VIBRATION OF A HUGE TANKER WITH SPECIAL CONSIDERATION TO ATHWARTSHIP FLEXIBILITY

KUMAI, Toyoji
Research Institute for Applied Mechanics, Kyushu University : Professor

出版情報：Reports of Research Institute for Applied Mechanics. 15 (51), pp.101-109, 1967. 九州大学応用力学研究所

Reports of Research Institute for Applied Mechanics
Vol. XV, No. 51, 1967

VIBRATION OF A HUGE TANKER WITH SPECIAL CONSIDERATION TO ATHWARTSHIP FLEXIBILITY*

By Toyoji KUMAI**

Abstract. The paper considers the eigen values and the corresponding modes of the vertical vibration of the rectangular box girder similar in plan to a huge tanker taking into account athwartship flexibility under an approximate solution. For confirming existence of spectra other than that in a simple beam some experiments by use of wooden model are attempted.

Introduction

If the hull of a huge tanker should be in resonance with the vertical vibration in the higher mode excited by the propeller blade force and moment, a half wave length of the longitudinal mode of the vertical vibration would become shorter than the breadth of the hull.

On the other hand, we will notice that the stiffness of the hull structure of a huge tanker per unit tank length in both the longitudinal and athwartship directions is approximately identical. Consequently, the athwartship flexibility should be considered in the calculation of the higher natural frequency of the huge tanker.

The present paper provides, as a preliminary work, the approximate solution of the fundamental equation of the flexural vibration of a homogeneous rectangular box approximating the hull of a huge tanker considering the shear deflection and the rotatory inertia in the athwartship as well as longitudinal direction. The eigen values are presented as numerical examples for two types of athwartship modes other than that of a Timoshenko beam. It is to be noted that the virtual mass of water in the vibration of a huge tanker with consideration to athwartship flexibility can be expected to be reduced compared to the case of vibration of a hull taken as a simple beam. Some results of the experiments by use of the box beam are shown for qualitatively confirming the above theory.

Fundamental equation and eigen values

The fundamental equation of the flexural vibration of a homogeneous rectangular plate, with consideration to the shear deflection and rotatory inertia of

*Published on *European Shipbuilding*, No. 3, Vol. XVI, 1967, and also on J. S. N. A. Seibu, No. 33, Feb. 1967.

**Professor of Kyushu University, member of Research Institute for Applied Mechanics, Kyushu University, Fukuoka, Japan.

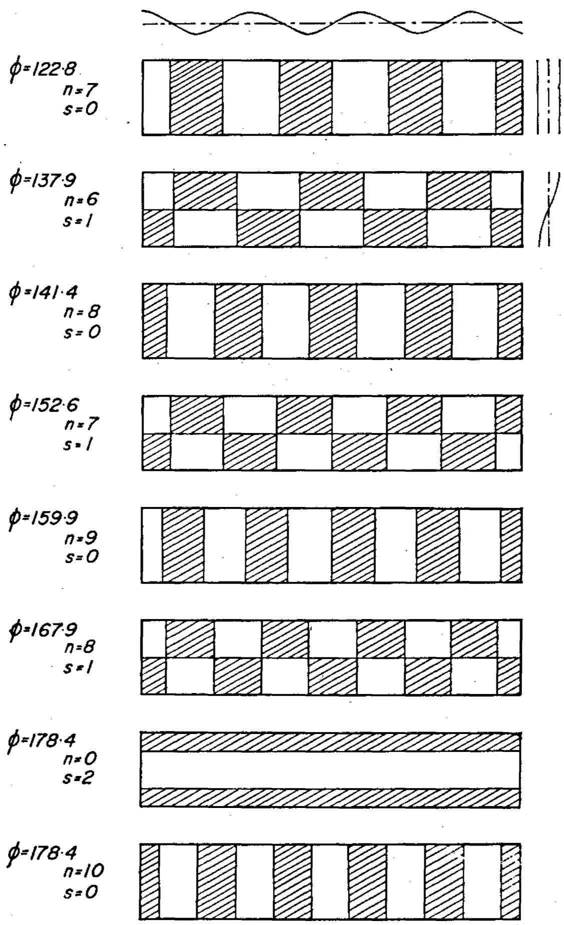

Fig. 2. Vibration pattern of rectangular hull.

