

CHAPTER 221

UTILIZATION OF MOORED VESSELS IN HYDRAULIC MODELS OF HARBORS.

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Abstract

The behavior of the model of a moored vessel in a harbor's hydraulic model (Froude's model) is studied, taking into account the difficulties arising from the fact that modelling a moored vessel requires to incorporate the Reynolds number. Experiments made with the model of a large grain carrier at a 1/175 scale are shown. The horizontal motions of this ship due to the waves were measured when it is moored in a zone in which the length scale along the vertical axis was modified (vertical distortion), and the influence of different depths on these motions was studied.

This study was performed in order to be able to use this ship model for the evaluation of different alternatives of harbor design, in the building of new facilities at a port placed in the Atlantic coast of Uruguay.

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Introduction

The amount of agitation in the hydraulic model of a harbor is a main fact to be taken into account when decisions about harbor design are to be made. The traditional way for the evaluation of agitation is directly measuring wave heights in those places of the berth which are considered important for its performance. Comparison between two harbor designs is done by comparing the wave heights associated to each design, and frequently, design criterium is not reaching certain values of wave height which are accepted as maximum for a safe and economical development of the port's activities. Nevertheless, this method, though useful to compare two harbor designs, does not provide an absolute criterium for the evaluation of the port. The port's ability to fulfill performance requirements must be based on experience about the influence of the action of waves. An absolute criterium for the evaluation of a harbor design must provide the prediction, by means of the model, of the motions to be experienced by the ships which will be moored to the prototype's docks. A good design will be that in which these motions allow cargo handling to be safe and economical. These motions shall also be safe to all the port's facilities. References related to the behavior of vessels' physical models and their utilization to harbor design is found in bibliography about harbor design [2,3,4,5].

Stating the problem

When harbor's modelling is performed to study the behavior of a moored vessel, a similitude problem arises. The harbor's hydraulic models are Froude's models with length scales e_L roughly varying from 1/100 to 1/200, and the modelling of a moored vessel makes it necessary to introduce the Reynolds number. Simulation of the motion of a body subjected to external forces produced by the fluid, the mooring lines and the fenders system must be performed. Oscillatory flows between the ship's hull, the sea bottom and the docks take place and in such flows, power dissipation per unit of surface area is a function of the Reynolds number of the flow. Reynolds number scale (e_{Re}) in a Froude's model is $e_{Re} = e_L^{3/2}$. Considering, for example, $e_L = 1/100$, it results $e_{Re} = 1/1000$. Reynolds number reduction from prototype to model is then very important, and therefore there will be no similitude between the model of a moored ship in an agitation hydraulic model, and its prototype.

As an exact theoretical solution for the complex dissipative process which takes place in the turbulent boundary layer placed between the ship's hull, the sea bottom and the docks, is still not available, the problem must be tackled experimentally. This will be performed taking into account recent results related to the turbulent boundary layer created by a wave. Following Blondeaux [1], it is seen that the dimensionless mean power D dissipated by the boundary layer generated by the wave,

per unit of surface area, usually decreases with the Reynolds number. D is defined as $D = D^*/\rho U_b^{*3}$, D^* being the mean power dissipated by the boundary layer per unit of surface area, and ρ is the water density. The Reynolds number is defined as $U_b^* a_b / \nu$, being U_b^* the maximum velocity of the irrotational wave in the bottom, $a_b = U_b^* t / 2\pi$, t being the period of the wave, and ν the kinematic viscosity of water. In the referred paper it is stated that in hydraulically smooth regime, D can be calculated, by means of the relationship: $D = 0.14 Re^{-0.23}$. Considering now the average total power per unit of surface area in a simple harmonic progressive wave, E^* , this power can be expressed as $E^* = H^2 \gamma / 8t$, where H is the wave amplitude, and t the wave period. Therefore $D^*/E^* = 8U_b^{*3} D / gH^2 T$. If model's and prototype's D^*/E^* values are compared and a D^*/E^* scale is defined it is obtained

$$e_{D^*/E^*} = (D^*/E^*)_m / (D^*/E^*)_p = e_v^3 e_t e_L^{-2} e_{Re}^{-0.23} = e_L^{-0.345},$$

being e_t the time scale, and e_v the velocity scale. Then

$$d(e_{D^*/E^*}) / d(e_L) < 0$$

If this result is applied to a ship moored in a harbor, the conclusion is that the D^*/E^* value in the model is higher than in the prototype. The precedent reasoning shows that if dissipation in the model is not reduced, results obtained about angular and linear displacements of a moored vessel in a hydraulic model of a harbor will be smaller, at prototype's scale, than real displacements.

