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VIBRATION OF THE DECK PANELS IN SHIP STRUCTURE

By Toyoji KUMAI

The investigations on the critical frequencies of the vibrations of the plate with stiffeners in ship structure are carried out in aid of the model experiments, and the results of the calculations are compared with which obtained by the measurements of the critical frequencies of some local vibrations in actual ships.

1. Introduction. An important problem in the local vibrations of ship's hull is the resonance of the flexural vibrations of the rectangular plate with stiffeners such as some parts of decks, house walls and engine casing panels etc.

It is necessary to predetermine the critical frequencies of the deck panels like as above mentioned for design of ship's hull. The practical method for the calculation of natural frequencies of the rectangular plate with the stiffeners and girders have already been presented by Okuda and Arima.[1)]

The formulae deduced by Okuda and Arima are very convenient in practical use. The effective breadth of the plate is taken into account in the computation of the moment of inertia of the section of deck panel. The experimental results obtained by them, however, are presented lower values of frequencies than the theoretical one.

The present paper gives some reasons for the discrepancies between the results of calculations and experiments or measurements in actual ship.

2. Coupled vibration of the deck panels. It is convenient to consider the effective beadth of the plate with the stiffeners in the calculations of the flexural rigidities of panel in the case of statical bending and buckling of the panels in ship structure. Such an assumption may also be applied in the calculation of the flexural vibrations of the panel. Subtructing the effective breadth, the remainder parts of the breadth of the plate is considered to be idle during the panel being under the action of statical loading.

In the case of flexural vibrations of the deck panels, however, the small panels of the plate subdivided by the stiffeners or beams and girders are

1) Okuda and Arima; "Strength and Frequency of Rectangular Thin Plates with Longitudinal and Lateral Stiffeners." Journal of Japan Soc. of Naval Architecture. No. 58, 1936.

considered to be the subsystem of the vibration of the deck panel. Consequently the vibration of the deck panel is supposed to be of the vibration system of two degree of freedom in a specified mode of the vibration of the panel.

Hence, the deck panel has two natural frequencies, the one is smaller than that of the system with one degree of freedom and the other is higher than it.

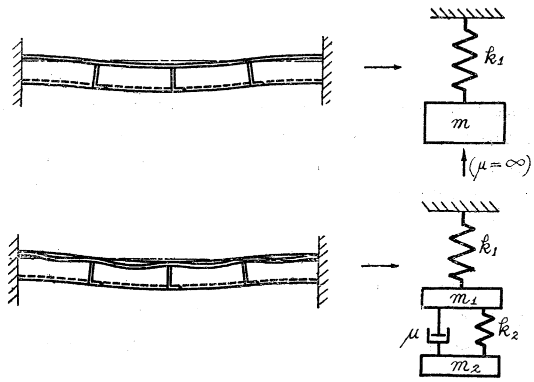

FIG. 1. Model presentation of the vibration system of the deck panel.

The model presentation of the vibration system of the deck panel is shown in fig. 1. If the damping coefficient of dash-pot μ presented in the figure takes an infinitely large value, or if k_2 becomes very large compared with k_1, the system is reduced to one degree of freedom. In the case of the vibration of the deck panel, however, the damping factor of which mentioned above is not infinite and the stiffness of the small panels of the plate between the stiffeners are not so high compared with that of the deck panel, thus the system of the vibration of the deck panel should be of two degree of freedom. That is to say, the plate of the deck panel contributes to the flexural stiffness of the vibration of the subsystem as well as that of the deck panel.

The combinations of the higher modes of the vibrations with two perpendicular directions will be possible in the system of the vibration of deck panel. Moreover, the coupled frequency of the combination of the higher modes of these two directions becomes comparable one with that of the fundamental mode of the panel, hence, it will be presumed that several peaks of the response occur in the narrow range of the frequencies.

The calculations of the natural frequencies of the coupled system above mentioned are obtained by the following equation

$$\frac{f_1}{f_2} = \frac{\gamma_1}{\gamma_2} \times f_{\mu=\infty} \tag{2.1}$$

where γ_1 and γ_2 are computed by the modified formulae of the coupled vibrations of the mass-spring system with two degree of freedom as follows,[2),3)]

$$\begin{matrix}\gamma_1\\ \gamma_2\end{matrix} = \sqrt{\frac{1+\beta}{2}\left[\{1+(1+\beta)\delta^2\} \mp \sqrt{\{1+(1+\beta)\delta^2\}^2 - 4\delta^2}\right.}, \quad (2.2)$$

β and δ in (2.2) and $f_{\mu=\infty}$ in (2.1) are easily computed by the use of given scantlings and the specified mode of the panel as shown in Appendix I. Numerical values of γ_1, the reduction factor of the frequency, are tabulated for the given value of δ^2 and the parameters β in Appendix II. The frequencies of the panel with the combinations of any higher modes in two directions are obtained by the use of corresponding value of k_1 of respective mode.

