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A Study on the Coupled Torsional-Axial Vibration of Marine Propulsion Shafting System using the Energy Method

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Abstract : Recently, the market trend for marine diesel engine has involved the lower running speeds, larger stroke/bore ratio and higher combustion pressure. Consequently, because of the flexible engine shafting system due to the larger mass, inertia and the more elasticity, the complicated coupled torsional-axial vibrations have occurred in the operating speed range. Also, the vibrations act as an excitation on the hull-structural vibration. To predict the vibration behavior with more accuracy and reliability, many studies have proposed the several kinds of method to calculate the stiffness matrix of crankshaft. However, most of these methods have a weak point to spend much time on three dimensional modeling and meshing work for crankshaft. Therefore, in this work, the stiffness matrix for the crankthrow is calculated using the energy method (Influence Coefficient Method, ICM), with the each mass having 6 degree of freedom. Its effectiveness is verified through the comparison with the stiffness matrix obtained by using the finite element method (FEM) and measured results for actual ships propulsion system.

Key words : *Coupled vibration, Propulsion shafting system, Marine engine, Stiffness matrix, ICM, FEM*

1. Introduction

The development of marine diesel engine toward the higher power and the lower fuel consumption has steadily increased the stroke/bore ratio and combustion pressure. Consequently, the gas excitation from combustion pressure on shafting system is increased, while the crankshaft and hull structure get more flexible. This has caused the shaft troubles due to axial or coupled torsional-axial vibration. In addition, the vibrations can excite under the adverse condition, the hull vibration in the longitudinal and vertical direction. To cope efficiently with this matter, it is necessary to predict the coupled torsional-

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axial vibration with reliability and accuracy.

The most of the studies, which have been done for the coupled torsional-axial vibration of crankshaft system have a focus on the modeling method of crankshaft causing the coupled vibration. Jeon, et al has modeled a unit crankthrow as an indeterminate beam with simple support bearing and restricted by the adjacent crankthrow and calculated the stiffness matrix by using the ICM (Influence Coefficient Method).[1][2][3] However, in the forced vibration calculation by this method, the radial force acting on the crank pin should be converted to axial forces on the journal. This is a quite troublesome work. S.B.Jakobsen, A.Kikuchi, Kim, et al have used 3-dimensional crankshaft model to more exactly calculate the stiffness matrix.[4][5][6] But this method has a weak point that calculating processes are complicated and to spend much time.

Therefore in this study, a half crankthrow is modeled as beam element with 6-degrees of freedom in the respective center of crank journal and pin. And a theoretical method to calculate the stiffness matrix of crankshaft is derived with the finite element method and influence coefficient method.

These methods are applied to the actual propulsion shafting system so as to calculate the natural frequencies and modes of coupled torsional-axial vibration. In addition these results were compared with measured ones to investigate the pros and cons of each method.

2. Modeling of Propulsion Shafting System

To analyze the vibration behavior in the propulsion shafting system, the crankshaft is modeled as an equivalent system as shown in Fig.1. Assuming that the external force operates at the center of crank pin and the reaction force operates at the center of crank journal, these were considered to be mass points. Coordinate of propulsion shafting system is defined to be origin at the mass point of the foremost main bearing.

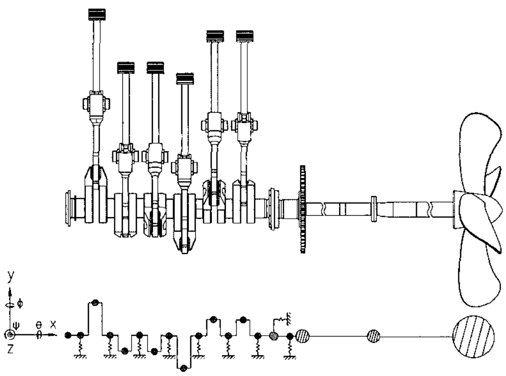


Fig. 1 Equivalent mass-elastic model of shafting system

The center line of the shaft is defined to be x-axis and y-axis is put on the plane vertical to x-axis and direction toward center line of the crank pin. z-axis is defined according to the right handed law. Every mass in the equivalent system has 6 degrees of freedom: translating displacement in x, y, z-axis direction and θ, ϕ, φ -motion around each x, y, z-axis. And the crankthrow is modeled to spring-mass system as shown in Fig.2.

