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DINAMICS OF CRANE VESSELS IN HEAVY LIFT OPERATION

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DINAMICS OF CRANE VESSELS IN HEAVY-LIFT OPERATION

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ABSTRACT.

The operability of crane vessels in waves is one of the important factors to engineer to take decisions on offshore operations. The effect of coupled motions between a vessel and the lifted load could result on amplification of responses in waves and could cause dangerous accidents.

Nowadays, the large crane vessels have been built to act in installations of large offshore jackets, modules and equipments on sea. Therefore, the analysis of dynamics of crane vessels in heavy operations becomes more important. Because of the natural period of the pendulum of the lifted cargo, the motions of vessels could be different from those without the crane with lifted cargo.

Therefore, the present paper shows the mathematical model of the eight-degree coupled motions of the crane vessel with lifted cargo and the comparison between calculated results and experimental results. The results of the experiment of the crane vessel in waves was obtained from wave tank tests.

INTRODUCTION

The dynamic of offshore crane vessels has been studied to analyse the behavior of the vessel during heavy lift operations in waves.[1],[2],[3].

The development of the offshore drilling and production system in Brazil to obtain more oil, brought us new technology and experience in design, fabrication,

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installation and operation of the Offshore structures.

The heavy lift operation of crane vessels is one of current operation in installation of jackets, modules and equipments on Campos Basin. Because of the large dimensions of these structures, this operation should be carefully analysed to be safe.

Today, large crane vessels as the "Micoperi 7000" could install 7000 tones modules and this lifting operations are pre-analysed considering the dynamic effects of the waves.

In this paper, the development of mathematical model of eight-degree freedom (six degrees of the vessel and two degrees of the lifted body) is presented. The analysed vessel was a crane barge from which the principal dimensions are presented in Table 1. The hydrodynamic coefficients and wave exciting forces are calculated using the Strip method.

The motions, accelerations and cable tensions were measured in wave tests [5],and the calculated results were compared to test results.

After the results have been analysed, the authors concluded that viscous damping and cross inertia terms consideration in the mathematical model are important to a better estimation of the dynamic of the crane barge in heavy lift operations in waves.

MATHEMATICAL MODEL

The additional two degrees of freedom are introduced into the dynamic equation of the vessel where there are six degrees of freedom. These two degrees of freedom correspond to angular motions φ_1 and φ_2 of the lifted body directed to y and x axis respectively as shown in the Fig.1.

The right-handed Cartesian coordinate system o-xyz with x and y axis on free surface and z axis positive upward were adopted as shown in the Figure 1 above. Therefore, there is a eight-degree of freedom system.

Considering the gravity center G of the system as a center of the rotation of the vessel, the velocity and

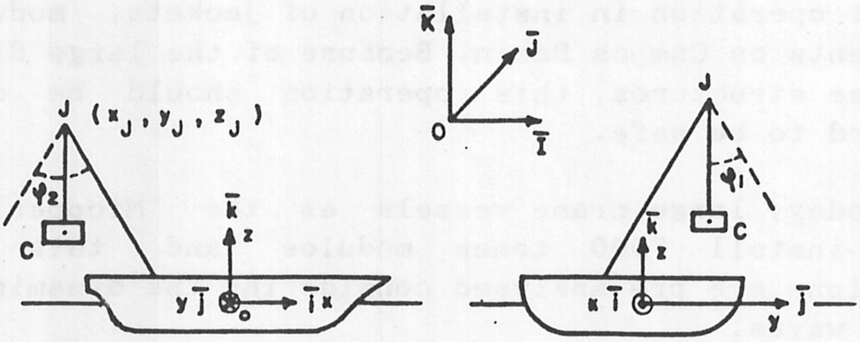

Figure 1. Coordinate System

acelleration of the tip of the crane J can be expressed by the following kinematic equation:

$$v_J = vg + \omega \wedge (\overline{GJ}) \qquad (1)$$

where vg is linear velocity of G
ω is angular velocity
$\overline{GJ}$ is the distance of crane tip from G

Then, the velocity on the tip of the crane can be written as:

$$v_J = (\dot{\xi}_1 i + \dot{\xi}_2 j + \dot{\xi}_3 k) +$$

$$(\dot{\xi}_4 i + \dot{\xi}_5 j + \dot{\xi}_6 k) \wedge (xi + yj + zk) \qquad (2)$$

The ξ_i are vessel motions, where i= 1,2 and 3 are linear motions x,y and z respectively and i=4,5 and 6 correspond to angular motions related to x,y and z axis respectively.

Where $\dot{\xi}_i$ is time derivatve of ξ_i.

On the tip of the crane, then, the velocity equation is, as follows,

$$v_J = (\dot{\xi}_1 + z_J\dot{\xi}_5 - y_J\dot{\xi}_6)i + (\dot{\xi}_2 + x_J\dot{\xi}_6 - z_J\dot{\xi}_4)j$$
$$+ (\dot{\xi}_3 + y_J\dot{\xi}_4 - x_J\dot{\xi}_5)k \qquad (3)$$

Assuming the small amplitude of the motions, the high order terms of the acelleration equation can be neglected and the acelleration on the crane tip J can be written as follows,

$$a_J = (\ddot{\xi}_1 + z_J\ddot{\xi}_5 - y_J\ddot{\xi}_6)i + (\ddot{\xi}_2 + x_J\ddot{\xi}_6 - z_J\ddot{\xi}_4)j$$
$$+ (\ddot{\xi}_3 + y_J\ddot{\xi}_4 - x_J\ddot{\xi}_5)k \qquad (4)$$

Where $\ddot{\xi}i$ is time derivative of the $\ddot{\xi}i$.

