A NASTRAN CORRELATION STUDY FOR VIBRATIONS OF A CROSS-STIFFENED SHIP'S DECK

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SUMMARY

To evaluate the effectiveness of NASTRAN for predicting the vibration modes of panels with bending-membrane coupling, a cross-stiffened ship's deck has been analyzed. In correlations with experimental data, one NASTRAN finite element representation gave results slightly more accurate than a previous analytical solution. Computational time was excessively long due to the Guyan method of reducing the eigenvalue problem. It is recommended that a more efficient method of matrix reduction be implemented for the lumped mass formulation.

INTRODUCTION

In studies concerning the shock environment of surface ships, the dynamic behavior of decks is of basic importance. Studies (References 1-4) performed at the Naval Ship Research and Development Center have established that the shock response of a deck can be predicted in terms of the input motions of the deck provided the modal characteristics of the deck are known.

The classical approach used to analyze the bending behavior of stiffened decks has been to use the concept of an equivalent orthotropic plate. After modification to include the effects of a few large, widely spaced stiffeners (References 1 and 2), the orthotropic plate approach has been used to successfully predict the vibration modes of a cross-stiffened ship's deck. In addition (References 3 and 4), the effect of local mass loadings on the vibration modes of the deck model has been studied by incorporating point masses in the modified orthotropic plate approach.

The application of this type of analytical representation is limited, of course, to highly idealized mathematical models of realistic ship structures. For this reason, it is of interest to evaluate the effectiveness of general purpose Finite Element programs such as NASTRAN by correlations with previous analytical and experimental studies.

A stiffened deck modeled with the finite element approach using a combination of plate and bar elements is characterized by a coupling between in-plane and transverse bending displacements. This behavior is characteristic of a number of other panel problems including corrugated panels which are among candidate thermal protection systems for the space shuttle. For this reason also, it is of interest to evaluate the effectiveness of NASTRAN for the prediction of vibration modes of stiffened decks.

The present paper describes a correlation study performed using NASTRAN to predict the vibration frequencies and modes of the 1/4-scale model of the cross-stiffened ship's deck studied in References 1-4. This structure was selected for investigation primarily because of availability of previous analytical and experimental results. The structure is also of further interest since its construction details are representative of a realistic ship structure. Thus, the study provides insight into the effectiveness of finite element modeling as used in NASTRAN when applied to a prototype ship structure.

The specific objectives of the study were to predict natural frequencies and nodal patterns of the model deck and correlate these with the available analytical and experimental results.

The study was performed from the viewpoint that the analyst would like to employ a detailed finite element representation of all details of the deck construction for accurate prediction of frequencies, as well as detailed predictions of mode shapes and modal stress distributions. Moreover, it would be desirable for the computational scheme to give a fast, accurate prediction of a large number of modes in one pass using an eigenvalue routine which protects the analyst from overlooking modes. The NASTRAN analysis was formulated and performed to satisfy these characteristics. The results of the NASTRAN analysis, after correlation with previous results, were evaluated in terms of these criteria.

STIFFENED DECK MODEL

The model ship's deck studied is the top deck of a 1/4-scale model of a compartment of a surface ship constructed and used by the Naval Ship Research and Development Center for shock and vibration studies. The details of the deck are shown in Figure 1. The center panel of the deck between the two interior bulkheads was the subject of the previous investigations reported in References 1-4. The center panel is stiffened in the transverse direction by a large number of closely spaced stiffeners and in the longitudinal direction by two widely spaced, deep stiffeners. Attempts were made in the design and construction of the model to provide clamped boundary conditions at the two edges of the panel supported by the interior bulkheads. The model was constructed in a Naval Shipyard using fabrication techniques representative of prototype ship construction.

PREVIOUS ANALYTICAL METHOD

The classical approach of representing the bending stiffness of ship's decks uses orthotropic plate theory. This approach was investigated in Reference 1. It was demonstrated that the presence of the two deep longitudinal stiffeners limited the accuracy of the orthotropic solution to only a few lower modes. As an improvement on the orthotropic theory, the flexure of the longitudinal stiffeners was considered separately which led to a modified version of the orthotropic plate equation. This approach, the Separated Stiffener Method, gave good agreement with the experimental frequencies for the EC-2 deck for up to nine half waves along the deep stiffeners.

