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MASTER

AN EXPERIMENTAL VIBRATION STUDY OF IN-AIR AND FLUID COUPLED CO-AXIAL CYLINDERS

by

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I. INTRODUCTION

The influence of a fluid environment on the dynamic characteristics of structures has been of interest for hundreds of years. In 1779 Pierre Louis Gabriel Du Buat (1) investigated the effects on the frequency response of pendulum bobs vibrating in water. Summaries of nineteenth century investigations can be found in the works of Lord Rayleigh (2) and Lamb (3). More recent experiments by Stelson (4), Keane (5), and Jones (6) were used to explore the virtual mass concept for beams in fluids. Analytical work on the interaction of a shell in a fluid medium is surveyed by Geers (7). Fritz (8), in an experimental investigation of a cylinder bounded externally by a fluid gap and a rigid shell, showed the use of the virtual mass approach for hydroelasticity and bouyancy effect, although damping and compressibility were neglected.

More recent experimental work which is of particular significance to this work is the coupled breathing vibrations of two long, thin coaxial cylinders in fluid by Levin and Milan (9), vibrations of a flexible cantilever cylinder surrounded by a water gap and a rigid cylinder by Au-Yang (10), the vibration of a thin cantilever cylinder surrounded by water and a rigid cylinder with varying gap sizes by Mulchay, et al (11), and the vibration study of a flexible set of cylinders by Au-Yang (12). In the Levin-Milan test, only two sets of flexible cylinders were evaluated with a large gap ((radius/gap)$\sim$5) and an inner cylinder with a variable thickness. The Ref (10) test consisted of a single cylinder test with about the same radius to gap ratio. The Mulchay tests included the variation of the gap size but did not include coupling of flexible cylinders.

Much analytical work has been performed since the 1960's. Some advantages and disadvantages of finite element (presently the most popular technique) eigenvalue solid-fluid methods with respect to flexible cylinders is discussed by Brown (13).

It is the objective of this study to develop and carry out an experimental program which would provide additional insight into the mechanics of fluid-solid interaction with respect to the response of a set of coaxial cylinders with water in the annulus. Such configurations are found in nuclear reactors in the vessel wall-thermal liner. The effects of cylinder thickness and the fluid filled annulus gap size on the resonant frequencies and mode shapes of the cylinders (either singly or coupled in air and a water environment) are presented in this paper; also included is an evaluation of damping for selected gaps and cylinder thicknesses. Details of the experimental setup and procedures are also outlined.

II. EXPERIMENTAL SET-UP

Test Cylinders - In order to assess the influence of the cylinder thickness, annulus gap size, and boundary conditions on the frequency response, a total of fourteen cylinders, twelve acrylic and two steel, of various diameters, thicknesses, and rigidities is tested in various combinations. Table 1 gives the nominal sizes of the test cylinders. Figure 1 shows several of the acrylic cylinders and the inner steel cylinder. The material properties of the acrylic and steel cylinders are given in Table 1. A study of the thickness variation and radius variation on the acrylic cylinders indicated that the variations were 0.4% to 5% and 0.08% to 3.4% respectively. The cast cylinders showed the best tolerances.

Figure 1 shows that the inner acrylic cylinders were manufactured with acrylic caps while the outer acrylic cylinders are fitted with flanges. The acrylic caps and flanges are 0.25 inches thick, with 6 bolt holes in the caps and 12 bolt holes in the flanges. Figure 2 shows a typical longitudinal section for an inner and outer cylinder pair as attached to the test frame.

The flanges and caps are designed to provide nearly rigid lateral restraints on the cylinder ends while allowing relatively high rotational freedoms. Metal disks are glued circumferentially at 10° increments around the circumference at three levels, namely 1/4, 1/2, and 3/4 of the height of each acrylic cylinder and at 1.5" increments along longitudinal lines at four locations, $\theta = 0$, 90, 180, and 270 degrees. These disks are used to provide monitoring points for eddy current probes for determining the mode shapes of vibration.

Test Set-Up - Figure 3 shows the test frame which consisted of two rigid steel stands. The stand on the right is used to mount the cylinders while the stand on the left housed the 10-lb shaker. Both stands are rigidly bolted on a 3 ton isolated vibration test table. Points of inner and outer cylinders are bolted to the frame at each end (except for certain tests for which the inner cylinder was left free at the top) using steel bolting rings and discs to circumferentially distribute the restraints about the flanges, as indicated in Figure 2. During the test series 13 and 14 (Table 2), the natural frequencies of a set of cylinders immersed in a tank of water is determined. The tank is constructed by applying four acrylic panels between each of the frame's four I-section support posts shown in Figure 3.

Instrumentation - Figure 4 gives an overall view of the instrumentation while Figure 5 is a schematic of the experimental procedure. As shown in Figure 5, the outer cylinder is "point" excited by a 10-lb shaker which is controlled by a B&K vibration exciter. Constant force excitation is maintained by a feedback signal from a load cell mounted in series with the exciting rod which instantaneously measured the excitation force. The frequency of oscillation of the cylinder is monitored by a B&K mini-accelerometer mounted on the surface of the cylinder and displayed on an electronic digital counter.

To obtain the mode shapes, the displacement responses of a cylinder is picked up by mapping eddy probes. Outer cylinders are mapped on the outer surfaces and inner cylinders are mapped on the inner surfaces. Time history responses of both inner and outer cylinders are simultaneously displayed on a dual trace scope for monitoring and phase relationship determination. The r.m.s. values of responses are instantaneously recorded by a B&K level chart recorder during frequency "sweeping" operation to determine the natural frequencies. The signals are also stored in analog form by an Ampex F. M. Recorder for future analysis.

III. EXPERIMENTAL PROCEDURES

Table 2 gives the series of tests performed in this study. As previously mentioned, these tests are designed to determine the influences of cylinder thickness and annulus gap size on the resonant frequencies and mode shapes of fluid coupled coaxial cylinder systems. The effect of damping is also assessed.

Resonant Frequencies - The approximate resonant frequencies of each cylinder system, i.e., an inner and outer cylinder combination, and individual cylinders is air and water are first determined using the standard "sweeping" techinque. The outer cylinder was excited from 5 Hz to 1K Hz at a slow sweeping rate (≈ 40 deg/min.) The peaks of the r.m.s. spectral plot of the response from a stationary eddy probe opposite the excitation point (anti-node) inside the inner cylinder provides the approximate locations of the resonant frequencies are then pinpointed by vernier scanning in the vicinity of the "peaks". Three to four of the lowest resonant frequencies are determined for the preliminary tests which was conducted in air.

Mode Shapes - After the resonant frequencies are determined for the coupled cylinder pairs, the phase relationship between the inner and outer cylinder responses at each particular frequency is determined by simultaneously displaying both signals on a dual trace scope. The mode shapes are determined by mapping with an eddy probe pointed at the metal disks which are distributed both circumferentially and axially.

Figure 7 shows the mode shapes of the inner cylinder 1 inside of outer cylinder 10 for the six lowest resonant frequencies.

Equivalent Viscous Damping Ratio (ε) - The half-power technique is used to determine the viscous damping ratio for a given natural mode. The damping ratio as developed for the half-power point techinque is:

$$\varepsilon = \frac{f_{b_1} - f_{b_2}}{2f_n}$$

where

f_n = natural frequency with a response x_n

f_{b_1}= lower side band frequency with a response,

$$x_{b_1} = 0.707\ x_n$$

f_{b_2}= higher sideband frequency with a response,

$$x_{b_2} = 0.707\ x_n$$

The response x_n is first measured by an eddy probe at the antinode for a given natural frequency f_n. Then the sideband frequencies f_{b_1} and f_{b_2} are obtained by "Vernier" scanning of both the left and right sides of the "peak" where the responses were 0.707 X_n as read through a digital meter.

IV. RESULTS AND OBSERVATIONS

Figures 9-23 represent data from the 81 test series 1 to 14 of Table 2. For each test approximately 4 resonant frequencies per cylinder and the corresponding mode shapes are determined beginning with the lowest frequency. The cylinder combinations are developed in the following manner. First, the resonant frequencies for each cylinder in air are determined (test series 1-3). Then the nine inner cylinders are combined with each of the three outer cylinders yielding 27 tests. A cross check of the shell modes is performed by allowing the inner cylinders to be simply supported during one phase (test series 4-6) and Cantilevered in the second phase (test series 7-9). In test series 4-9, the annular gap between the outer and inner cylinder was filled with water. In all tests the outer cylinders are supported at both ends.

Figures 20-22 represent the results of test series 10-12 in which the damping characteristics of 7 cylinder combinations were studied. Test series 10 consisted of the rigid outer cylinder with flexible cylinders #3, #6, #9 of 0.125 inch-thickness of different diameters. Test series 11 involved the rigid inner cylinder and the #12, 0.125 inch thick outer cylinder, while test series 12 used the #12 of 0.125 inch thick outer diameter and the 0.125 inch thick #3, #6, #9 inner cylinders with different diameters.

During test series 4-9, it was observed that the cylinder combinations vibrated either in-phase or out-of-phase. During in-phase motion, the inner and outer cylinders both experienced the same sign on their radial component of motion at the same circumferential location. During out-of-phase motion, the inner and outer cylinders experienced different signs on their radial component of motion at the same circumferential location.

In Air Variation of Frequency with Thickness - Figures 9-12 illustrate the variation of frequency vs thickness for constant diameter of the 12 acrylic cylinders tested in air. We observe an increase in frequence as a function of increasing thickness and the greatest variation occuring for smaller diameter cylinders. Included in figures 9-12 is analytical comparisons to the experiments provided by a finite element idealization of the cylinders. An illustration of the SAP6 plate element (5 DOF per mode) representation of an inner cylinder is provided in figure 13. The finite element results (of ideal shell dimensions) in Figures 9-12 shows good correlation with the experimental data.

Variation of Frequency with Annular Gap - Figures 14-19 show the variation in the resonant frequency with gap for various model responses.
Figures 14-16 cover the out-of-phase data and Figures 17-19 cover the in-phase data. These results show that as the gap size decreases the resonant frequency of the cylinder system decreases for the out-of-phase modes, while for in-phase modes the resonant frequency increases as the gap size decreases. It is also seen that the resonant frequency is a nonlinear function of the gap size. Interestingly, a close correspondence of results as found for similar inner-outer and outer-inner cylinder thickness, e.g. the 0.25 inch outer - 0.187 inch inner cylinders have similar resonant frequencies as the 0.187 inch outer - 0.25 inch inner combination. Similar results also exist for other cylinder combinations for the out-of-phase as well as for the in-phase modes.

Variation of Frequency with Inner Cylinder Thickness - Figures 14-19 show the variation in the resonant frequencies with the thickness of the inner cylinder for constant gap size. These results indicate that for both in-phase and out-of-phase modes the resonant frequency of the cylinder combinations tend to decrease with decreasing thickness. These results also show that the in-phase modes have significantly higher resonant frequencies associated with them.

Another observation which can be made involves the frequency change as a function of the inner cylinder thickness. It is noted for the thinest outer cylinder 0_3, thickness of 0.125 inch, that the variation in the resonant frequency does not depend on the inner cylinder thickness.

Variation of Damping Ratios with the Annular Gap - Figures 20 and 21 show the variation of the damping ratio with gap for the out-of-phase and in-phase modes, respectively, of two flexible concentric cylinders with a 0.125 inch thicknesses and a varying gap. In Figure 20 the upper curve is labeled with ε_2 and ε_3 since the resonant frequencies ω_2 and ω_3 were the same for this particular thickness. These results indicate that for the out-of-phase modes the damping ratios increase from about 5% to about 15% as the gap decreases, while for the in-phase modes the damping ratios generally remain within the 4 - 5% range as the gap size becomes smaller.

Figure 22 shows the variation of damping ratios with gap for a rigid outer cylinder concentric with flexible inner cylinders. Again it is observed that the damping ratio increases drastically from about 4% to 32% as the gap between the two cylinders decreases.

Cylinder Frequency versus Thickness (immersed in tank of water) - In Figure 23 is presented the results of test series 13 and 14 which is cylinders #7, #8, #9 shown in Table 1. A comparison of Figure 11 and 23 clearly shows the reduction of cylinder frequencies of those obtained in air versus those performed in air. It should be pointed out that the 31" x 31" x 24" tank takes a cruciform shape because of the 6 1/4" x 6 1/4" column sections occupying each corner.