Therefore, if the acelleration of the crane tip is described, the acelleration of the lifted body ca be written as a function of the cable length L as follows,

$$a_T = (\ddot{\xi}_1 + z_J\ddot{\xi}_5 - y_J\ddot{\xi}_6 - \ddot{\varphi}_2 L)i$$
$$+ (\ddot{\xi}_2 + x_J\ddot{\xi}_6 - z_J\ddot{\xi}_4 + \ddot{\varphi} L_1)j$$
$$+ (\ddot{\xi}_3 + y_J\ddot{\xi}_4 - x_J\ddot{\xi}_5)k \qquad (5)$$

Introducing the potential and kinematic energy concept, the equations of dynamics of crane vessels with heavy lift cargo were obtained.

Then, the differential equation system to be solved,in which the ξ_i, φ_1 and φ_2 are unknow parameters, will be as follows,

$$\sum_{j=1}^{6} (M_{1j}\ddot{\xi}_j + B_{1j}\dot{\xi}_j + C_{1j}\xi_j) - mL\ddot{\varphi}_2 = F_1$$

$$\sum_{j=1}^{6} (M_{2j} \ddot{\xi}_j + B_{2j} \dot{\xi}_j + C_{2j} \xi_j) + m L \ddot{\varphi}_1 = F_2$$

$$\sum_{j=1}^{6} (M_{3j} \ddot{\xi}_j + B_{3j} \dot{\xi}_j + C_{3j} \xi_j) = F_3$$

$$\sum_{j=1}^{6} (M_{4j} \ddot{\xi}_j + B_{4j} \dot{\xi}_j + C_{4j} \xi_j) - m L z_J \ddot{\varphi}_1 = F_4$$

$$\sum_{j=1}^{6} (M_{5j} \ddot{\xi}_j + B_{5j} \dot{\xi}_j + C_{5j} \xi_j) - m L z_J \ddot{\varphi}_2 = F_5$$

$$\sum_{j=1}^{6} (M_{6j} \ddot{\xi}_j + B_{6j} \dot{\xi}_j + C_{6j} \xi_j) + mLx_J \ddot{\varphi}_1 + mLy_J \ddot{\varphi}_2 = F_6$$

$$m L^2 \ddot{\varphi}_1 + W L \ddot{\varphi}_1 + m L \ddot{\xi}_2 - m L z_J \ddot{\xi}_4 + m L x_J \ddot{\xi}_6 = 0$$

$$m L^2 \ddot{\varphi}_2 + W L \ddot{\varphi}_2 - m L \ddot{\xi}_1 - m L z_J \ddot{\xi}_5 + m L y_J \ddot{\xi}_6 = 0 \qquad (6)$$

In the equations (6), the M_{ij} are inertia and added inertia terms of the vessel , B_{ij} and C_{ij} are linear damping terms and restoring terms of the vessel respectively and F_j are wave exciting forces of an harmonic wave acting on vessel.

CALCULATION & MEASURED DATA

The typical crane barge in installation operation of module on jacket is adopted as the calculation model to compare with results of experimental model on tank tests as can be shown in Fig 2. The model scale of the barge is 1:48.

Principal dimensions of the barge and lifted module are shown in Table 1.

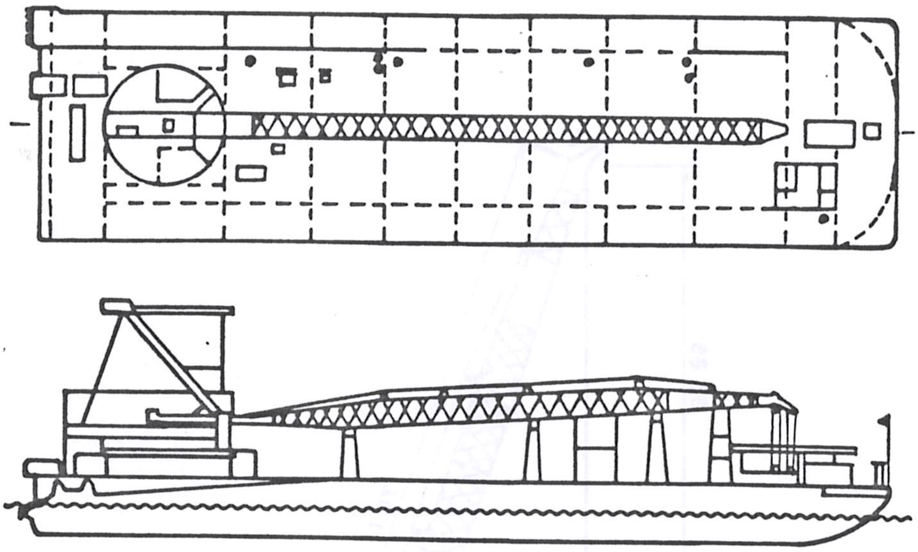

Figure 2. Adopted Crane barge

Table 1. Principal dimensions

displacement	14610,.0 ton
draft	4,33 m
KG	6,72 m
LCG	4,37 m AFT
Kxx	10,52 m
Kyy	35,90 m
Kzz	35,00 m
GM_T	17,28 m
module weight	400 ton
cable lenght	53,00 m

The detailed dimensions of the crane and lifted module are shown in Fig.3. The calculated parameters were six degrees of freedom of barge, two degrees of lifted body, acelleration on crane tip and cable tension for each wave frequency. The measured parameters of [5] were Heave, Pitch, Roll, Acelleration on the crane tip, Cable tension, Vertical crane tip displacement and Wave height as shown in Fig.4 .

