SHIP VIBRATION ANALYSIS USING MODAL SYNTHESIS TECHNIQUE

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ABSTRACT

MSC/NASTRAN's modal synthesis technique was applied to the study of the complex vibratory phenomena of the afterbody of the ship. Calculated results were compared with those obtained by exciter test on the actual ship structure. Advantages and disadvantages of the modal synthesis technique, comparing to the ordinary direct method, were discussed in terms of computer resources, quality of the analysis model, and understanding of the complicated vibratory phenomena.

1. INTRODUCTION

The ship vibrations are the vibratory phenomena caused by excitation forces of the propeller and the main engine. These vibrations are the coupled vibrations of main hull girder and substructures as composed of superstructure, double bottom in engine room and main engine-propeller shafting system and so on. Due to the difficulty in the estimation of excitation forces and the complexity of the ship hull construction, the simple estimation method for the dynamic response of a ship has not been established yet. Furthermore, the main hull girder shows no longer the beam like vibration in the higher frequency range, therefore, the large three dimensional finite element model is applied to the ship vibration analysis instead of the beam model. From the complication of vibratory phenomena and the necessity of the large finite element model in the analysis, the modal synthesis technique has been introduced to the ship vibration analysis as an efficient method in recent year [1-3].

In this paper, MSC/NASTRAN's modal synthesis technique is applied to the study of the vibration of the ship afterbody in order to evaluate the efficiency of this technique from the practical design point of view and the modal synthesis analysis is compared to the ordinary eigenvalue analysis. The vibration analysis is carried out for a 80,000 DWT tanker and calculated results are compared with the results obtained by exciter test.

2. MODAL SYNTHESIS IN MSC/NASTRAN

Modal synthesis is a technique of dynamic substructuring and reduction of the number of degrees of freedom for large-sized finite element problems. Many different formulations for modal synthesis have been developed by many authors [4-10]. In MSC/NASTRAN, the standard modal synthesis technique (SOL63) is the extension of Generalized Dynamic Reduction (GDR), but the exact eigenvectors are used in the modal synthesis technique instead of the approximate eigenvectors in GDR. Therefore the formulation of this technique in MSC/NASTRAN is also based on the Rayleigh-Ritz method. Modal synthesis technique in

MSC/NASTRAN is similar to the method called the fixed interface method but the improvement, which takes into account the arbitrary interface condition, i.e. fixed and/or free interface condition, is added to this formulation [10,11].

The procedure of MSC/NASTRAN's modal synthesis technique (SOL 63) is divided into the following three distinct phases [12].

Phase 1 - Reduction phase

Obtain the uncoupled component eigensolutions in each substructure and reduce the stiffness and mass matrices.

Phase 2 - System solution phase

Synthesize the reduced matrices of each substructure and matrices of residual structure and obtain system solution.

Phase 3 - Data recovery phase

Extract eigenmodes from system solution and calculate eigenmodes of each substructure.

In addition to SOL 63, MSC/NASTRAN provides a capability of assembling the modal data for components prepared by several organizations from test data and from MSC/NASTRAN analysis or other finite element analysis. This capability is formulated on the basis of MacNeal's method [8] and is programmed into the modal synthesis rigid formats as SOL 41,42 and 43 [13].

In this paper, the standard modal synthesis technique, SOL 63, is applied to the ship vibration analysis, because SOL 63 provides the simple manipulation of MSC/NASTRAN.

3. CALCULATION OF SHIP VIBRATION

The example of calculation is a 80,000 DWT tanker built in Oppama shipyard of our company, SHI. The principal particulars and the general arrangement of this tanker are shown in Fig. 1. To estimate the efficiency of MSC/NASTRAN's modal synthesis technique for the ship vibration analysis, the natural frequency and response calculation using the modal synthesis technique is carried out by partitioning the afterbody of the ship into six substructures. (see Figs. 2 and

3). The results of this technique are compared with those of the ordinary direct technique (SOL 3) for the entire afterbody model. In the direct technique solution, the technique of Generalized Dynamic Reduction is used to reduce the processing time and cost. Furthermore, caluclated results are compared with the exciter test results.

3.1 CALCULATION MODEL

The calculation model is the three dimensional finite element model of the afterbody of the ship including main engine. This model is the cantilever model clamped at the transverse bulkhead which is 1 frame space apart from the front wall of the superstructure and is one half model from its symmetry. The boundary condition at the center line is assumed to be anti-symmetric because the transverse vibration is investigated. The structural mass is distributed over the idealized elements as weight density and the non-structural mass of the cargo and other major components, for example, propeller and rudder is located at the nearest grid points. The added mass of the surrounding water is distributed at the side shell and the bottom plate as lumped mass. The entire afterbody model is shown in Fig. 4. Total number of grid points is 824 and total number of elements is 1883 in this model. The calculation using the direct technique is carried out for this entire afterbody model.

In the calculation using the modal synthesis technique, the entire afterbody model shown in Fig. 4 is divided into six substructures or superelements, i.e. superstructure, funnel, main engine, double bottom in engine room, overhang and engine room excluding the double bottom. Fig. 5 shows these superelement models. The interior boundary points are placed in B-set (fixed) for all superelements except the double bottom in engine room. In the double bottom in engine room, some boundary points are placed in B-set (fixed) and others are placed in C-set (free) to satisfy the simply supported condition as a whole. Number of interior grid points, number of elements and number of boundary grid points of each superelement are given in Table 6.

3.2 CALCULATION PROCEDURE FOR SUPERELEMENT MODEL

There are two types of substructuring technique, i.e. single level substructuring and multilevel substructuring. Multilevel substructuring allows step-by-step investigations of the coupled effect of the vibration of each components. Therefore multilevel substructuring is preferable to single level substructuring. However, in this study, single level substructuring is adopted for its simplicity. The actual calculation procedure for superelement model is performed by the following three steps [12,14].