V. CONCLUSIONS AND RECOMMENDATIONS

The results of this study have provided significant data and understanding on the mechanics of solid-fluid interactions of two concentric flexible cylinders with water in the annular gap. The most significant findings of this study can be briefly stated as follows:

1) The frequency varies nonlinerarly with gap size. For out-of-phase modes, the resonant frequency increases as the gap size decreases and for in-phase modes the resonant frequency approaches a limiting value as the gap size decreases.
2) The resonant frequency decreases as the thickness of the inner cylinder decreases for both the in-phase and out-of-phase modes.
3) There is a significant increase in the damping ratios of the cylinder system as the gap decreases for the out-of-phase modes but only a moderate increase for the in-phase modes.
4) There is significantly greater damping associated with flexible-to-rigid cylinder pairs compared to flexible-to-flexible cylinder pairs.

From the results of this study, it is clear that the gap size is an extremely important parameter in either increasing or decreasing the frequency response and/or the damping ratios of the system. The selection of cylinder thickness is important in tuning the coaxial cylinders. The results of this study indicate that, in designing reactor systems composed of coaxial cylinders which reside in a fluid environment, it is imperative that analytical formulations incorporate the additional features which account for the effects of the fluid. It is felt that the work presented here provides benchmark problems which can be used in the verification of computer codes. Further studies on the seismic response of fluid-coupled coaxial cylinders and the effects of fluids with varying viscosities on the resonant characteristics of coupled systems will be reported in later papers.

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Table 1 TEST CYLINDERS

Cylinder	Material	OD(in)	ID(in)	t(in)	R/t	L(in)	Type*	Code**
1	acrylic	12.0		0.250	24	24	E	I_{11}
2	acrylic	12.0		0.187	32.1	24	E	I_{12}
3	acrylic	12.0		0.125	48	24	E	I_{13}
4	acrylic	13.0		0.250	26	24	R	I_{21}
5	acrylic	13.0		0.187	34.8	24	R	I_{22}
6	acrylic	13.0		0.125	52	24	R	I_{23}
7	acrylic	13.625		0.250	27.3	24	R	I_{31}
8	acrylic	13.625		0.187	36.4	24	R	I_{32}
9	acrylic	13.625		0.125	54.5	24	R	I_{33}
10	acrylic		13.875	0.250	28.8	24	R	O_1
11	acrylic		13.875	0.187	38.1	24	R	O_2
12	acrylic		13.875	0.125	56.5	24	R	O_3
13	steel	13.625		0.25		24	R	I_R
14	steel		13.875	0.25		24	R	O_R

* E = extruded (catalogue item)

R = rolled (special order)

** I = inner cylinder

O = outer cylinder

Material Properties @ 70°F

A) Acrylic Cylinders

E (Young's Modulus) = 450,000 psi

ν (Poisson's ratio) = 0.375

ρ (density) = 0.04297 #in^3

B) Steel Cylinders

E = 30 x 10^6psi

ν = 0.3

ρ = 0.2841 #/in^3

C) Water

K (bulk modulus) = 319,000 psi

ρ = 0.03605 #in^3

Table 2 TESTING PROGRAM

Test Series	Description	Purpose	Boundary Conditions**	Tests
1	I_{ij} in air	Frequencies*	SS	9
2	I_{ij} in air	Frequencies	C	9
3	0_i in air	Frequencies	SS	3
4	0_1 with I_{ij} (water in gap)	Frequencies	0_1(SS):I_{ij}(SS)	9
5	0_2 with I_{ij} (water in gap)	Frequencies	0_2(SS):I_{ij}(SS)	9
6	0_3 with I_{ij} (water in gap)	Frequencies	0_3(SS):I_{ij}(SS)	9
7	0_1 with I_{ij} (water in gap)	Frequencies	0_1(SS):I_{ij}(C)	9
8	0_2 with I_{ij} (water in gap)	Frequencies	0_2(SS):I_{ij}(C)	9
9	0_3 with I_{ij} (water in gap)	Frequencies	0_3(SS):I_{ij}(C)	9
10	0_R with I_{i3} (water in gap)	Damping Ratios	0_R(SS):I_{i3}(SS)	3
11	0_3 with I_R (water in gap)	Damping Ratios	0_3(SS):I_R(SS)	1
12	0_3 with I_{i3} (water in gap)	Damping Ratios	0_3(SS):I_{i3}(SS)	3
13	I_{3i} in a 31" tank of water	Frequencies	SS	3
14	I_{3i} in a 31" tank of water	Frequencies	C	3

* Frequencies denote not only the magnitude of the frequencies but also the associated mode shapes.

