MSC/NASTRAN Implementation of Coupled Structural-Acoustic Response for Aircraft Cabin Noise Prediction

by A. Mera*, T. F. Yantis*, G. SenGupta**, A. Landmann**

Low frequency noise transmission into an aircraft cabin (airborne and structure-borne) is becoming an increasing factor for advanced propeller powered aircraft. Analysis of this problem with finite element methods requires coupling the motion of a flexible structure (fuselage) with the pressure in the acoustic volume (cabin interior). The matrix formulation takes advantage of certain underlying similarities between the structural and acoustic equations and solves the 3-D coupled structural-acoustic problem for free and forced vibrations.

In this presentation, the basic principles of this method will be outlined, and results will be presented to illustrate its potential.

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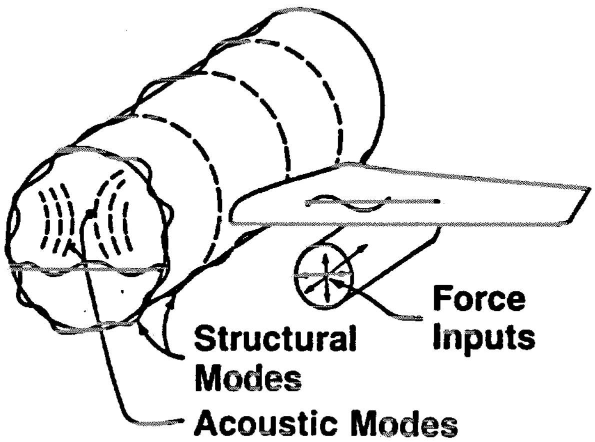
OUTLINE

1. INTRODUCTION
2. MATRIX AND F.E. FORMULATION OF STRUCTURE, FLUID, AND COUPLING
3. DYNAMIC REDUCTION FOR LARGE MODELS
4. MSC/NASTRAN IMPLEMENTATION OPTIONS
5. APPLICATIONS, RESULTS
6. CONCLUSIONS, FURTHER DEVELOPMENTS

GENERATION OF STRUCTURE-BORNE NOISE



STRUCTURES	ACOUSTICS
ELASTIC CONSTANTS (E,G)	$(\text{DENSITY})^{-1}$ $(1/\rho)$
DENSITY (ρ_s)	BULK MODULUS $(x)^{-1}$
DISPLACEMENT (U_X)	PRESSURE (P)
NORMAL STRESS (σ_X)	(-)PARTICLE ACCELERATION IN X-DIRECTION $\ddot{\xi}_X = \frac{1}{\rho}\frac{\partial P}{\partial X}$
SHEAR STRESSES (τ_{XY}) (τ_{ZX})	NEGATIVE ACCELERATIONS $\ddot{\xi}_Y = \frac{1}{\rho}\frac{\partial P}{\partial Y}$ $\ddot{\xi}_Z = \frac{1}{\rho}\frac{\partial P}{\partial Z}$

STRUCTURAL-ACOUSTIC ANALOGY

F.E. DISCRETIZATION OF EQUATIONS OF MOTION

Structure:

$$M^s\ddot{u} + B^s\dot{u} + K^s u = F + A^T p$$

Mass — Damping — Stiffness — External Force — Fluid Pressure at Interface

Fluid:

$$M^f\ddot{p} + K^f p = -A\ddot{u}$$

Fluid "Mass" — Fluid "Stiffness" — "Load" Due to Boundary Acceleration

COUPLED FLUID-STRUCTURE SYSTEM

$$\begin{bmatrix} M^s s^2 + K^s & -A^T \\ s^2 A & M^f s^2 + K^f \end{bmatrix} \begin{Bmatrix} u \\ p \end{Bmatrix} = \begin{Bmatrix} F \\ o \end{Bmatrix}$$

Transient Response $\quad S = \partial / \partial t$

Frequency Response, Complex Eigenvalues $\quad S = i\omega$

DYNAMIC REDUCTION IN THE STRUCTURE

$$u = \Phi^s \xi^s$$

$$\begin{bmatrix} m^s s^2 + k^s & -\Phi^{sT} A^T \\ s^2 A \Phi^s & M^f s^2 + K^f \end{bmatrix} \begin{Bmatrix} \xi^s \\ p \end{Bmatrix} = \begin{Bmatrix} \Phi^{sT} F \\ o \end{Bmatrix}$$

Where:

$$\left[m^s \right] = \Phi^{sT} M \Phi^s \qquad \left[k^s \right] = \Phi^{sT} K \Phi^s$$

DYNAMIC REDUCTION IN THE FLUID

$$p = \Phi^f \xi^f$$

$$\begin{bmatrix} M^s s^2 + K^s & -A^T \Phi^f \\ s^2 \Phi^{fT} A & \lceil m^f \rfloor s^2 + \lceil k^f \rfloor \end{bmatrix} \begin{Bmatrix} u \\ \xi^f \end{Bmatrix} = \begin{Bmatrix} F \\ o \end{Bmatrix}$$

Where:

$$\lceil m^f \rfloor = \Phi^{fT} M^f \Phi^f \qquad \lceil k^f \rfloor = \Phi^{fT} K^f \Phi^f$$

DYNAMIC REDUCTION IN BOTH

$$u = \Phi^s \xi^s$$
$$p = \Phi^f \xi^f$$

$$\begin{bmatrix} \lceil m^s \rfloor s^2 + \lceil k^s \rfloor & -\Phi^{sT} A^T \Phi^f \\ s^2 \Phi^{fT} A \Phi^s & \lceil m^s \rfloor s^2 + \lceil k^s \rfloor \end{bmatrix} \begin{Bmatrix} \xi^s \\ \xi^f \end{Bmatrix} = \begin{Bmatrix} \Phi^{sT} F \\ o \end{Bmatrix}$$