The differential equation describing the eigenfunctions ϕ of the stiffened deck by the Separated Stiffener Method has the form

$$L[\phi] = \lambda M[\phi] \qquad (1)$$

where the eigenvalues λ are related to the unknown natural frequencies ω, by $\lambda = \omega^2$. In this differential equation, the stiffness operator L is given by

$$L = D_x \frac{\partial^4}{\partial x^4} + 2H \frac{\partial^4}{\partial x^2 \partial y^2} + [D_y + \Sigma D_{yi} \delta(x-a_i)] \frac{\partial^4}{\partial y^4} \qquad (2)$$

and the mass operator

$$M = \rho + \Sigma \mu_{yi} \delta(x-a_i) \qquad (3)$$

In the above equations, D_x, H, D_y are the orthotropic plate stiffnesses, D_{yi} is the bending stiffness of the ith longitudinal stiffener and associated plating, ρ is the orthotropic plate mass per unit area, μi is the beam mass per unit length, and δ is the Dirac delta function.

For the boundary conditions of the EC-2 deck, this equation was solved by separation of variables in the form,

$$\phi_{mn} = X_{mn}(x)\ Y_n(y) \qquad (4)$$

where m and n are integers. Beam mode shapes were used for the Y_n functions in connection with an energy approach to obtain the X_{mn} functions. These results are tabulated in Reference 1. It may be noted from the form of equation (4) that the Separated Stiffener Method always predicts straight nodal lines. The results of the Separated Stiffener analysis will be presented later for correlation with experimental findings and the results of the NASTRAN analysis.

NASTRAN ANALYSIS

NASTRAN Finite Element Models

Two NASTRAN analyses were performed. In the first analysis, NASTRAN Model 1, the center panel of the deck was analyzed for direct comparison with the previous analytical predictions. In the second analysis, NASTRAN Model 2, the entire deck of the EC-2 deck was represented in hopes of obtaining improved agreement with the experimental results.

The finite element mesh for the two NASTRAN models is shown in Figure 2. In the NASTRAN Model 1 analysis, the interior deck panel was modeled using 1/2 symmetry. The deck plating was represented with 220 CQUAD2 plate elements. All stiffeners were represented with offset CBAR elements; 240 bar elements were used. The entire deck was represented as NASTRAN Model 2 using 1/4 symmetry. In the NASTRAN Model 2 analysis, 143 CQUAD2 elements and 132 CBAR elements were employed. Element properties for these two models are tabulated in Table I. Before constraints, NASTRAN Model 1 had 1265 degrees of freedom and NASTRAN Model 2 had 840 degrees of freedom.

NASTRAN Computations

All NASTRAN computations were performed using the lumped mass formulation. Prior to eigenvalue extraction, rotational and in-plane degrees of freedom were omitted using the NASTRAN Guyan reduction method. Eigenvalue extraction was performed for both analyses using the Givens method. Degrees of freedom at each stage in the analyses are shown in Table II.

Plots of the nodal patterns for the NASTRAN analyses were obtained from a separate FORTRAN program. During each NASTRAN execution, printed and punched output for the eigenvectors was requested. After execution the eigenvectors were copied from the punchfile onto a tape. This tape was subsequently used as input to the FORTRAN program which calculated nodal points by linear interpolation between grid point displacements. Nodal points were calculated by first making sweeps along lines parallel to the x-axis and then along lines parallel to the y-axis. The resulting nodal points were then plotted using a DDI plotter to yield the nodal patterns.

Significant computational times in various modules as well as total times are tabulated in Table III. All computations were performed on LRC, CDC 6600 computers. A salient characteristic of both analyses is that a very large amount of time was required to perform the Guyan reductions.

DISCUSSION OF RESULTS

Comparisons of the experimental and predicted frequencies are given in Table IV. Measured frequencies are compared with the analytical solution previously described and the results of the two NASTRAN analyses. Measured nodal patterns are compared with NASTRAN predicted patterns in Figure 3.