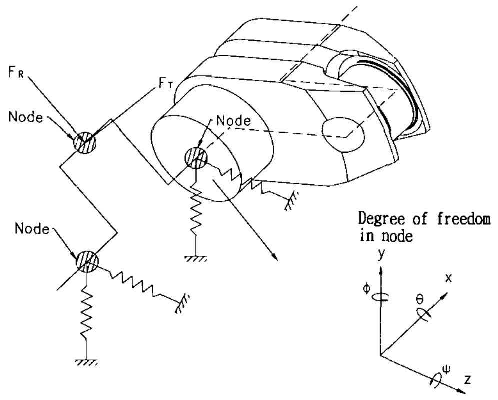


Fig. 2 Modeling of crankthrow

For the distribution of mass, a crankthrow is divided into 3 equivalent masses and each mass is put on main bearings and crank pin. The mass moments of inertia around x-axis located at the centre of main bearing is determined by summing a half of equivalent mass moments of inertia of piston and connecting rod reduced to shaft line from the adjacent cylinder respectively. The mass moment of inertia of crankthrow is allocated into crank pin. The mass moments of inertia around y, z-axis are defined by a half of mass moments of inertia around x-axis that is, $I_y = I_z = I_x/2$.

The crankshaft consists of a unit crankthrow assembled with angles. Considering that the crankthrow is symmetric at the center of crank pin, a half crank throw is taken as a unit element for calculating the crankshaft stiffness matrix.

2.1 Coupled Stiffness Matrix of Crankthrow

2.1.1 Analysis of Coupled Stiffness Matrix by FEM [4]

Based on a finite element model of a half crankthrow shown in Fig.3, the stiffness matrix of crankshaft is generated. In order to have 6 degrees of freedom at main journal and crankpin center (Fig.2), the following procedure is applied

$$[S]\{\delta\} = \{F\} \tag{1}$$

where $[S]$ is the stiffness matrix and $\{F\}$ is the external force vector. Displacement vector $\{\delta\}$ consists of translating motion u, v, w at all nodes and then is divided into the master DOF $\{\delta_m\}$ and slave DOF $\{\delta_s\}$. Namely, displacement vector of the nodes on the vertical plane exposed to external and reaction force is defined as master DOF, the rest is defined as slave DOF.

The stiffness matrix $[S]$ in eq.(1) is partitioned into submatrices $[S_{mm}]$, $[S_{ms}]$, $[S_{sm}]$ and $[S_{ss}]$ and rearranged as Eq.(2)

$$\left[\frac{[S_{mm}]\ [S_{ms}]}{[S_{sm}]\ [S_{ss}]}\right]\left\{\frac{\{\delta_m\}}{\{\delta_s\}}\right\} = \left\{\frac{\{F_m\}}{\{0\}}\right\} \tag{2}$$

where $\{F_m\}$ contains external force for the master DOF and $\{0\}$ contains only zeros. Rearranging Eq. (2) for master DOF, the reduced stiffness matrix in Eq.(3) is obtained

$$\begin{aligned} &[K_{Ansys}]\{\delta_m\} = \{F_m\} \\ &[K_{Ansys}] = [[S_{mm}] - [S_{ms}][S_{ss}]^{-1}[S_{sm}]] \end{aligned} \tag{3}$$

where $[K_{Ansys}]$ is considered as the master DOF stiffness matrix

In order to calculate the stiffness matrix for beam element, the master DOF in Eq.(3) is further reduced to 6 degrees of freedom at each main journal and crankpin center in Eq.(4). The details for procedure are not more described in this paper.

$$[K]\{\triangle_{bm}\}=\{F_{bm}\} \tag{4}$$

Where $[K]$ is considered as the stiffness matrix for beam element with 6 degrees $\delta_x, \delta_y, \delta_z, \theta, \psi, \varphi$) of freedom at each mass.