** SS = Simple support

C = Clamped

Figure 1 Acrylic and Inner Steel Cylinder

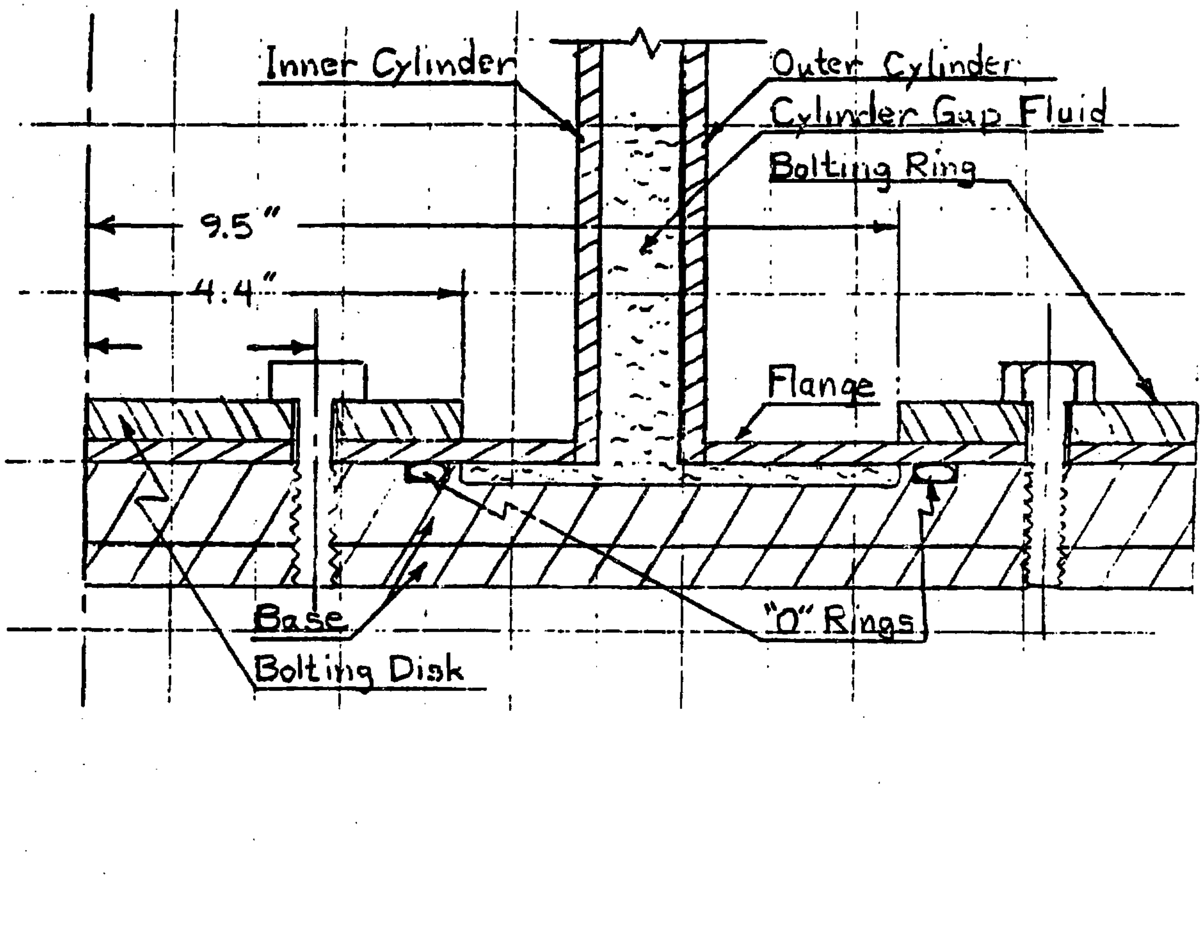


Figure 2 Cylinder Support Configuration

Figure 3 Inner Cylinder Mounted In Test Frame

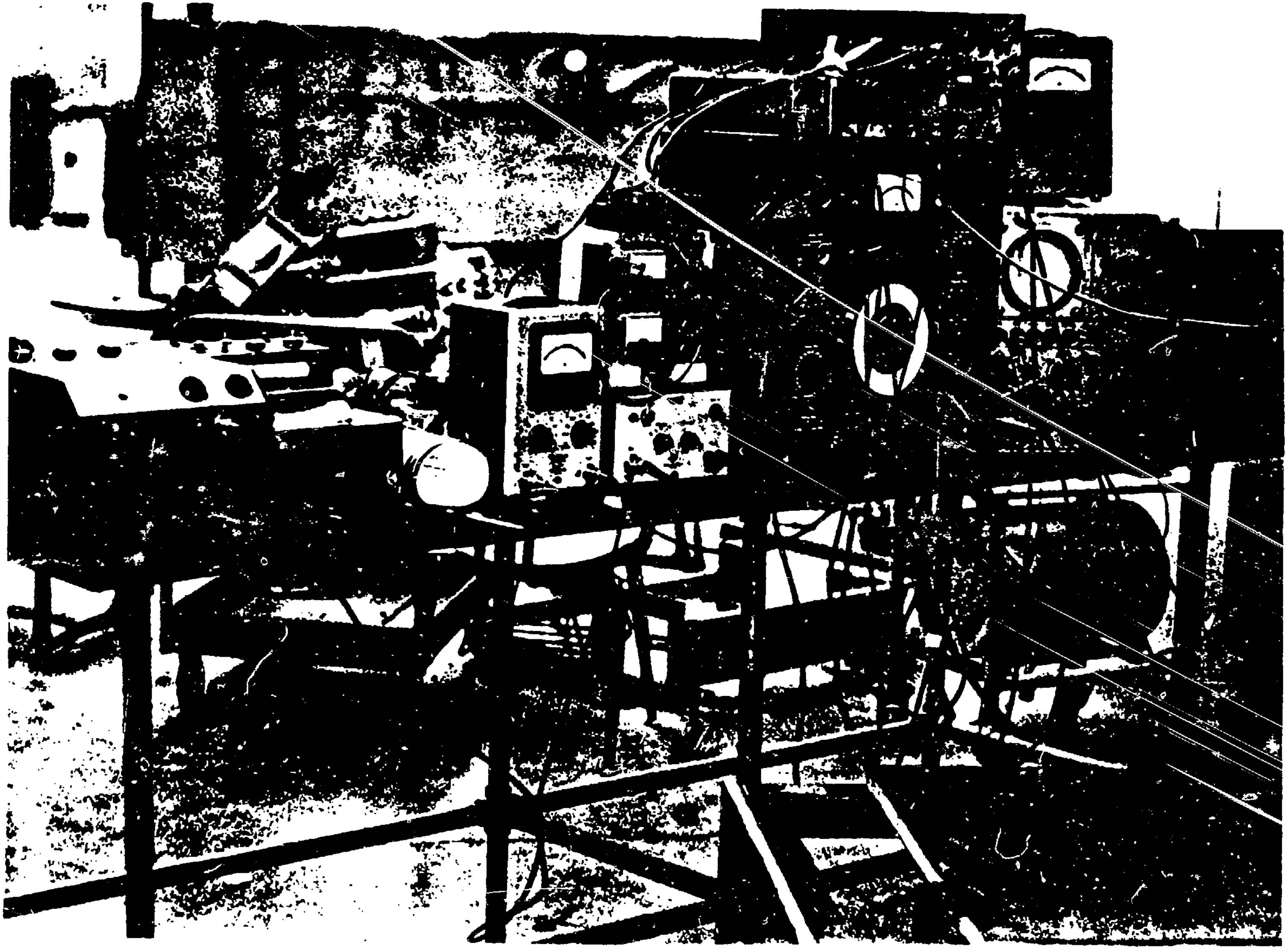
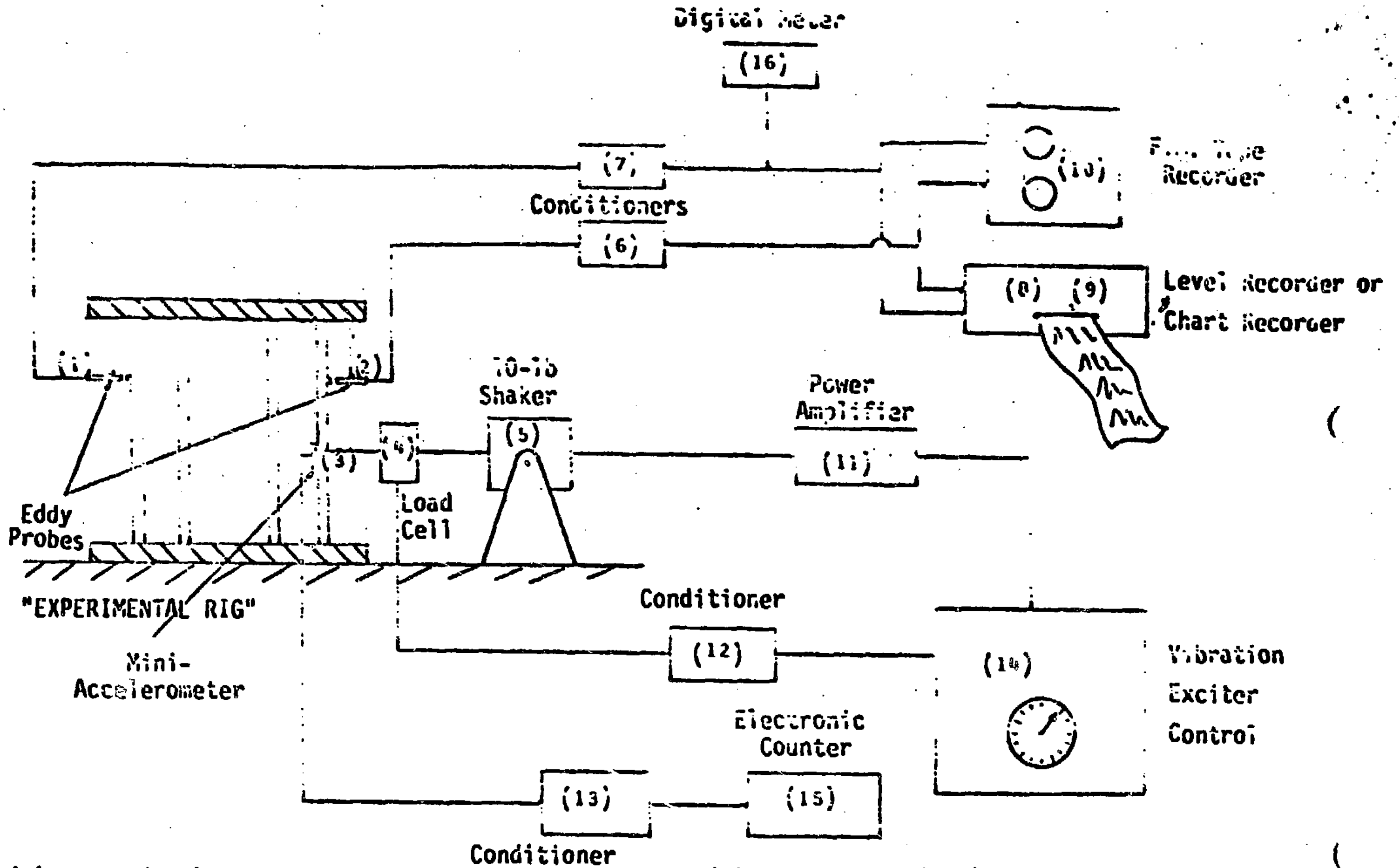


Figure 4 Electronic Recording Equipment for Resonant Frequency Study



(1) DYMAC (M60)
(2) DYMAC (M60)
(3) B & K (8307)
(4) B & K (8200)
(5) B & K (4809)
(6) DYMAC (M600)
(7) DYMAC (M600)
(8) B & K (2305)
(9) GOULD-BRUSH (440)
(10) AMPEX (FR 1300-A)
(11) B & K (2706)
(12) B & K (2622)
(13) B & K (2622)
(14) B & K (1019)
(15) HP (5512-A)
(16) KEITHLEY (160)

Figure 5. Instrumentation for Natural Frequency and Mode Shape Determination

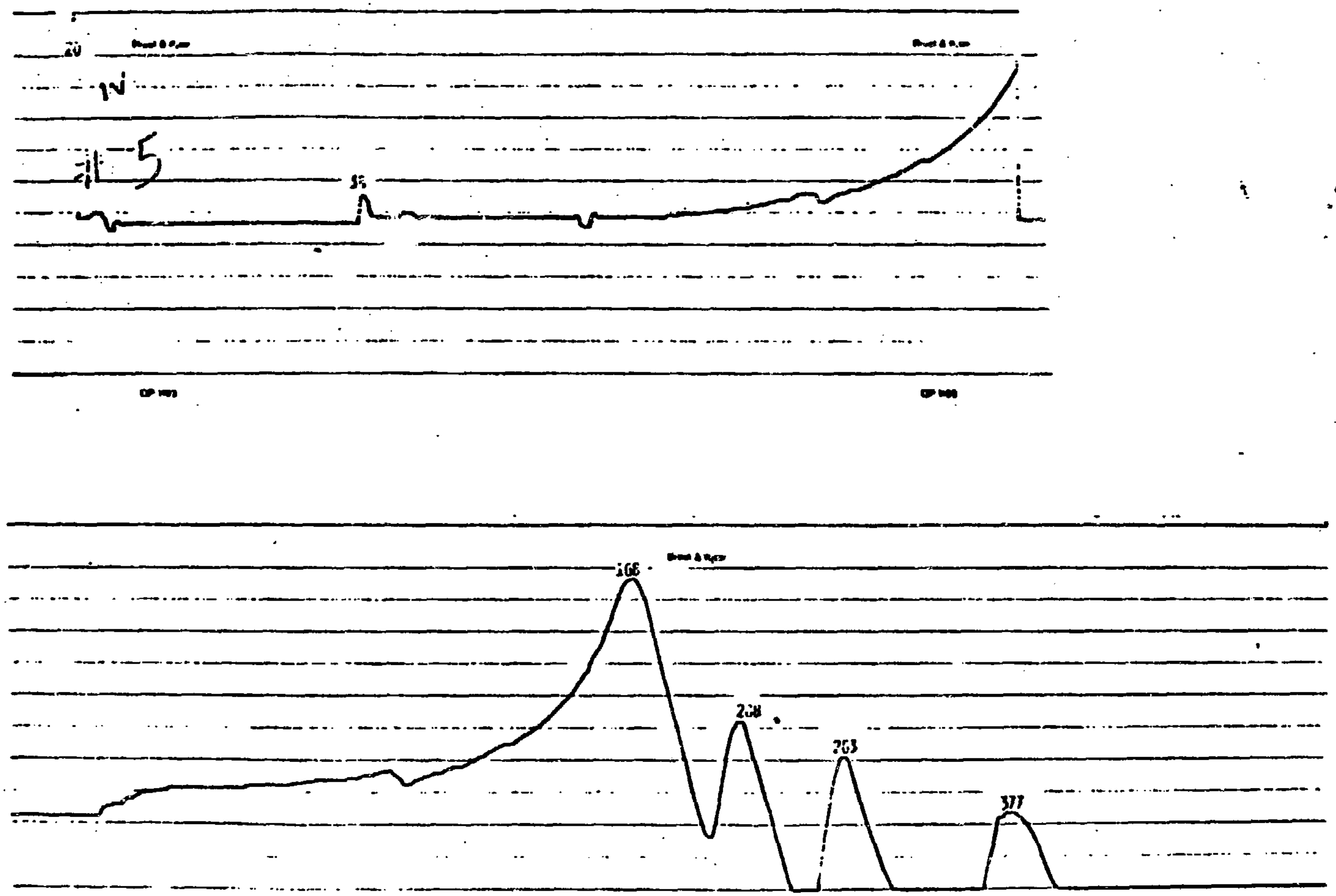
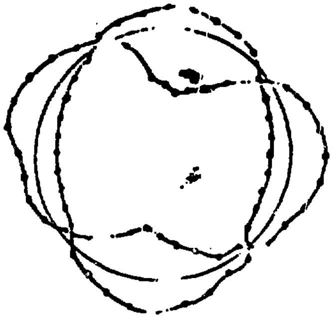
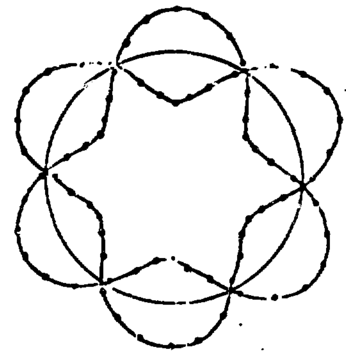


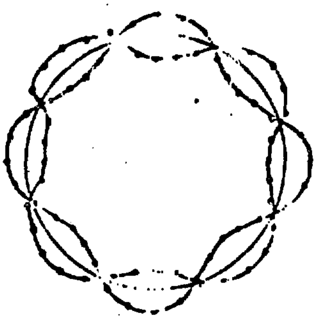
Figure 6. Strip Chart Recording for Simple Supported Inner Cylinder 5 in Air



