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DEVELOPMENT OF SMALL DIAMETER HYDRAULIC COILED TUBING TECHNOLOGY

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JUNE 1981

Technical Report NADC-79217-60
Final Report for Period June 1980 through June 1981

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NAVAL AIR DEVELOPMENT CENTER
Aircraft and Crew Systems Technology Directorate
Warminster, Pennsylvania 18974

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1. REPORT NUMBER: NADC-79217-60
2. GOVT ACCESSION NO.: AD-A
3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle): DEVELOPMENT OF SMALL DIAMETER HYDRAULIC COILED TUBING TECHNOLOGY
5. TYPE OF REPORT & PERIOD COVERED: Final Technical Report, June 1980 thru June 1981
6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s): K. E. Whitfill, R. B. Olsen
8. CONTRACT OR GRANT NUMBER(s): N62269-80-C-0205
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Vought Corporation, P. O. Box 225907, Dallas, Texas 75265
10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS: Naval Air Development Center (Code 30211), Aircraft and Crew Systems Directorate, Warminster, Pennsylvania 18974
12. REPORT DATE: June 1981
13. NUMBER OF PAGES: 126
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)
15. SECURITY CLASS. (of this report): UNCLASSIFIED
15a. DECLASSIFICATION DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report): Approved for public release; distribution unlimited
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)
18. SUPPLEMENTARY NOTES
19. KEY WORDS (Continue on reverse side if necessary and identify by block number):
    - coiled tubing
    - flexible plumbing
    - titanium tubing
    - hydraulic system plumbing
20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The objective of this program was to develop design equations for coiled tubing as a helical extension or compression spring which has the capability of acting as a flexible fluid connector in aircraft hydraulic systems. The purpose of the coiled tube is to replace hose, swivels, or extension units on moving hydraulic actuator barrels which have linear

or limited angle rotary motion.

The use of 3Al-2.5V titanium cold worked stress relieved tubing is recommended for coiled tubing applications. Also recommendations are made for wall thickness for three titanium tube alloys for operating pressures of 1500, 3000, and 8000 psig. Design nomographs are provided to allow use in establishing coil tube design parameters such as number of coils and force at maximum deflection.

Four coiled tube assemblies in two configurations were subjected to a 100,000 cycle endurance test at 100% deflection of the coils. There were no failures. A vibration survey was made to determine natural frequencies and transmissibility with a random input vibration spectrum.

PREFACE

From the Lightweight Hydraulic System program it was determined that system reliability would be improved if flexible plumbing such as extension units and swivels which contain elastomeric seals were replaced. It was recognized that coiled tubing was a desirable replacement but there were limitations with use of ARP 584 design information. A choice of mean coil diameters is not offered; the material recommended is CRES with a high torsional modulus, and the minimum tube diameter is 1/4 inch. Perhaps the most severe handicap is that the configurations analyzed are most applicable to limited angle rotary motion with no provisions for linear motion which occurs with moving barrel power control actuators.

From these needs, the goals of this program were established as follows: (1) to develop new configurations which were adaptable to installation envelope, (2) to develop design equations which allowed choice of coil diameter and tube diameter, (3) to determine the feasibility of the designs in tests.

These goals have been met. The helical and tri-coil configurations presented in this report will work on moving barrel actuators which have either linear or limited angle motion about a pivot point.

This report describes the analysis and testing conducted on the two new configurations. Technology for the design and fabrication of coiled tubes has been extended by inclusion of smaller tube diameters, choice of coil diameters, high strength, low modulus titanium alloy tube materials and recommendations for tube wall thickness in applications up to 8000 psig operating pressure. Derivation of the design equations was based in part on work disclosed in report WADC TR55-121 (1955).

The program was sponsored by Naval Air Systems Command and administered by the Naval Air Development Center, with Mr. Douglas Bagwell, Project Engineer (30211).

Mr. K. E. Whitfill, Vought Corporation was principal investigator. The final report was written by Mr. Olsen. Appreciation is expressed to J. A. Bird, W. L. Breaux, W. V. Brewer, G. K. Fling and other personnel who participated in this effort.

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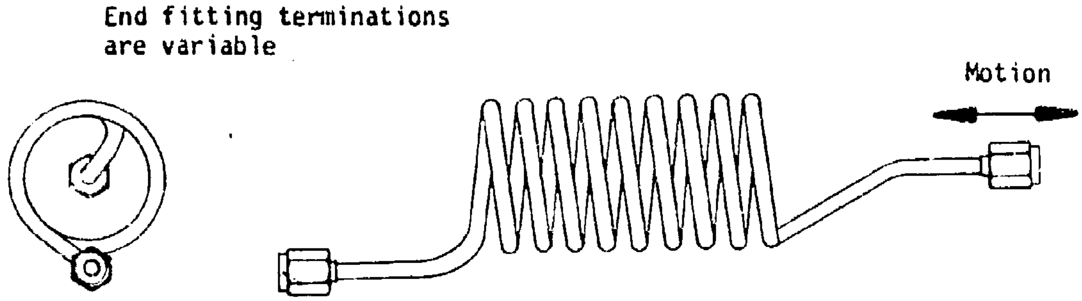
1. INTRODUCTION

For many years the only devices available to transmit hydraulic power across moving joints were hose, swivels, extension units, and combinations of these devices. While great strides in reliability and compactness have been made in these devices, there remained a need for a device that had no moving joints, required no elastomeric seals, was compact, and low in weight. In 1955 and 1957 significant work was done to define the stress analysis and design criteria for straight tubing loaded in torsion and coiled tubing loaded in bending. The tubing material was corrosion resistant steel. Many successful plumbing installations have been based upon these efforts and other pioneering work conducted in the early 1940's.