Experimental solution of the problem.

The installations:

The experiments described in the paper were performed in a hydraulic agitation model*. The ship which was modelled is the largest grain carrier that will be able in the future to operate in the new facilities to be built at this port. The model is shown in figure 1, in working position, with the mooring lines and fenders system described later in this paper. The length scale was $e_L = 1/175$. The main dimensions of the ship's model are:

*built in the Institute of Fluid Mechanics and Environmental Engineering, of the Faculty of Engineering of Uruguay.



Figure 1. Ship's model moored in harbor's hydraulic model.

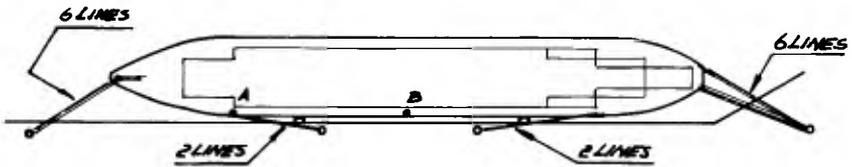


Figure 2. Arrangement of mooringlines in ship's model.

Length between perpendiculars:.....Lppm =1.41m
 Maximum moulded breadth:.....Bm =0.184m
 Displacement (maximum):.....Dm =146N
 Draught (maximum):.....Tm =0.071m
 Displacement (ballast condition):...D'm =75.2N
 Draught (ballast condition):.....T'm =0.039m
 Block coefficient:.....Cb=0.84

In addition to the hull's shape, longitudinal distribution of the weight (longitudinal position of the center of gravity, LCG) was simulated. The model's behavior was studied in ballast condition, and for this case, transverse distribution of weight, and metacentric radius were simulated. Being the time scale $e_t = e_l^{1/2}$, as it happens in Froude's models, the natural period of roll, which is of 8s in the prototype, resulted of 0.605s in the model.

This model was built to study the new breakwaters location in the port adjacent to the city of La Paloma, in the Atlantic coast of the country.

Mooring of the model and fenders system

Figure 2 shows the mooring of the vessel. The lines in the bow and in the stern simulate six synthetic-fibres lines of 80 mm diameter in each case, as required for this kind of ships. The lines on the sides simulate the behavior of two of these 80mm diameter lines, for each mooring. In order to measure the motions of the ship's model, the moorings of bow and stern were pretensioned with a force of 0.137 N, correspondent to 1.23×10^5 N for each line in the prototype. Forces on the lines were measured in the model by joining them to two force transducers which operate with strain-gages. In these transducers, a variable current is obtained when a horizontal force, perpendicular to them, is exerted. When forces vary from 0 to 0.98 N, the intensity of this current vary from 4 to 20 mA, and the variation is linear. Pretensioning of the lines meant therefore an increase of 2.24 mA in the intensity of the current generated at each transducer. The simulation of the mooring lines was performed using textil fibres which have a force-deformation law reproducing, at model's scale, the behavior of real mooring lines. The force-deformation law of several 80 mm diameter lines are shown in figure 3, as function of a reference force F_0 of 0.98×10^5 N in the prototype. In this figure, δ is a relative deformation, expressed in percentage ($100 \times \Delta L/L$), and δ_0 is the relative deformation corresponding to F_0 . With regard to the fenders system, simulation was achieved by using two force transducers of the same kind used to measure the forces on the mooring lines. Two springs were added in parallel to these transducers (figure 4). The elasticity constant of the device formed by each transducer and its spring is 1.23×10 N/m, so that both of them, when placed in the model's docks simulate the behavior of a fenders system adequate for the reception of ships of the size of the one which was modelled.