The NASTRAN frequency predictions, on the average, are in slightly better agreement with the experimental results than the analytical solution. NASTRAN Model 1, which has the same boundary conditions as the analytical solution, has an average percentage difference of 8.5% whereas the analytical solution has an average percentage difference of 10.9%. The second NASTRAN model, which predicted the fundamental frequency almost exactly, generally predicted lower frequencies than the experimental results.

The fact that the second NASTRAN model predicted frequencies which were generally too low may be attributed to the simply supported boundary condition assumed on the interior bulkheads. The model was designed for the interior bulkheads to represent clamped edges. The experimental nodal patterns show, however, that some rotation is permitted. From the second NASTRAN analysis it may be concluded that the boundary conditions at the bulkheads are most nearly represented as fully clamped since assuming simple supports predicts frequencies consistently much too low.

The nodal patterns predicted by NASTRAN Model 1 show good agreement with the experimental results. Although the analytical prediction of nodal lines were unavailable these results consist of intersecting straight lines. NASTRAN predicted nodal lines which were generally nonintersecting and curved.

The disappointing feature of the NASTRAN analyses was the long computer times required to reduce the degrees of freedom prior to eigenvalue extraction. In the NASTRAN Model 1 analysis, the Guyan reduction required approximately 3000 CPU seconds out of a total of 5300 CPU seconds. Of the 3000 seconds required in the Guyan reduction, over 70% of the time was required in reduction of the mass matrix.

One of the reasons that NASTRAN was relatively inefficient in the present analyses is that no attempt is made to take advantage of the lumped mass formulation. Considerable time-savings would have been accomplished if a distinction was made between the reduction used for the lumped mass and consistent mass formulations. For a considerable number of vibration problems (see Reference 5), the gain in computational efficiency offered by the lumped mass formulation more than offsets advantages of the increased accuracy and bounded nature of the consistent mass formulation.

CONCLUDING REMARKS

To evaluate the effectiveness of NASTRAN for predicting the vibration modes of panels with bending-membrane coupling, a cross-stiffened ship's deck has been analyzed. A fine mesh of beam and plate elements was used. To obtain a large number of modes and to insure that all modes were obtained, the matrix eigenvalue problem was reduced by a Guyan reduction and solved by the Given's method.

For one NASTRAN finite element model, the matrix reduction for the stiffness matrix required about 850 CPU seconds, and the mass matrix reduction required about 2200 CPU seconds. The long computational time was required because of large matrix multiplications in the Guyan reduction.

In correlations with experimental data one NASTRAN finite element model was slightly more accurate for frequency predictions and nodal patterns than a previous analytical method. Agreement with experimental results was good.

It can be concluded that NASTRAN was effective in meeting all of the evaluation criteria with the exception of computational time. Excessively large computer time was required because of the Guyan method of reducing the mass matrix. It is recommended that the NSMO consider investigating and implementing other more efficient methods of mass matrix reduction for the lumped mass formulation.

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Table I

Bar Element Properties

Stiffener	Components of Offset Vector			Area	Moments of Inertia			Torsional Constant
	Z1 cm	Z2 cm	Z3 cm	A cm^2	I1 cm^4	I2 cm^4	I12 cm^4	J cm^4
Transverse y = constant	0.0	0.4298	3.167	1.785	3.694	0.9121	-1.061	.0420
Longitudinal x = 129.5 cm	-1.183	0.0	8.659	9.149	157.9	36.06	-43.75	.6919
x = 281.9 cm	1.183	0.0	8.659	9.149	157.9	36.06	43.75	.6919

Table II

Degrees of Freedom (DOF) During Computations

	NASTRAN Model 1 (one-half symmetry)	NASTRAN Model 2 (one-quarter symmetry)
DOF Before Constraints	1265	840
DOF After SPC	1028	648
DOF Omitted	798	516
DOF for Eigenvalue Problem	230	132

Table III

Computational Times for NASTRAN Analyses

Operation (Module)	CPU Time (seconds)	
	NASTRAN Model 1	NASTRAN Model 2
Generate Stiffness Matrix (SMA1)	210	136
Generate Mass Matrix (SMA2)	9	4
Impose SPC (SCE1)	26	17
Reduction of Stiffness Matrix (SMP1)	852	} 911
Reduction of Mass Matrix (SMP2)	2198	
Eigenvalue Extraction (READ)	1453	348
Eigenvector Recovery for 25 Modes (SDR1)	249	118
Total CPU for Above Operations	4997 Seconds	1534 Seconds
Total CPU for all Operations	5344 Seconds	1674 Seconds
Total PPU for all Operations	3260 Seconds	927 Seconds