2.1.2 Analysis of Coupled Stiffness Matrix by ICM [2]

In this section, the calculation of coupled stiffness matrix by ICM is described. A half crankthrow is divided into three parts such as crank journal, arm and pin as shown on Fig.4. The symbols in Fig.4 are defined as follows. J_1, J_2 : moments of inertia of journal and pin respectively. J_{JP}, J_{PP} : polar moments of inertia of journal and pin respectively.

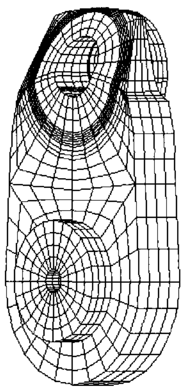


Fig. 3 Modeling of a half crankthrow

J_X, J_Y, J_Z : moments of inertia of arm in each cross sectional area respectively. $J_Y = cwh^3$, where c being a constant depending on the ratio w/h . And the elastic deformation that occurs in acting a unit load on each element is easily obtained by applying Castigliano's theorem.

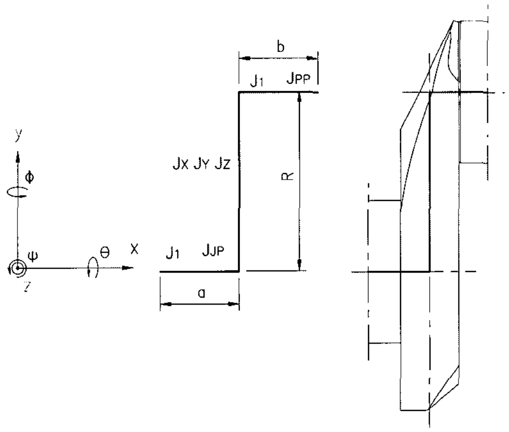


Fig. 4 Equivalent beam model for a half crankthrow

1) Influence numbers under $F_{XX}=1$ in X-direction

As shown in Fig.5, when a unit force $F_{XA}=1$ in the X-direction acts on A-end with B-fixed end, then X-displacement of A-end x_A is derived and leads to the expression of the influence number in X-direction, $f_{XX}=x_A/F_{XA}$ shown in Eq.(5).

$$f_{XX}=\frac{R^3}{3EJ_Z}+\frac{R^2 b}{EJ_2} \tag{5}$$

Considering the shear deformation of crank arm, the compression displacement of crank journal and pin, the resultant influence number $\overline{f}_{XX}$ is calculated as follows

$$\overline{f}_{XX}=\frac{6}{5Gwh}(R-(D_J+D_P)/2)+\frac{L_J}{EA_J}+\frac{L_P}{EA_P}+f_{XX} \tag{6}$$

where D_J, D_P are the outer diameter of crank journal and pin, w and h are the width and height of the rectangular arm section parallel to Z - and X -axis respectively. L_J, A_J, L_P, A_P denote the length and sectional area for crank journal, pin respectively. Similarly, the influence numbers for $f_{YX}, f_{\psi X}$, are calculated as follows.

$$f_{YX} = -\frac{R^2 a}{2EJ_Z} - \frac{R}{2EJ_2}\{(a+b)^2 - a^2\} \qquad (7)$$

$$f_{\psi}X = \frac{R^2}{2EJ_Z} + \frac{Rb}{EJ_2} \qquad (8)$$

Further, the relation $f_{ij} = f_{ji}$ holds for any i and j according to Maxwell-Betti's reciprocal theorem.