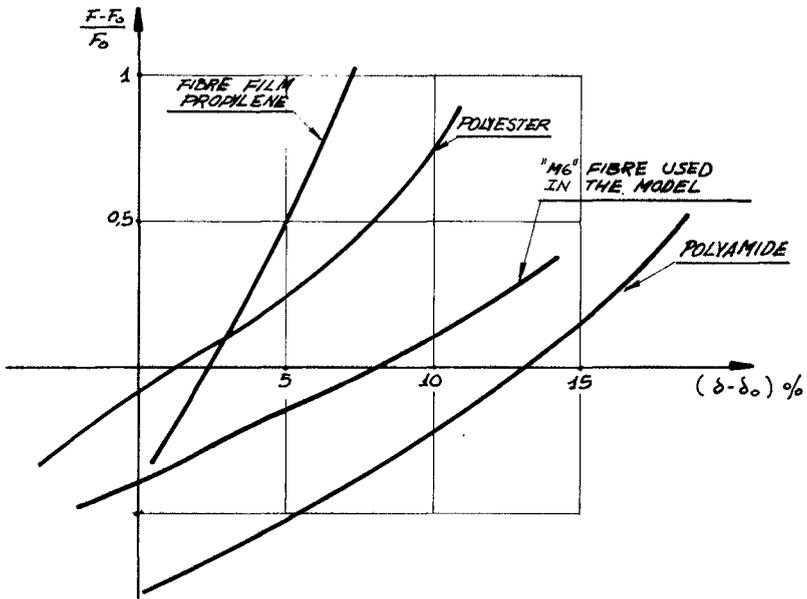


Figure 3. Force-deformation laws of 80 mm diameter lines, and of line used in the experiments.

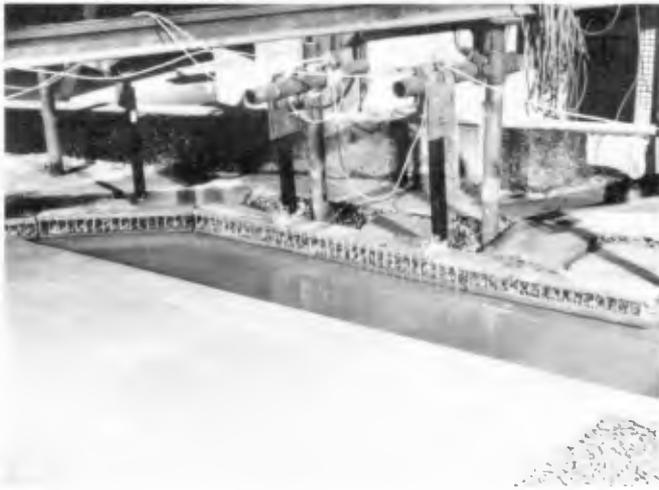


Figure 6. View of the distorted zone.

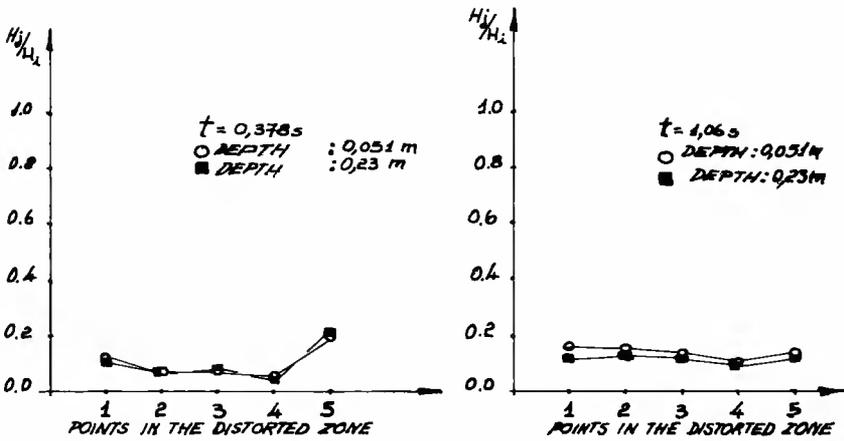


Figure 7. Measurement of wave heights in the distorted for different depths.

Horizontal motions of points A and B (figure 2), were measured visually with the aid of graduated scales

With the aim of reducing power dissipation in the model, and taking into account that power transfer from the mean flow to turbulent fluctuations is proportional to the magnitude of the components of the mean deformation velocities tensor [6], the harbor's model depth in the ship's neighborhood was distorted. The distortion was made in a zone in which reception of large vessels is planned. The area of this distorted zone is 3.72% of the total area of the port, and with regard to wave heights, no change in the behavior of the hydraulic model of the port was detected, either inside or outside the distorted zone, when measuring these wave heights with resistive devices.

In figure 5, the location of this zone is shown, and in figure 6 an overall view of it is provided. In figure 7, wave heights in five points named 1,2,3,4 and 5, in the distorted region are shown. Wave heights are referred to the amplitude of the wave in the generation zone, and they were measured with the normal depth of the port design (without any distortion), and with a depth of 0.23 m in the model, correspondent to the maximum distortion made. In both cases, wave periods in the model were 1.06s, and 0.378s. In figure 8, location of the points above mentioned in the distorted zone, is indicated.