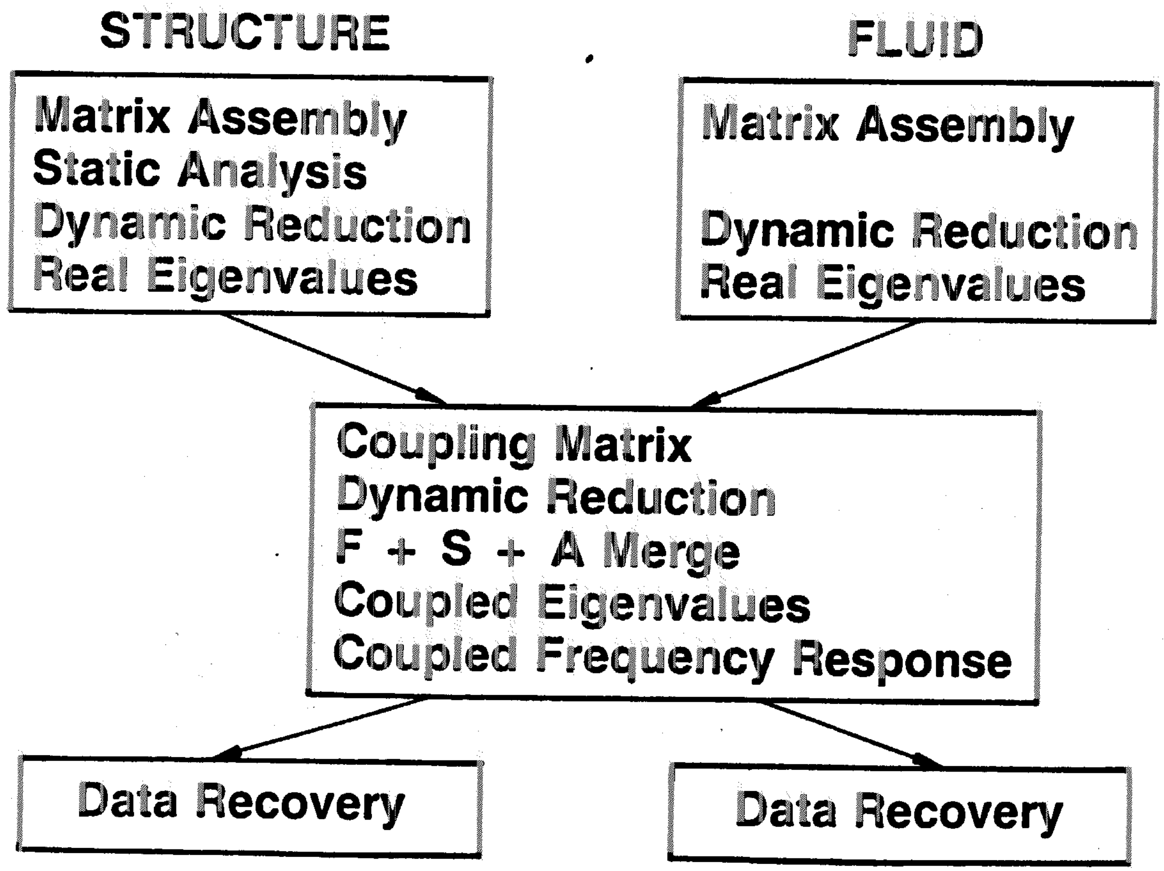
MSC/NASTRAN IMPLEMENTATION CONCERNS

- o USE DMAP WHENEVER POSSIBLE: CANNOT MODIFY MSC/NASTRAN AT FORTRAN LEVEL
- o MAINTAINABILITY, UPWARD COMPATIBILITY: DMAP BETTER THAN ALTER PACKET
- o USER FRIENDLINESS TO NON-EXPERT
- o SIZE LIMITATIONS: SUPEREMENT TECHNIQUES, SPLIT DATA BASES
- o DYNAMIC REDUCTION VIA:
  - GDR OR LANCZOS
  - GUYAN
  - MODAL TRANSFORMATION

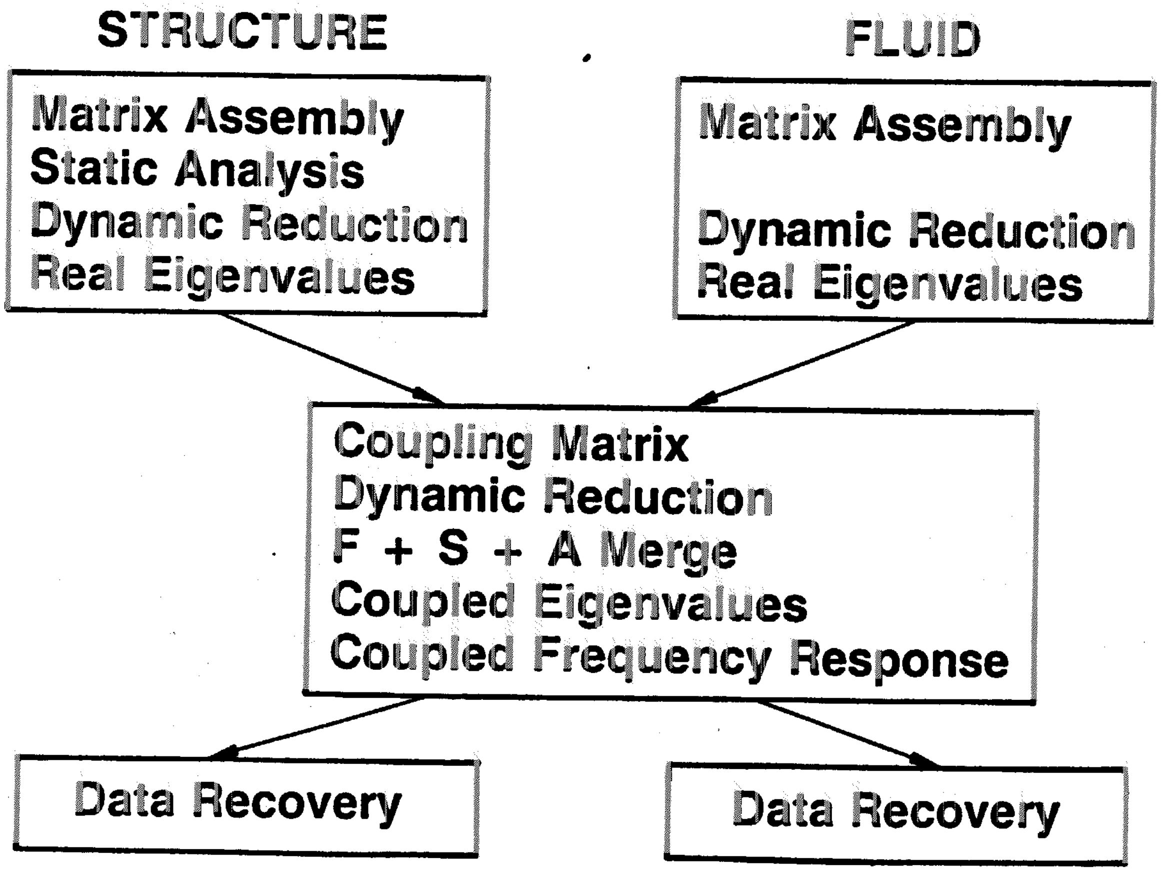
PROJECT TEAM QUALIFICATIONS

- FINITE ELEMENT THEORY
- STRUCTURAL DYNAMICS: CMS
- ACOUSTICS
- MSC/NASTRAN EXPERTISE: MODELING, DMAP
- NUMERICAL METHODS: EFFICIENCY, ERROR ANALYSIS
- SOFTWARE/HARDWARE INTERFACE: LARGE PROBLEMS, PRE/POST PROCESSING