Step 1. Initial data storage to database (SOL 60)

Step 2. Reduction of matrices of each superelement (phase 1)

Step 3. System solution of residual structure (phase 2) and data recovery (phase 3)

Fig. 7 shows the assembly of superelements and the use of multiple database in this procedure. Computation time and size of database of the individual superelement in each phase are presented in Table 6 together with the computation time required in the direct analysis. Computation time for the modal synthesis analysis was two times that of the direct analysis in this case study.

3.3 NATURAL FREQUENCY CALCULATION

Natural frequencies of each superelement obtained in the reduction phase and natural frequencies of the residual structure obtained in system solution phase are presented in Table 8. Table 8 also shows the calculated results of the entire afterbody model using the direct technique. The typical uncoupled component modes of each superelement are shown in Fig. 9.

Generally, in a complex structure as ship, where substructures are the major vibrating parts, it is difficult to identify modes from the obtained mode shapes only. MSC/NASTRAN provides a capability of calculation of strain energy fraction of each superelement using parameter SESEF on PAPAM card. This strain energy fraction is very useful for characterizing the principal modes of the individual superelemnt. The strain energy fractions of the individual superelement in each frequency are shown in Table 10.

From these strain energy fractions and the mode shapes, the following resonant frequencies are identified in the modal synthesis analysis. The lowest horizontal frequency of cantilever vibration of the ship afterbody is 5.11 Hz and the lowest torsional frequency of the ship afterbody is 7.6 Hz. H-type vibration of main engine is 5.63 Hz and X-type vibration is 14.96 Hz and 15.63 Hz. The lowest horizontal frequency of the superstructure is 11.64 Hz and that of the funnel is 13.05 Hz. The mode shapes of the entire system corresponding to these resonant frequencies are shown in Fig. 11. The results obtained by the lateral exciter test of main engine on the actual ship structure show that H-type vibration is 6.9 Hz and X-type vibration is 14.9 Hz.

3.4 RESPONSE CALCULATION

The response due to the lateral excitation at the aft top of main engine is calculated by post-processing the results of natural frequency calculation. The reference points of the response calculation are the fore top of main engine and the front end of the navigation bridge deck in the superstructure. In the strict meaning, the different damping value in each substructure such as main hull, superstructure and main engine should be applied, but for the simplicity the modal damping ratio is assumed to be 3 % of critical damping in the response calculation of main engine and 1.5 % in the calculation of the superstructure. These results of the response calculation are plotted in Figs. 12 and 13 together with the exciter test results.

3.5 CONSIDERATION OF RESULTS

From the results of the natural frequency and response calculation, in the lower frequency range the results of the modal synthesis analysis coincide with those of the direct analysis, but in the higher frequency range there is some difference in the number of eigenvalues and the response level. This difference is considered to be caused by the free interface condition assumed in the double bottom in engine room and the insufficient number of uncoupled component eigenmodes used for the synthesis.

It can be seen in Figs.12 and 13 that the general tendency of the response characteristics of the structure is predicted well by the analysis. However, several quantitative differences are also observed. The calculated resonant frequency of H-type vibration of main engine is slightly lower than the exciter test results, which may indicate the insufficient stiffness given to the side shell model. The calculated response levels are generally higher than the exciter test results. This difference may be caused either by assuming the too small damping ratio or utilizing the insufficient extent of the ship structure model. If these points are improved in the idealization, it is reasonable to expect the better correlation between calculations and exciter test to be obtained.

4. CONCLUSION

MSC/NASTRAN's modal synthesis technique was applied to the vibration analysis of the afterbody of the ship and this technique was compared with the direct technique. And calculated results were also compared with the results obtained by exciter test. From these comparisons, the following conclusions are obtained.

(1) In general, the modal synthesis technique is applied for the vibration analysis in order to reduce the computation time and cost. But in this moderate-sized model which has about 5,000 degrees of freedom, computation time of the modal synthesis analysis was two times that of the direct analysis.

(2) MSC/NASTRAN's modal synthesis technique requires a large amount of disk spaces for the data base. On IBM computers, user must pre-allocate the enough space but it is sometimes difficult to evaluate the space correctly. This is undesirable for unskilled user to use this technique.

(3) By using the modal synthesis technique, the separate operations, i.e.

preparation and modification of the finite element data and calculation of the uncoupled eigenmodes are performed for each superelement. This independency enables us to estimate the quality of the finite element model through the dynamic characteristics of each substructure. As a result, the modal synthesis technique is proved to be much more effective than the direct technique.

(4) The dynamic characteristics and the coupled effect of the vibration of each component substructure would surely become clear by the modal synthesis technique. From this reason, this technique is very useful to study the vibratory phenomena of the complex structure as ships.

Finally, from these points mentioned above, it is concluded that, in spite of minor inconvenience, MSC/NASTRAN's modal synthesis technique is a very powerful tool applicable to the ship vibration analysis.

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Length (B.P)	232.00 m
Breadth (MLD)	42.00 m
Depth (MLD)	18.70 m
Designed draft	12.23 m
Deadweight (designed draft)	81,944 MT
Main Engine	Sumitimo Sulzer 5RLA90
MCO/MCR	17,000 ps/90 rpm
Normal output/Normal rpm	15,300 ps/87 rpm
Service speed	15.00 kt
Number of Propeller Blades	4 (c.p.p)
Class	LR

