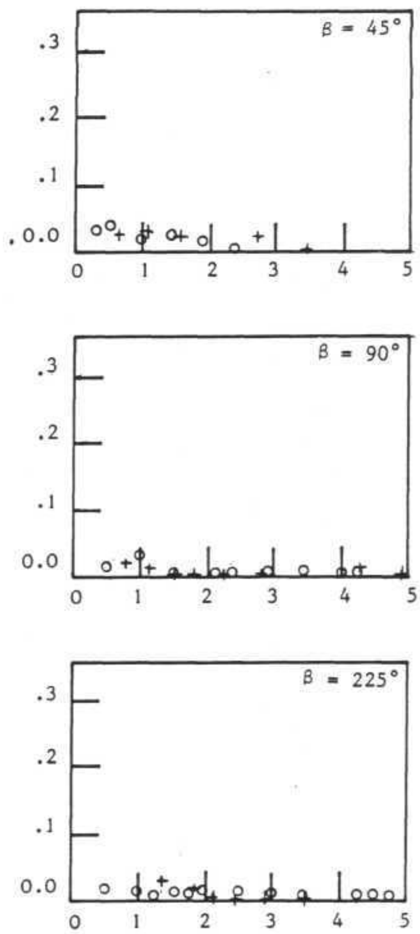


Figure 13 - Nondimensional Vertical Axial (Yaw) Moment Transfer Function versus Wavelength/Barge Length for the Tug Barge at Various Headings.

o experimental results
+ numerical results

- Thirdly the variation in immersion (or emmersion ?) of the propeller introduce distorsion ef fects on the measured forces.

- Fourthly the directionnal course stability in self-steered tests present serious difficulties particularly in following seas (the same problem is known in full scale).
These instabilities involve supplementary forces in roll and yaw.

All these reasons explains certain differences between measurements and calculations specially for the longitudinal pin force (see Fig. 10)

An indirect effect of the non-linearities are perhaps the hemichrone roil which appear in certain figures. In particular to our mind the peak responses at 45 degrees of heading forpitch (Fig.4), roll (Fig.5). tug pilot house vertical acceleration (Fig.6), vertical transversal forces and in the moments (Fig.8, 9, 10,11,12,13) are due to this previously mentionned effect.

This phenomenon due to the change in metacentric height with time is increased by the lack of bilge keels on the model which involve a small roll damping. In addition a coupling bet ween heaving and rolling exist (See. Fig.3 ; secondary peaks for 0,45 and 90 degrees), this accentuate the variation in metacentric height and thus the hemichrone roll.

In connection with this phenomenon due to non-linearly roll responses, two other effects contribute probably to the differences between experiments and calculations at 45° ; of heading

- firstly the induced non-linearity in pitch (the peak response in Fig.4 is very high)

- secondly the maximum excitation forces occur for λ / LWL =1 or λ / LB = 1, 14 and are in the same frequency game than t'ne hemichrone roll.

In spite of these remarks on the whole the agreement between theory and experiment is correct.

VII. POSSIBLE EXTENSION OF THE METHOD

The connection hydraulic cushions have in fact non linear stiffness characteristics.

In this program the approximation is made by linearizing these characteristics around a me— dium point of operation.

We envisage to solve the mechanical part thanks to the ised simulation program. In this case we shall be in a position to follow-up the evolution in course of time of the forces on the bearings whatever their ty of stiffness charac- teristics.

The equations of the mechanics as non linear should be written down again.

Like wise, the non linear hydrostatic cha- racteristics that we already possess, have to be integrated with this program.

The resolution of the hydrodynamic problem will, unfortunately remain linear till new theo retical developments are obtained.

An other more simple extension is the modeli zation of flexible bearings at 6 degrees of free dom.

So, this results in a local stiffnees matrix of 6 x 6, and not any more of 3 x 3.

In this case, the bearings are not any lon- ger punctual and it could forthermore be possi- ble to envisage to take the local frictions in- to account.

VIII. ANTICIPATED EXTENSIONS

A second program is envisaged to solve the problem of the tug-barge for which certain re lative movements in relation to the barge have been released.

In this case, equation (20) is not any more true and we must solve both the hydrodynamic problem and the coupled equations of the twin body mechanics.

The n body hydrodynamic with forward run- ning speed is already solved at E.N.S.M.

The n body mechanics on elastic bearing is written, remaining in the assumption of lineari ty. This should permit to solve the problem of calculation of the efforts in the case of an assembly of several barges articulated toge- ther.

Besides, in the same assumptions, we could solve the problem of traditional towing with one or several floats.

The problems of the non linear dynamic cha- racteristics of the connections should not be underestimated.

In a first step, it will be possible to haw recourse to simulation models such as indicated in 7.

In this paper we have not shown any compari- son of calculations/basin tests in random waves. This will be done later on.

Finally, thanks to the technology of the ACS Hydraulic cushions, it will be possible ot make calculation/sea trials comparisons providing that we have, at the same time as the measure- ments of the efforts, a correct evaluation of the meteoceanic conditions.

IX. CONCLUSION

The present comparisons between experimen- tal results and numerical results shows that the computer program DYNAPLOUS 81 permits to forgeast, with a good accuracy, the first order motions of a floating body, the accelerations at any point and the forces between two parts of this body.

The qualities of the numerical results given by the computer program are well adapted to stu dy the forces in each connections point between the two parts of a rigid ocean-going tug barge system.

The results of mesurement on real tug-barge system may give us the possibility to make new comparisons without scale effects or problems binded with basin tests.

The computer program is now adapted for an industrial use and must allow any development of hydrodynamic studies concerning all the sys- tem of rigid floating bodies at sea.

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