2) Influence numbers under $F_{YA} = 1$ in Y -direction

As shown in Fig. 6, when a unit force $F_{YA} = 1$ in the Y -direction acts on A-end with B-fixed end, the influence numbers, f_{YY} and $f_{\psi Y}$ are calculated as follows.

$$f_{YY} = -\frac{a^3}{3EJ_1} + \frac{Ra^2}{EJ_Z} + \frac{1}{3EJ_2}\{(a+b)^3 - a^3\} \qquad (9)$$

$$f_{\psi Y} = -\frac{a^2}{2EJ_1} + \frac{Ra}{EJ_Z} + \frac{1}{2EJ_2}\{(a+b)^2 - a^2\} \qquad (10)$$

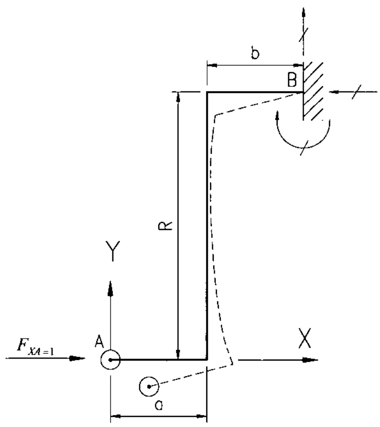


Fig. 5 Journal center under loading in X - direction

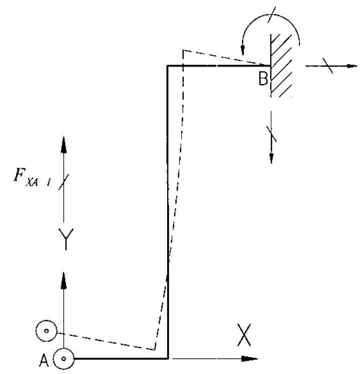


Fig. 6 Journal center under loading in Y-direction

Considering the shear deformation for crank journal and pin, the compression displacement of arm, the resultant influence coefficient $\overline{f}_{YY}$ is calculated as follows.

$$\overline{f}_{YY} = \frac{(R-(D_J+D_P)/2)}{Ewh} + \frac{10}{9G}\left(\frac{L_J}{A_J} + \frac{L_P}{A_P}\right) + f_{YY} \qquad (11)$$

3) Influence numbers under $F_{ZA} = 1$ *in Z-direction*

As shown in Fig.7, when a unit force $F_{ZA} = 1$ in the Z-direction acts on A-end with B-fixed end, the corresponding influence number of f_{ZZ}, $f_{\Theta Z}$ and $f_{\Theta Z}$ are calculated as follows.

$$f_{ZZ} = \frac{a^3}{3EJ_1} + \frac{R^3}{3EJ_X} + \frac{Ra^2}{GJ_Y} + \frac{1}{3EJ_2} (a+b)^3 - a^3\} + \frac{R^2 b}{GJ_{PP}} \qquad (12)$$

$$f_{\Theta Z} = -\frac{R^2}{2EJ_X} - \frac{RB}{GJ_{PP}} \qquad (13)$$

$$f_{\Theta Z} = \frac{a^2}{2EJ_1} + \frac{Ra}{GJ_Y} + \frac{1}{2EJ_2}\{(a+b)^2 - a^2\} \qquad (14)$$

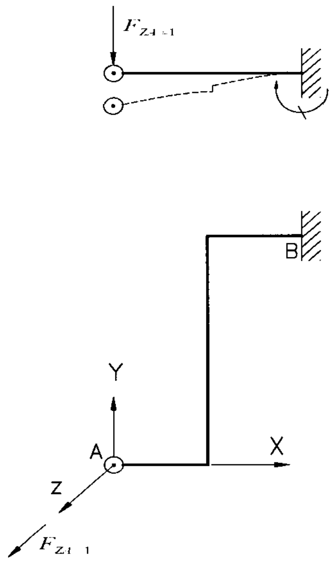


Fig. 7 Journal center under loading in *Z*-direction

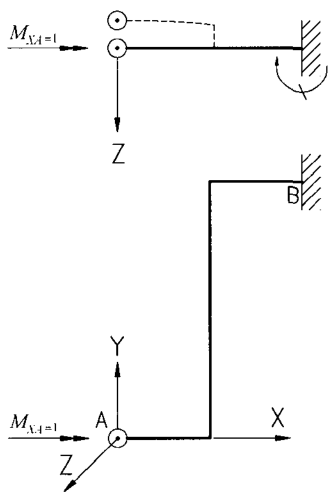


Fig. 8 Journal center under loading around *X*-axis

Considering the shear deformation of crank journal, pin and arm, the resultant influence number, $\overline{f}_{ZZ}$ is calculated as follows.