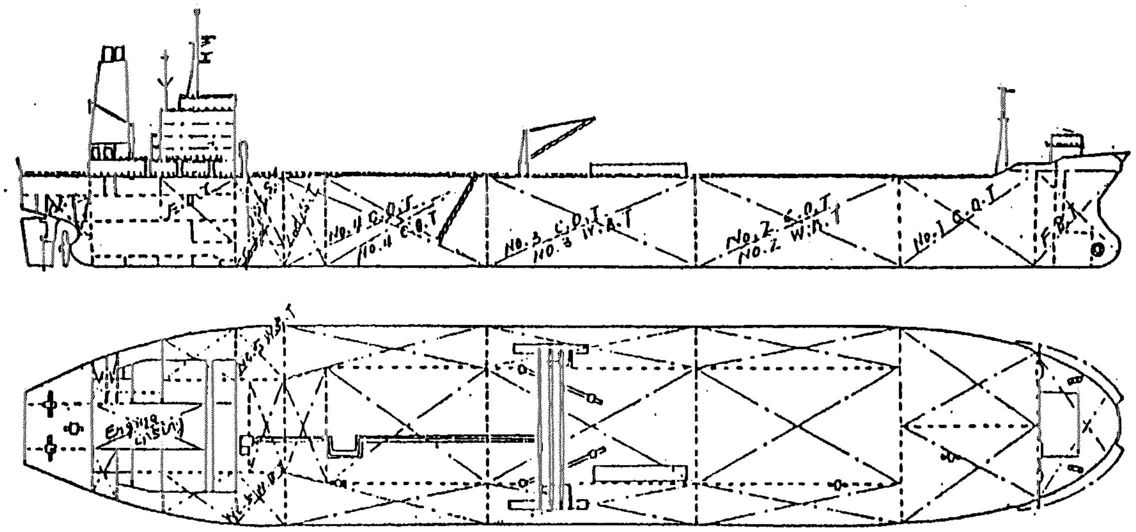


Fig.1 Principal Particulars and General Arrangement

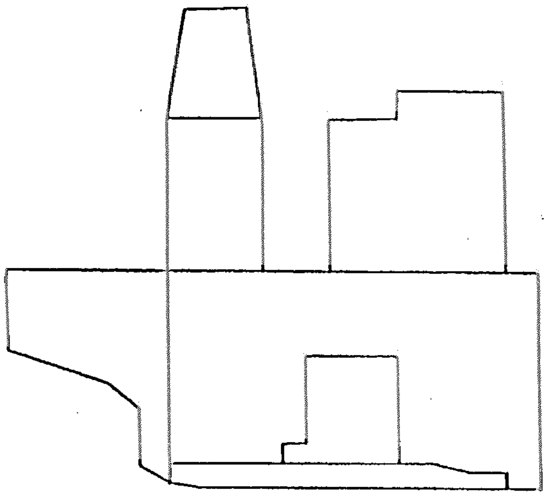


Fig.2 Global Model for Direct Technique

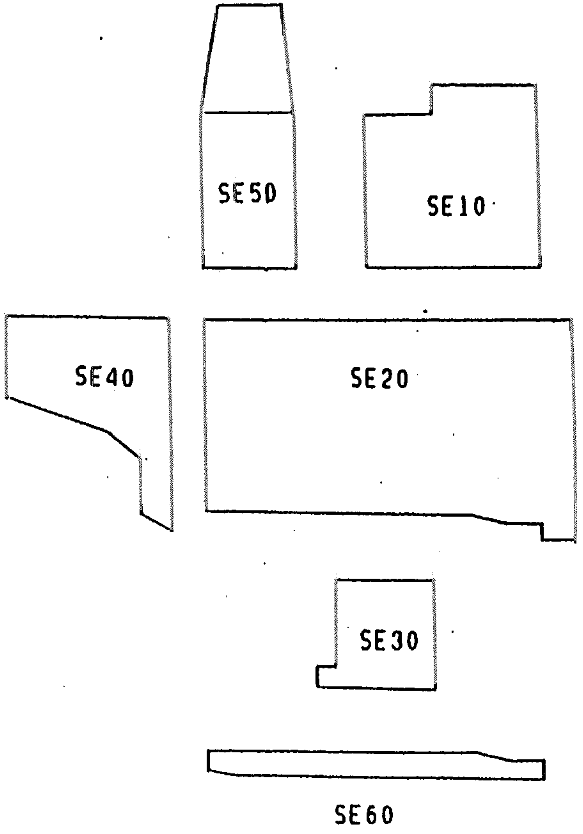


Fig.3 Superelement Models for Synthesis Techniques

QUADA : 982
TRIA3 : 283
BAR : 461
ROD : 200
ELAS2 : 7

Total Number of Grid Points : 824

Total Number of Elements : 1883

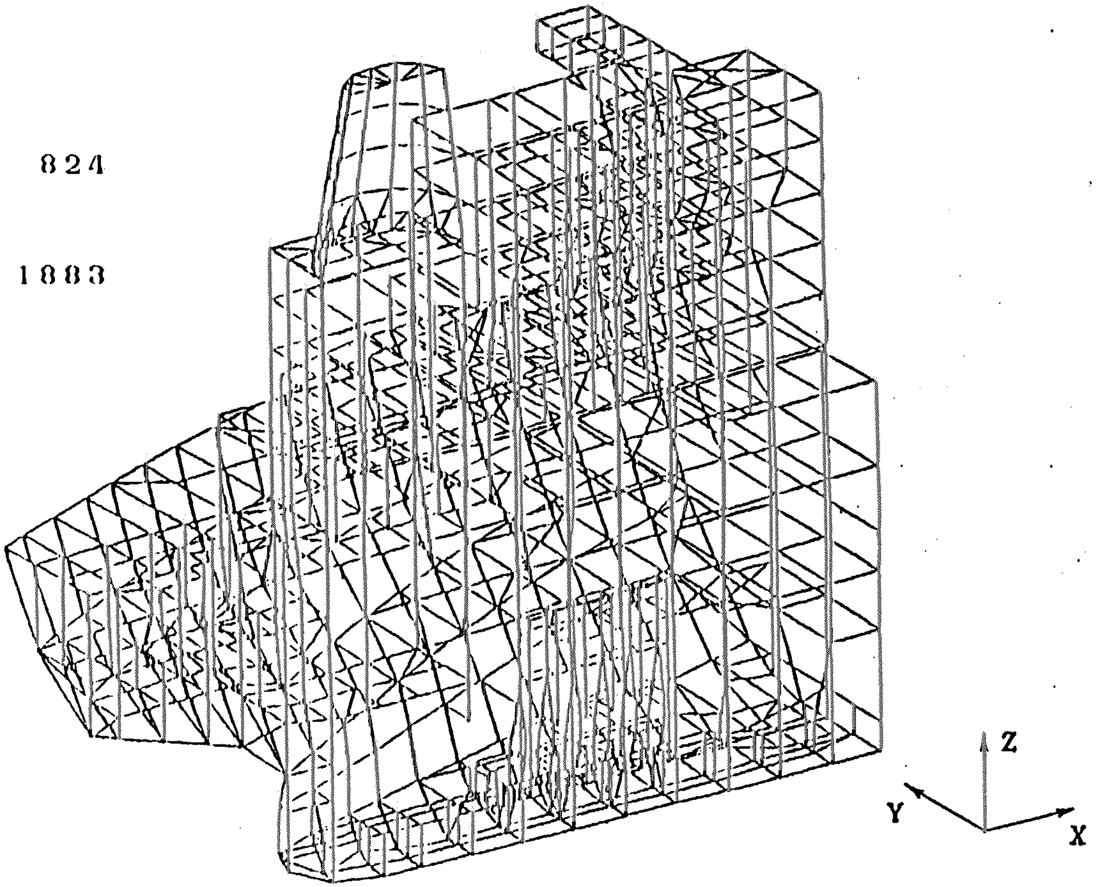
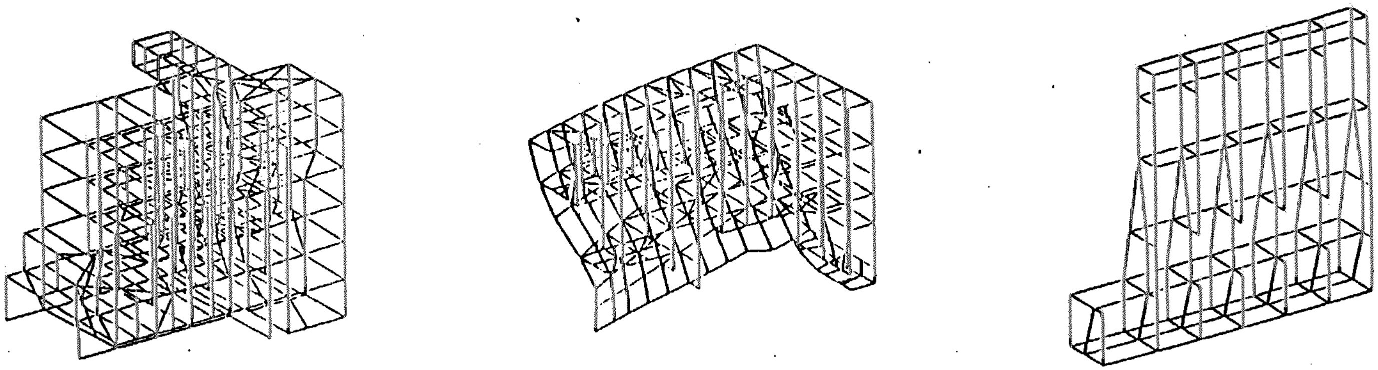
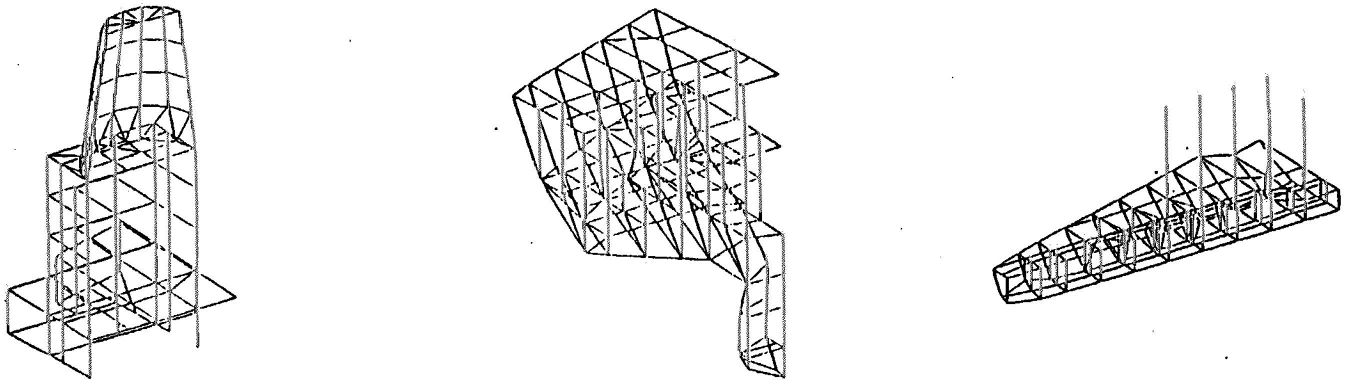


Fig.4 Global Model



Superstructure (SE 10)   Engine Room (SE 20)   Main Engine (SE 30)



Funnel (SE 50)   Over Hang (SE 40)   Double Bottom (SE 60)

Fig.5 Superelement Models

Table 6 Superelement Identification

Name	Superelement	Number of Grid Points	Number of Elements	Number of Boundary Grid Points	Band Width	CPU (*) Time(Sec)	Database (**) Size (Blocks)
Superstructure	SE 10	221	348	32	40	157	259
Engine Room	SE 20	244	596	79	16	254	385
Main Engine	SE 30	103	120	7	15	24	74
Over Hang	SE 40	108	228	27	16	46	121
Funnel	SE 50	121	238	21	15	33	103
Double Bottom	SE 60	123	212	40	12	50	158
Residual Structure	SE 0	110	141			596	522
Summary		824	1883			1160	1622
Global Model	NON SE	824	1883		71	505	