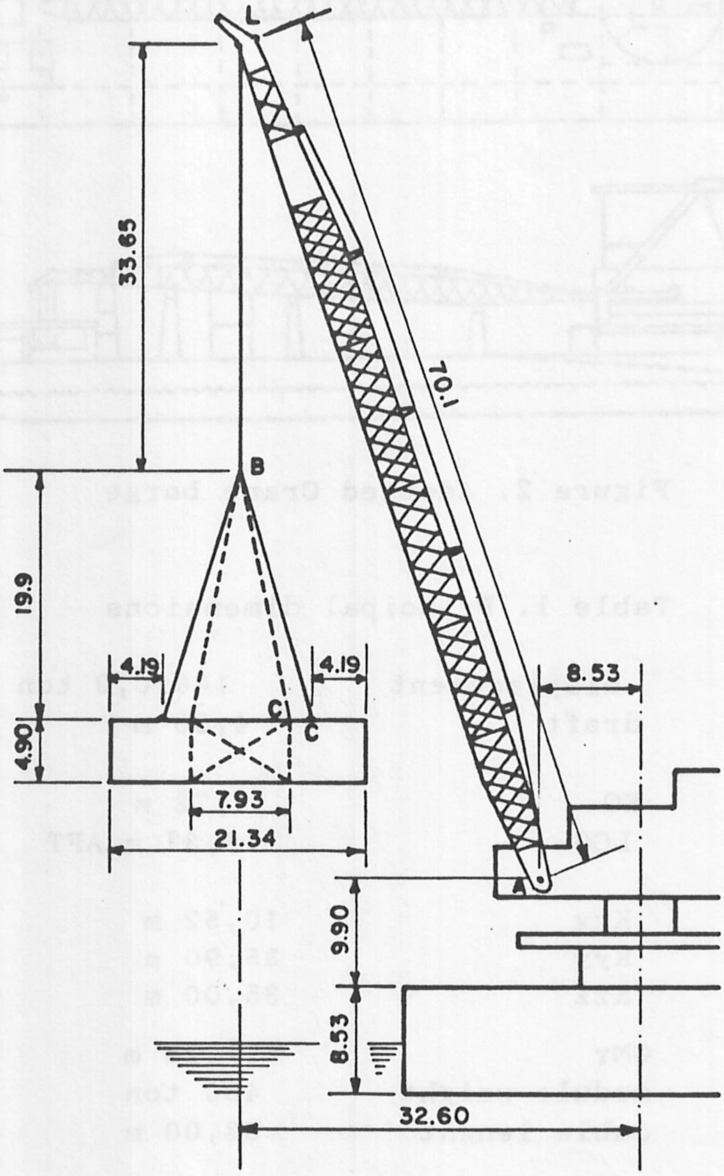

Figure 3. Principal dimensions of crane and lifted module [5]

1-cable tension
2-crane tip vertical motion
3-crane tip vertical acelleration
4-"pitch" motion
5-"heave" motion
6-"roll" motion
7-wave

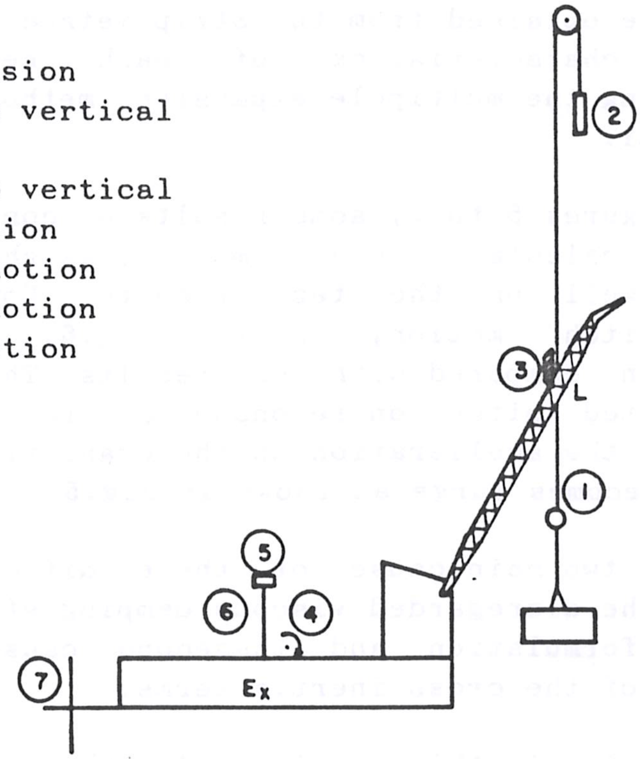

Figure 4. Measured data [5]

ANALISIS OF RESULTS

To verify better operational conditions of thecrane vessel, several conditions were tested and compared with experimental results. The crane direction to install the module on the jacket could be parallel to x axis or parallel to y axis. Therefore, the following three principal conditons are presented in this paper:

Condition 1- the crane is directed to aft and the wave comes from fore, i.e., head sea.

Condition 2-the crane is directed to starboard and the wave comes from fore, i.e., head sea.

Condition 3-the crane is directed to starboard and the wave comes from oblique direction (45 deg.)

In all the conditions, the considered wave periods are ocean frequency range periods.

The wave exciting forces and the hydrodynamic parameters were obtained from the Strip method in which the hydrodynamic characteristics of each section were calculated using the multipole expansion method idealized by Ursell-Tasai.

In the Figures 5 to 7, some results of condition 1 are presented. The calculated "heave" motion, as shown in the Fig.7, fits well on the test results. However, the calculated "pitch" motion, in the Fig.6, shows some difference when compared with test results. The peak value of the calculated "pitch" on resonance ωp is very large. Consequentlly, the acelleration on the crane tip near this frequency ωp becomes large as shown in Fig.5.