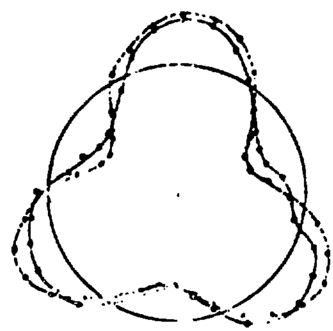
29.2 Hertz



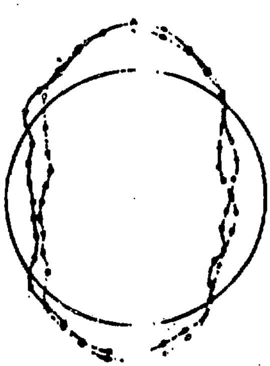
38.5 Hertz



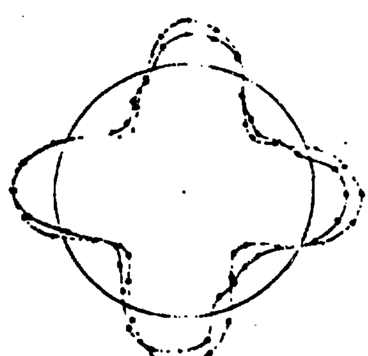
78.0 Hertz



140.4 Hertz

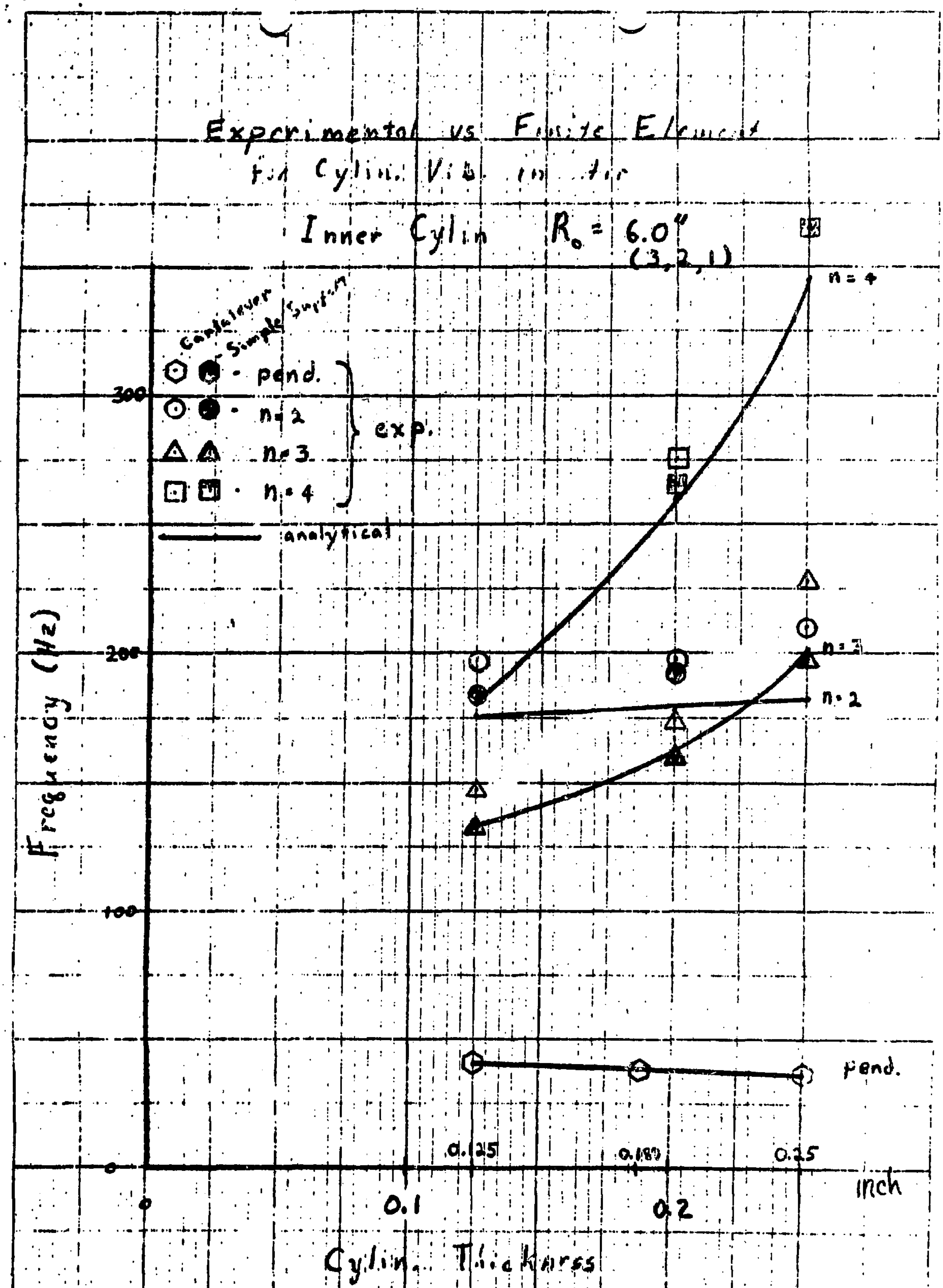


153.3 Hertz



226.8 Hertz

Figure 7 Mode Shapes of Simple Supported Inner Cylinder 1 with Outer Cylinder 10



Exprimental vs Finite Element
for Cylin. Vib. in Air
Inner Cylin R_o = 6.0"
(3,2,1)
Cantilever
Simple System
pend.
n = 2
n = 3
n = 4
exp.
analytical
Frequency (Hz)
300
200
100
0
0.125
0.187
0.25
0
0.1
0.2
inch
n = 4
n = 3
n = 2
pend.
Cylin. Thickness

Figure 9

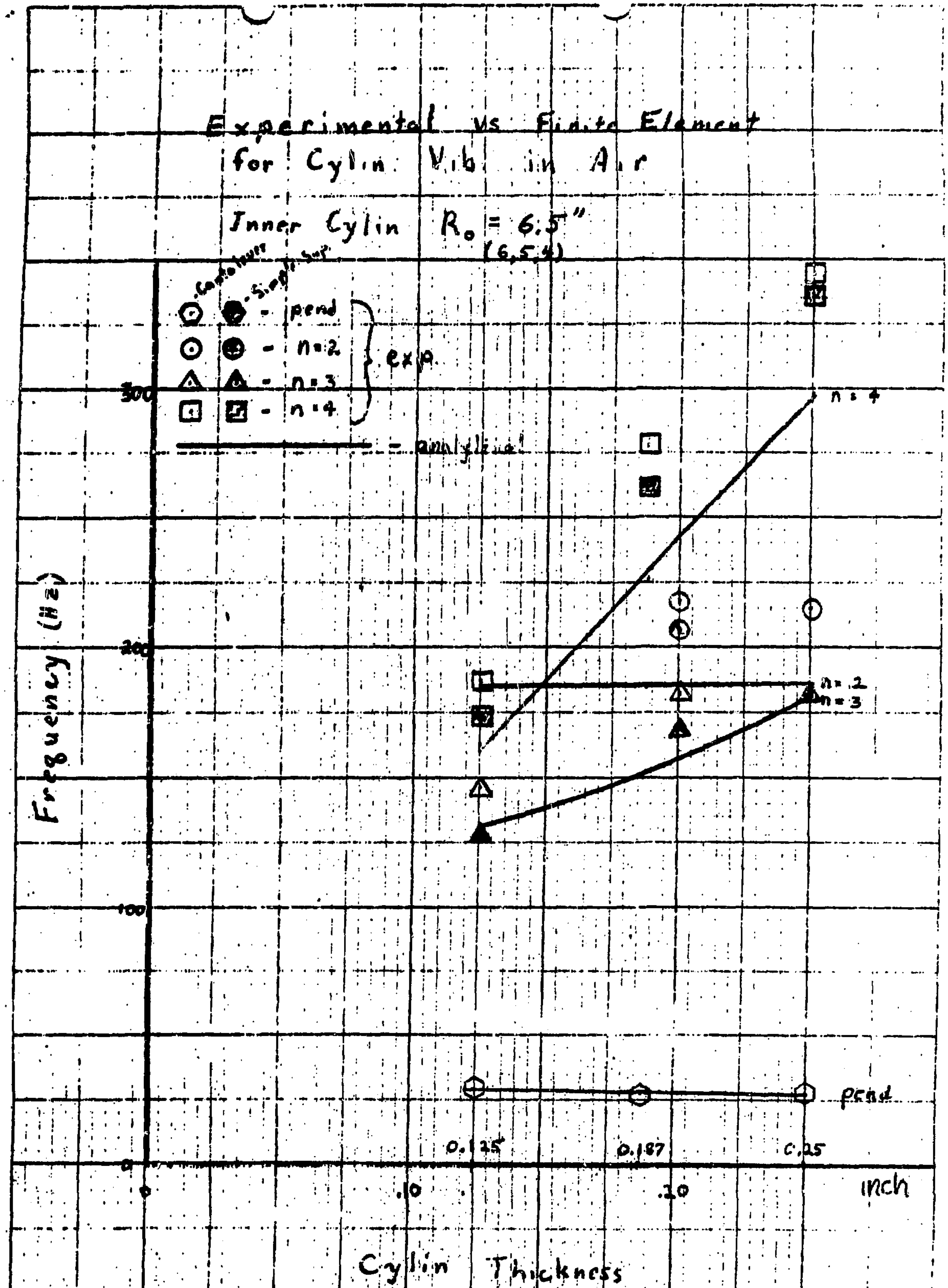
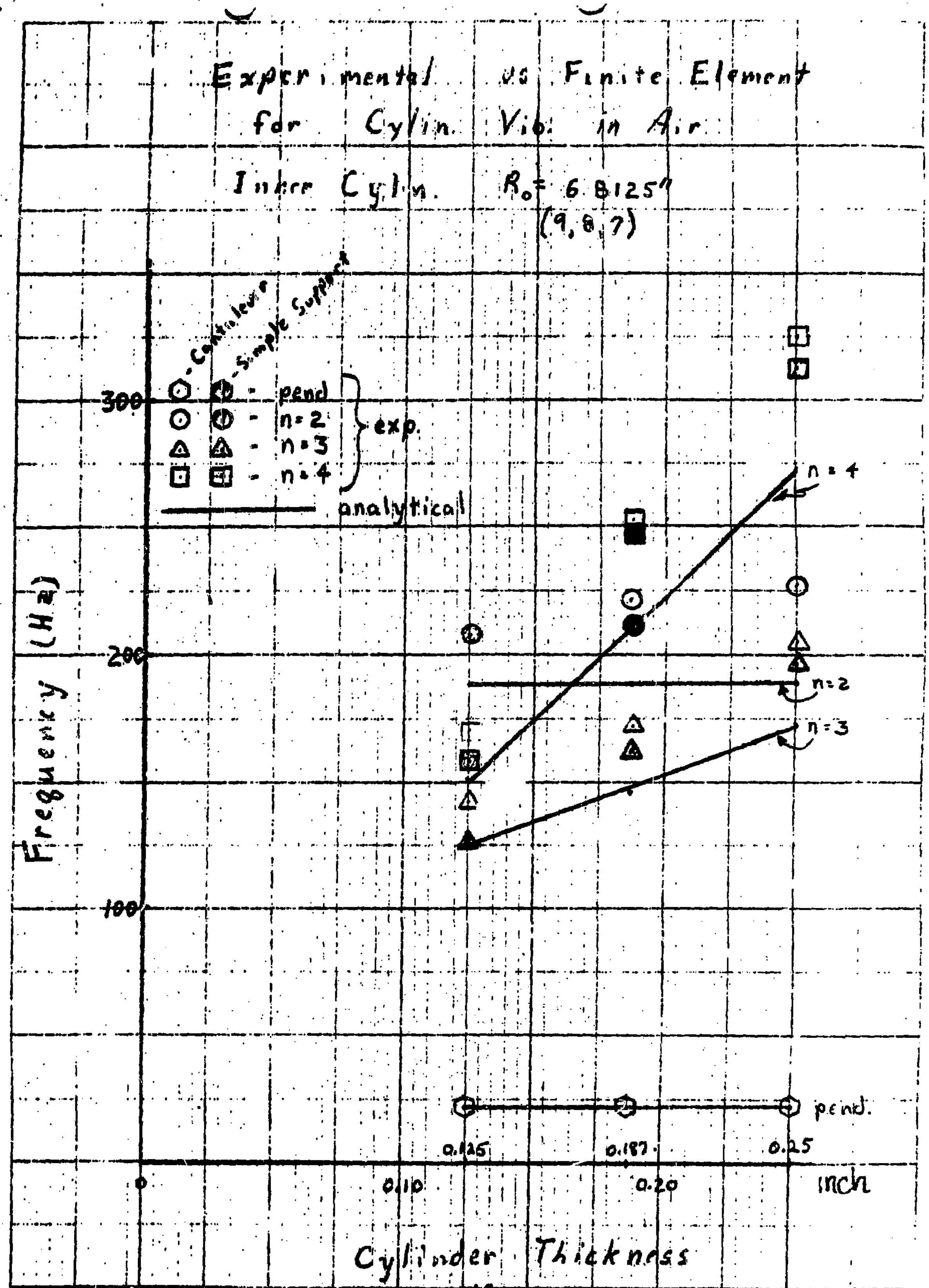
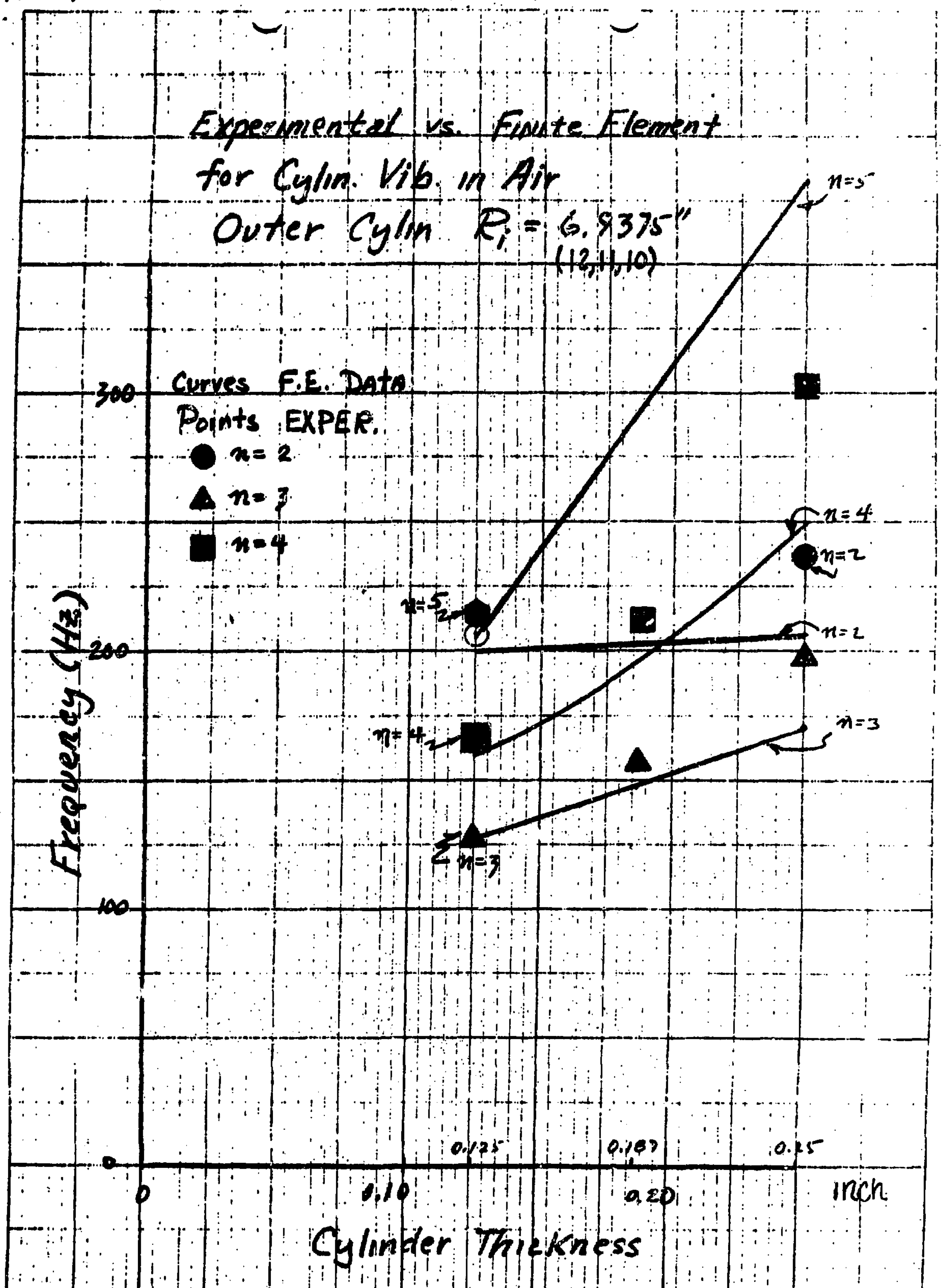

