Some Notes on the Local Vibrations of Ships

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NOTES

Some Notes on the Local Vibrations of Ships

By Toyoji KUMAI

1. Vibration Amplitude of The Deck Panels Stiffened in One Direction.

When the deck panels having only transverse stiffeners like as that of the hatch side deck in ship structure vibrate under the forcing action, the several numbers of peaks of the response will occur in the incomprehensive range of the frequencies of the panel. It will be taken place that the vibration system of the panel of this type has several combinations of the modes in two perpendicular directions of the rectangular panel. Since the bending stiffness of the stiffener is very high compared with the panel plate in the other direction, the higher mode of the vibration will easily occur in the plate.

Now, put p_{xm} the critical frequencies of the stiffener with the plate of unit or effective breadth, and p_{yn} is that of the panel plate in the perpendicular direction of the stiffeners. Since the frequency p_{xm} is considered to be that of the forced vibration of the panel, the amplitude factor a_{mn} is presented by the well known form,

$$a_{mn} = \frac{1}{\sqrt{\left\{1-\left(\frac{p_{yn}}{p_{xm}}\right)^2\right\}^2 + 4\,h^2}}$$

where h is the factor depending on the damping property of the system, but this is ignored in the present discussion. The forcing frequency p_{yn}, however takes from the fundamental to several higher modes of the vibrations. Thus the amplitude ratios a_n $(n=1, 2, 3, \cdots)$ are obtained by assort the given values of p_{yn} $(n=1, 2, 3, \cdots)$ with the fundamental mode of p_{xm} $(m=1)$. As well known property, the maximum value of the vibration amplitude of the panel is found out when p_{x1} closed up to p_{yn}. The amplitude ratios of the panel are thus easily be made by assorting p_{x1} with p_{y1}, p_{y2}, $\cdots$, hence, the mode of the vibration n which present the maximum amplitude of the panel is obtained.

The numerical value of p_{x1} and p_n $(n=1, 2, 3, \cdots)$ are computed by the method of author's previous paper.[1] The results of the model experiments will be seen in fig. 1 a) and b). The apparatus for model experiments has already be shown in author's previous paper.

In conclusions, the envelope of the peaks in the response curve of the panel considered shows the same type as resonance curve of forced vibration system. The maximum amplitude of the vibration of the deck panel stiffened in one direction occur not always in the fundamental mode of the panel, but it occurs at the n-th mode when the frequency p_{yn} closed up to p_{x1}.

2. Critical Frequencies of The Panels Partly Immersed in Water.

The critical frequencies of the panels of the shell plate of ship's hull, for instance, whose boundaries are limited by margin angle and second deck and engine room bulkheads are required in the design stage of ship's hull. The estimation of the criticals

[1] T. Kumai, Reports of Res. Inst. Appl. Mech. Kyūshū Univ. Vol. II. No. 8 1953.