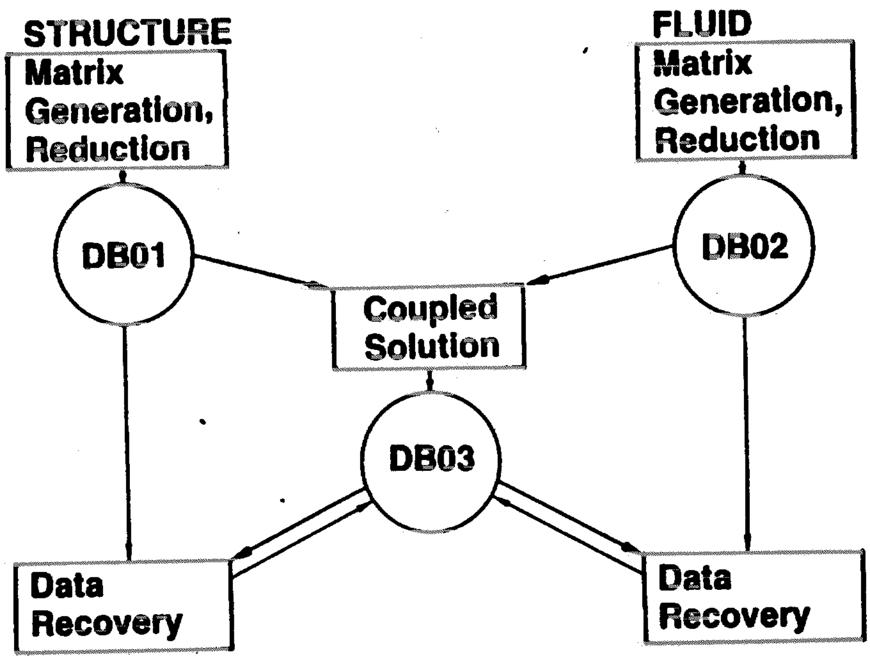
SOLUTION FLOW DIAGRAM



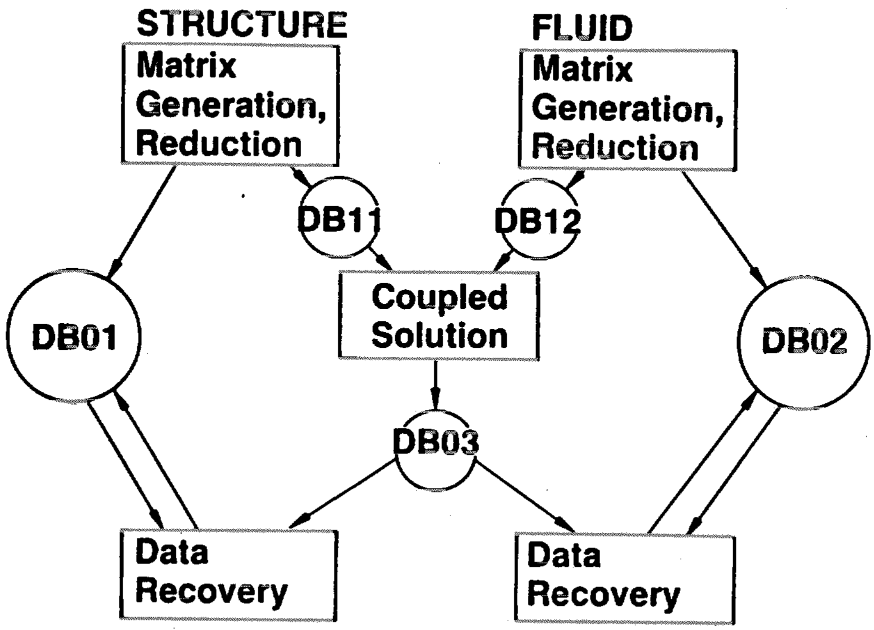
SOLUTION FLOW DIAGRAM



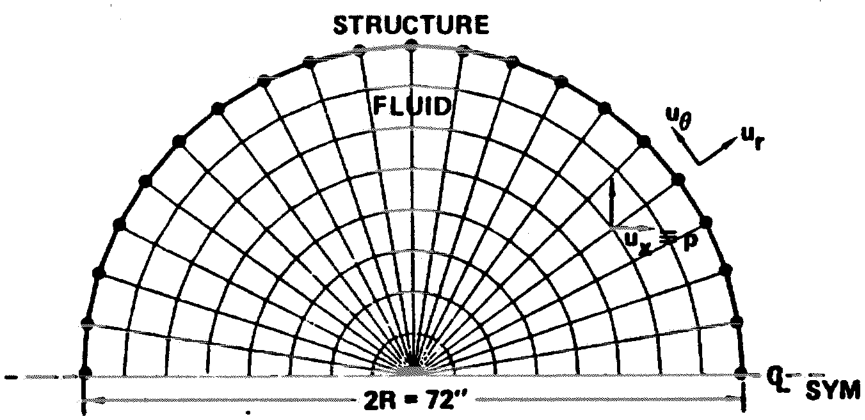
DATA BASE MANAGEMENT



DATA BASE SUBDIVISION FOR LARGE PROBLEMS



2-D CYLINDER, SYMMETRIC HALF MODEL



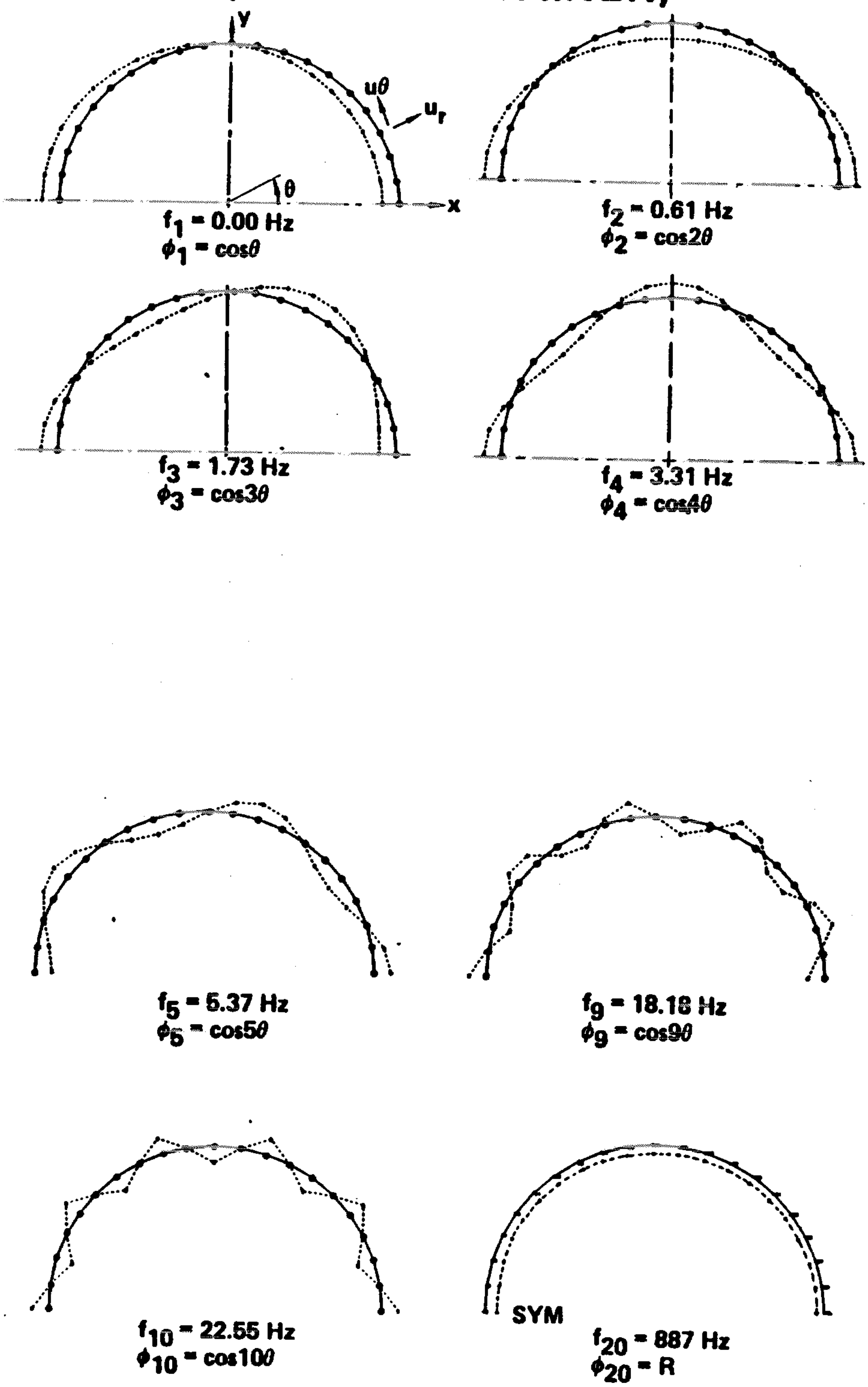
2D Structural Vibrations, Uncoupled (Empty Container)

$U = \cos(n\,\phi)$

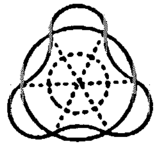
MODE n	THEORETICAL (Hz)	NASTRAN (Hz)
1	0.00	0.00
2	0.85	0.61
3	2.03	1.73
4	3.70	3.32
5	5.84	5.37
.		
.		
.		
20	95.22	91.22

.26

UNCOUPLED 2-D STRUCTURAL VIBRATIONS (EMPTY CONTAINER)

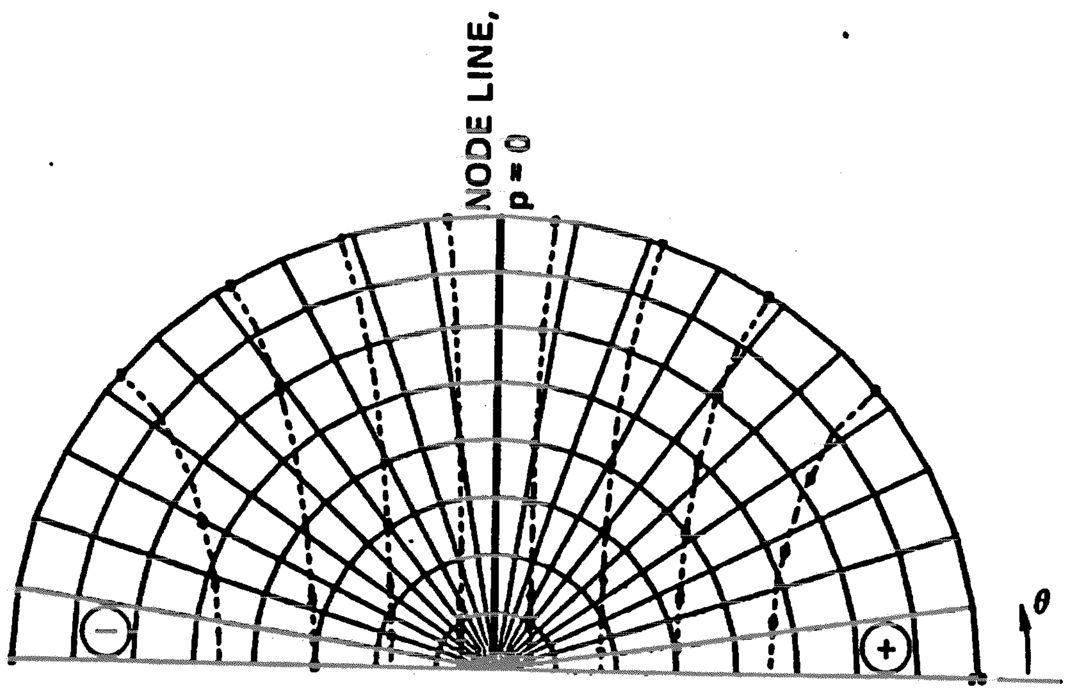


UNCOUPLED ACOUSTIC VIBRATIONS
(RIGID WALL)