Table IV

Measured and Computed Natural Frequencies of Deck

Analytical Half-Wave Numbers		Measured Frequency	Analytical Solution (Ref. 1)		NASTRAN Model 1		NASTRAN Model 2	
n*	m*	HZ	HZ	%Diff.	HZ	%Diff.	HZ	%Diff.
1	1	29	35.0	20.7	33.2	14.5	28.9	0.3
	2	47	44.4	5.5	43.1	8.3	36.5	22.3
	3	67	57.5	14.2	58.6	12.5	51.1	23.7
3	1	88	96.3	9.4	86.7	1.5	83.1	5.6
	2	84	79.8	5.0	79.9	4.9	69.0	17.9
	3	62	57.8	6.8	68.1	9.8	64.5	4.0
5	1		105.0		97.5		94.4	
	2		87.0		91.2		84.0	
	3	76	64.6	15.0	81.9	7.8	76.2	0.2

*n denotes the number of half waves along the y-axis; m denotes the number of half waves along the x-axis.

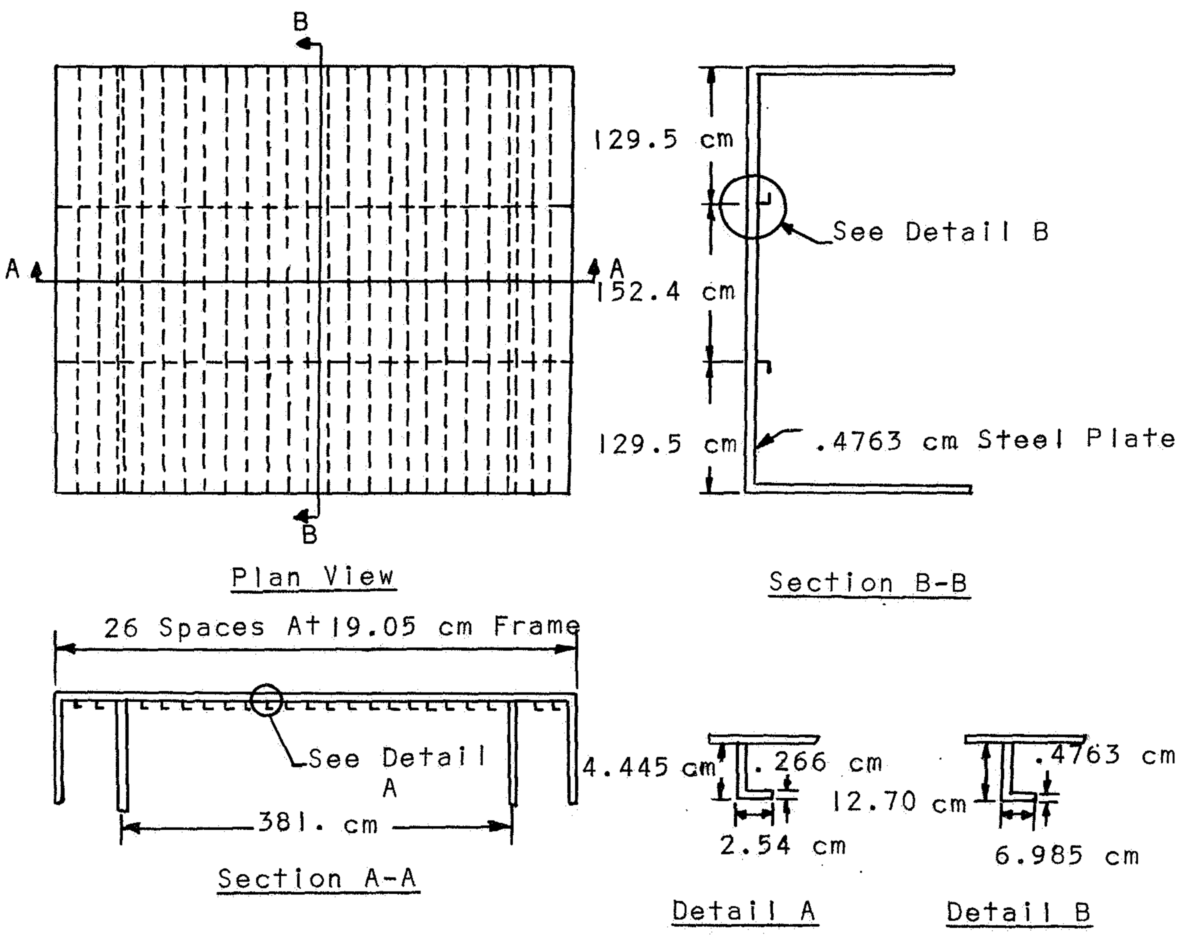

Figure 1.- Cross-stiffened model of ship's deck.