There are two main cause of these differences, the first one is the disregarded viscous damping effect on the mathematical formulation and thesecond cause is the consideration of the cross inertia terms.

Particularly, in this condition 1, the effect of the viscous damping effect could be the main cause of the discrepancy of the results.This viscous damping, normally, is disregarded in motions calculations of conventional barges or ships. However, this effect seems to be important when it is coupled with a heavy-lifted body.

The cross inertia terms effect seems to be small, because, in the condition 1, only Kzx has effective value. However, in this study, the Kzx value is not calculated and more analisis is requered.

In the Figures 8 to 11, the results of condition 2 are shown. The Fig.10 presents the comparison between calculated "heave" and experimental heave results and,as can be noted, the coincidence of both results is very good. However, the same differences presented in condition 1 appear on results of condition 2. The calculated "pitch" motion near the resonance frequency ωp is very large and presents two peaks, but the experimental result shows only one peak, Fig.9. Consequently, the vertical acelleration on the crane tip position presents the same differences, as shown in Fig.8.

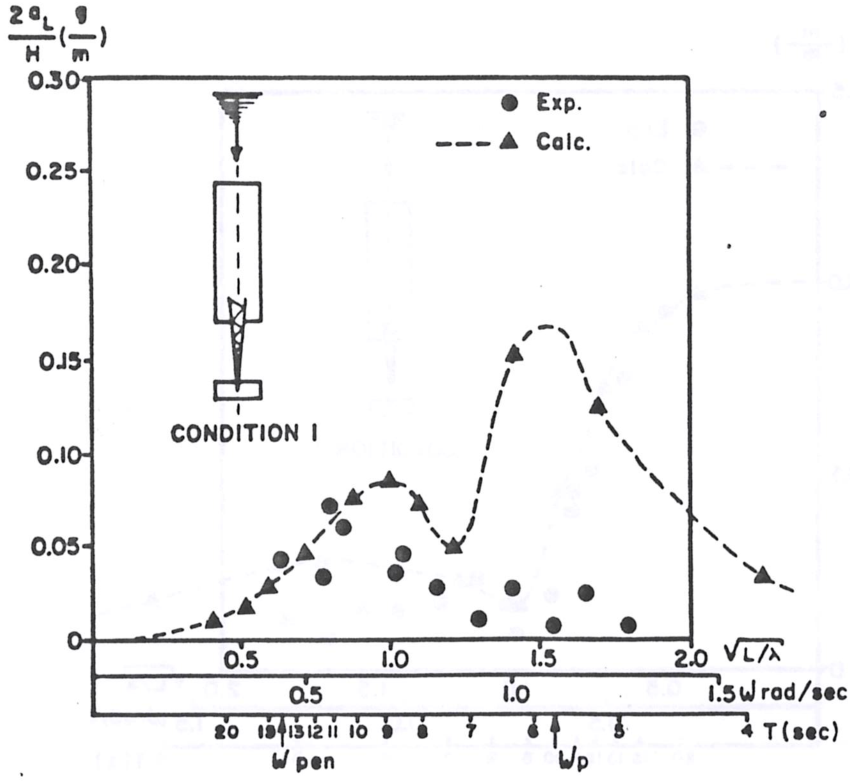

Figure 5. Vertical acelleration of crane tip (condition 1)

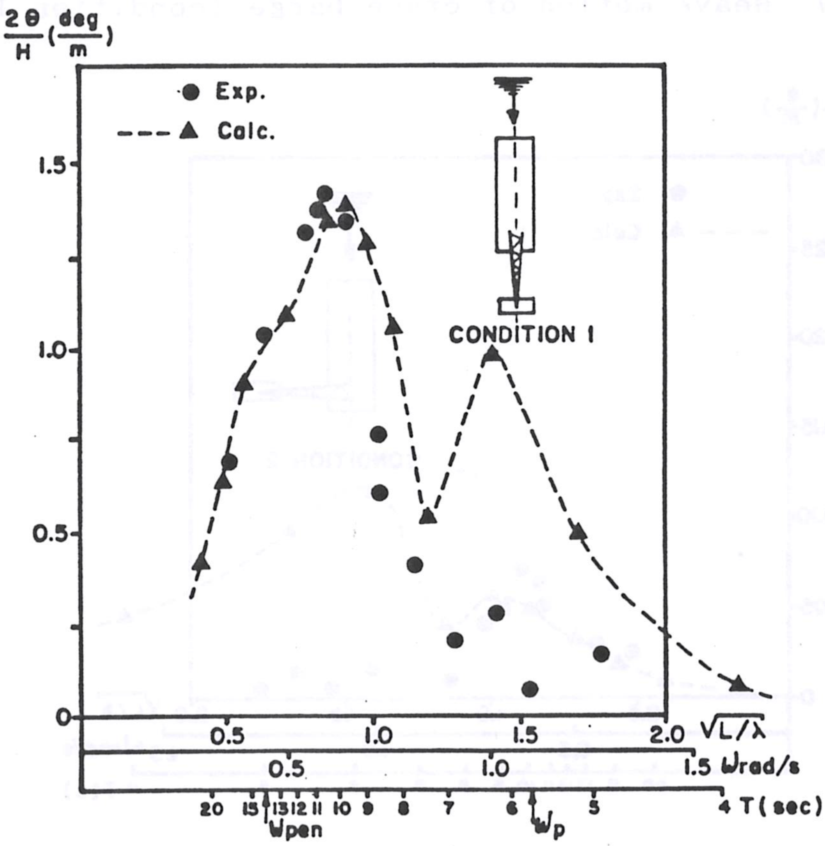

Figure 6. Pitch motion of crane barge (condition 1)