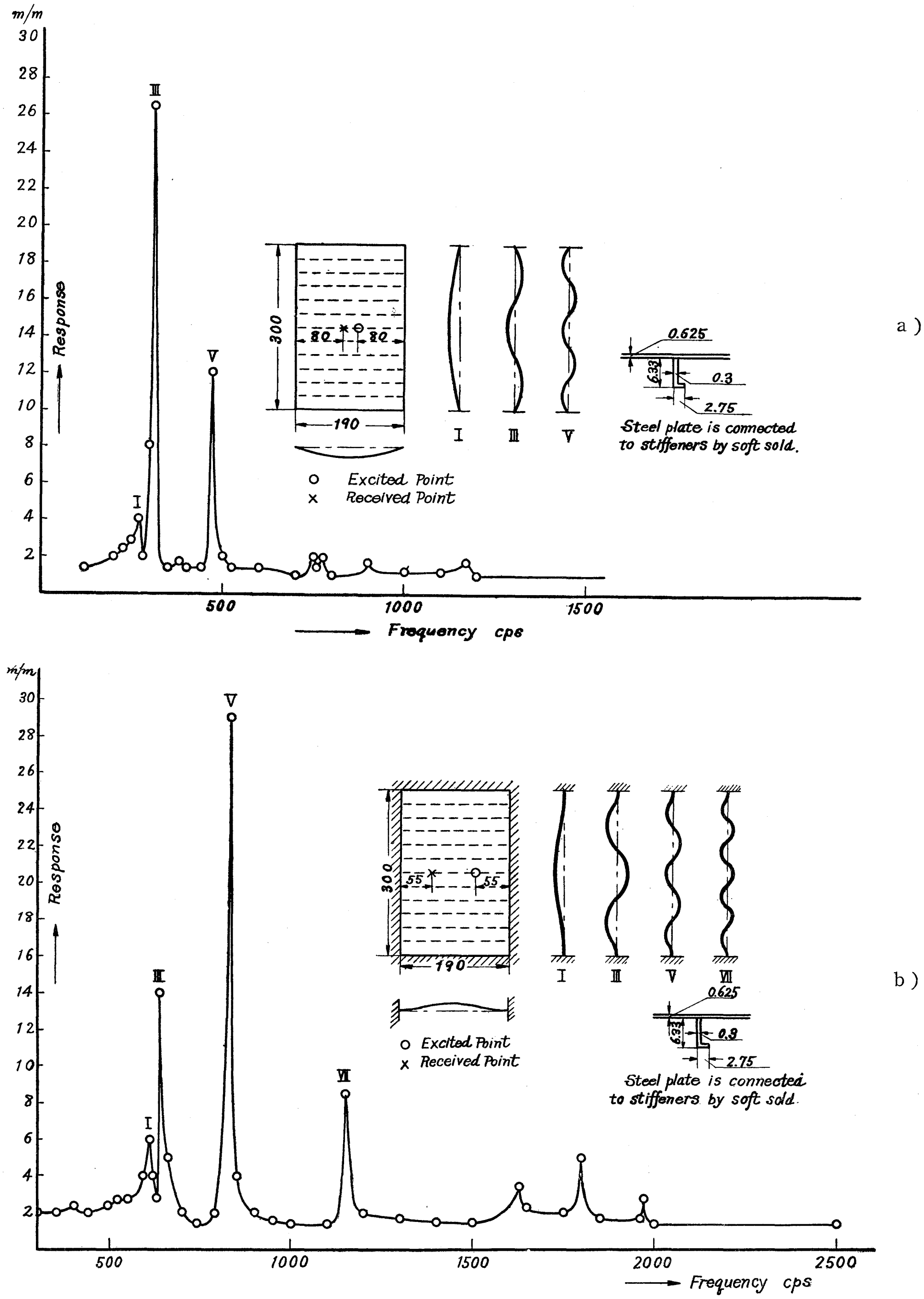

FIG. 1. Response curve of the vibration of the panel.
a) all edges supported b) all edges clamped

of the panels in *vacuo* are easily be obtained by use of K. Okuda-T. Arima's formula[2] or the author's formula in previous paper. Also in the case of the panel vibration with virtual mass of water surrounding all surface of the panel, the effects of the virtual mass on the criticals are computed by use of F. Kidô's formula.[3] In actual ships, however, the panel like as mentioned above is almost in the state of partly immersed in water in the case of the half load or ballast conditions. Hence, the effect of entrained water on the critical frequencies of the panels partly immersed in water is necessary when the criticals of these panels of the shell plate are required.

The present note shows the approximate calculations and the experimental verification on the vibration of the plate partly immersed in water, and also the simple formula for calculating added virtual mass ratio of water and of vibrating plate mentioned above is proposed.

i) Approximate Calculation.

The vibration mode of the panel is assumed to be of the form

$$W = A \cos my \cos nz \sin \omega t \tag{1}$$

where y, z two perpendicular axes with the origin at the centre of the plate respectively,
m, n numbers of half wave in directions x, y respectively, and $m = \pi/a$, $n = \pi/b$ when a, b takes breadth and depth of the plate respectively,
ω critical frequency of the plate,
t time,
A constant.

The velocity potential function is approximately assumed to be of the form[4)5)]

$$\phi = B \cos my \cos nz\, e^{-\alpha x} \cos \omega t\,, \tag{2}$$

where x coordinate perpendicular to yz-plane,
B constant.

Now ϕ satisfies the following equation,

$$\frac{\partial^2\phi}{\partial x^2} + \frac{\partial^2\phi}{\partial y^2} + \frac{\partial^2\phi}{\partial z^2} = 0\,, \tag{3}$$

substitute from (2) in (3), α becomes

$$\alpha = \frac{\pi}{a}\sqrt{1-\left(\frac{a}{b}\right)^2}\,, \tag{4}$$

on the surface of the plate,

$$\frac{\partial\phi}{\partial x} = \frac{\partial W}{\partial t} \quad \text{at} \quad x = 0\,, \tag{5}$$

thus, the constant B in (2) is obtained

$$B = -\frac{\omega}{\alpha} A\,. \tag{6}$$

2) K. Okuda-Arima, Jour. Japan Soc. of Naval Architecture No. 58, 1936.
3) F. Kidô, Jour. Japan Soc. of Naval Architecture No. 266, 1944.
4) T. Terada, Proc. Tokyo Math. Phys. Soc, 1906.
5) *loc. cit.* 3).

