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Dynamic Modeling and Model Reduction for a Large Marine Engine

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Key words : Dynamic model, Marine engine structure, Top bracing

Abstract

This article provides a dynamic modeling methodology of engines to be accurate with a small number of degrees of freedom for an active vibration control using a top bracing. First, a finite element (FE) model for the engine structure is constructed so that the size of model is as small as possible where the dynamic characteristics of engine are ensured. Second, a technique is studied to obtain the exact mass and stiffness matrices of the FE model. The size of matrices from the FE model is still too large to apply. Finally, a model reduction is, therefore, conducted to make an appropriate dynamic model for designing and simulating a top bracing. In this article, a dynamic model of a large 9 cylinder engine is constructed and reviewed by comparing its natural frequencies and steady state reponses with those of experimental data provided by manufacturer.

1. Introduction

Although the global modes of large marine engines such as H mode, X mode and L mode have been generated in the low frequency range, the vibrations in the modes may result in fatigue damages and stress concentration in bolts between the engine and the floor plate of ship due to the big inertia forces of heavy engines. A top bracing has been applied to suppress the vibrations with its active force.

In the design step of top bracing, its mathematical model has to be simulated with the engine's model. The dynamic model of engine is required to be accurate with a small number of degrees of freedom(DOF) for an active vibration control using a top bracing. Since the structure of engine is made up of several

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substructure, it is difficult to be modeled as a simple vibrating system. So the dynamic models of most engines, in general, have been conducted by the finite element(FE) method[1],[2] and the substructure synthesis method.[3],[4] FE models of practical engine structures have often very large number of DOF, while the models by substructure synthesis method are not enough to be accurate.[5] This is a dilemma for a good model of engine structure to simulate and design a top bracing.

This article provides a dynamic modeling methodology to overcome the dilemma. First, a FE model for the engine structure using a commercial software ANSYS is constructed so that the size of model is as small as possible where the dynamic characteristics of engine are ensured. Second, a technique is studied to obtain the exact mass and stiffness matrices of the model from ANSYS. The ANSYS model still may be too large to apply. Finally, a model reduction is, therefore, conducted to make an appropriate dynamic model for designing and simulating a top bracing.

In this article, an ANSYS(6)FE model for a large 9 cylinder engine is constructed and reviewed by comparing the natural frequencies of the model with those of experimental data provided by manufacturer. Then an extraction procedure of mass and stiffness matrices from ANSYS was presented with points to be considered in modeling. Finally the matrices are reduced by using Guyan's method(7) and the reduced model is reviewed.

2. ANSYS FE model for 9 cylinder engine

A 9 cylinder engine structure is simplified to be its dynamic model for analysis in low frequency range near global modes. The characteristics of modeling are to:

(1) keep the external dimensions and maintain the total volume and the total mass of the engine, if possible.
(2) use 3 dimensional 8 node brick element(solid45 element of ANSYS, 3 DOF per node) for overall model.
(3) simplify the crank shaft to a quadrilateral cross-section beam.
(4) simplify the connecting rod, piston and components in driving part to a hexahedron.
(5) simplify the thickness of bearing support and web plate to their average
(6) simplify the thickness of cylinder box and adjacent parts to their average.
(7) assume to be welded among the substructures.
(8) apply the boundary conditions in bolts, which mount the engine to main body of ship.

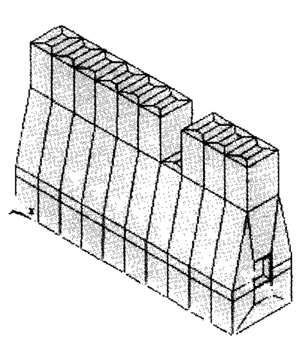

Fig. 1 A simplified FE model for 9 cylinder engine

Fig. 1 shows a simplified FE model for 9 cylinder engine structure using a commercial software ANSYS. This model is conducted for a modal analysis. The H and X mode shapes, two of the analysis results, are shown as Fig. 2 and 3, respectively.