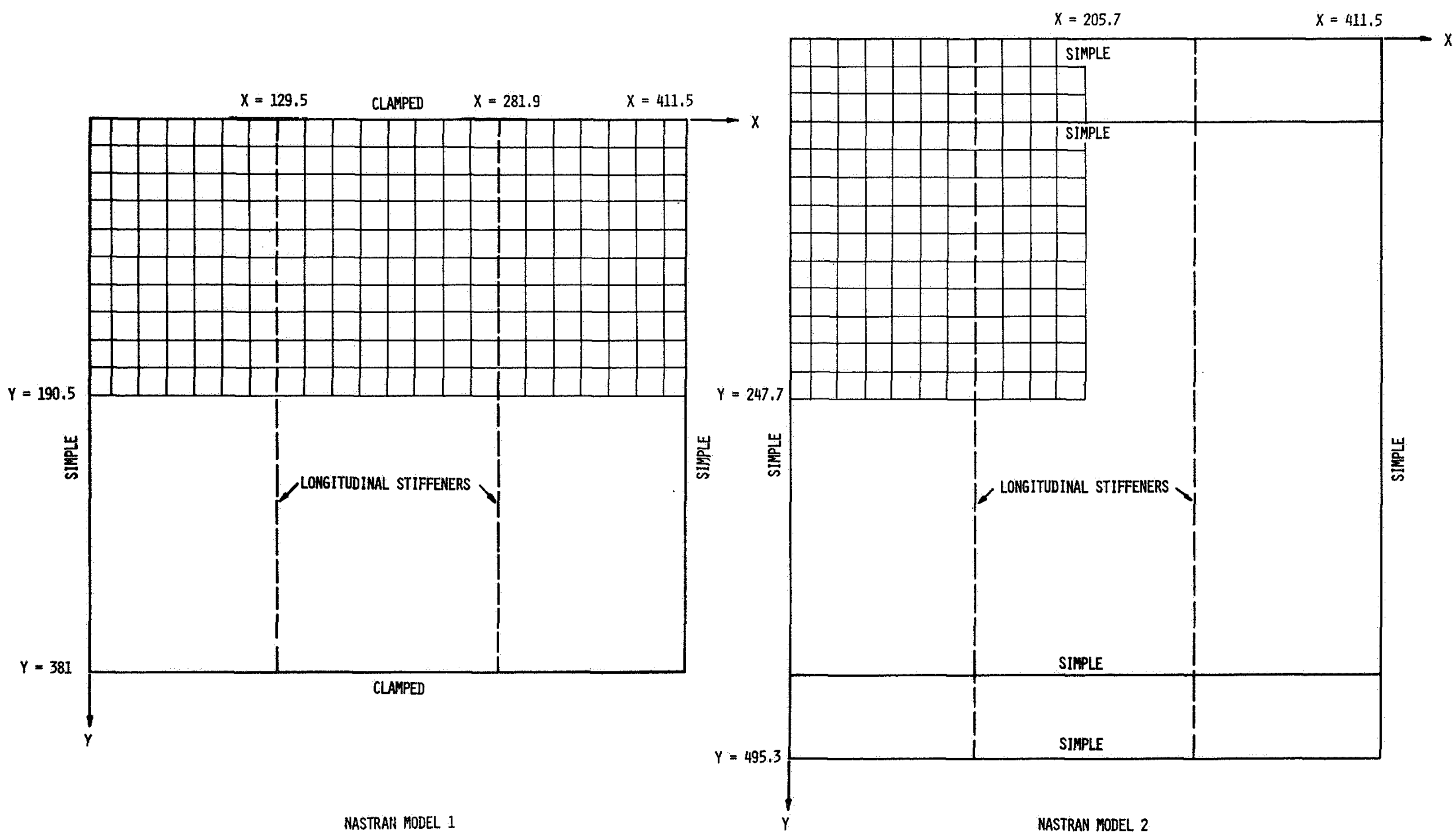

Figure 2.- NASTRAN finite element models.