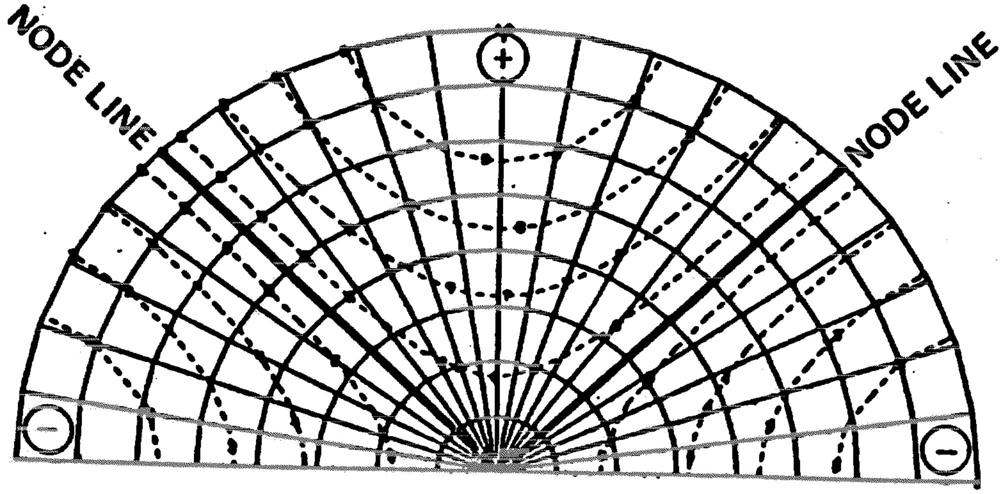
$\Diamond\ u = \cos(n\,\phi)\, Be_k(R)$

MODE n.k.	THEORETICAL (Hz)	NASTRAN (Hz)
0.0	0.00	0.00
1.0	109.05	109.09
2.0	180.83	180.07
0.1	226.86	224.65
3.0	248.74	246.09
4.0	314.82	308.92
1.1	315.64	313.41