$$\overline{f}_{ZZ}=\frac{6(R-(D_J+D_P)/2)}{5Gwh}+\frac{10}{9G}\left(\frac{L_J}{A_J}+\frac{L_P}{A_P}\right)+f_{ZZ} \tag{15}$$

4) Influence numbers under $M_{XA}=1$ around X-axis

As shown in Fig.8, when a unit torque $M_{XA}=1$ around X-direction acts on A-end with B-fixed end, the influence number, $f_{\Theta\Theta}$ is calculated as follows.

$$f_{\Theta\Theta}=\frac{a}{GJ_{JP}}+\frac{R}{EJ_X}+\frac{b}{GJ_{PP}} \tag{16}$$

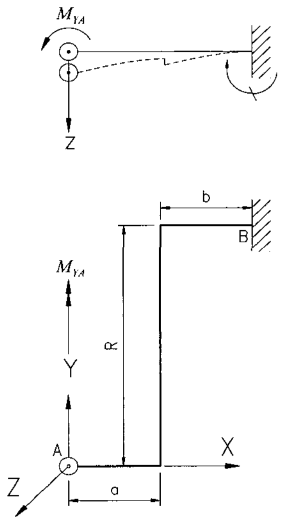


Fig. 9 Journal center under loading around *Y*-axis

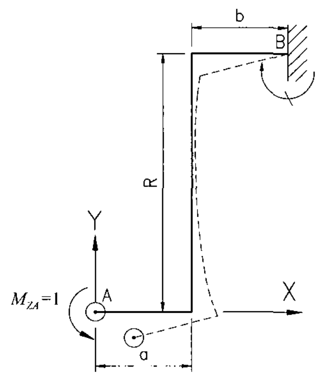


Fig. 10 Journal center under loading around *Z*-axis

5) Influence numbers under $M_{YA}=1$ around Y-axis

As shown in Fig.9, when a unit

moment $M_{YA}=1$ around Y-direction acts on A-end with B-fixed end, the influence number, $f_{\Theta\Theta}$ is calculated as follows.

$$f_{\Theta\Theta}=\frac{a}{EJ_1}+\frac{R}{GJ_Y}+\frac{b}{EJ_2} \tag{17}$$

6) Influence numbers under $M_{ZA}=1$ around Z-axis

As shown in Fig.10, when a unit moment $M_{ZA}=1$ around Z-direction acts on A-end with B-fixed end, the influence number, $f_{\Psi\Psi}$ is calculated as follows.

$$f_{\Psi\Psi}=\frac{a}{EJ_1}+\frac{R}{EJ_Z}+\frac{b}{EJ_2} \tag{18}$$

2.1.3 Stiffness Matrix of a Unit CrankThrow

Using the influence numbers derived in the previous section, the relation between forces and displacements can be presented in the form of matrix as follows

$$\{X_A\}=[F_{AA}]\{F_A\},\ [F_{AA}]=$$

$$\begin{bmatrix} f_{XX} & f_{XY} & 0 & 0 & 0 & f_{X\psi} \\ f_{YX} & f_{YY} & 0 & 0 & 0 & f_{Y\psi} \\ 0 & 0 & f_{ZZ} & f_{Z\Theta} & f_{Z\Phi} & 0 \\ 0 & 0 & f_{\Theta Z} & f_{\Theta\Theta} & 0 & 0 \\ 0 & 0 & f_{\Phi Z} & 0 & f_{\Phi\Phi} & 0 \\ f_{\psi X} & f_{\psi Y} & 0 & 0 & 0 & f_{\psi\psi} \end{bmatrix} \tag{19}$$

where $[F_{AA}]$ is a symmetric matrix. $[K_{AA}]$ is a stiffness matrix for beam element loaded at A-end with B-fixed end. This is the inverse matrix of $[F_{AA}]$. Accordingly, the Eq.(19) can be expressed in terms of stiffness matrix as follows.