calculated by equation (4) using the above numerical values, and the results are shown in Figure 1. As will be seen in the figure, it should be noted that the spectra in the case of $s=1$ and $s=2$ of the vibration of the hull appear comparatively in the range of lower eigen values. Figure 2 illustrates some of the mode configurations corresponding to the above natural vibrations from $n=7$ with $s=0$ to $n=10$ with $s=0$, successively for the eigen values. It is to be noted that there appear several spectra of $s=1$ or $s=2$ between the successive modes of $s=0$.

Experimental study by the use of a wooden box beam

In order to confirm the above theory, some vibration tests of a box beam with the longitudinal and transverse bulkheads were carried out in air and in water. The length, breadth and depth of the box beam are 150 cm, 30 cm and 15 cm, respectively. The model is made of wooden sheet of 0.55 cm thickness, except 0.30 cm in the deck plate. Two rows of longitudinal bulkheads and fourteen sheets of transverse bulkheads are set up at equal distances in this box beam.

The model is put on two soft bolsters and the exciting force is applied at a corner of the beam to give vertical flexual vibration of the model. This excitation means that the model is subject to a vertical force as well as a twisting moment, acting at one end of the box beam. The exciting couple is applied at the end of the beam in order to determine the spectra of the torsional vibration of the beam. The natural frequencies and the corresponding modes of the vibrations

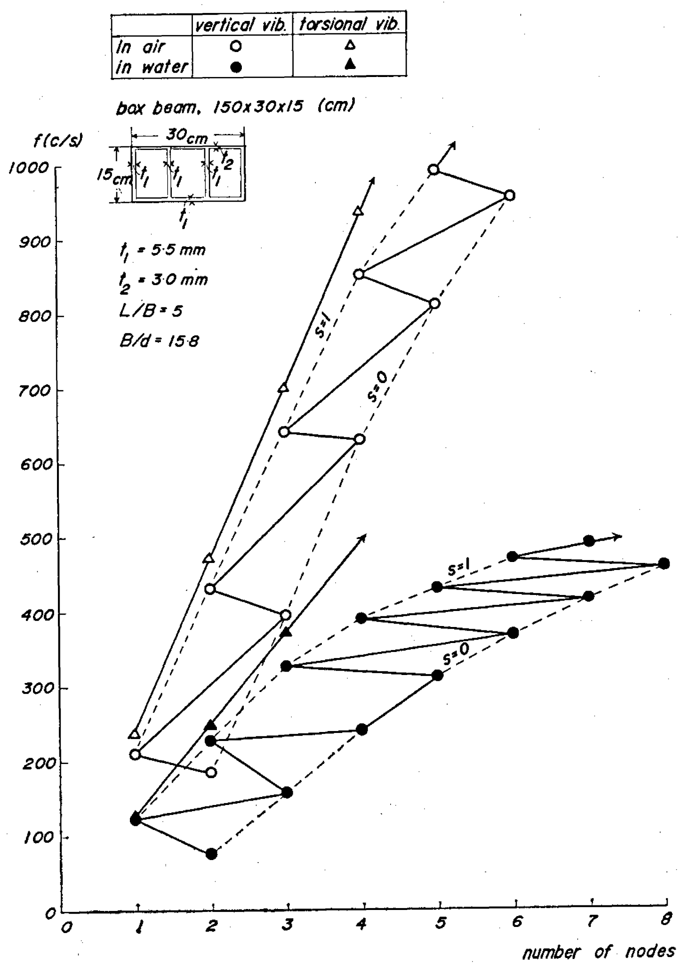

Fig. 3. Measured natural frequencies of wooden model of rectangular hull in air and in water.