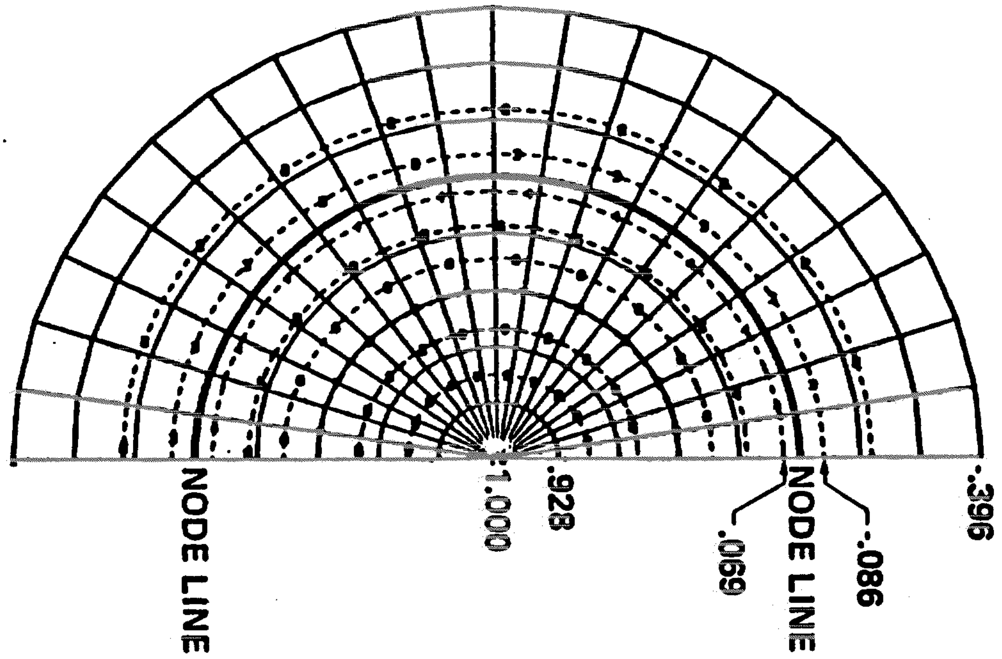
f_2 = 109.09 Hz

$\phi_2 = \cos\theta$

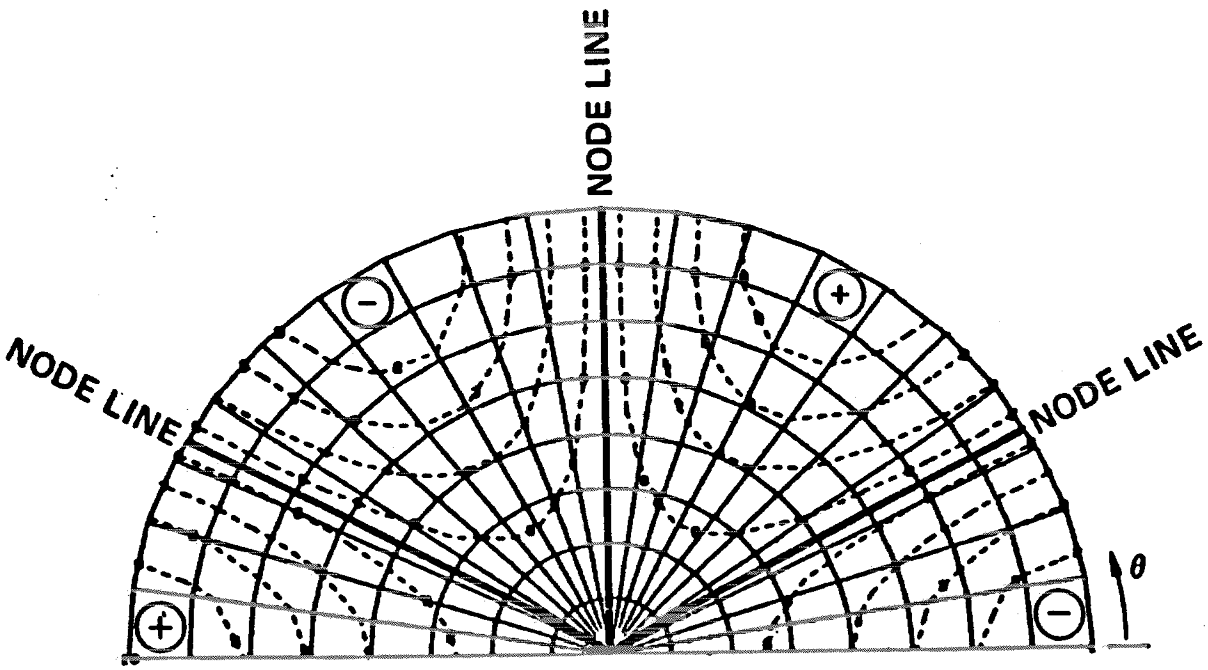


f_3 = 180.07

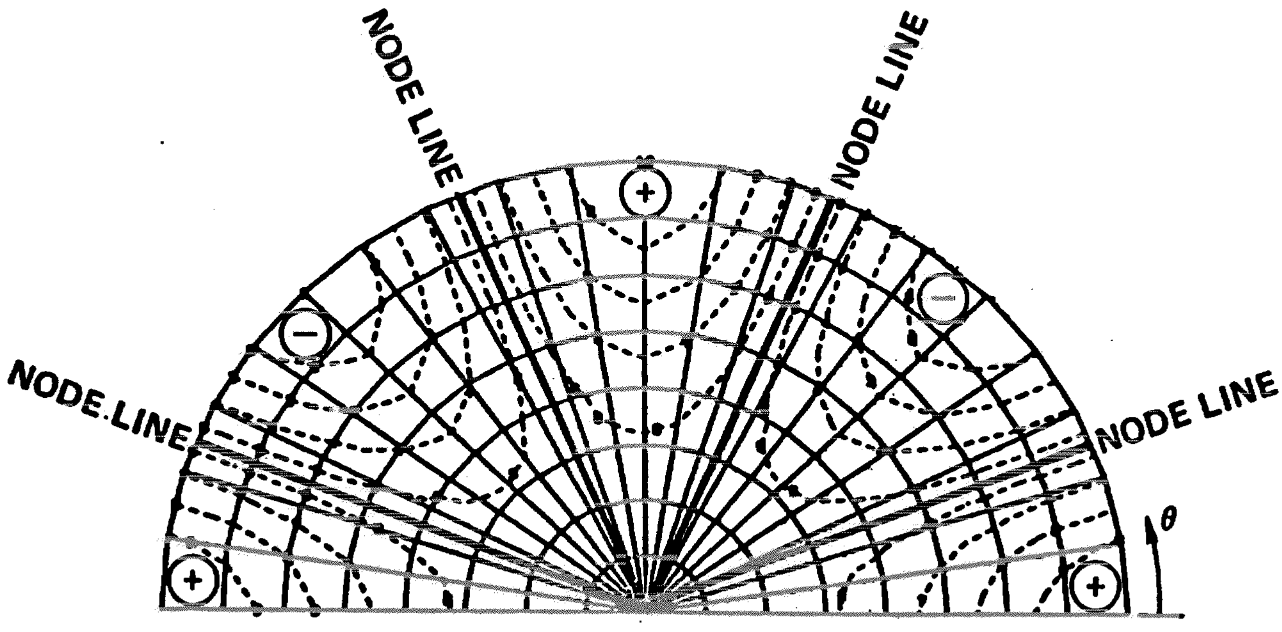
$\phi_3 = -\cos 2\theta$



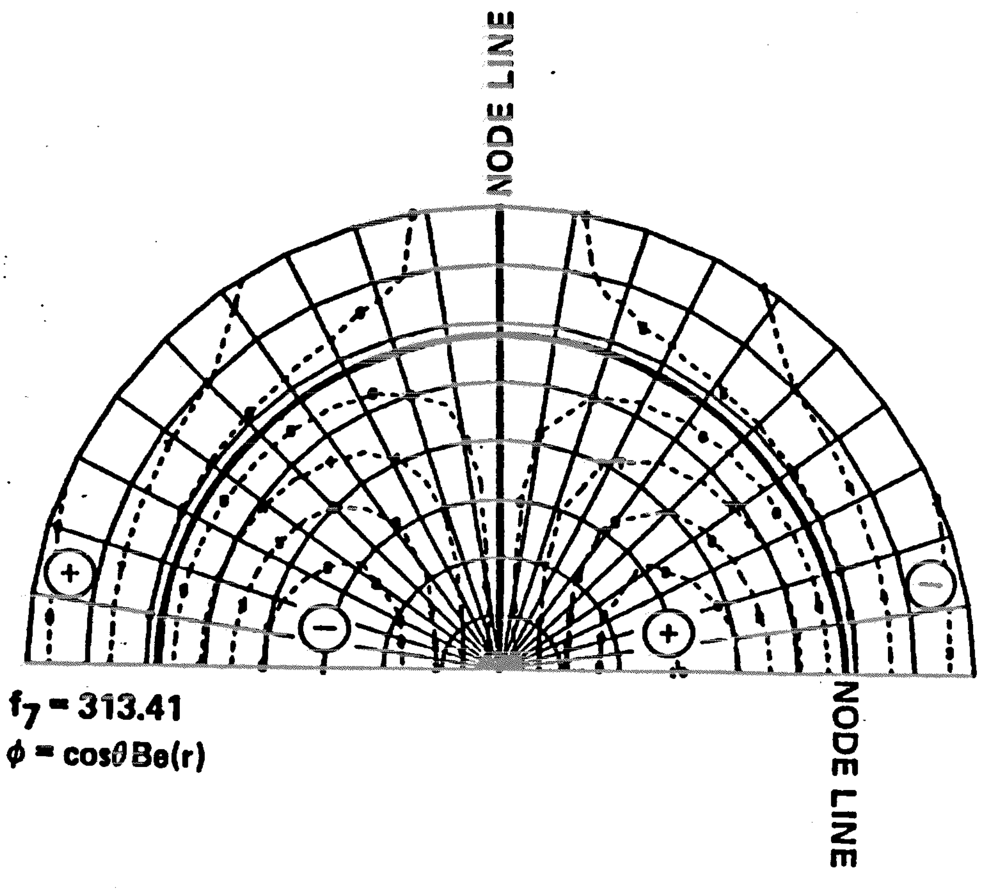
$f_4 = 224.65$ RADIAL MODE $\phi_6 = -Be(R)$



$f_5 = 246.09$ $\phi_5 = -\cos 3\theta$

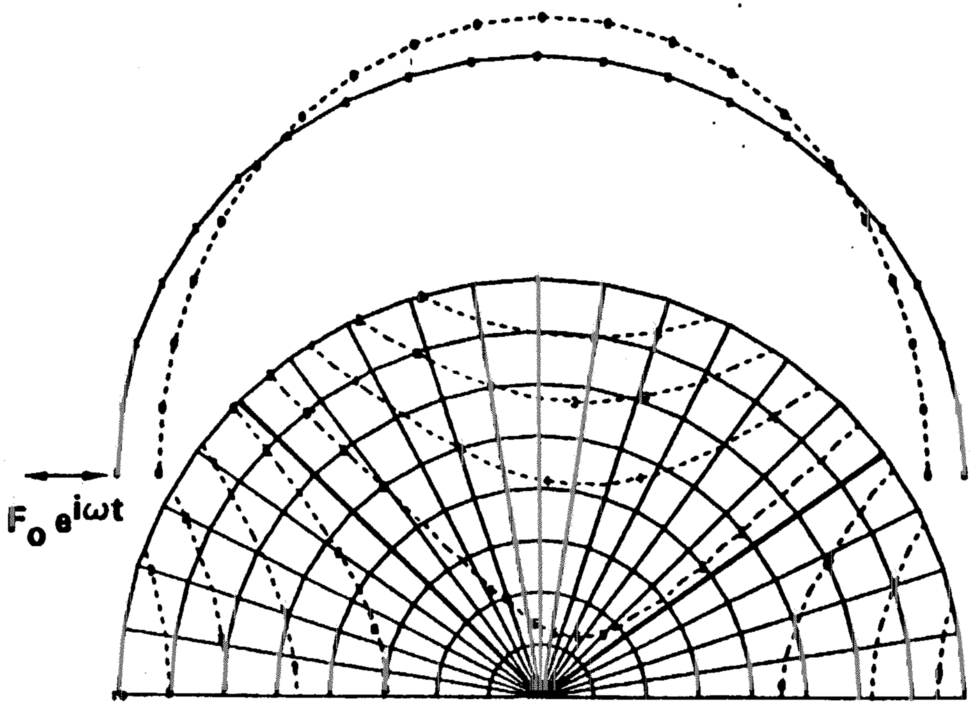


$f_6 = 308.92$ $\phi_6 = \cos 4\theta$

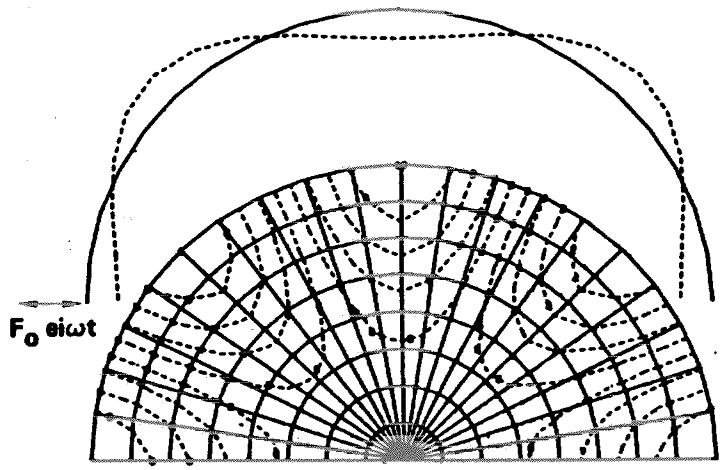


$f_7 = 313.41$
$\phi = \cos\theta \, Be(r)$

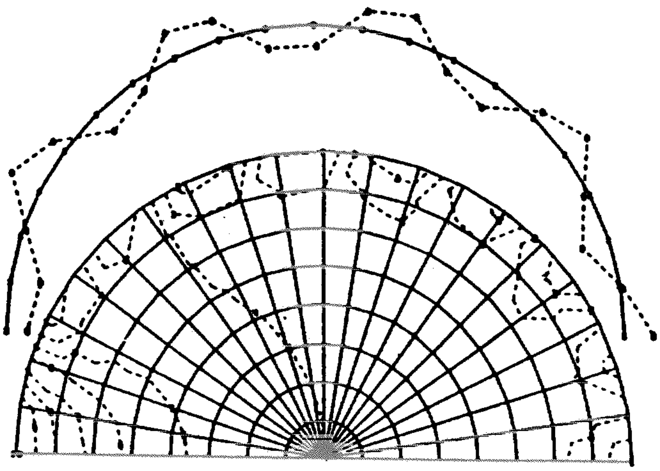
COUPLED FREQUENCY RESPONSE: SPATIAL DISTRIBUTION AT 0.612 Hz