3. Experimental Verifications of the coupled vibration of deck panel. The model experiments on the vibrations of the deck panel were carried out by the use of the one fifth scale model of the Okuda-Arima's specimen of the deck panel. The model was forced by means of magnetic exciter which be derived by the beat oscillator in the range of $50 \sim 5000$ cps of frequencies and the response was obtained from the reading of the amplitude of the wave pattern presented on Brown-tube oscillograph connected to the out put device of the capacitance pick up[4)] which set up with narrow gap to the surface of a model. The calibrations of the frequency measurements of the vibration of deck panel were carried out by the use of the twelve standard tuning forks of 440 (a^1) to 830.6 ($g^2\#$) cps. The schematic diagram of the apparatus for measurment of the frequencies and the response are shown in fig. 2.

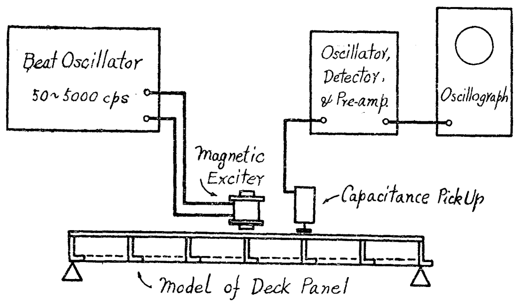

FIG. 2. Experimental apparatus for measurement of frequencies and response of the model of a deck panel.

Four peaks of the response were observed up to 1000 cps, of the frequencies and whose respective combinations of the modes of the vibration of each perpendicular directions were measured as shown in fig. 3.

2) Den Hartog; "Mechanical Vibrations" p. 111, 1940.
3) Timoshenko; "Vibration Problems in Engineering" p. 240, 1937.
4) Heteny; "Hand Book of Experimental Stress Analysis" p. 295, 1950.

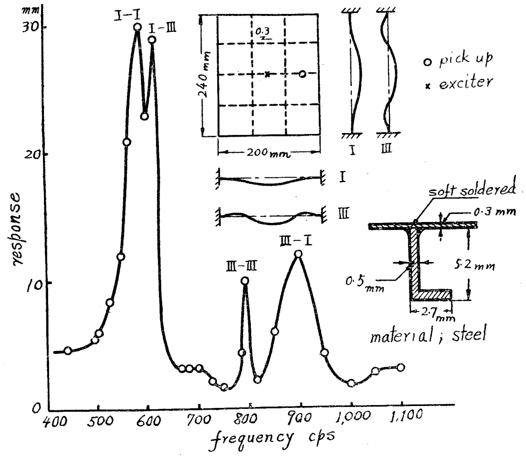

FIG. 3. Characteristic curve of the frequency of the deck panel. The scale of the model is 1/5 of the prototype.

The frequencies of peaks of the response obtained by the experiments and of which computed by the formulae explained in the previous article are shown in the Table 1.

TABLE 1.

Comparisons of the frequencies of vibration of the model obtained by the calculations and the measurements.

combination of modes in x, y directions	number of frequency cps			
	by the author		by Okuda-Arima	
	theoretical	experimental	theoretical	experimental
I - I	589	590	745	615
I -III	628	620	—	—
III-III	758	795	—	—
III- I	859	900	—	—

The frequencies of four peaks measured by the experiments are found in good agreements with which obtained by the theory as will be seen in Table 1. The result observed from Okuda-Arima's experiment is shown in the same table. Since the result of Okuda-Arima's experiment has been obtained from the record of the wave pattern of the free vibration, it will be supposed that only the natural frequency of the fundamental mode of the vibration has been picked up as the critical one in their results.

4. Some local vibrations in actual ship. Some of the results obtained by the measurements of the frequencies of local vibrations in actual ships[5] are compared with those computed, as shown in Table 2.

TABLE 2.