Experimental vs Finite Element
for Cylin Vib in Air
Inner Cylin R_o = 6.5"
(6.5 4)
Cantilever
Simple Sup
pend
n=2
n=3
n=4
exp.
analytical
n=4
n=2
n=3
pend
300
200
100
0
Frequency (Hz)
0.125
0.187
0.25
0
.10
.20
inch
Cylin Thickness

Figure 10



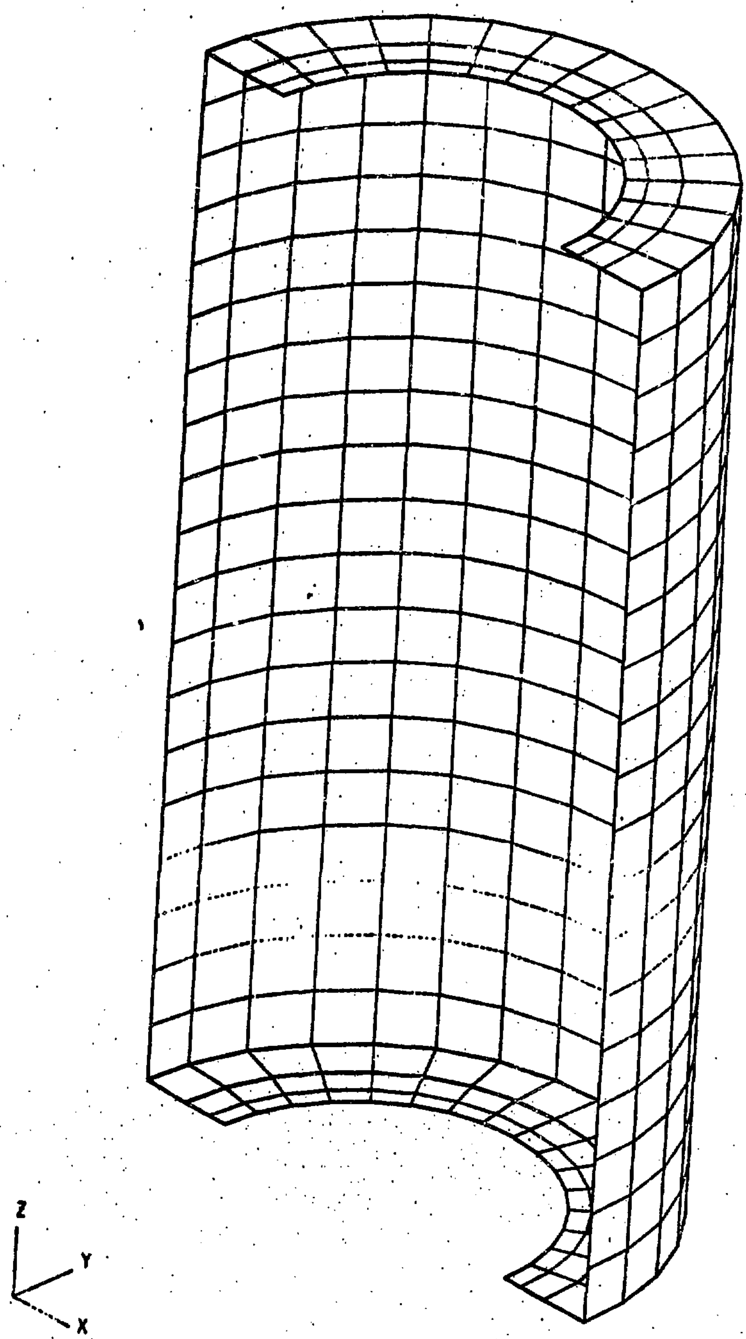
Experimental vs Finite Element
for Cylin. Vib. in Air
Inner Cyln. R_o = 6.8125"
(9, 8, 7)
- Cantilever
- Simple Support
- pend
- n = 2
- n = 3
- n = 4
exp.
analytical
n = 4
n = 2
n = 3
pend.
Frequency (Hz)
300
200
100
0
0.10
0.20
0.125
0.187
0.25
inch
Cylinder Thickness

Figure 11



Experimental vs. Finite Element
for Cylin. Vib. in Air
Outer Cylin R_i = 6.9375"
(12,11,10)
Curves F.E. Data
Points EXPER.
n = 2
n = 3
n = 4
n=5
n=4
n=2
n=2
n=3
n=3
300
200
100
0
0.125
0.187
0.25
0
0.10
0.20
inch
Frequency (Hz)
Cylinder Thickness

Figure 12



[redacted]0

Figure 13

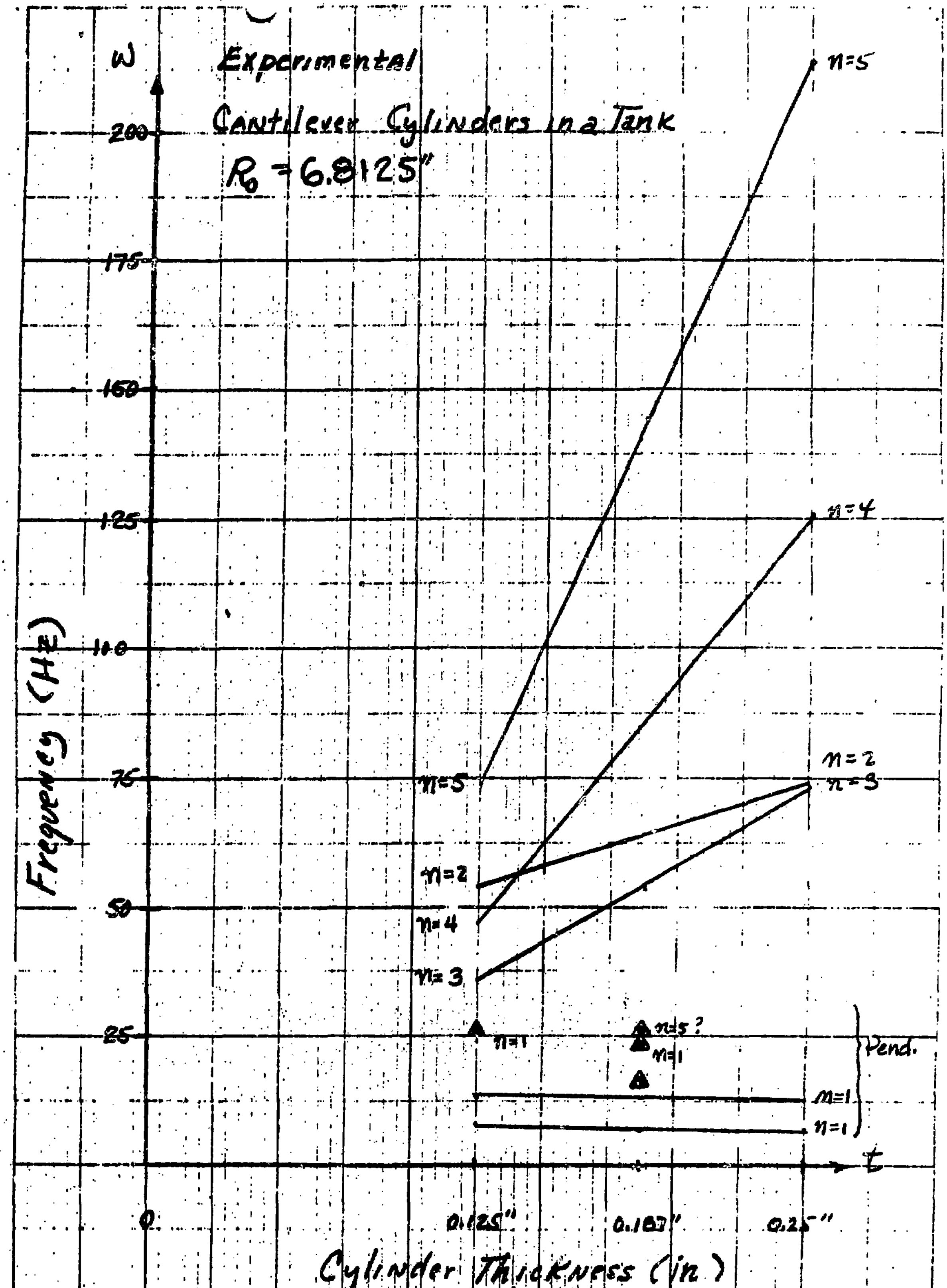
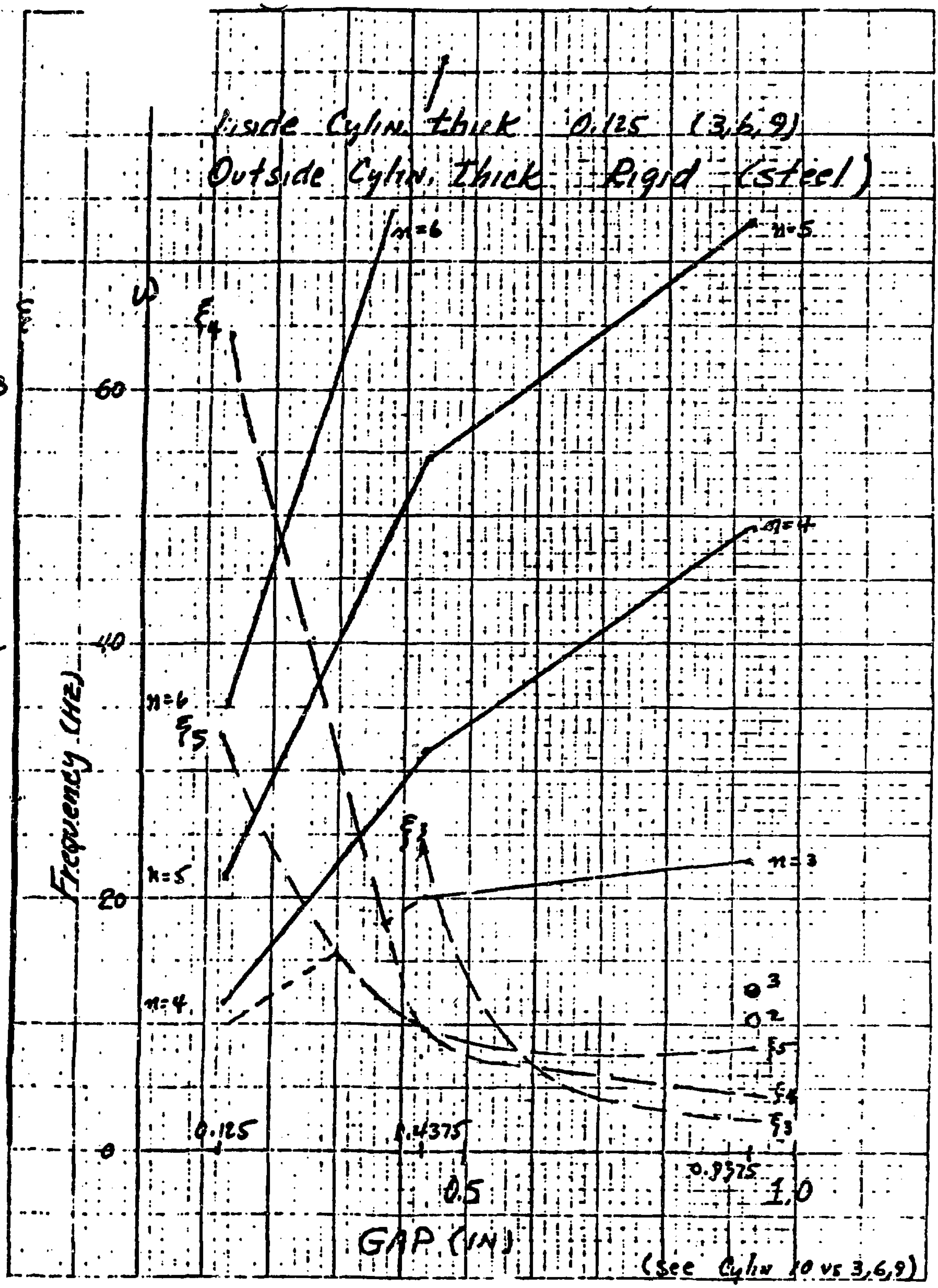