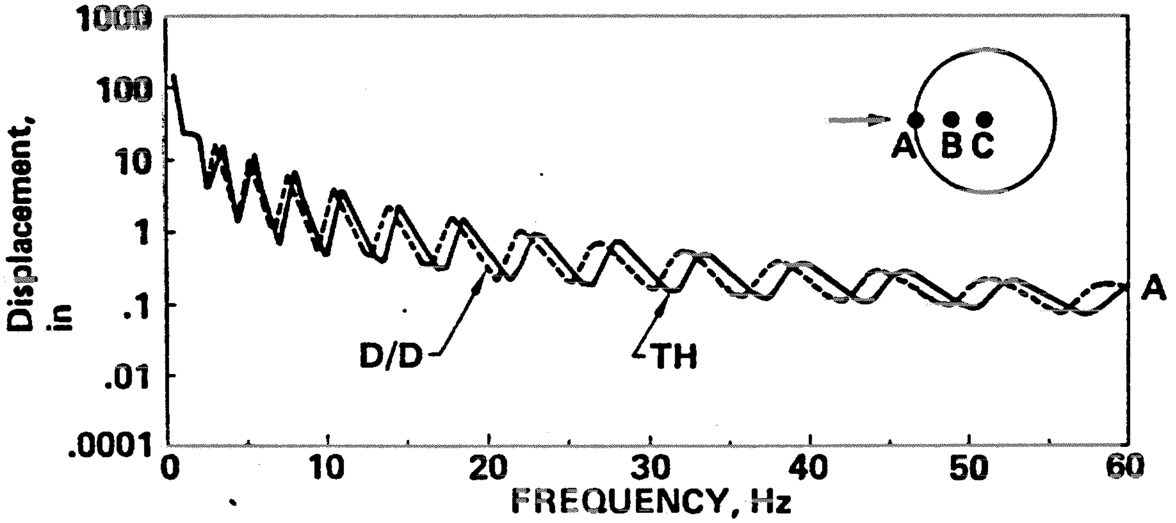
COUPLED FREQUENCY RESPONSE AT 3.20 Hz

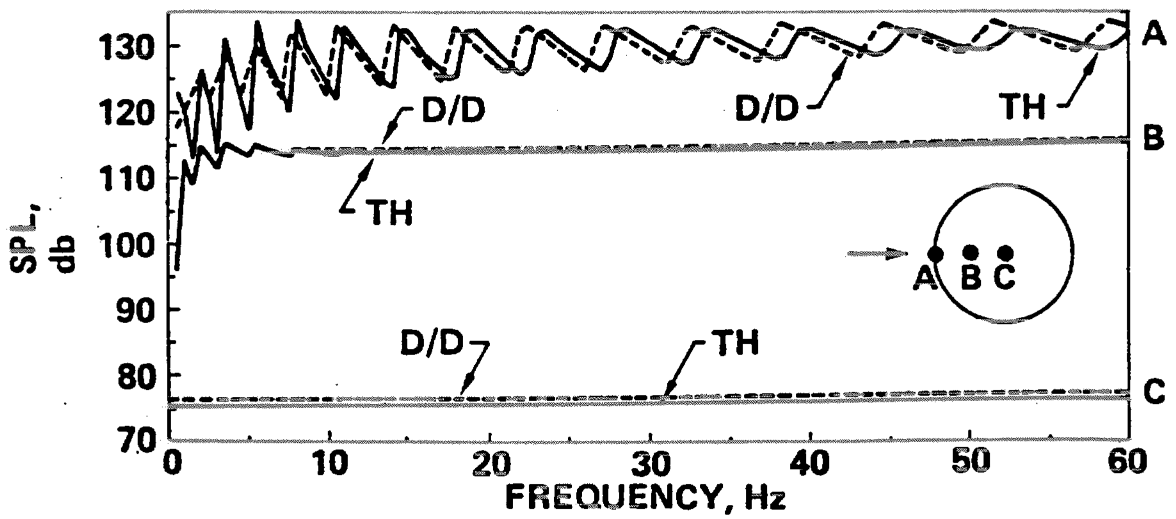


COUPLED FREQUENCY RESPONSE AT 20 Hz

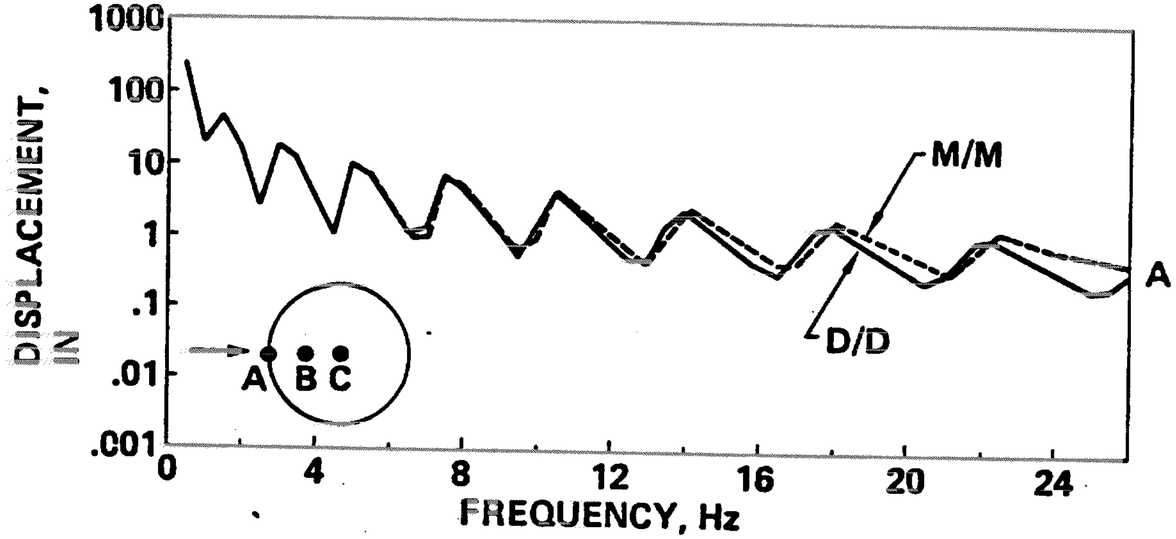


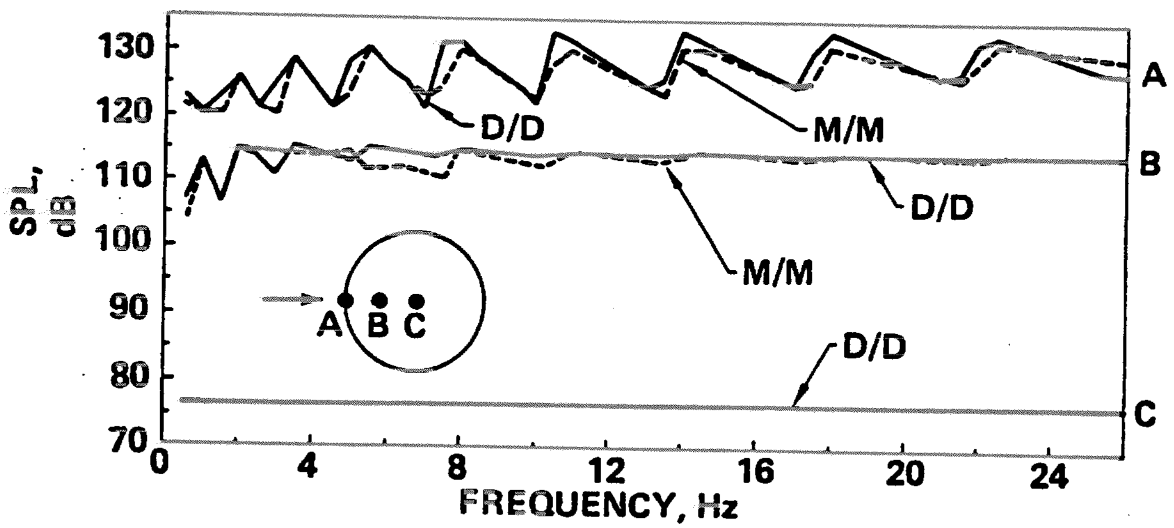
COUPLED FREQUENCY RESPONSE AT SELECTED LOCATIONS