Comparisons of the frequencies of some local vibration of actual ship obtained by calculations and measurement. Marks (s) and (c) in the table show the conditions of simply supported edges and of clamped edges respectively in the calculations of the frequencies of vibrations of the panels. Ships A and B type are both D. W. 10,000 ton motor cargo vessels.

name of ship	measured position	scantlings of the panel (mm)	β	δ^2	number of frequency (cpm) theoretical $f_{\mu=\infty}$	$\gamma_1 \times f_{\mu=\infty}$	measured
A	upper dk.	9600×6000×22	6.25	30.5 (s)	750	≒750	700
	bridge dk.	9600×6000×18.5	5.26	22.3 (s)	780	≒780	690
B	eng. casing wall	2350×2000×6	8.89	0.0527 (c) 0.273 (s)	2625 1145	1543 982	1270
	pantry wall	2350×2050×6	6.06	0.130 (c) 0.672 (s)	3800 1660	2660 1500	1260

It will be supposed that one of the reasons of the discrepancies of both results is the disagreement of the boundary conditions of the four edges of the panel considered between the calculations and the actual state. The effect of the coupled vibrations of the plate upon the frequencies of the panel, however, is remarkably found in the case of such panels constructed with relatively thin plate as the house walls and engine casing walls, but little effect is found in the panels with thick plate like as the upper deck panels in the large type of cargo ship. The tendencies of this phenomena are explained by the characteristics of the coupled vibration.

5. Conclusive Remarks and Acknowledgement. The effect of the coupled vibrations of the plate subdivided by the stiffeners upon the vibration of the deck panel were investigated. One of the reason of the discrepancies of the frequencies of the vibrations between the results of the calculations and the experiments was clearly explained by the present investigation. The model expriments on the vibrations of the deck panels were carried out and the results were compared with which obtained by Okuda and Arima. The actual data measured from the local vibrations in ship structure were also compared with the results of calculations.

[5] T. Okabe; The Report of Sub-committee of West Japan of the Ship Structure Committee of Japan Soc. of N. A., No. 2.15.3/6, 1952.

The results of the present calculations of the frequencies of the deck panels show the tendencies that approache to the actual one as will be seen in table 2.

The results presented in this paper are based on the Reports of Subcommittee of West Japan of Ship Structure Committee of Japan Society of Naval Architecture. Author wishes to acknowledge to assistance rendered in the investigation through a grant from the Committee. To Professor Y. Watanabe, Chief of Subcommittee, the author wishes to express his obligation for helpful suggestions in regard to the problems discussed in this paper, also he is greatly indebted to the assistance of Mr. H. Hiyama and Miss. S. Aoki, assistants of the Institute in the phases of the experiments and the numerical calculations respectively. He also wishes to thank Assistant Prof. T. Suhara for his criticisms upon reading the manuscript.

Appendix I

1. Calculations of $f_{\mu=\infty}$;

$$f_{\mu=\infty} = \frac{1}{2\pi}\sqrt{\frac{k_1}{m_1 + m_2}}, \tag{1}$$

$$k_1 = k_x + k_y + k_p, \tag{2}$$

where, k_x, k_y and k_p are the spring stiffness equivalent to those of the stiffeners of two perpendicular directions and of the plate taking the same neutral planes as that of the deck panel.

m_1, m_2; mass of the stiffeners and of the plate respectively.

The deflection of the deck panel during vibration of the rectangular planel with the conditions of simply supported edges, for an example, is shown by the following form

$$w = \delta_0 \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{b}, \tag{3}$$

where,

w; deflection of panel vibration,
δ_0; unit deflection of panel,
a, b; length and breadth in directions x, y respectively,
m, n; numbers of half wave in both directions x, y.

k_x and k_y are yield to the forms of[6]

$$\left.\begin{aligned} k_x &= \frac{m^4\pi^4 g E I_x(q_a+1)}{a^3} \\ k_y &= \frac{n^4\pi^4 g E I_y(q_b+1)}{b^3} \end{aligned}\right\} \tag{4}$$

[6] Okuda-Arima; *loc. cit.* p. 231.

where,

E; Young's modulus.