ω
Experimental
Cantilever Cylinders in a Tank
$R_0 = 6.8125''$
200
175
150
125
100
75
50
25
0
Frequency (Hz)
n=5
n=4
n=2
n=3
n=5
n=2
n=4
n=3
n=1
n=5 ?
n=1
Pend.
m=1
n=1
t
0.125"
0.187"
0.25"
Cylinder Thickness (in)

FIGURE 23



Inside Cylin. thick 0.125 (3,6,9)
Outside Cylin. Thick Rigid (steel)
n=6
n=5
n=4
n=3
Damp Ratio
Frequency (HZ)
0.3
0.2
0.1
0
60
40
20
0
0.125
.4375
0.9375
0.5
1.0
GAP (IN)
(See Cylin 10 vs 3,6,9)

FIGURE 22

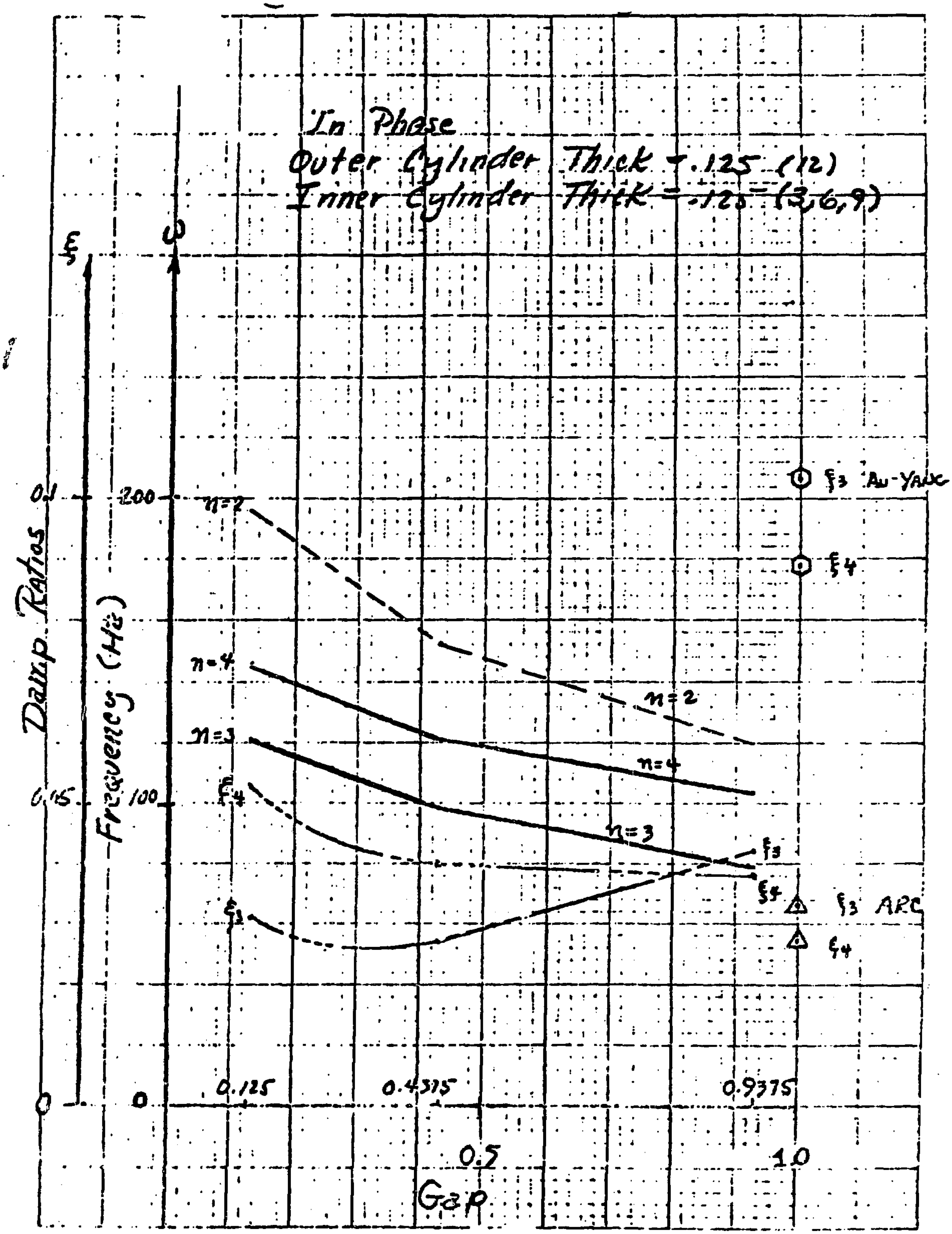
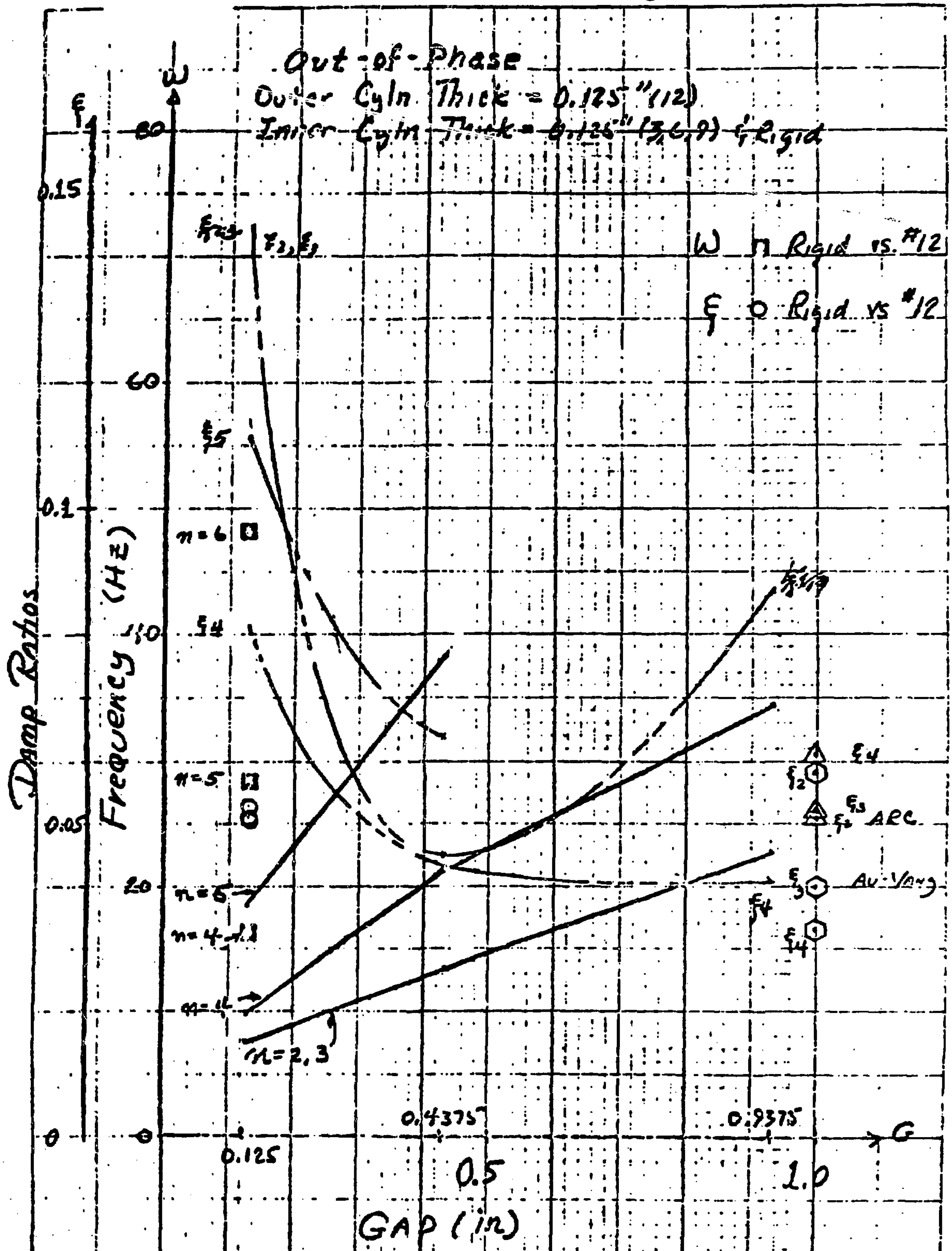
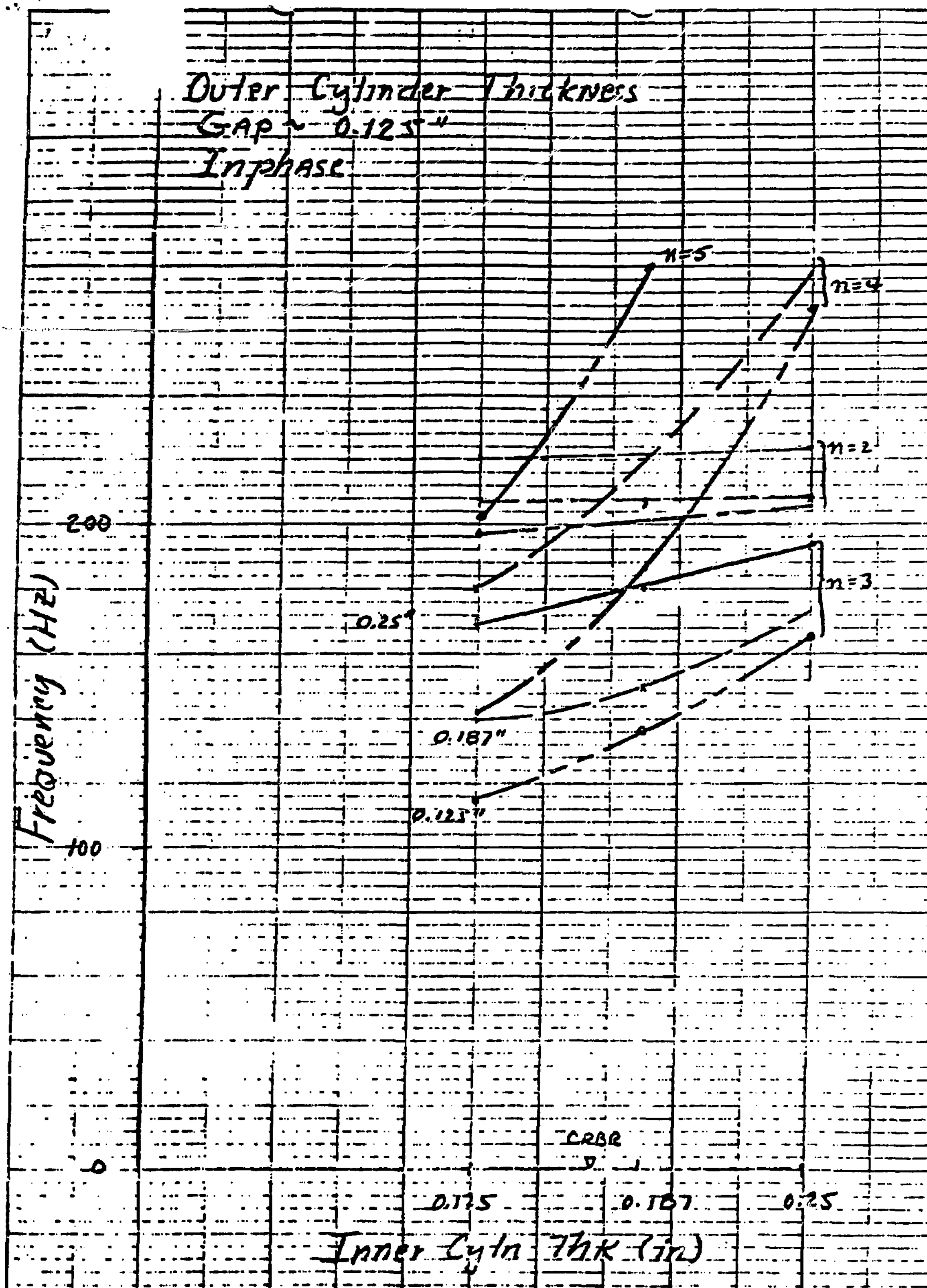