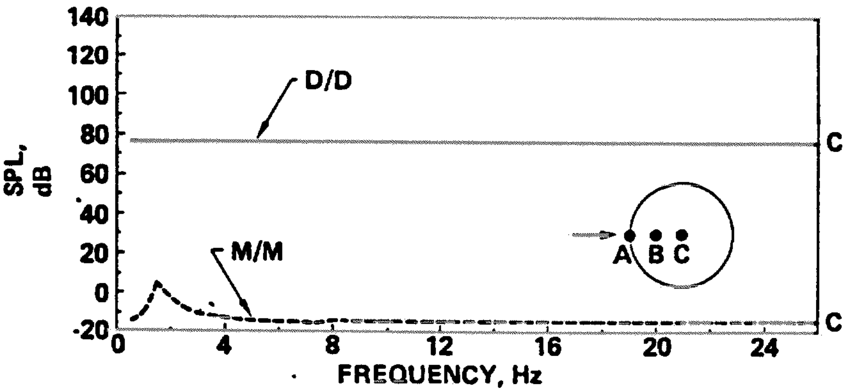




NASTRAN MODAL/MODAL VS. NASTRAN DIRECT/DIRECT



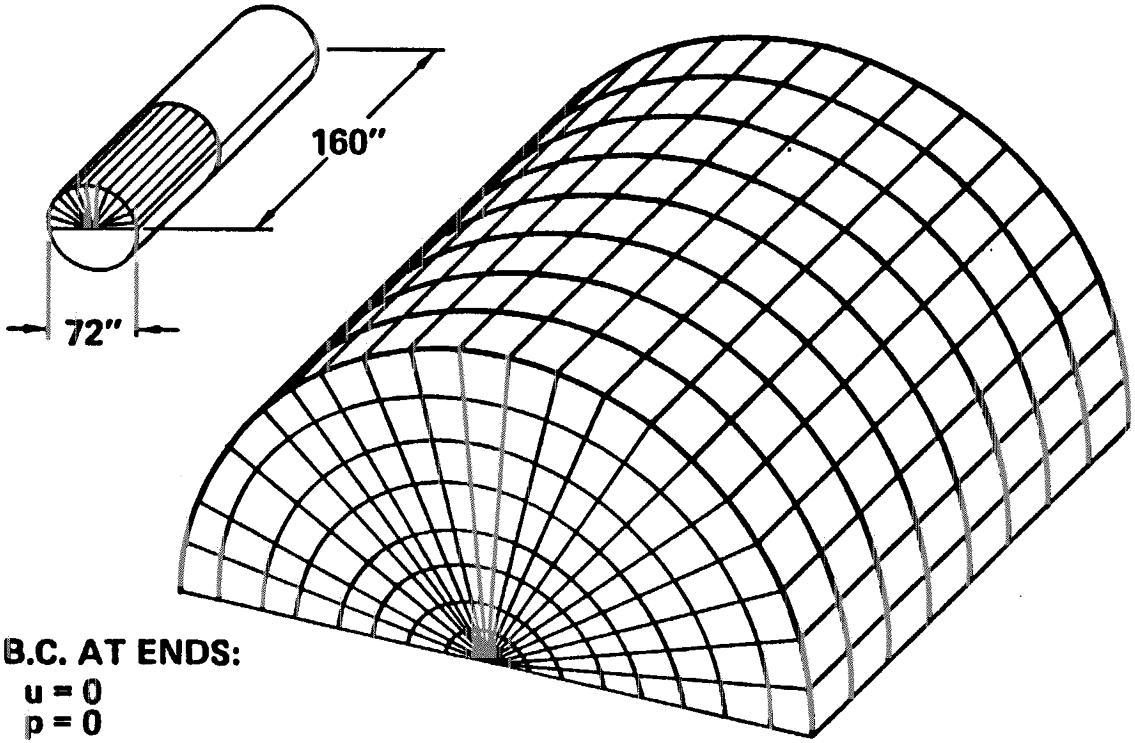




2-D CYLINDER CONVERGENCE STUDIES

- ELEMENT SELECTION: QUAD2 VS. QUAD4
- MESH REFINEMENT
- MODAL TRUNCATION
- MODAL CONTENT OF APPLIED LOAD

3-D CYLINDER, SYMMETRIC QUARTER MODEL



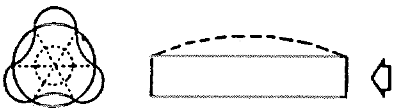
B.C. AT ENDS:
$u = 0$
$p = 0$

3D Structural Vibrations
Uncoupled (Empty Container)

$$u = \cos n\phi \sin \frac{m \pi x}{L}$$

MODE n.m.	THEORETICAL (Hz)	NASTRAN (Hz)
6.1	14.71	14.68
5.1	14.84	14.85
7.1	16.73	16.66
4.1	18.28	18.26
8.1	20.05	19.99
9.1	24.21	24.21
3.1	26.95	26.59

UNCOUPLED ACOUSTIC VIBRATIONS (RIGID WALL)



$$u = \cos (n\phi) \sin \frac{m \pi x}{L} Be_k (R)$$

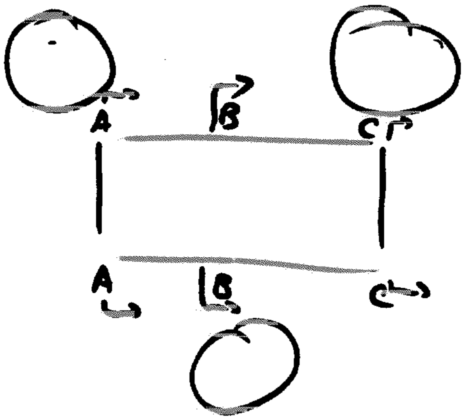
MODE n.m.k.	THEORETICAL (Hz)	NASTRAN (Hz)
1.0.0	41.85	41.89
1.1.0	116.77	117.06
3.0.0	125.55	126.71
3.1.0	166.27	167.27
1.2.0	185.61	187.55
5.0.0	209.25	214.66

UNCOUPLED 3-D STRUCTURAL VIBRATIONS (EMPTY CONTAINER)

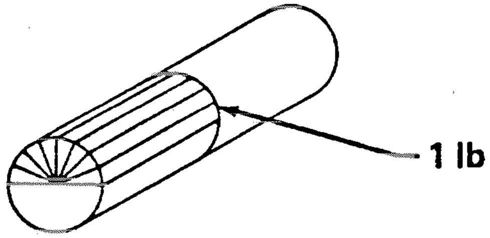
$f_1 = 15.73$ Hz
$\phi_1 = \cos 6\theta * \sin \frac{\pi z}{\ell}$

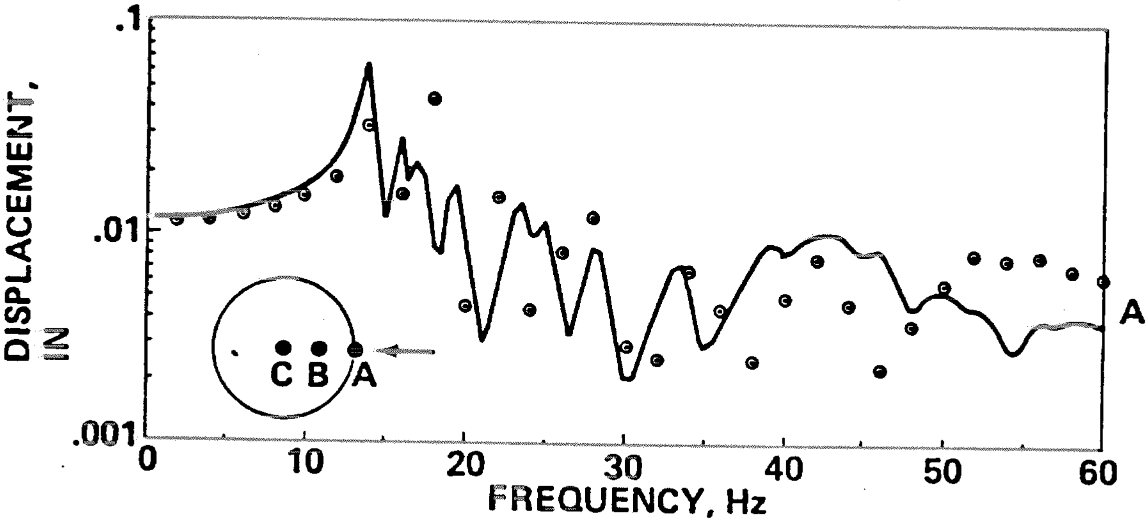
UNCOUPLED 3-D FLUID VIBRATIONS (RIGID CONTAINER)