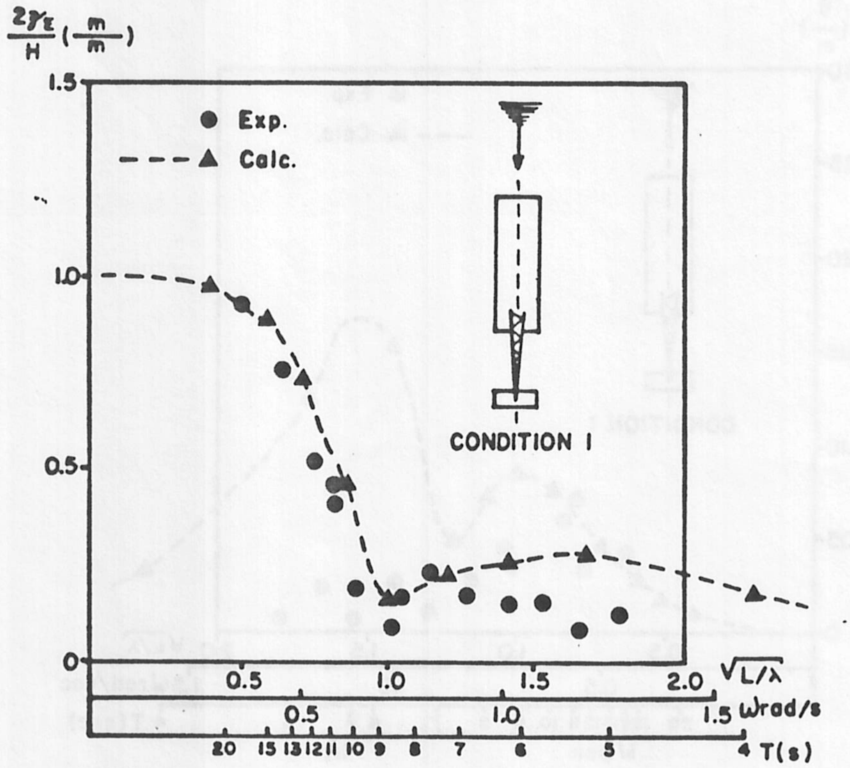

Figure 7 Heave motion of crane barge (condition 1)

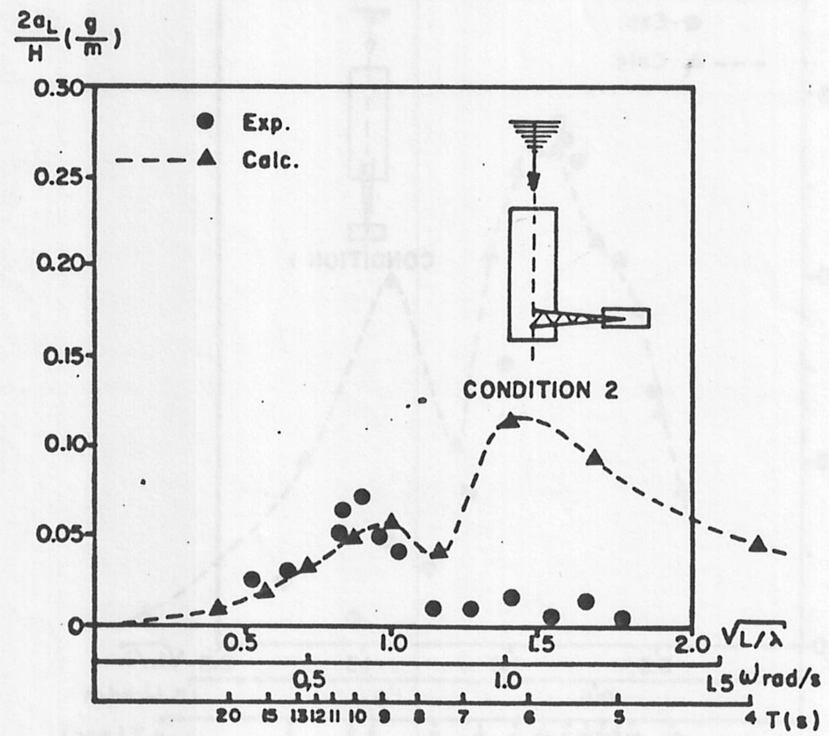

Figure 8. Vertical acelleration of crane tip (condition 2)