In the present discussion, the boundary condition at the water aurface is ignored. The total kinetic energy of the vibration of the particles of water is presented by[6]

$$T_1 = -\rho \int_{-b/2}^{d-b/2} \int_{-a/2}^{a/2} \phi \frac{\partial W}{\partial t} \, dy \, dz \,, \tag{7}$$

where ρ density of water,
d draught of immersed plate.

As an extreme case, if the upper limit of integration with respect to z takes $b/2$ or $d=b$ in (7), energy of completely immersed plate is obtained. On the other hand, the kinetic energy of the vibrating plate is shown by

$$T_2 = \frac{1}{2} \rho_m h \int_{-b/2}^{b/2} \int_{-a/2}^{a/2} \left(\frac{\partial W}{\partial t}\right)^2 dy \, dz \,, \tag{8}$$

where ρ_m density of the plate,
h thickness of the plate,

equating (7) and (8), the ratio of virtual mass and vibrating mass of the plate immersed in water with the draught d is obtained as follows,

$$\varepsilon_d = \left\{ \frac{d}{b} + \frac{1}{2\pi} \sin\left(\frac{2d}{b} - 1\right) \right\} \varepsilon_1 \,, \tag{9}$$

where,

$$\varepsilon_1 = \frac{\rho}{\rho_m} \frac{a}{h\pi} \frac{1}{\sqrt{1 - \left(\frac{a}{b}\right)^2}} \,, \tag{10}$$

this is the formula of which in the case of the half side immersed plate in water.

Consequently, the frequency ratio of the plate immersed in water with draught d and of which in *vacuo* is cast into as follows,

$$\frac{N_d}{N_0} = \sqrt{\frac{1}{1 + \varepsilon_d}} \,. \tag{11}$$

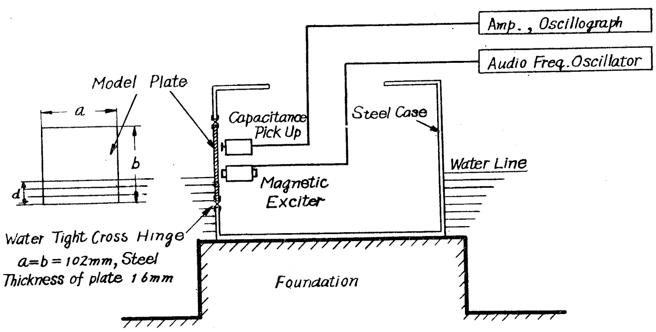

FIG. 2. Apparatus for measuring the critical frequency of the plate immersed in water.

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[6] H. Lamb, Hydrodynamics,

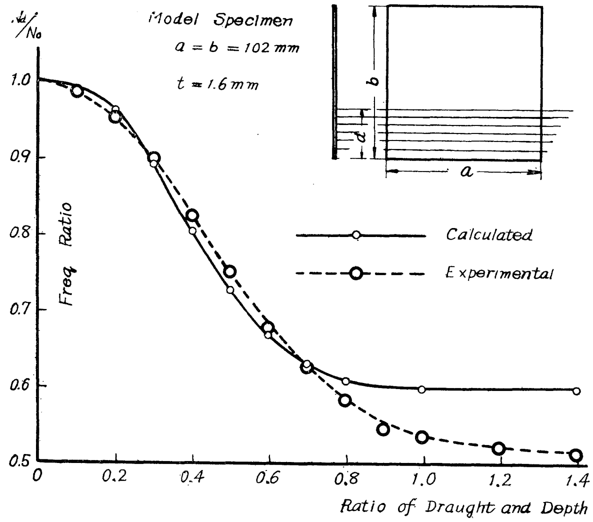

FIG. 3. Ratio of frequency of the immersed square plate with given draught d and of which in *vacuo*.

ii) Model Experiment.

The apparatus for measuring the critical frequency is shown in Fig. 2. The tests were taken place in water tank of 700 mm $\times$ 700 mm $\times$ 2800 mm and the draught of the model plate is varied by adjustment of the water surface in tank. The mean values of the results of five repeat tests on the same draught are shown in Fig. 3. as an illustration. As will be seen in the figure, the effect of the entrained water mass upon the critical frequencies of partly immersed plate is clearly shown and is compared with the results of the approximate calculations obtained above. The agreement of both results is fairly good for practical use.

Two notes described above are quoted from the Reports of West Japan Sub-committee of Ship Structure of Japan Society of Naval Architecture.

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