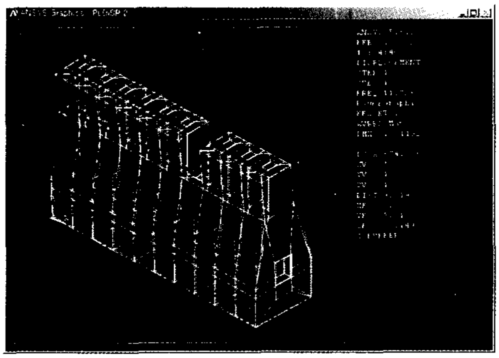

Fig. 2 H mode shape after modal analysis

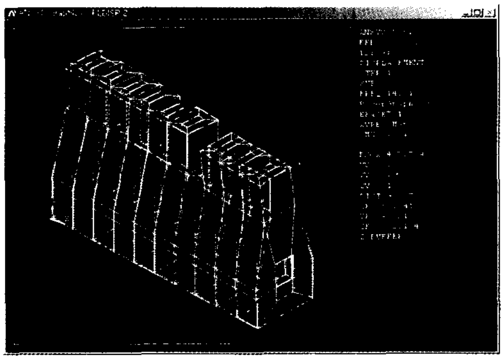

Fig. 3 X mode shape after modal analysis

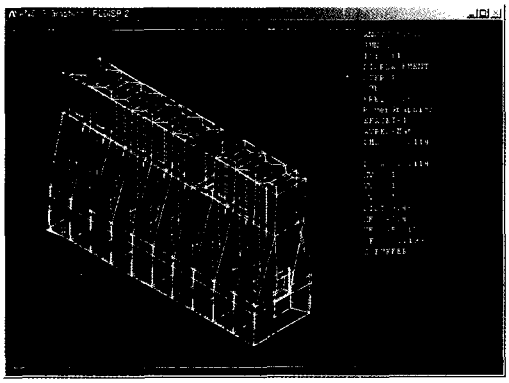

Fig. 4 L mode shape after modal analysis

Table 1. Comparison of natural frequencies

	ANSYS Model	Test Data from manufacturer
H-mode	11.726 Hz	11 Hz
X-mode	14.513 Hz	14 Hz
L-mode	17.896 Hz	17 Hz

Table 1 shows the natural frequencies obtained by the modal analysis of ANSYS model and those from test data provided by the manufacturer of engine. The mode shapes and natural frequencies leads to the accuracy of the ANSYS FE model.

3. Mass and stiffness matrix extraction from ANSYS

The ANSYS model is still too large to apply for simulating and designing a top bracing. So the model has to be reduced. Before reducing, a technique should be developed to obtain the exact mass and stiffness matrices of the model from ANSYS. Fig. 5 shows a procedure for extracting mass and stiffness matrices of the model from ANSYS.

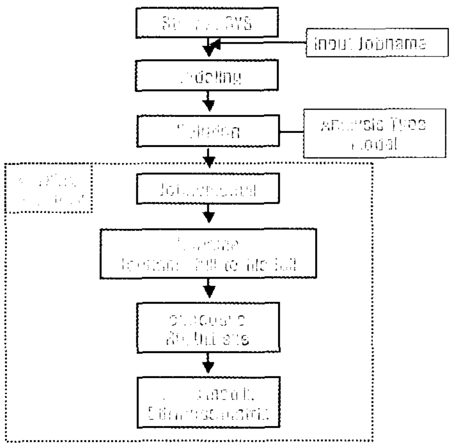

Fig. 5 A procedure for extracting mass and stiffness matrices from ANSYS

After the modal analysis for a model is performed, a file called 'jobname.full' is created in the present working directory. 'rdfull.f' inside ANSYS is a file which is used to extract the mass and the stiffness. Since the input file of 'rdfull.f' is set to 'file.full', 'jobname.full' should be renamed to 'file.full'. After then 'rdfull'is executed, 'mass.matrix' and 'stiffness. matrix' are created.

As shown in Fig. 6, these two files consist of several lines of column, row and element value for only non-zero element of matrix, and have no information for the matrix elements of all nodes at boundary conditions. Mass and stiffness matrices with crisscross out for all nodes at boundary conditions should be completed using the data.

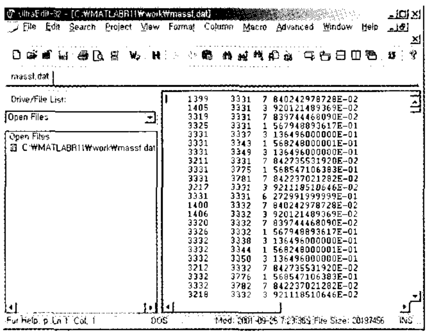

Fig. 6 An example of 'mass.matrix'

Then, the dynamic equations of engine structure are represented as

$$M_{ub}\ddot{x}_{ub} + K_{ub}x_{ub} = f_{ub} \tag{1}$$

where M and K are mass matrix and stiffness matrix, respectively. f is the external force vector and the subscript ub represents all nodes at no boundary condition. The size of M_{ub} or K_{ub} is the DOF per node number of ub, which is 2658 in this article.

To review the accuracy of M_{ub} and K_{ub}, an eigenvalue problem of Eq. (1) might be solved. Natural frequencies of (M_{ub}, K_{ub}) system are exactly same as those of ANSYS in Table 1. Therefore the extraction of M_{ub} and K_{ub} must be accurate.

Note that the Mass21 element and the coupling command in ANSYS should not be used in modeling. The mass and stiffness extraction of the model with them may not be accurate.