The vertical distortion proportionally reduces the vertical gradients of mean velocities, which are the most important in this case, for turbulent dissipation. The effect of different depths in the distorted zone, on the motions of points A and B of the ship's model was studied. The depths used were $d = 0.051\text{m}$, $d = 0.15\text{m}$, and $d = .23\text{m}$, and referring these depths to the model's draught in ballast condition, values of d/T' were 1.32, 3.9 and 5.88. In figure 9 (a,b), results of the measurement of horizontal motions' magnitudes at points A and B (denoted as HA and HB) are summarized. In order to make comparison possible, HA and HB were divided by the height H_i of the generated wave in the hydraulic model of the harbor. Wave periods utilized were $t_1 = 1.06\text{s}$, $t_2 = 0.907\text{s}$, $t_3 = 0.756\text{s}$, $t_4 = 0.605\text{s}$ and $t_5 = 0.454\text{s}$, which are correspondent to periods of 14s, 12s, 10s, 8s and 6s in the prototype. In figure 10 (a,b), the ratios HA/H_i and HB/H_i are plotted against increasing values of L_w/L_{ppm} , L_w being the wave longitude in the generation zone, and, as already stated, L_{ppm} is the length between perpendiculars of the ship's model.

It is found that when depth increases in the distorted zone, the amplitudes of oscillations of the ship's model also increase. It's found also that the effect of depth on these amplitudes diminishes when distortion is increased, and that a condition is reached in which, beneath the appreciation of the measurements performed, no more effects on these motions are observed when depth is increased. the same behavior was observed by Giraudet [4], when studying berthing speeds in vessels' models, and energy transfer from ships to the fenders system.

These results are interpreted as a limit situation in which power transfer from mean flow to turbulent

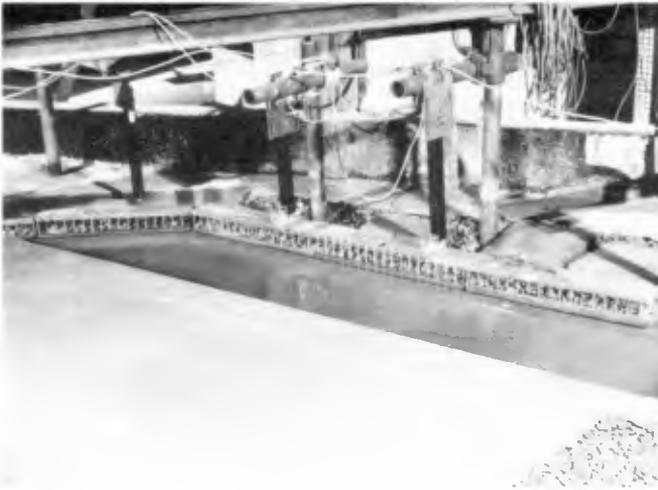


Figure 6. View of the distorted zone.

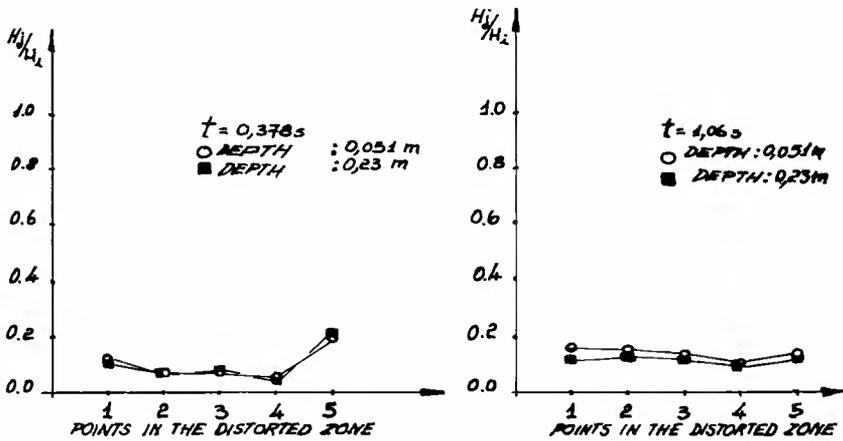


Figure 7. Measurement of wave heights in the distorted for different depths.

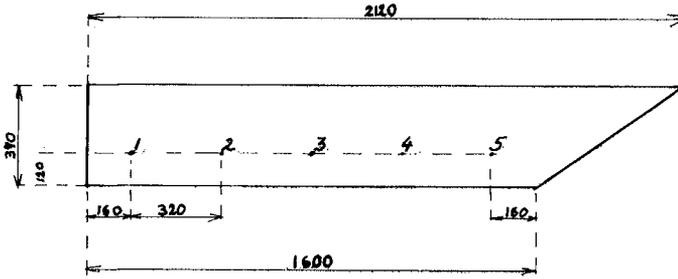


Figure 8. Location of points 1 to 5 in the distorted zone.