(*) IBM 3081-K

(**) BUFFER SIZE = 5860 (3380 DISK)

```
ASSGIN  SE10.DB  DB01
ASSGIN  INPT.DB  DB15

NASTRAN FILES=(DB01,DB15)
        DBSET 15=(DB15)
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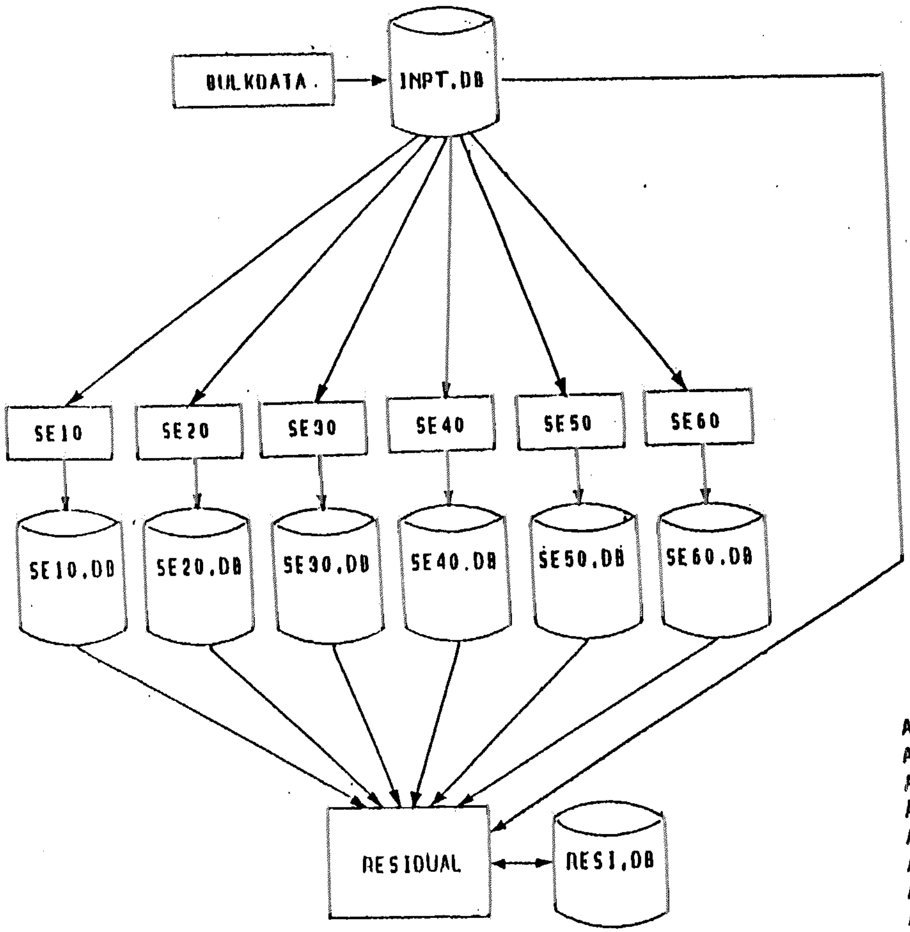


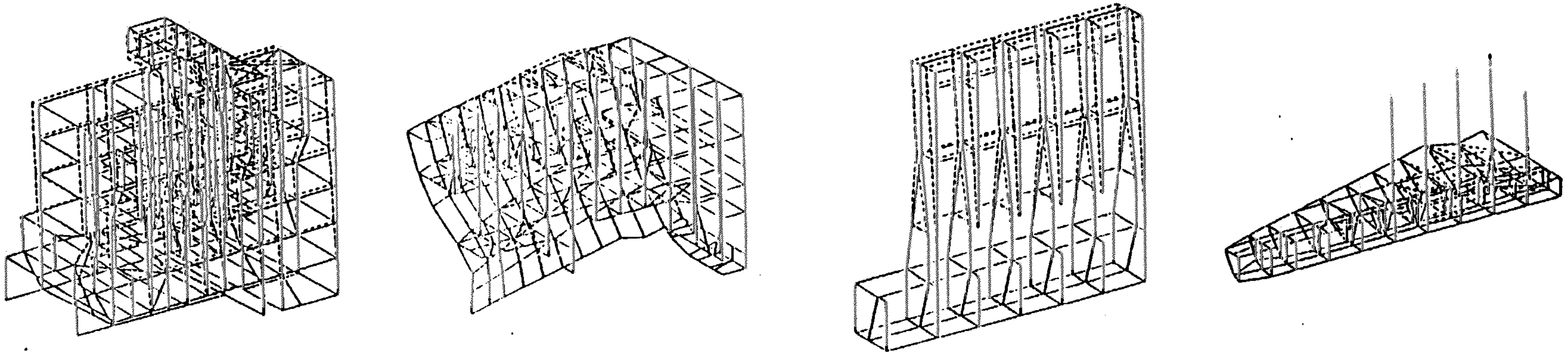
```
ASSIGN  RESI.DB  DB01
ASSIGN  SE10.DB  DB02
ASSIGN  SE20.DB  DB03
ASSIGN  SE30.DB  DB04
ASSIGN  SE40.DB  DB05
ASSIGN  SE50.DB  DB06
ASSIGN  SE60.DB  DB07
ASSIGN  INPT;DB  DB15