In Phase
Outer Cylinder Thick = .125 (12)
Inner Cylinder Thick = .125 (3,6,9)
ξ
ω
Damp Ratios
Frequency (Hz)
0.1
200
0.05
100
0
0
n=2
n=4
n=3
ξ4
ξ3
n=2
n=4
n=3
ξ3
ξ4
ξ3 AU-YANG
ξ4
ξ3 ARC
ξ4
0.125
0.4375
0.9375
0.5
1.0
Gap

FIGURE 21



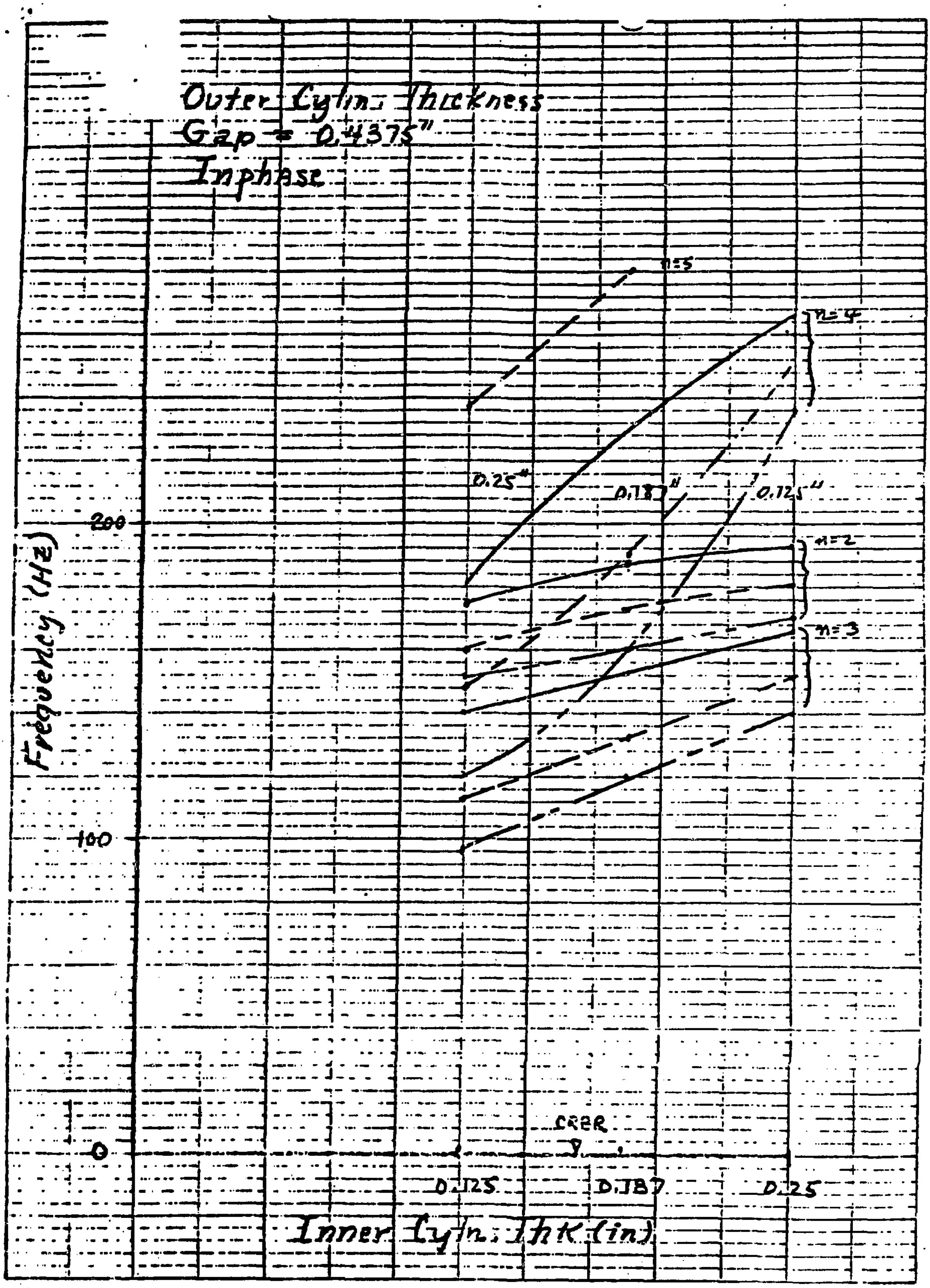
Out-of-Phase
Outer Cyln Thick = 0.125"(12)
Inner Cyln Thick = 0.125" Rigid
ω □ Rigid vs. #12
ξ ○ Rigid vs #12
Damp Ratios
Frequency (HZ)
n=6
n=5
n=4
n=2,3
ARC
Av-Vang
0.125
0.4375
0.9375
0.5
1.0
GAP (in)
G

FIGURE 20



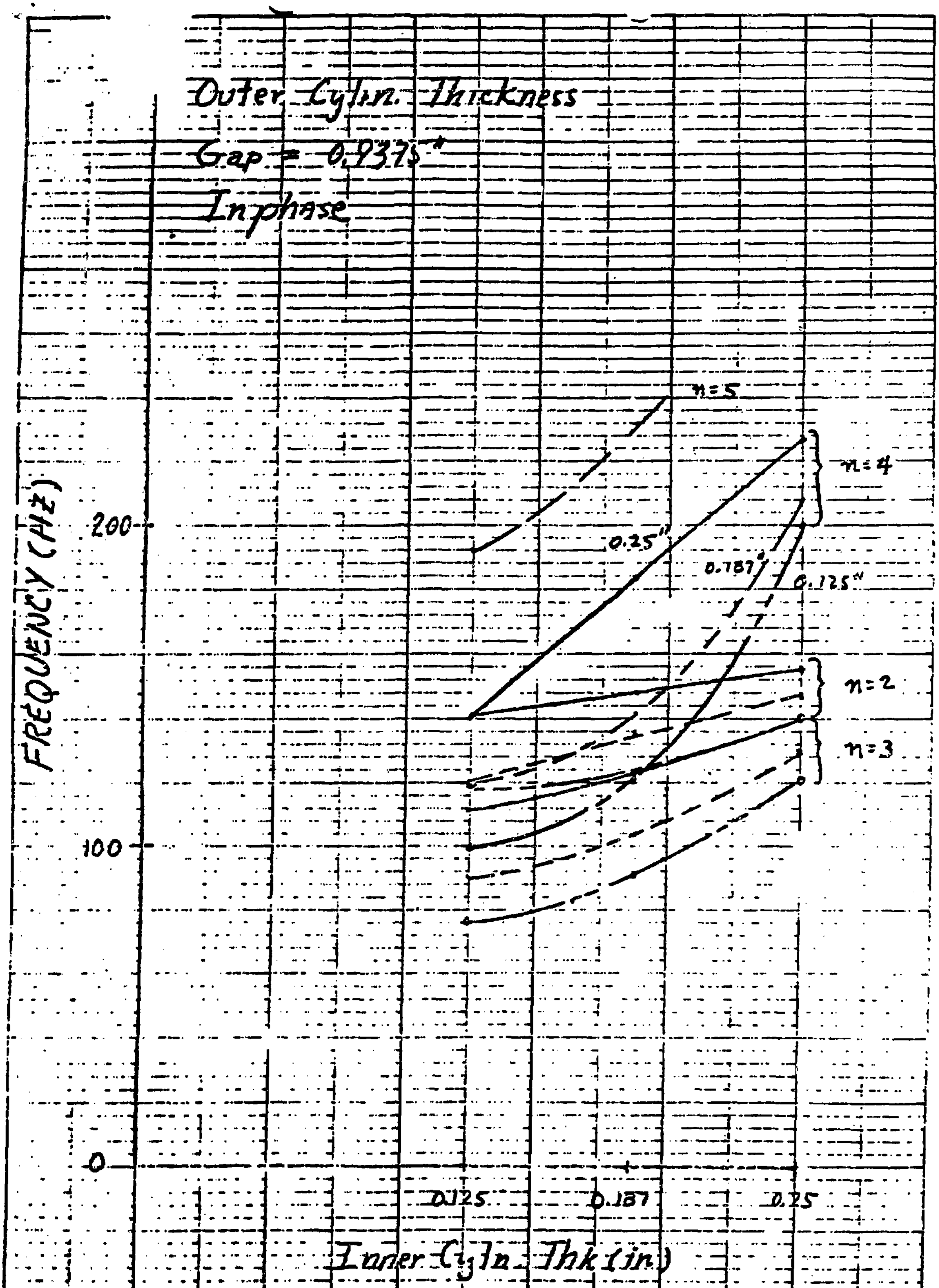
Outer Cylinder Thickness
GAP ~ 0.125"
Inphase
n=5
n=4
n=2
n=3
200
0.25"
0.187"
0.125"
Frequency (Hz)
100
0
CRBR
0.125
0.187
0.25
Inner Cyln Thk (in)