4. Reduced order modeling

M_{ub} and K_{ub} might be still large to apply for simulating and designing a top bracing. A reduced dynamic model should be required. Guyan presented a good model reduction methodology.[7] Nodes may be classified as significant nodes such as forcing nodes and insignificant nodes.

Consider that the significant nodes are selected for reduced model. Then, M_{ub} and K_{ub} in Eq. (1) can be partitioned according to displacements for selected nodes denoted by x_s and those for unselected nodes x_u. This yields

$$\begin{bmatrix} M_{ss} & M_{su} \\ M_{us} & M_{uu} \end{bmatrix}\begin{bmatrix} \ddot{x}_s \\ \ddot{x}_u \end{bmatrix} + \begin{bmatrix} K_{ss} & K_{su} \\ K_{us} & K_{uu} \end{bmatrix}\begin{bmatrix} x_s \\ x_u \end{bmatrix} = \begin{bmatrix} f_s \\ f_u \end{bmatrix} \tag{2}$$

Consider the potential energy of the system, V defined by

$$V = \frac{1}{2}\begin{bmatrix} x_s \\ x_u \end{bmatrix}^T \begin{bmatrix} K_{ss} & K_{su} \\ K_{us} & K_{uu} \end{bmatrix}\begin{bmatrix} x_s \\ x_u \end{bmatrix} \tag{3}$$

Since no force in unselected nodes exists, $\partial V/\partial x_u = 0$. This yields

$$\frac{\partial}{\partial x_u}(x_s^T K_{ss} x_s + x_s^T K_{su} x_u + x_u^T K_{us} x_s + x_u^T K_{uu} x_u) = 0 \quad (4)$$

and the condition that $K_{su} = K_{us}$ yields

$$x_u = -K_{uu}^{-1} K_{us} x_s \quad (5)$$

If Eq. (5) is substituted into Eq. (2), Eq. (2) is transformed to a new reduced order system of the form as

$$M_s\ \ddot{x}_s + K_s x_s = f_s \quad (6)$$

where

$$M_s = M_{ss} - K_{us}^T K_{uu}^{-1} M_{us} - M_{su} K_{uu}^{-1} K_{us} + K_{us}^T K_{uu}^{-1} M_{uu} K_{uu}^{-1} K_{us} \quad (7)$$

and

$$K_s = K_{ss} - K_{su} K_{uu}^{-1} K_{us} \quad (8)$$

If the harmonic forces $f_s = f\cos\omega t$ are applied to the model, the steady state responses are calculated as

$$x_s = X_s \cos\omega t \quad (9)$$

where the amplitude of the response is

$$X_s = (K_s - \omega^2 M_s)^{-1} f \quad (10)$$

The reduced model is reviewed by comparing its natural frequencies and steady state responses with those of ANSYS FE model. For this, 22 nodes at external parts, which maintain the shape of engine body and 1 node near the guide shoe inside first cylinder, the point A in Fig. 7, are selected and the 2658 DOF model is reduced to 69 DOF model.

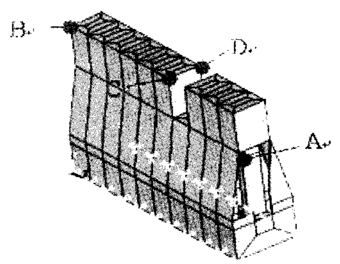

Fig. 7 Forcing and measuring points

The natural frequencies of reduced model are determined as 11.759 Hz, 14.567 Hz, 18 Hz, which are very similar to those in Table 1. When 100coc $50\pi t$(N) in the z direction is applied to point A in Fig. 7, Table 2 shows the amplitudes of steady state response in the z direction at points B, C and D in Fig. 7. The steady state responses of reduced model have good agreement with those of ANSYS FE model.

Table 2. Comparison of steady state responses

Points	Ansys FE Model	Reduced model
B	0.42761E-08 m	0.412628E-08 m
C	0.17301E-08 m	0.166953E-08 m
D	0.17294E-08 m	0.166881E-08 m

5. Conclusion

(a) A dynamic model of 9 cylinder engine was constructed for analysis in the low frequency range near global modes. Natural frequencies obtained by modal analysis of the model have good agreement with those from test data provided by manufacturer. The modeling technique will be very helpful.

(b) A technique was presented to obtain the exact mass and stiffness matrices of the model from ANSYS. An important notice was also stated, that is, the Mass21 element and the coupling command in ANSYS should not be used in modeling.

(c) The ANSYS FE model was reduced to the selected DOF model. The reduced model was reviewed by comparing to ANSYS FE model with the results for the eigenvalue analysis and forced vibration analysis. It is provided to construct an accurate dynamic model with small number of degrees of freedom for simulating and designing a top bracing.

(d) The dynamic modeling and model reduction technique may be used effectively for a forced vibration analysis of engine structures with the sum of appropriate excitation sources caused by the different phase of each cylinder.

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