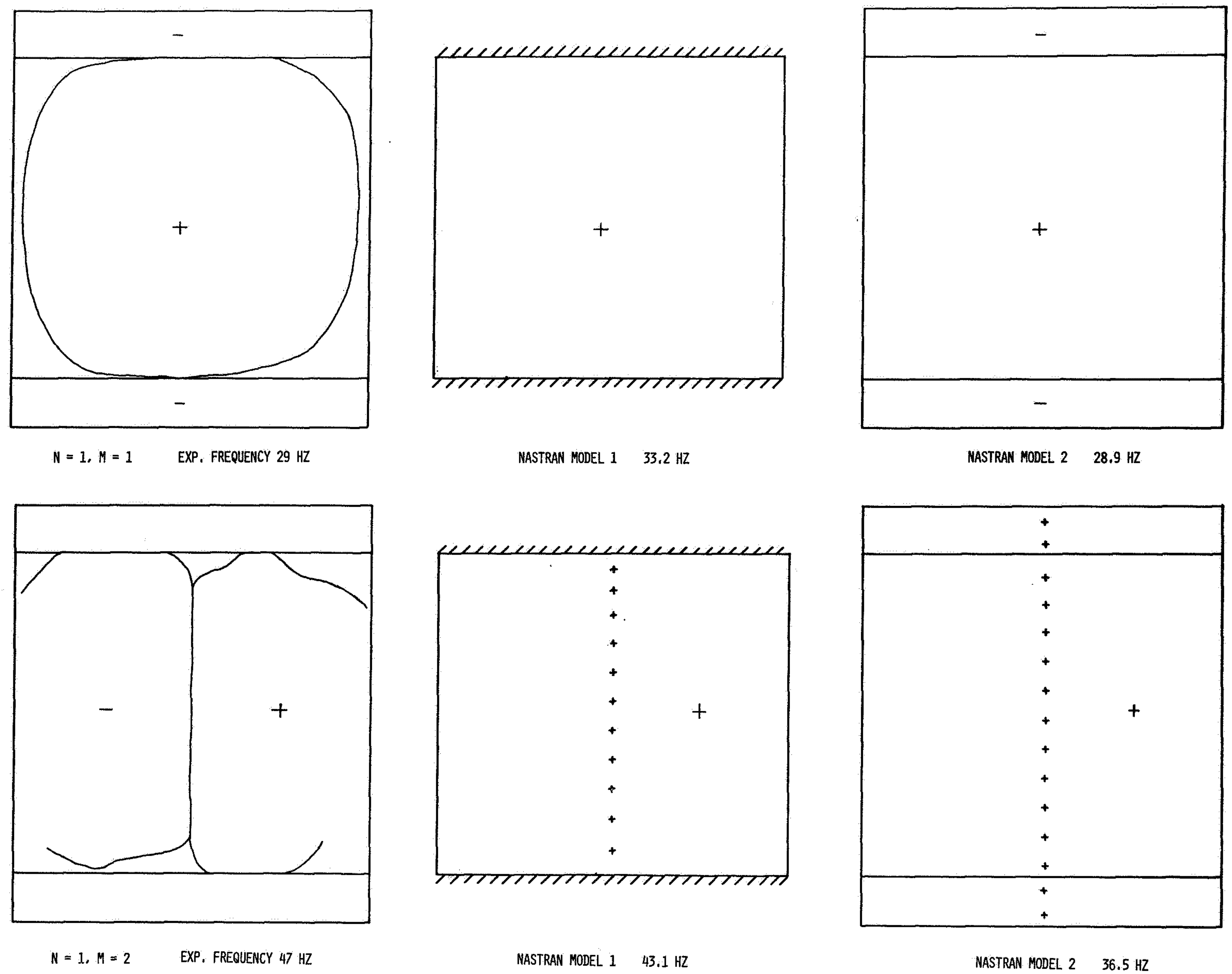

Figure 3.- Experimental and NASTRAN nodal patterns.