I_x, I_y; moment of inertia of the sections of stiffeners along x and y directions and whose neutral planes coinside with those of panel respectively,

q_a, q_b; numbers of stiffeners of panel along the directions x and y respectively,

g; gravitational acceleration,

and k_p is presented as the following form after some calculations of the potential energy and of the kinetic energy of the plate considered with the neutral plane of the deck panel,

$$k_p = g D \pi^4 \frac{b}{a^3}\left[\alpha m^4 + \{(\alpha+\beta)\nu + 2\gamma(1-\nu)\} m^2 n^2 \frac{a^2}{b^2} + \beta n^4 \frac{a^4}{b^4}\right], \quad (5)$$

where,

$D = \dfrac{E t^3}{12(1-\nu^2)}$; stiffness of plate with thickness t and Poisson's ratio ν,

$$\alpha = 1 + 12\left(\frac{\eta_x}{t}\right)^2,$$

$$\beta = 1 + 12\left(\frac{\eta_y}{t}\right)^2,$$

$$\gamma = 1 + 12\left(\frac{\eta_x \eta_y}{t^2}\right),$$

η_x, η_y; distance from middle plane of the plate to the neutral planes of the panel in both directions of x, y respectively.

As an extreme case, if the panel has no stiffeners in both directions, α, β and γ take unity because of $\eta_x = \eta_y = 0$, thus k_p yield to ordinary form of the plain rectactangular plate, namely,

$$k_p = g D \pi^4 a \cdot b \left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)^2.$$

2. Calculations of β and δ^2;

$$\beta = \frac{m_2}{m_1} = \frac{\text{mass of the plate of deck panel}}{\text{mass of the stiffeners in both directions}}, \quad (6)$$

$$\delta^2 = \frac{1}{\beta}\frac{k_2}{k_1}, \quad (7)$$

k_1; equivalent spring stiffness of the stiffeners with the combination of given modes of both directions, (see (2))

k_2; sum of the stiffness of the rectangular plain plate subdivided by the stiffeners in both directions, k_2 is easily computed by the approximate formulae, for instance, in Sezawa's paper.[7]

[7] K. Sezawa: "On the Lateral Vibration of a Rectangular Plate Clamped at Four Edges." Rep. of Aero. Res. Inst. No. 70, April, 1931.

Appendix II

The reduction factors (γ_1) for obtain the lower value of the coupled frequency of the vibration of the deck panel for given value of β and δ^2 were computed and are tabulated as follows. The table of the numerical results of the values of γ_2 (higher value of the frequency of coupled vibration) are omitted in the present paper.

δ^2	$\gamma_1(\beta=0.5)$	$\gamma_1(\beta=1)$	$\gamma_1(\beta=1.5)$	$\gamma_1(\beta=2)$	$\gamma_1(\beta=2.5)$	$\gamma_1(\beta=3)$	$\gamma_1(\beta=4)$
0.001	0.0387	0.0447	0.0500	0.0561	0.0600	0.0648	0.0724
0.005	0.0865	0.1000	0.1118	0.1230	0.1326	0.1407	0.1573
0.010	0.1227	0.1414	0.1571	0.1736	0.1877	0.1974	0.2196
0.015	0.1491	0.1742	0.1915	0.2128	0.2302	0.2395	0.2659
0.020	0.1734	0.2000	0.2204	0.2445	0.2581	0.2745	0.3041
0.03	0.2100	0.2414	0.2677	0.2913	0.3121	0.3316	0.3653
0.04	0.2424	0.2773	0.3067	0.3328	0.3595	0.3770	0.4143
0.05	0.2702	0.3085	0.3402	0.3675	0.3934	0.4195	0.4577
0.06	0.2953	0.3360	0.3701	0.3998	0.4259	0.4492	0.4895
0.10	0.3770	0.4244	0.4633	0.4962	0.5245	0.5497	0.5913
0.15	0.4549	0.5064	0.5465	0.5800	0.6081	0.6324	0.6723
0.20	0.5172	0.5685	0.6081	0.6403	0.6670	0.6893	0.7266
0.25	0.5678	0.6180	0.6559	0.6861	0.7111	0.7321	0.7656
0.30	0.6123	0.6587	0.6939	0.7220	0.7451	0.7635	0.7949
0.40	0.6819	0.7207	0.7507	0.7745	0.7941	0.8105	0.8363
0.5	0.7344	0.7653	0.8032	0.8109	0.8278	0.8417	0.8652
0.6	0.7745	0.7987	0.8189	0.8375	0.8516	0.8757	0.8838
07	0.7984	0.8243	0.8423	0.8577	0.8699	0.8812	0.8981
0.8	0.8481	0.8445	0.8599	0.8729	0.8848	0.8944	0.9096
0.9	0.8500	0.8607	0.8741	0.8862	0.8964	0.9050	0.9188
1.0	0.8660	0.8740	0.8858	0.8965	0.9058	0.9246	0.9262
1.5	0.9140	0.9230	0.9221	0.9291	0.9354	0.9408	0.9494
2.0	0.9373	0.9364	0.9410	0.9462	0.9508	0.9549	0.9616
2.5	0.9408	0.9492	0.9526	0.9567	0.9609	0.9637	0.9695
3.0	0.9592	0.9577	0.9604	0.9637	0.9668	0.9696	0.9741
3.5	0.9658	0.9638	0.9660	0.9688	0.9694	0.9738	0.9777
4.0	0.9603	0.9684	0.9702	0.9726	0.9749	0.9770	0.9804
4.5	0.9727	0.9719	0.9745	0.9756	0.9777	0.9795	0.9825
5.0	0.9765	0.9747	0.9756	0.9781	0.9800	0.9816	0.9843
6.0	0.9806	0.9790	0.9801	0.9816	0.9832	0.9845	0.9869
7.0	0.9835	0.9820	0.9827	0.9842	0.9855	0.9873	0.9888
8.0	0.9856	0.9842	0.9850	0.9862	0.9873	0.9884	0.9902
9.0	0.9872	0.9860	0.9867	0.9877	0.9887	0.9897	0.9912
10.0	0.9885	0.9874	0.9880	0.9889	0.9899	0.9907	0.9920