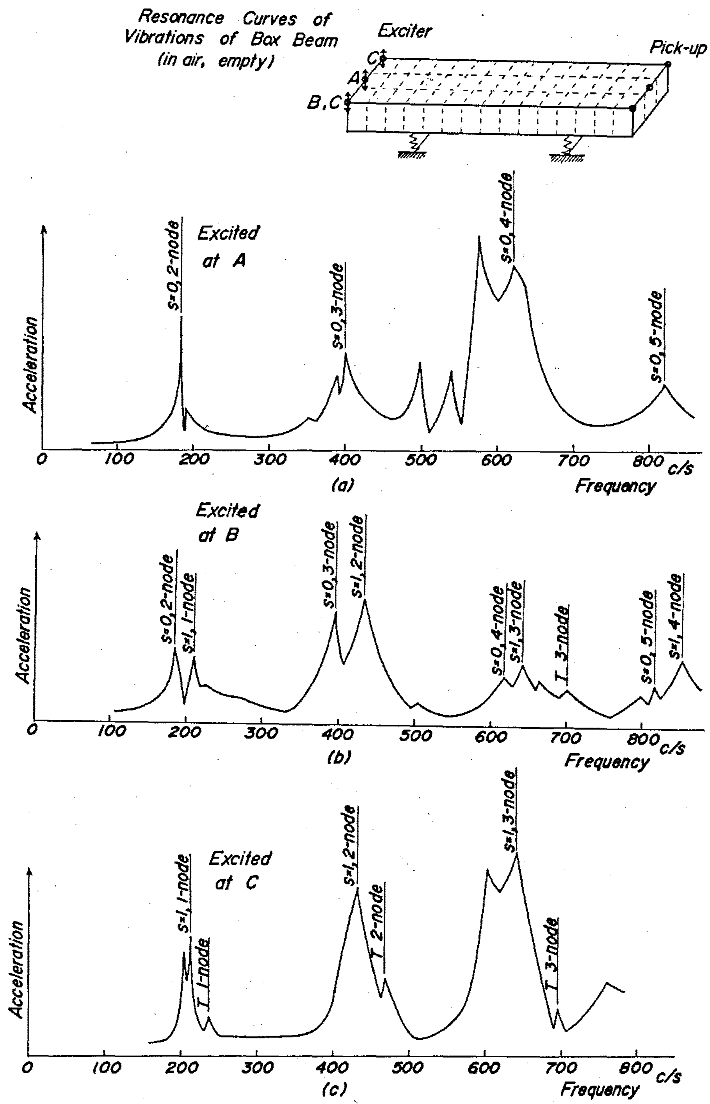

Fig. 4. Resonance curves of vertical flexural and torsional vibrations of wooden model hull.

were measured and the present theoretical approach was qualitatively confirmed as shown in Figures 3 and 4, and Table I.

In the results of the vibration tests on the above box beam, it is to be noted that the natural frequencies of the torsional vibration of the beam clearly departs from that of the vertical flexural vibration associated with athwartship flexibility, or in the case of $s=1$, as seen in the Figures 3 and 4 (c).

Table I Measured natural frequencies and virtual inertia coefficients of entrained water.

$\tau=(f_a/f_w)^2-1$

B/d	s	n	f_a(c/sec)	f_w(c/sec)	τ	τ_{n1}/τ_{n0}
15.8	0	2	189	75.7	5.233	
		3	399	159.9	5.225	
		4	619	241	5.595	
		5	815	313	5.780	
		6	960	369	5.768	
		7		415		
	1	1	212	123.4	1.951	
		2	432	231	2.497	0.4772
		3	643	328	2.842	0.5439
		4	854	390	3.795	0.6783
		5	995	467	3.541	0.6126
		6		474		
	1T	1	235	123	2.648	
		2	469	247	2.605	
		3	698	370	2.557	
		4	936			
		5	1221			
		6	1405			

If the vertical exciting force should be given at the centre line of the beam, the spectra corresponding to $s=1$ would not appear as seen in Figure 4 (a). In the vibration on board actual ships, however, the propeller excitation always consists of force and moment in both the vertical and horizontal directions. Furthermore, in a twin-screw ship, an additional exciting vibratory moment will be expected to occur on board a huge tanker. So the spectra $s=1$ other than $s=0$ will possibly also occur on board actual ships as seen in Figure 4 (b).