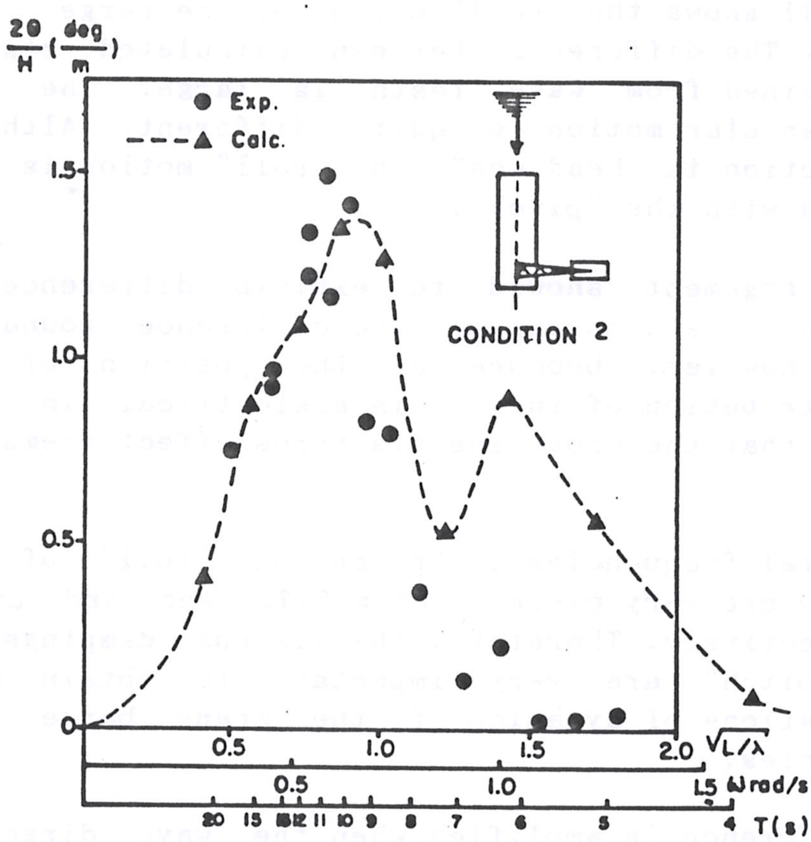

Figure 9. Pitch motion of crane barge (condition 2)

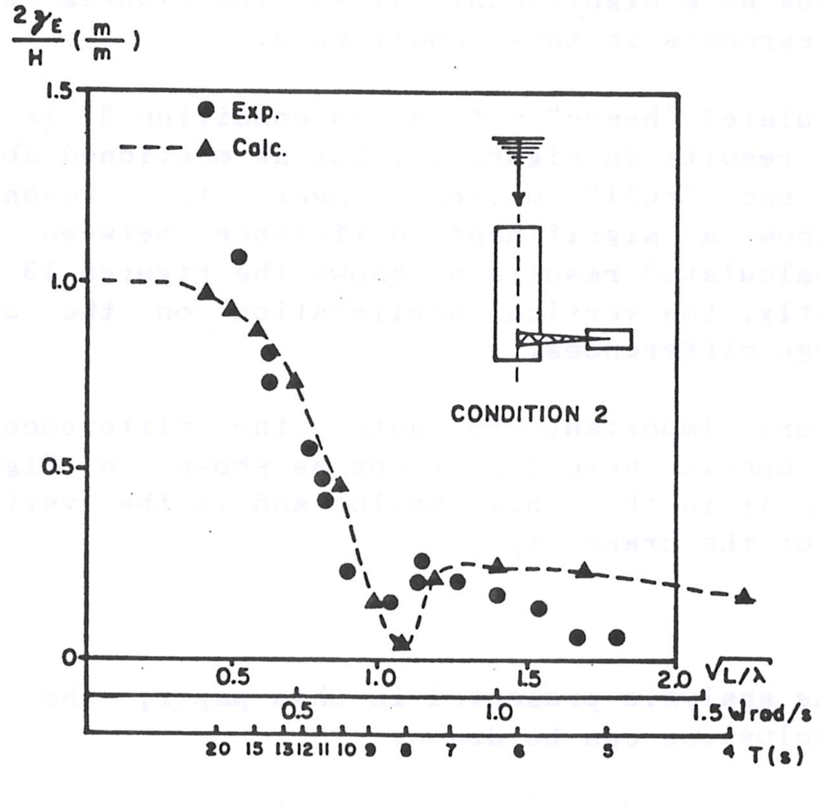

Figure 10 Heave motion of crane barge (condition 2)

The Fig.11 shows the "roll" motion of the barge with a lifted body. The difference between calculated results and those obtained from wave tests is large. The peak positions of angular motion is quite different. Although the wave direction is "head sea", the "roll" motion is very large compared with the "pitch".

The same argument showed to explain difference in codition 1 can be used to explain the difference found in condition 2. However, because of the position of the crane,the distribution of inertia is assimetrical in this condition, so that the cross inertia terms effect seems to be important.

The natural frequencies of "pitch" and "roll" of the barge in study are very close to ωp = 5.76 sec and ωr = 6.85 sec respectively. Therefore, the viscous dampings of "roll" and "pitch" are very important to obtain more precise estimations of dynamics of the crane barge near these frequencies.

This difference is amplified when the wave direction is oblique wave (condition 3), because both "roll" and "pitch" motions have significant values. The Figures 12 to 15 show the responses in this condition 3.

The calculated "heave" motion, in condition 3, is very close to test results in Figure 14, but as mentioned above, the "pitch" and "roll" motions near the resonance frequencies show a significant difference between test results and calculated results as shows the Figures 13 and 15. Consequently, the vertical acelleration on the crane tip shows large differences.

It is very important to note, the difference of responses for oposite wave directions as shown in Figures 12 to 15, mainlly in the "roll" motion and in the vertical acelleration of the crane tip.