$$\{F_A\}=[K_{AA}]\{X_A\} \tag{20}$$

The reaction matrix at B-fixed end, $[K_{BA}]$ is derived from the equilibrium condition under the external loads.

$$\{F_B\}=[A_R]\{F_A\}=[A_R][K_{AA}]\{X_A\}=[K_{BA}]\{X_A\}$$

$$[A_R]=\begin{bmatrix} -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & -1 & 0 & 0 & 0 \\ 0 & 0 & R & -1 & 0 & 0 \\ 0 & 0 & -L & 0 & -1 & 0 \\ -R & L & 0 & 0 & 0 & -1 \end{bmatrix} \tag{21}$$

When a unit load acts on B-end with A-fixed end, $[F_{BB}]$ is calculated by the same procedure. The stiffness matrix of $[F_{BB}]$. The reaction matrix at A-end, $[K_{BA}]$ can be calculated from the equilibrium condition. Combination of these matrices gives an overall stiffness matrix as follows.

$$[K^{AB}]=\left[\frac{K_{AA}}{K_{BA}}\ \ \frac{K_{AB}}{K_{BB}}\right] \tag{22}$$

where $[K^{AB}]$ presents the stiffness matrix for beam element AB.

3. Coupled Free Vibration of Actual Ships Propulsion Shafting System

In this chapter, the crankshaft stiffness matrix for Ship(A) and Ship(B) were calculated using both the FEM and the ICM. And the calculated coupled free vibrations are compared with the measured values. Table 1 shows the general particulars of the actual ships shafting system. The dimensions of crank throw are shown in Fig.11 and 12.

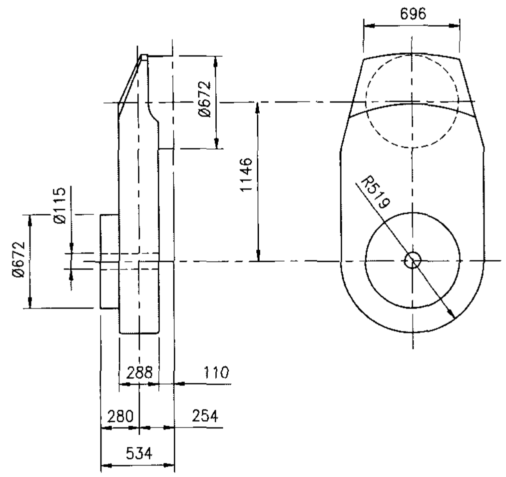


Fig. 11 Crankthrow of ship(A)

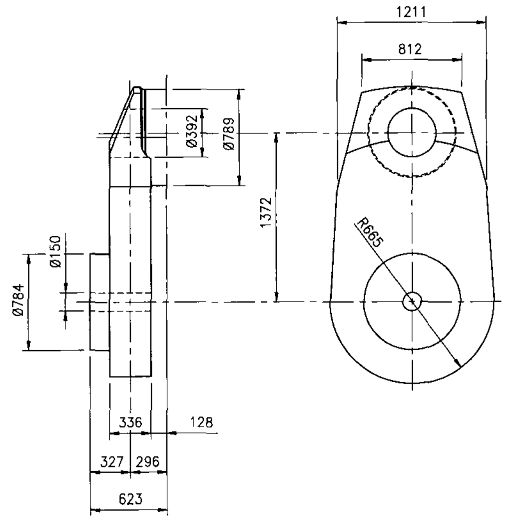


Fig. 12 Crankthrow of ship(B)

3.1 Stiffness Matrix Analysis by Finite Element Method and Influence Coefficient Method

For the calculations of the coupled stiffness matrix for crankthrow shown on Fig.11 and 12, the solid model for crankthrow was meshed by hexahedron elements with 3 degree of freedom at each node. The number of elements with respect to a half crankthrow is of about 5300 for ship(A), of about 4200 for ship(B). Based on these crankthrow, the coupled stiffness matrix was analyzed by using both the FEM and ICM.