Conclusions

The natural flexural vibration of a rectangular box girder similar to a huge tanker in the plan was considered. It is to be noted that there appear spectra other than that of a simple beam, because the athwartship flexibility of the ship hull should be considered in the higher modes of natural hull vibration of a huge tanker. The theoretical results were qualitatively confirmed by the observation of vibration tests by use of wooden box girder.

There are no ship data as yet on the vibration measurements on board huge tanker concerning above spectra, however, it is to be expected that the above spectrum may be one of the numerous measured peaks other than that of a simple beam on board actual ship.

References

1) Timoshenko, S.: Vibration Problems in Engineering. 1937, p. 338.
2) Timoshenko, S.: Theory of Plates and Shells. 1940, p. 85.

(Received January 9. 1968)

Appendix

Fundamental equation of the vibration of a plate considering shear deflection

Consider u the deflection of the vibrating plate due to bending moment and $(w-u)$ the shear deflection of the plate in the cartesian co-ordinate, the component of shearing force Q_x and Q_y in an element of the plate is thus written by

$$\left.\begin{aligned} Q_x &= k'Ga\left(\frac{\partial w}{\partial x} - \frac{\partial u}{\partial x}\right), \\ Q_y &= k'Ga\left(\frac{\partial w}{\partial y} - \frac{\partial u}{\partial y}\right). \end{aligned}\right\} \tag{a}$$

The equation of the shearing vibration of the plate is shown that,

$$\frac{\partial Q_x}{\partial x} + \frac{\partial Q_y}{\partial y} = \rho\frac{\partial^2 w}{\partial t^2}. \tag{b}$$

Substituting from (a) into (b), the above equation becomes

$$\nabla^2 w - \nabla^2 u = \frac{\rho}{k'Ga}\frac{\partial^2 w}{\partial t^2}. \tag{c}$$

The equation of motion of the element due to bending moment in x and y directions are respectively given by[2)]

$$\left.\begin{aligned} -\frac{\partial M_{yx}}{\partial y} - \frac{\partial M_x}{\partial x} + Q_x &= \rho r^2\frac{\partial^2}{\partial t^2}\left(\frac{\partial u}{\partial x}\right), \\ \frac{\partial M_{xy}}{\partial x} - \frac{\partial M_y}{\partial y} + Q_y &= \rho r^2\frac{\partial^2}{\partial t^2}\left(\frac{\partial u}{\partial y}\right), \end{aligned}\right\} \tag{d}$$

where,

$$\left.\begin{aligned} M_x &= -D\left(\frac{\partial^2 u}{\partial x^2} + \nu\frac{\partial^2 u}{\partial y^2}\right), \\ M_y &= -D\left(\frac{\partial^2 u}{\partial y^2} + \nu\frac{\partial^2 u}{\partial x^2}\right), \\ M_{xy} &= -M_{yx} = D(1-\nu)\frac{\partial^2 u}{\partial x\partial y} \end{aligned}\right\} \tag{e}$$

Substituting (e) into (d) and also eliminating $\left(\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y}\right)$ and $\nabla^2 u$ from (c), the following equation is obtained

$$\nabla^4 w + \frac{\rho}{D}\frac{\partial^2 w}{\partial t^2} - \frac{\rho}{k'Ga}\frac{\partial^2}{\partial t^2}\nabla^2 w + \frac{\rho r^2}{D}\frac{\partial^2}{\partial t^2}\left(\nabla^2 w - \frac{\rho}{k'Ga}\frac{\partial^2 w}{\partial t^2}\right) = 0 \tag{1}$$

The above equation correspond to the Timosenko equation of the vibration of an elastic beam.[1] If we assume $k'Ga \to \infty$ and also $\rho r^2 \to 0$, we have the well known equation of the vibration of the plate only due to the bending moment that,

$$\nabla^4 w + \frac{\rho}{D}\frac{\partial^2 w}{\partial t^2} = 0 \tag{f}$$