CONCLUSIONS

After the analysis presented in this paper, the following conclusions can be drawn:

1) The dynamics of Crane vessel in heavy-lift

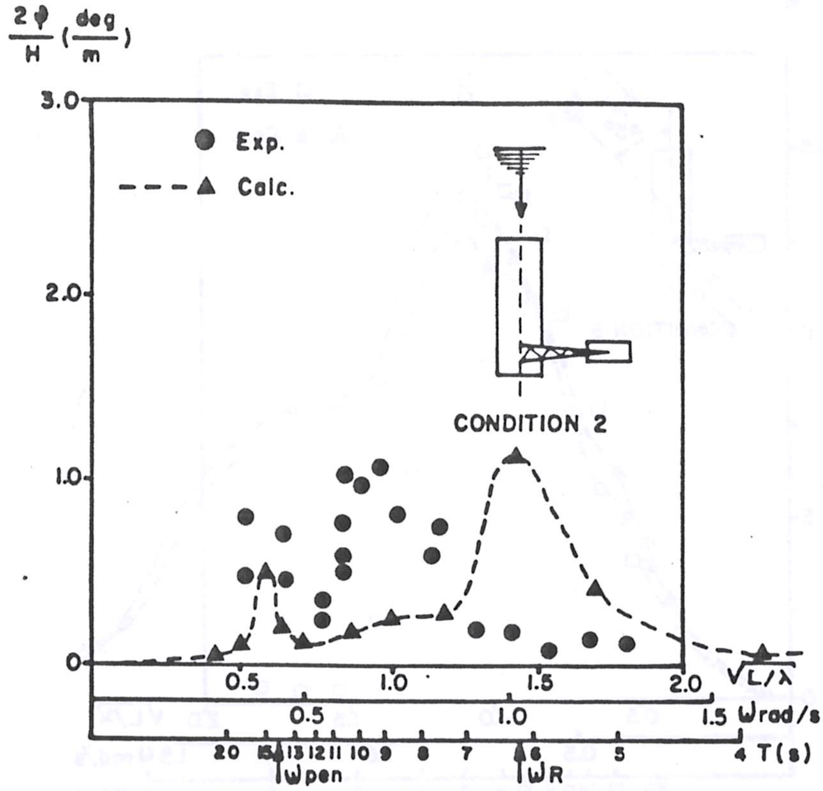

Figure 11 Roll motion of crane barge (condition 2)

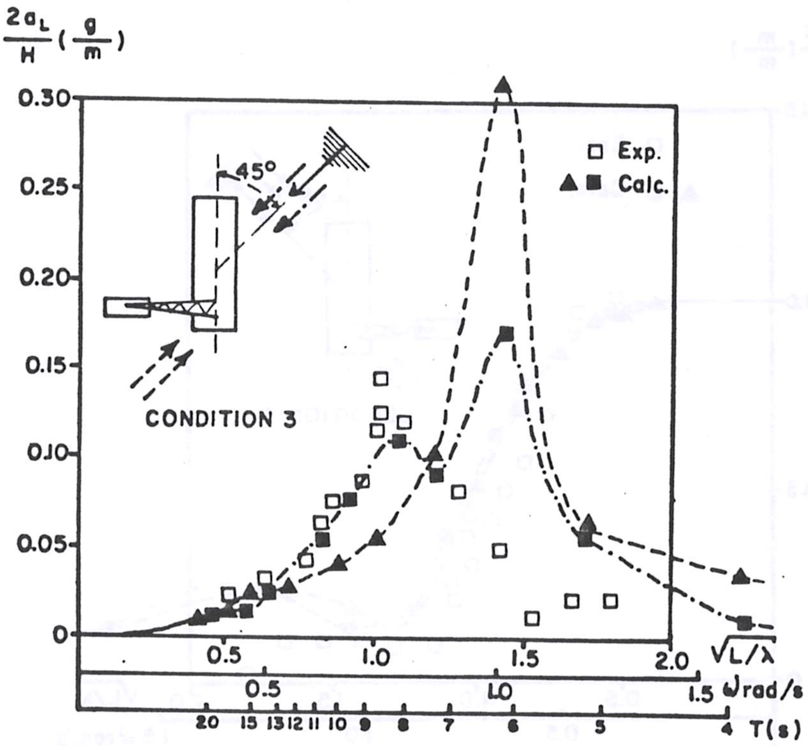

Figure 12. Vertical acelleration of crane tip (condition 3)

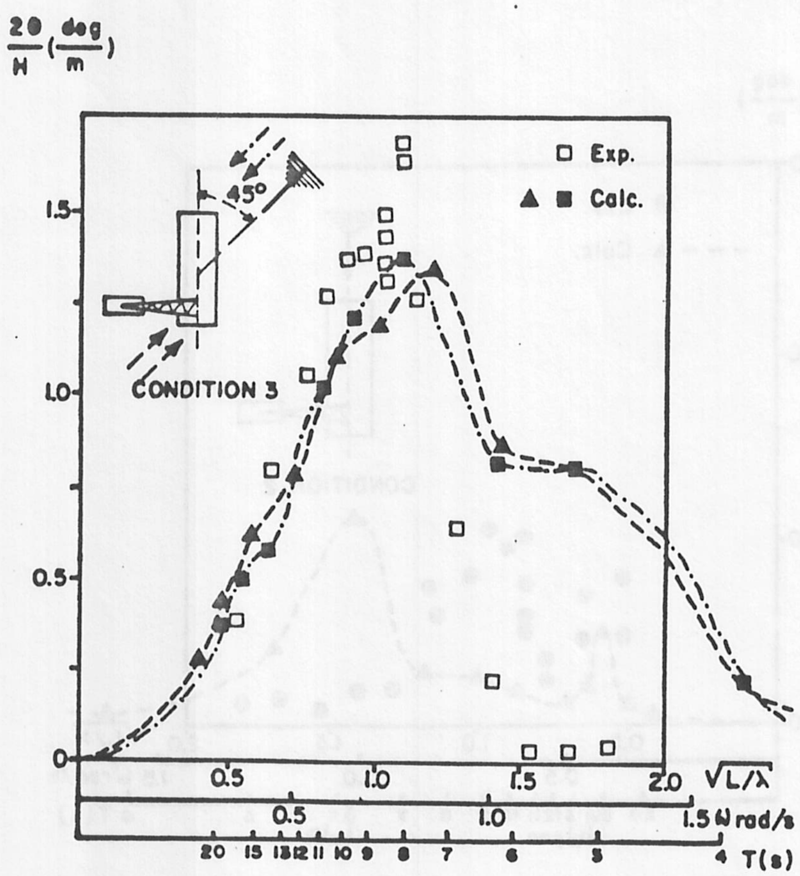

Figure 13. Pitch motion of crane barge (condition 3)