Table 1 General particulars of the actual ship's shafting system

Name of ship	Ship(A)	Ship(B)
Engine Type	MAN B&W 6S60MC	MAN B&W 7S70MC
Max. Output	10959kW	19667kW
MCR	94 rpm	91 rpm
M.E.P	18 bar	18 bar
Bore	600 mm	700 mm
Stroke	2250 mm	2674 mm
Firing Order	1-5-3-4-2-6	1-7-2-5-4-3-6
Propeller Type	FPP	FPP
Propeller Dia.	7.3m	8.4m
Blade No.	4	4

The stiffness coefficients having a big effect on the torsional and axial vibration are shown on Table 2 and 3 respectively. And A denotes the results by FEM, B denotes the results by ICM. Comparisons between the results show a good agreement except for the stiffness coefficient k_{xx} in X- direction and $k_{x\varphi}$ coupled with φ-displacement around Z-axis. It is thought that the deviation by more than 50% in the stiffness coefficient k_{xx} and $k_{x\varphi}$ is caused by the modeling method of crank fillet part.

3.2 Comparison of Coupled Free Vibration of the Actual Propulsion Shafting System

In order to investigate the effect of coupled stiffness matrix on the natural frequencies and modes of torsional and axial vibration, the coupled free vibrations were analyzed and compared with the measured ones. Table 4 and 5 are shown the lower natural frequencies gained by the FEM and ICM and also, the measured results. The results by the conventional uncoupled method are in addition presented. There are shown that the results made by two methods are in a good agreement with the measured ones. Therefore, two methods are applicable for actual shafting system.

Fig. 13 and 14 are shown the relative amplitude curves corresponding to both methods and also, curves derived from the conventional method using axial and torsional stiffness given by engine

Table 2 Comparison of calculated stiffness of crankthrow-Ship(A)

Method	$k_{xx}\times 10^9$ N/m	$k_{x\varphi}\times 10^9$ N/rad	$k_{\theta\theta}\times 10^9$ Nm/rad	$k_{\theta\phi}\times 10^8$ Nm/rad	$k_{\phi\phi}\times 10^8$ Nm/rad	$k_{\varphi\varphi}\times 10^9$ Nm/rad
A	4.144	-1.882	4.182	-1.016	9.657	2.862
B	2.724	-1.039	3.630	-8.916	8.621	3.346
A/B	1.52	1.18	1.15	1.14	1.12	0.86

Table 3 Comparison of calculated stiffness of crankthrow-Ship(B)

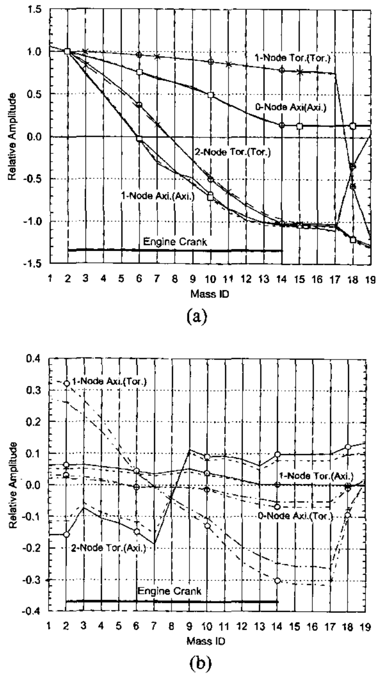
Method	$k_{xx}\times 10^9$ N/m	$k_{x\varphi}\times 10^9$ N/rad	$k_{\theta\theta}\times 10^9$ Nm/rad	$k_{\theta\varphi}\times 10^9$ Nm/rad	$k_{\phi\phi}\times 10^9$ Nm/rad	$k_{\varphi\varphi}\times 10^{10}$ Nm/rad
A	4.144	-1.882	4.182	-1.016	9.657	2.862
B	2.724	-1.039	3.630	-8.916	8.621	3.346
A/B	1.52	1.18	1.15	1.14	1.12	0.86