FIGURE 19



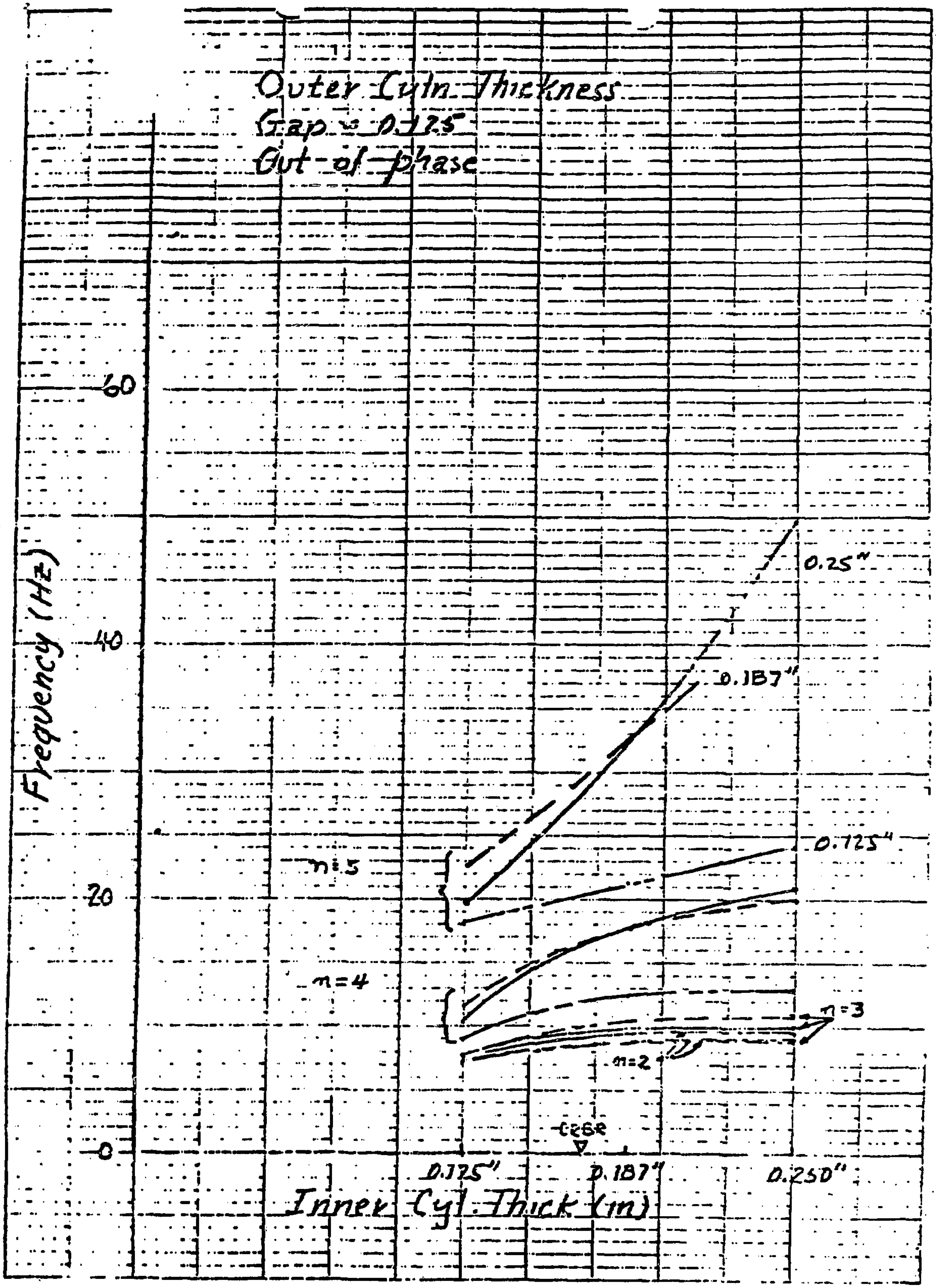
Outer Cylin. Thickness
Gap = 0.4375"
Inphase
Frequency (Hz)
200
100
0
n=5
n=4
n=2
n=3
0.25"
0.187"
0.125"
0.125
0.187
0.25
Inner Cyln. Thk (in)

FIGURE 18



Outer Cylin. Thickness
Gap = 0.9375"
Inphase
FREQUENCY (HZ)
200
100
0
n=5
n=4
n=2
n=3
0.25"
0.187"
0.125"
0.125
0.187
0.25
Inner Cyln. Thk (in)

FIGURE 17



Outer Cyln Thickness
Gap = 0.125
Out-of-phase
Frequency (Hz)
60
40
20
0
0.25"
0.187"
0.125"
n=5
n=4
n=3
n=2
0.125"
0.187"
0.250"
Inner Cyl Thick (in)

FIGURE 16

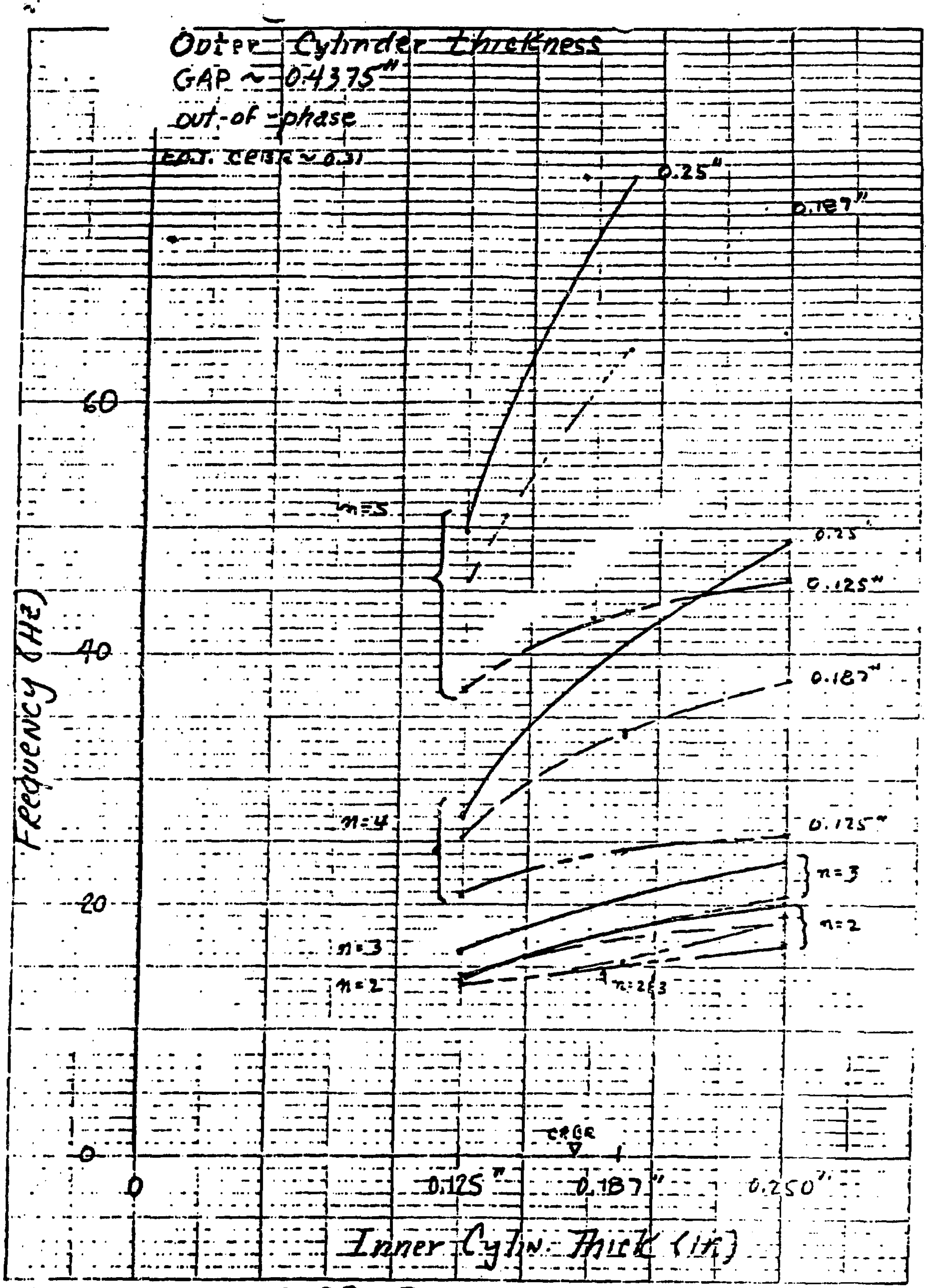
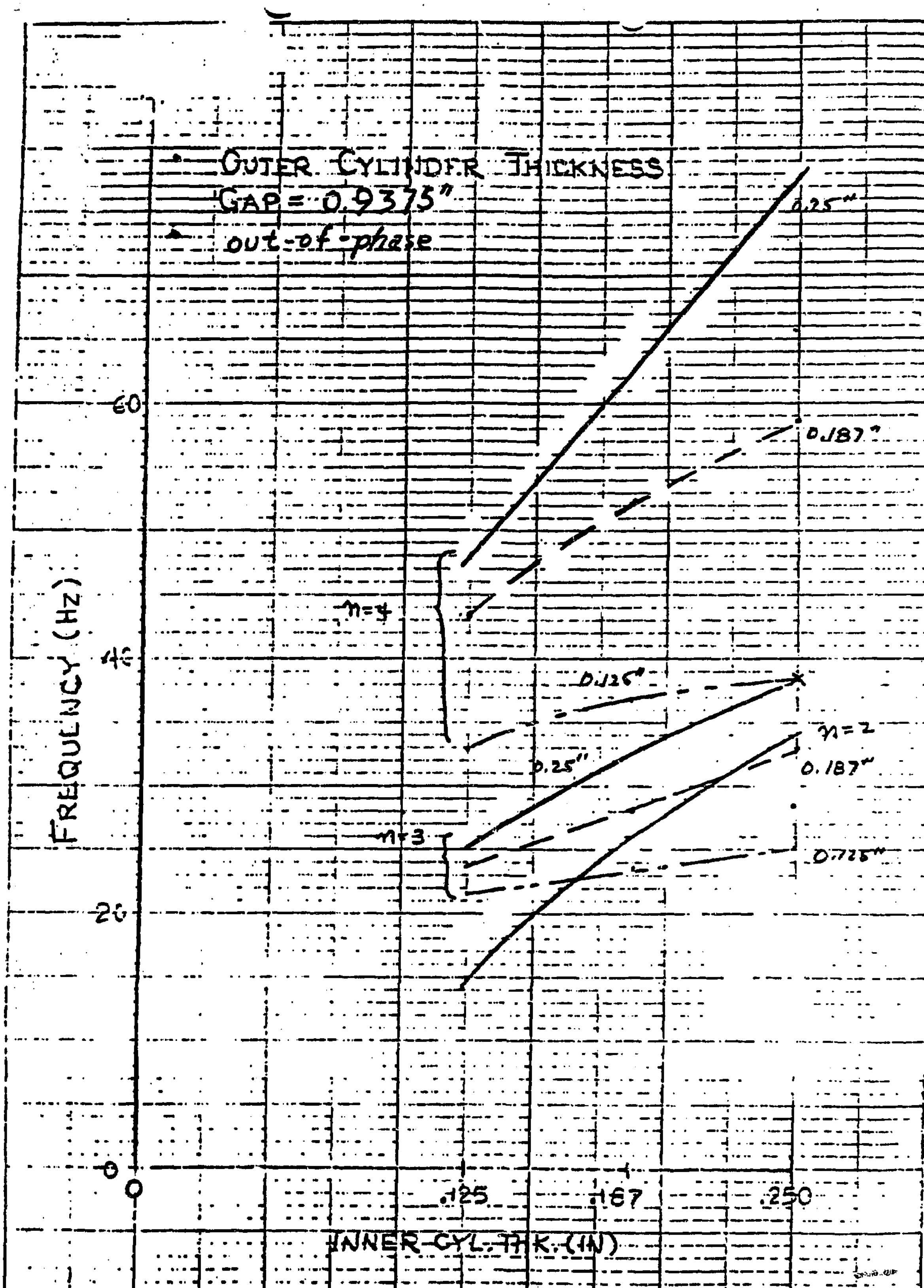

Outer Cylinder Thickness
GAP ~ 0.4375"
out-of-phase
0.25"
0.187"
60
n=5
0.25
0.125"
40
0.187"
n=4
0.125"
n=3
20
n=3
n=2
n=2
n=2&3
0
0
0.125"
0.187"
0.250"
Frequency (Hz)
Inner Cylin. Thick (in)

FIGURE 15



OUTER CYLINDER THICKNESS
GAP = 0.9375"
out-of-phase
FREQUENCY (Hz)
60
40
20
0
0
.125
.187
.250
INNER CYL. THK. (IN)
0.25"
0.187"
n=4
0.125"
n=2
0.25"
0.187"
n=3
0.125"

FIGURE 14