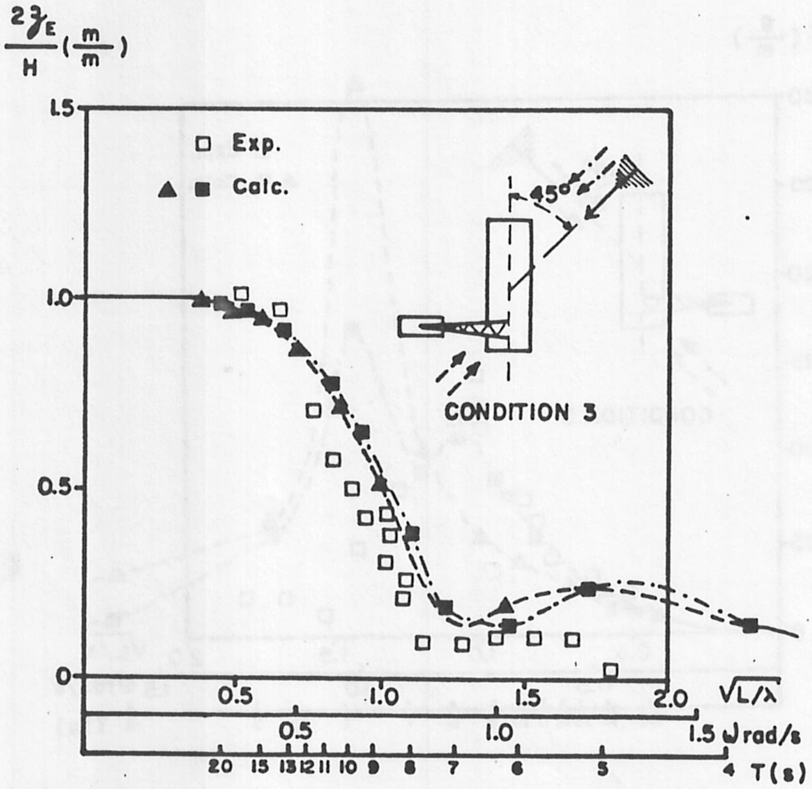

Figure 14 Heave motion of crane barge (condition 3)

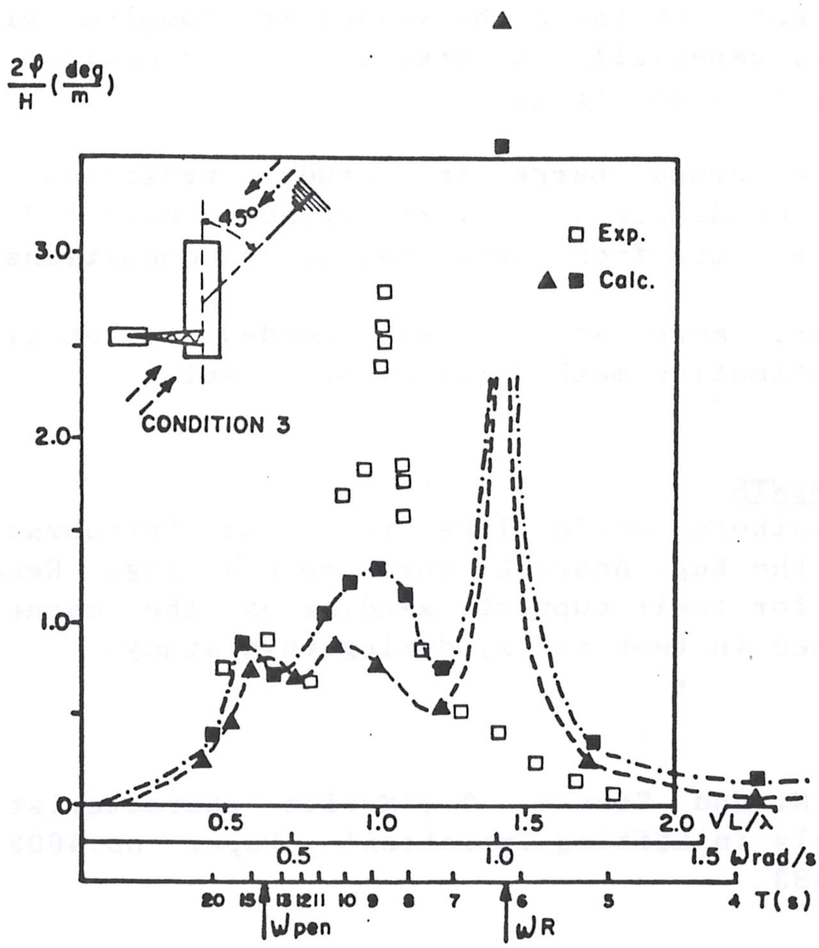

Figure 15 Roll motion of crane barge (condition 3)

operations are very different from vessel motions alone. Therefore, the coupled motions of crane vessel and lifted body were developed and shows good results, if compared with test results.

2) The viscous damping and cross inertia terms are very important when the crane vessel is coupled with the litfed body, especially, to predict the dynamics of the vessel near resonant frequency.

3) The crane barge in study presented better behavior in condition 1, i.e. the crane is directed to aft and the wave comes from fore, than others conditions.

However, more studies are needed to obtain more accurate estimation method for crane vessels.

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