NASTRAN FILES=(DB01,DB02,...,DB15)
        DBSET 15=(DB02......DB15)
```

Fig.7 Superelement Models Reduction and Solution

Table 8 Frequency Comparison (unit;HZ)

Direct Method (SOL 3)	Modal Synthesis Method (SOL. 63)							
	RESI	SE 10	SE 20	SE 40	SE 50	SE 30	SE 60	SE 30 + 60
5.08	5.11							
5.60	5.63					① 9.12		① 7.60
7.62	7.68							
10.03	10.03		① 10.07					
10.49	10.49		② 10.53					
10.91	10.91		③ 10.99					
11.46	11.47		④ 11.63					
11.48	11.64	① 16.88						
11.85	12.13		⑤ 12.19					
12.14	12.15		⑥ 12.22					
12.16	12.22		⑦ 12.33					
12.22	12.76		⑧ 12.77					
12.82	13.05				① 18.44			
13.37	13.63							
14.00	13.98		⑨ 13.97					
14.65	14.77					② 17.37		② 16.09
15.25	14.96			① 14.88				
15.36	15.02		⑩ 15.19					
15.37	15.22		⑪ 15.24					
15.84	15.31			② 15.31				
	15.56		⑫ 15.54					
	15.63							
	15.91		⑬ 15.83					
	16.66			③ 16.67				
	16.79							
	16.82		⑭ 16.83					
							① 28.09	③ 26.40

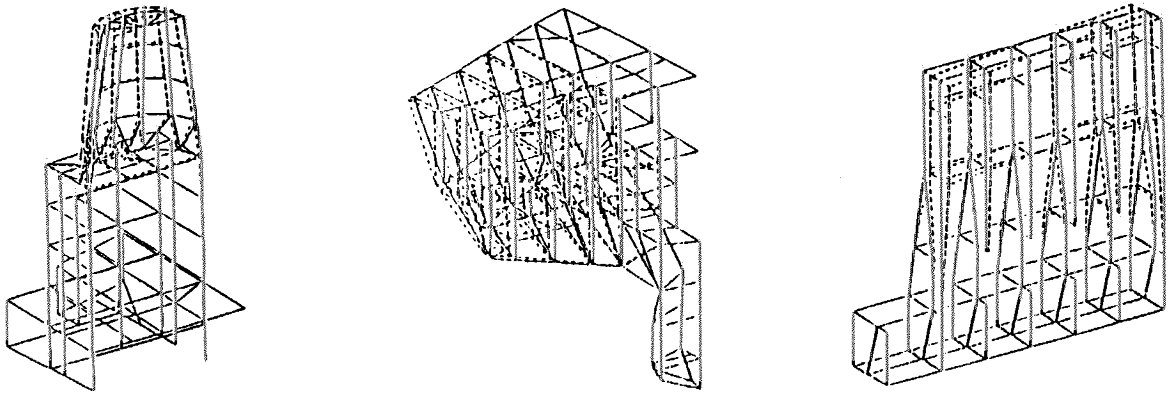