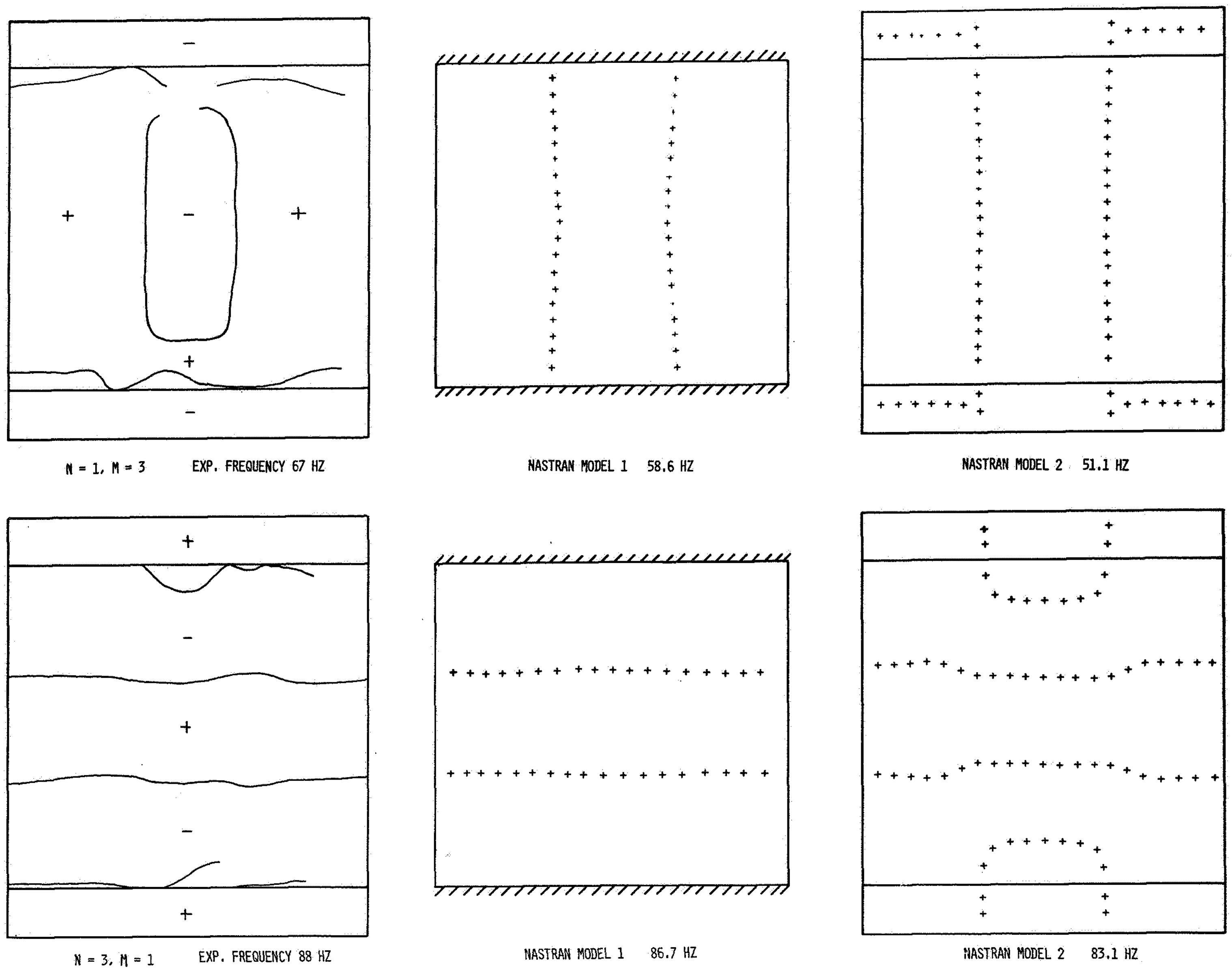

Figure 3.- Experimental and NASTRAN nodal patterns — Continued.