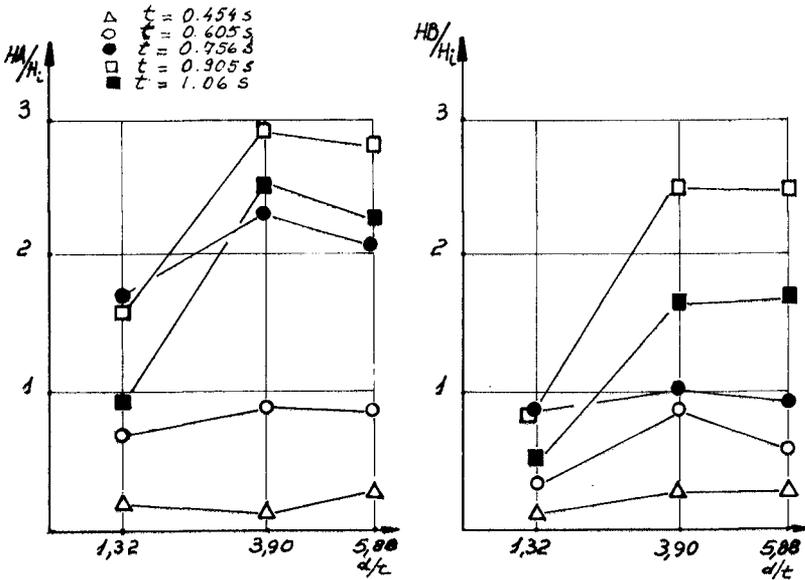
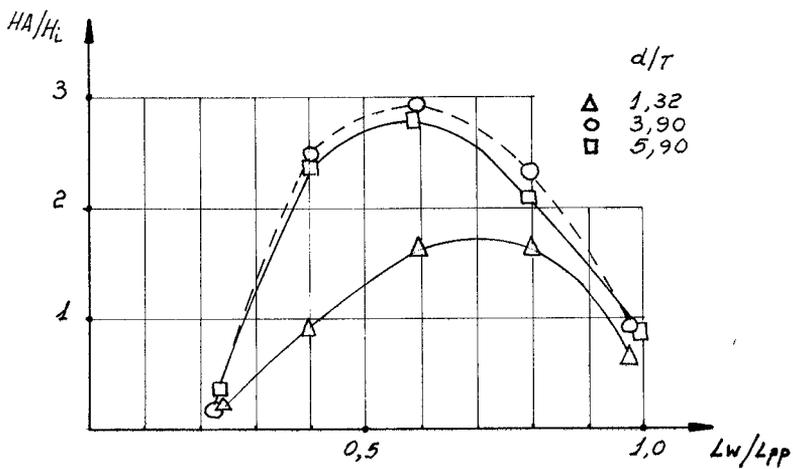
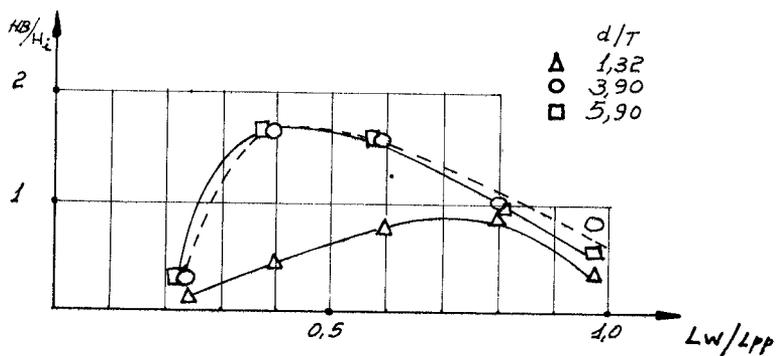


Figure 9(a,b). Displacement of the ship's model for different depths.



a



b

Figure 10(a,b). Displacement of ship's model as function of incident wave length.

fluctuations does not depend now on geometry, and, thus on the Reynolds number. Therefore the model operates in an "automodelling condition" with regard to Reynolds number. As the magnitude of the motions increases with depth, this condition places the model in a situation in which these motions will not be of less magnitude, at model's scale than those of the prototype, and results related to harbor's agitation by means of these ship's motions are on the safety side.

Conclusions

A method for evaluating agitation in harbors' hydraulic models is studied, which is more representative of port operation than measuring only wave heights in a mesh of points of the berth. It is shown that by using this method, safety factors are introduced if the bathymetry of the model is modified in a reduced area, which in view of it's small size, does not change the behavior of the port.

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