δ^2	$\gamma_1 (\beta=5)$	$\gamma_1 (\beta=6)$	$\gamma_1 (\beta=7)$	$\gamma_1 (\beta=8)$	$\gamma_1 (\beta=9)$	$\gamma_1 (\beta=10)$
0.001	0.0793	0.0857	0.0916	0.0972	0.1000	0.1048
0.005	0.1714	0.1852	0.1979	0.2088	0.2190	0.2297
0.010	0.2393	0.2571	0.2734	0.2893	0.3033	0.3163
0.015	0.2893	0.3108	0.3292	0.3472	0.3633	0.3788
0.020	0.3300	0.3529	0.3746	0.3934	0.4117	0.4221
0.03	0.3949	0.4211	0.4445	0.4524	0.4801	0.5029
0.04	0.4459	0.4750	0.4983	0.5202	0.5403	0 5583
0.05	0.4880	0.5167	0.5418	0.5640	0.5843	0.6024
0.06	0.5233	0.5524	0.5786	0.6000	0.6200	0.6379
0.10	0.6254	0.6536	0.6776	0.6974	0.7165	0.7326
0.15	0.7037	0.7293	0.7507	0.7689	0.7845	0.7980
0.20	0.7551	0.7778	0.7967	0.8127	0.8261	0.8377
0.25	0.7954	0.8117	0.8284	0.8423	0.8541	0.8642
0.30	0.8181	0.8366	0.8560	0.8640	0.8746	0.8834
0.40	0.8557	0.8710	0.8834	0.8934	0.9019	0.9091
0.50	0.8804	0.8893	0.9037	0.9124	0.9195	0.9256
0.60	0.8979	0.9092	0.9182	0.9256	0.9303	0.9369
0.70	0.9109	0.9208	0.9288	0.9353	0.9407	0.9453
0.80	0.9210	0.9299	0.9370	0.9429	0.9476	0.9517
0.90	0.9291	0.9370	0.9436	0.9486	0.9531	0.9569
1.0	0.9377	0.9430	0.9488	0.9536	0.9576	0.9609
1.5	0.9560	0.9610	0.9651	0.9684	0.9713	0.9743
2.0	0.9666	0.9704	0.9736	0 9762	0.9782	0.9799
2.5	0.9731	0.9762	0.9787	0.9815	0.9825	0.9838
3.0	0.9774	0.9801	0.9822	0.9840	0.9853	0.9866
3.5	0.9807	0.9830	0.9846	0.9861	0.9874	0.9883
4.0	0.9830	0.9849	0.9867	0.9879	0.9889	0.9800
4.5	0.9848	0.9867	0.9881	0.9892	0.9902	0.9911
5.0	0.9863	0.9879	0.9898	0.9904	0.9912	0.9919
6.0	0.9886	0.9899	0.9911	0.9920	0.9927	0.9933
7.0	0.9903	0.9913	0.9923	0.9931	0.9937	0.9941
8.0	0.9915	0.9924	0.9933	0.9940	0.9944	0.9949
9.0	0.9924	0.9932	0.9939	0.9947	0.9952	0.9955
10.0	0.9931	0.9939	0.9945	0.9951	0.9974	0.9960

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