Superstructure (SE 10)

Engine Room (SE 20)

Main Engine (SE 30)
(H-type)

Double Bottom (SE 60)



Funnel (SE 50)

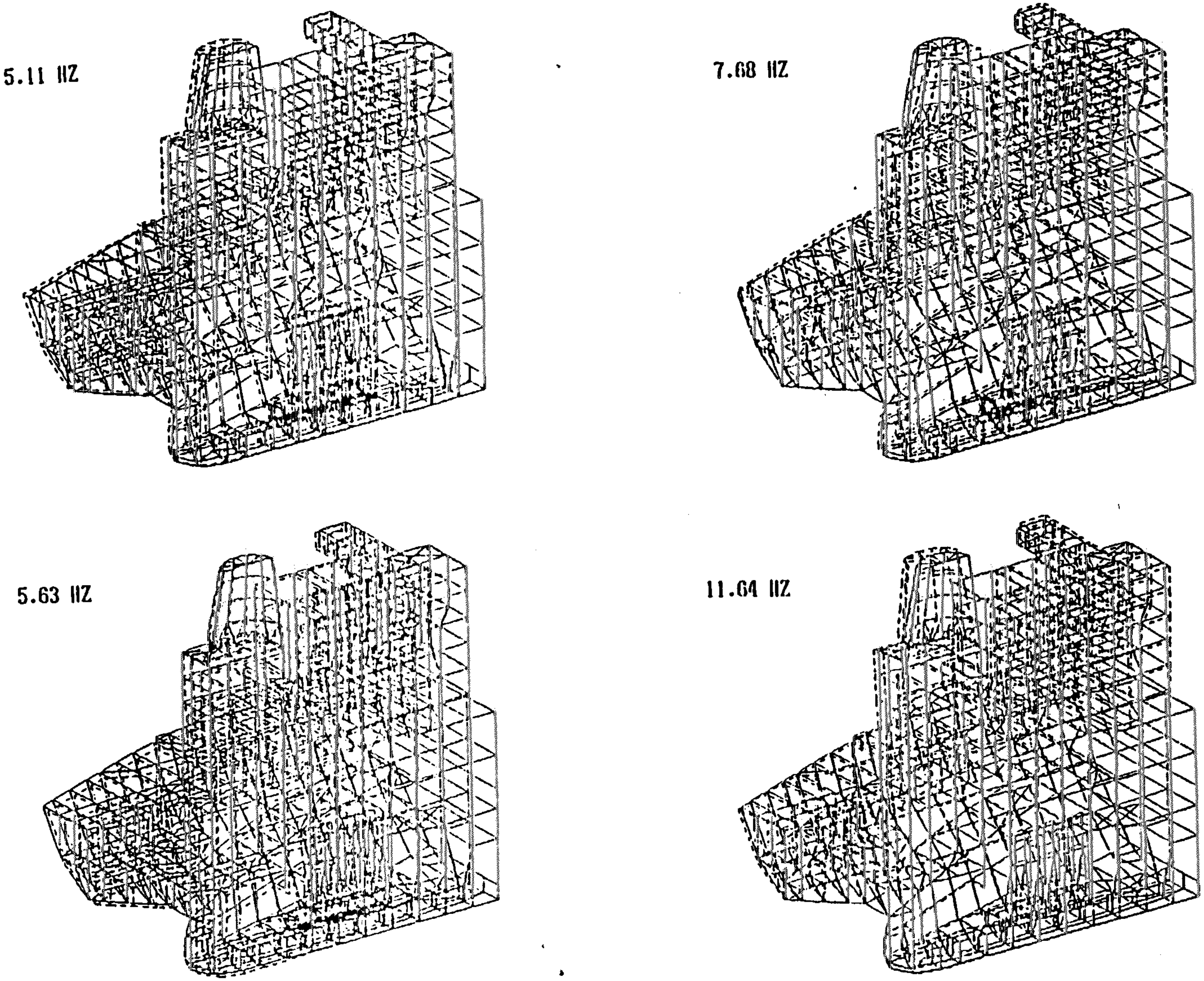
Over Hang (SE 40)

Main Engine (SE 30)
(X-type)

Fig.9 Mode Shape of Uncoupled Component Modes

Table 10 Superelement Strain Energy Fraction

Mode	Frequeuey(HZ)	SE 0	SE 10	SE 20	SE 30	SE 40	SE 50	SE 60
1	5.11	5.6	8.1	48.2	15.8	1.5	3.9	16.9
2	5.63	4.6	4.0	51.5	10.8	2.5	1.0	19.6
3	7.68	3.2	13.7	57.4	8.3	1.1	5.3	11.2
4	10.03	0.1	0.1	99.8	0.0	0.0	0.0	0.0
5	10.49	0.2	0.1	99.7	0.0	0.0	0.0	0.0
6	10.91	0.3	0.1	99.6	0.0	0.0	0.0	0.0
7	11.47	0.8	0.2	99.0	0.0	0.0	0.0	0.0
8	11.64	5.2	25.3	47.1	0.3	5.0	14.5	2.5
9	12.13	0.3	0.0	98.1	0.0	1.6	0.0	0.0
10	12.15	0.0	0.0	99.6	0.0	0.0	0.0	0.4
11	12.22	0.0	0.0	99.9	0.0	0.0	0.0	0.0
12	12.76	0.2	0.2	99.3	0.0	0.1	0.2	0.1
13	13.05	11.6	10.4	54.5	5.8	5.8	6.5	5.5
14	13.63	4.0	5.9	65.0	7.3	5.8	4.0	7.9
15	13.98	2.3	1.7	74.0	6.9	6.6	3.5	4.9