Table 4 Comparison between the measured and calculated natural frequencies of propulsion shafting system of Ship(A)

Mode of vibration	Measured (A)	Conventional Method(B)		Coupled-ICM(C)		Coupled-FEM(D)	
		Torsional	Axial				
	C.P.M	C.P.M	C.P.M	C.P.M	(A,B)/C	C.P.M	(A,B)/D
1-Node Tor.	288.0	284.6	-	284.2	1.01	284.2	1.01
0-Node Axi.	462.0	-	462.9	467.0	0.99	478.7	0.97
1-Node Axi.	-	-	986.9	984.4	1.00	986.8	1.00
2-Node Tor.	-	1143.3	-	1124.8	1.02	1151.3	0.99

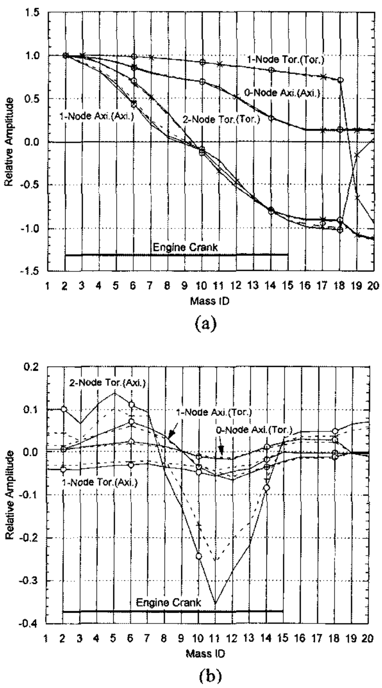
Table 5 Comparison between the measured and calculated natural frequencies of propulsion shafting system of Ship(B)

Mode of vibration	Measured (A)	Conventional Method(B)		Coupled-ICM(C)		Coupled-FEM(D)	
		Torsional	Axial				
	C.P.M	C.P.M	C.P.M	C.P.M	(A,B)/C	C.P.M	(A,B)/D
1-Node Tor.	278.4	274.9	-	274.9	1.01	275.5	1.01
0-Node Axi.	409.0	-	400.7	416.3	0.98	413.1	0.99
1-Node Axi.	848.1	-	837.7	844.3	1.00	841.0	1.00
2-Node Tor.	1071.0	1121.5	-	1104.9	0.97	1120.6	0.96



(○: FEM, □: Conventional, ×: ICM)

Fig. 13 Relative amplitude curve for ship(A)



(○: FEM, □: Conventional, ×: ICM)

Fig. 14 Relative amplitude curve for ship(B)

designer. Fig. 13-(a) and 14-(a) present the mode shapes for the respective torsional and axial vibration. Fig. 13-(b) and 14-(b) show the vibration modes accompanied by axial or torsional vibration. These results have almost the same tendency regardless of the applied methods. Furthermore vibration modes of Fig. 13-(b) and Fig. 14-(b) are generated by the coupled vibration effect.

4. Conclusion

In order to analyze the coupled torsional-axial vibration for propulsion shafting system, a half crankthrow is modeled as a beam element with 6 degrees of freedom at each journal and pin center. Based on a beam model, the method to calculate the stiffness matrix is derived by the finite element method and the influence coefficient method. The methods are applied to the actual shafting system to calculate the coupled free vibration. The calculated results are analyzed and compared with the measured ones. From the above, the conclusions are drawn as follows.

(1) Theoretical method to derive the stiffness matrix of crankshaft is obtained by using the finite element method and influence coefficient method.

(2) The coupled free vibrations for the actual shafting system are calculated with the stiffness matrix obtained in (1). And the results are compared with the measured ones. These show a good agreement.

(3) The influence coefficient method to calculate the stiffness matrix for crankshaft is more simple and easier than the finite element method.
(4) In case the stress analysis caused by the coupled torsional-axial vibration is required, the finite element method is desirable.

Acknowledgement

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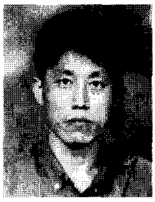

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