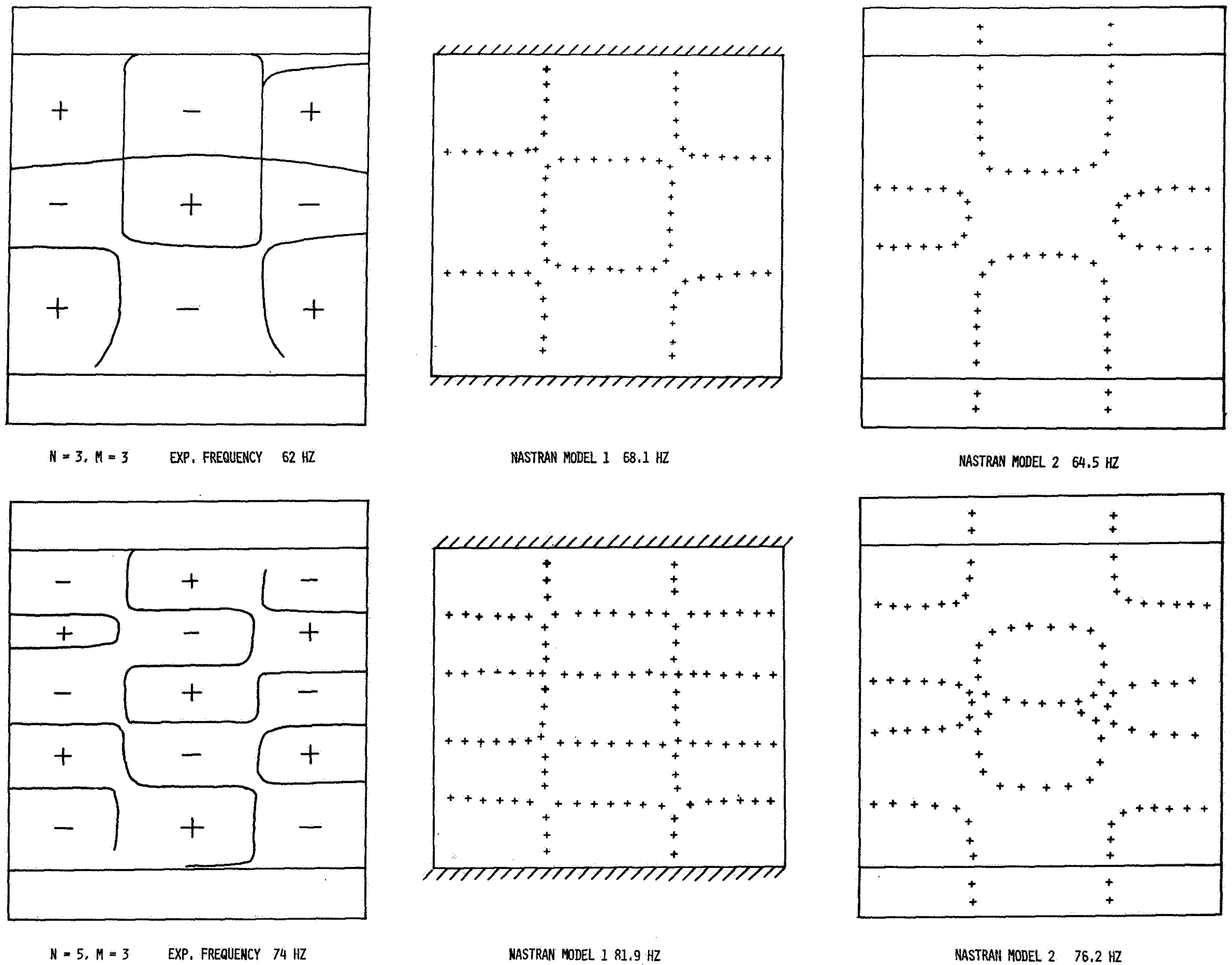

Figure 3.- Experimental and NASTRAN nodal patterns — Concluded.