16	14.77	4.6	0.7	32.3	39.8	10.0	0.4	12.2
17	14.96	0.6	0.3	12.8	0.8	84.1	0.2	1.3
18	15.02	0.4	0.0	96.1	0.1	3.1	0.0	0.2
19	15.22	0.4	0.0	74.8	0.7	1.5	0.0	22.6
20	15.28	0.1	0.1	36.7	1.3	59.9	0.1	1.8
21	15.31	0.7	0.1	58.1	0.8	38.6	0.4	1.4
22	15.56	0.4	0.1	95.5	1.4	1.8	0.6	0.4
23	15.63	3.7	1.8	30.0	39.0	7.5	4.3	13.6
24	15.91	0.4	0.0	68.3	0.1	0.0	0.0	31.2
25	16.66	0.1	0.0	0.2	0.1	99.4	0.2	0.0
26	16.79	3.8	0.0	52.9	0.0	0.0	0.0	43.3
27	16.82	0.1	0.0	99.6	0.0	0.1	0.1	0.0



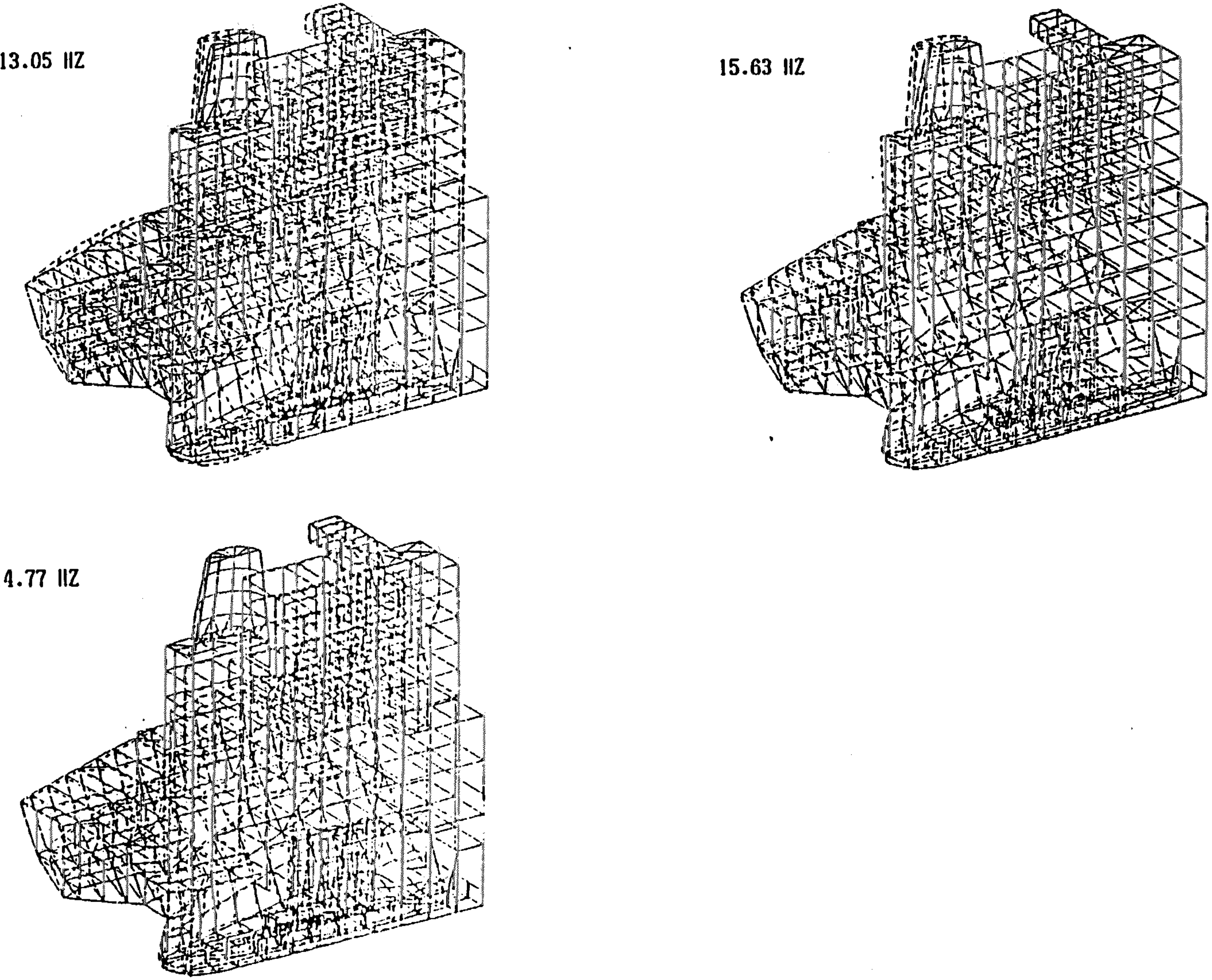
5.11 HZ

7.68 HZ

5.63 HZ

11.64 HZ

Fig.11 Mode Shape of Synthesis of Component Modes



13.05 HZ

15.63 HZ

14.77 HZ

Fig.11 Mode Shape of Synthesis of Component Modes (continued)

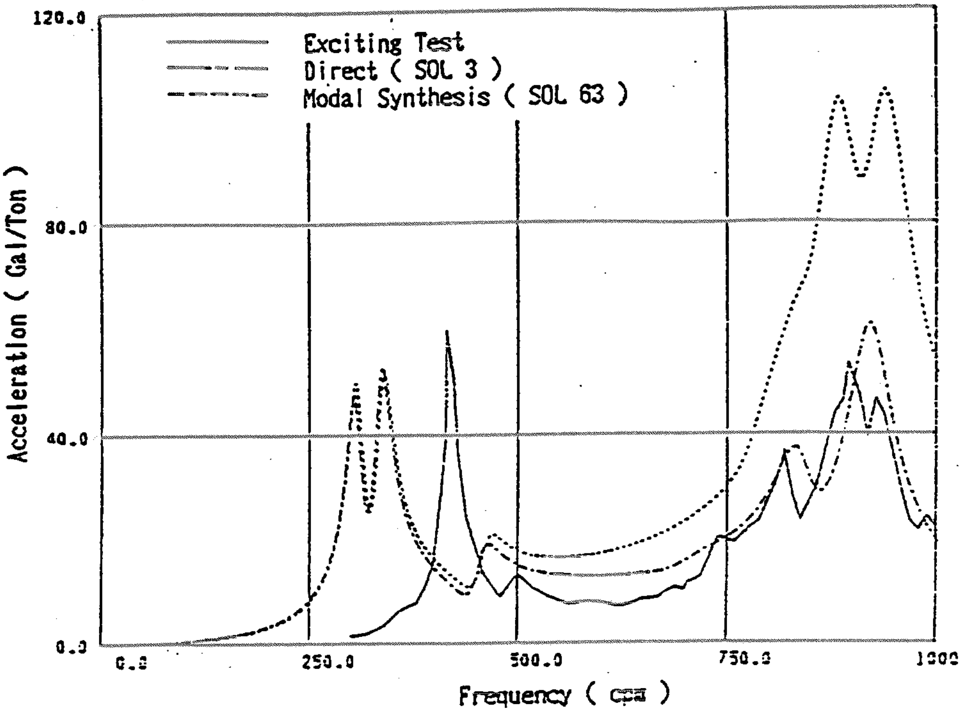


Fig.12 Resonance Curve of Transverse Vibration of Main Engine at Lateral Exciting of Engine

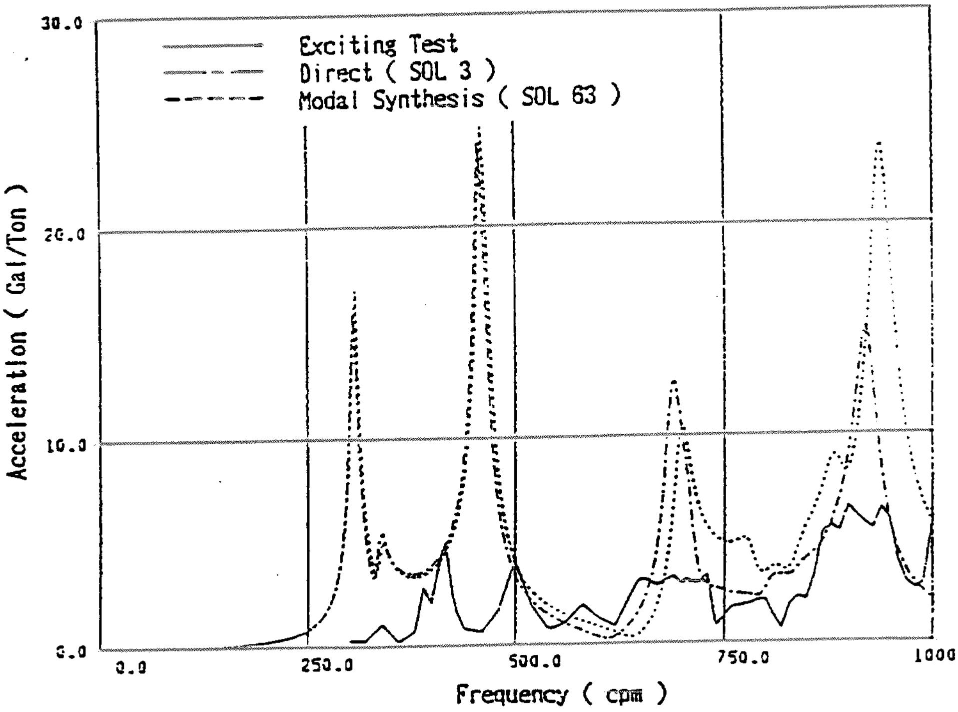


Fig.13 Resonance Curve of Transverse Vibration of Superstructure at Lateral Exciting of Engine