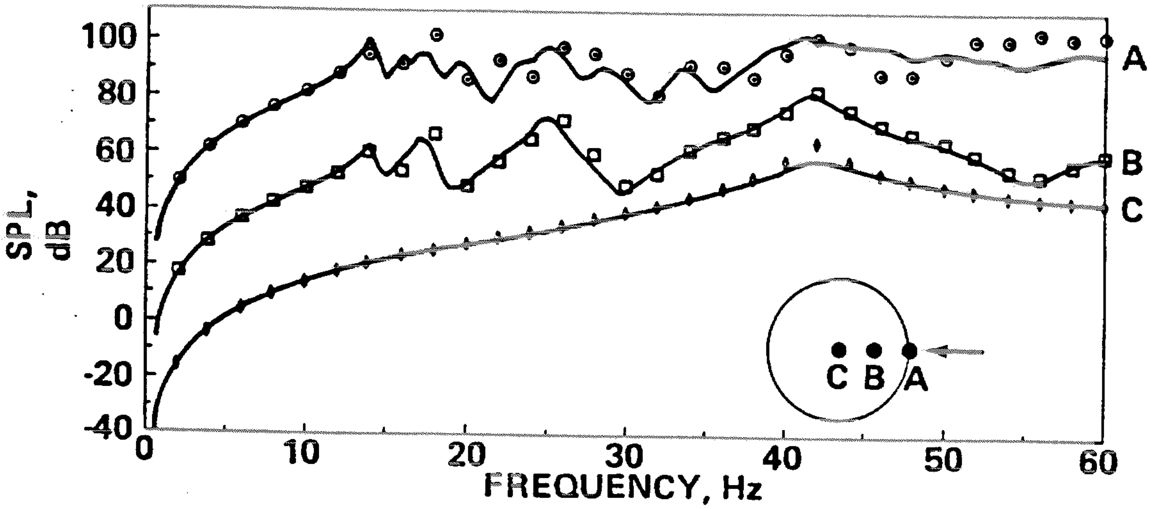
SEE COLOR CONTOUR PLOTS



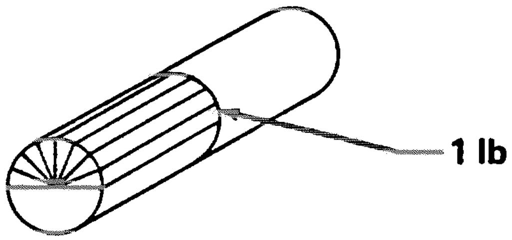
COUPLED 3-D FREQ RESPONSE
NASTRAN D/D VS. CLOSED FORM

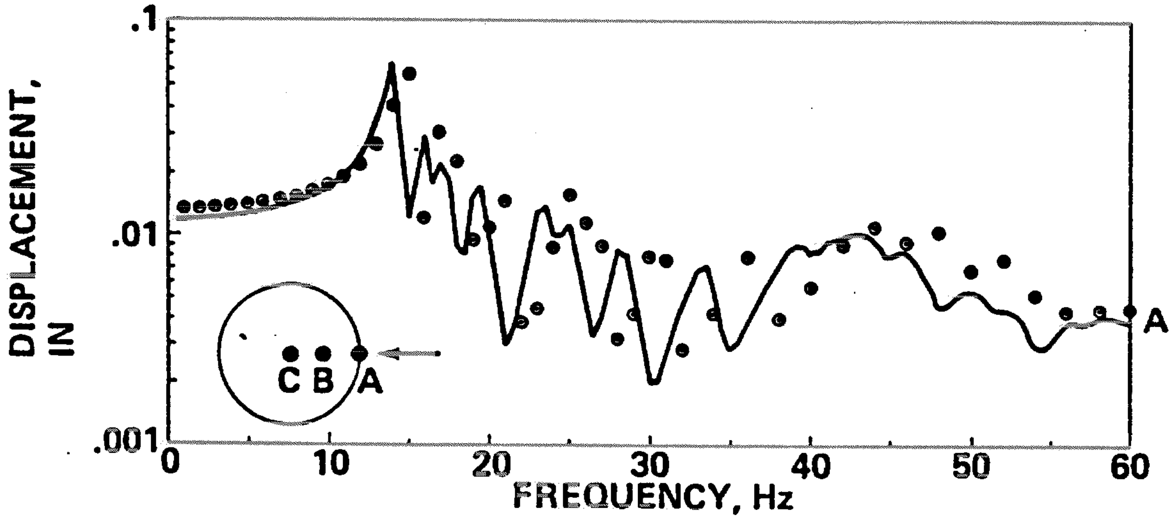


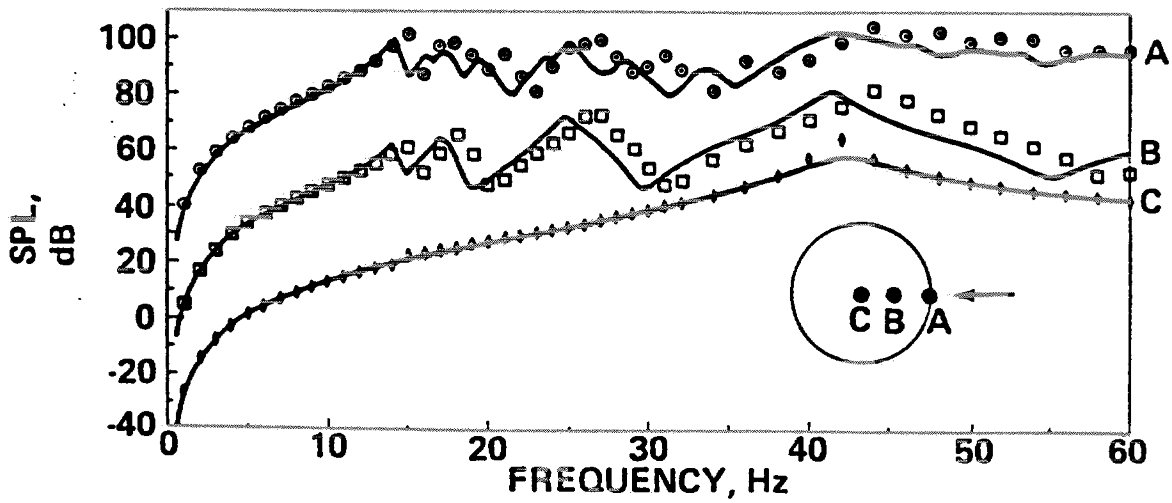




COUPLED 3-D FREQ RESPONSE
UNCOUPLED D/D VS. CLOSED FORM







CONCLUSIONS

- FSI CAPABILITY GAVE GOOD RESULTS FOR TEST PROBLEMS
- SIMPLIFYING ASSUMPTIONS
- COST VS. ACCURACY TRADE-OFF
- SIZE LIMITATIONS
- MSC/NASTRAN DMAP IMPLEMENTATION LIMITATIONS
- FURTHER PLANS FOR ENHANCEMENT

FURTHER PLANS FOR ENHANCEMENT

- INCLUDE RESIDUAL FLEXIBILITY OF TRUNCATED HIGHER ORDER MODES
- DISCRETE DAMPING
- RELAX MODELING RESTRICTIONS AT INTERFACE
  - NON-IDENTICAL MESH (INTERPOLATION)
  - NON-NORMAL STRUCTURAL u_1
- MORE DATA RECOVERY OPTIONS
- VALIDATION VS. VIBRATION TEST