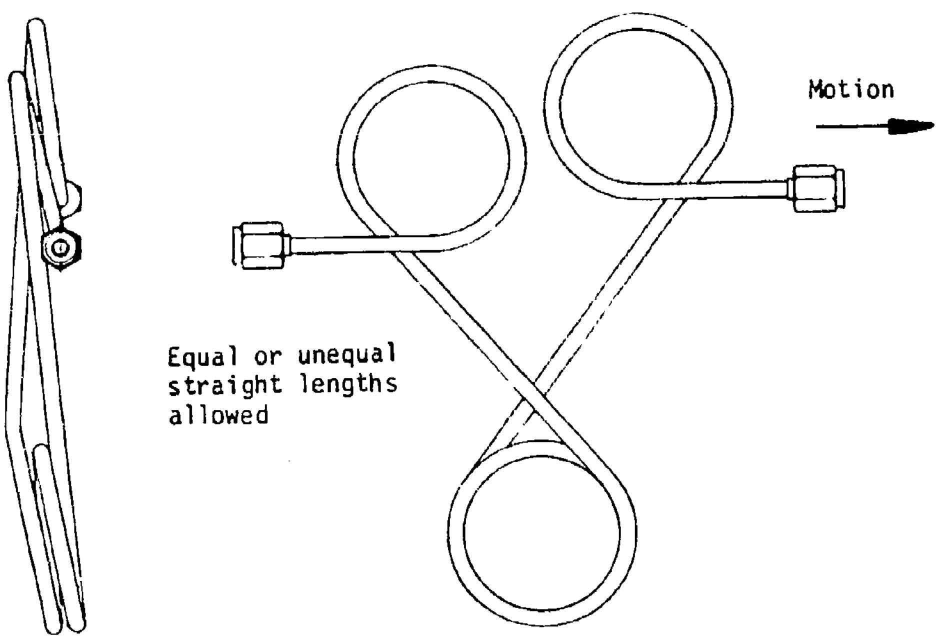
The Lightweight Hydraulic System program determined that system reliability would be improved if extension units and swivels, both containing elastomeric seals, were replaced. The extension units provide a flexible fluid connector for moving barrel actuators with linear motion. Swivels provide a flexible fluid connector for moving barrel actuators with motion about a pivot. The coiled tube configurations available from the 1955 work and documented in ARP 584 are applicable only to motion about a pivot. The material is 18-8 CRES steel which has a relatively high torsional modulus. Only a single coil diameter is recommended for each tube size. The minimum tube size considered was 1/4-inch.

This program extends coiled tube technology by introduction of new configurations useable with linear or pivotal motion, use of 3/16-inch diameter tube, recommendations for three titanium alloy tube materials, and design equations allowing choice of coil mean diameter. Also, safe tube wall thickness is recommended for system pressure up to 8000 psig.

See Figure 1 for identification of the basic configurations analyzed in this report.



Helical Configuration



Tri-Coil Configuration

Figure 1. Helical and Tri-Coil Configurations
Evaluated in the Program

2. MATERIAL SELECTION RATIONALE

2.1 Rationale for the Helical Configuration. The selection of materials for coiled tubing is controlled by parameters that will maximize reliability and minimize weight. Reliability is maximized in a mechanical system when stress is minimized. This is accomplished by using material that will deflect with a minimum of stress. In a coiled tubing spring the load is carried by torsional shear stress and the Modulus of Rigidity, "G", is the material parameter that controls this stress. Weight is minimized in a hydraulic system when the tubing wall thickness is minimized. This is accomplished by using a high strength, light weight material. The ultimate tensile strength, F_{tu}, is the parameter that measures this strength.

To maximize reliability and minimize weight we want to minimize G and maximize Ftu. The material that has the highest ratio of Ftu/G is the material that accomplishes this and is the one that is the most desirable for use in a coiled tubing spring.

2.2 Rationale for the Tri-Coil Configuration. For the tri-coil design the load is carried as a bending stress. Again, a material is desired that will deflect with a minimum of stress. The material parameter that controls bending stress is "E", the modulus of elasticity. Weight in a hydraulic system is reduced when the tubing wall thickness is minimized. This is accomplished by using a high strength, light weight material. The ultimate tensile strength F_{tu} is the material parameter that limits this strength.

To maximize reliability and minimize weight of the tri-coil design, "E" should be a minimum and F_{tu} should be a maximum. The material that has the highest ratio of F_{tu}/E is the most desirable for use in the tri-coil configuration.

2.3 Material Properties. Table 1 shows the material properties for a number of metals successfully used in hydraulic systems. The ratio Ftu/G is shown at the right of the table. Note that the three titanium alloys selected for this program show from 3:1 to 2:1 improvement in the ratio Ftu/G when compared to CRES 321 (1/8 H) x or CRES 304 (1/8 H).

Table 2 gives the recommended wall thickness for three titanium tube alloys for the helical and the tri-coil configuration.

For the helical configuration the allowable tensile stress was as follows:

Ti 6Al-4V	65000 psi
Ti 3Al-2.5V CWSR	62500
Ti 3Al-2.5V Annealed	45000

These values represent the maximum allowable when all stresses are considered using the distortion energy theory of failure and a fatigue requirement of 100,000 cycles of full stroke reversed stress.

For the tri-coil configuration the allowable tensile stress was as follows:

Ti 6Al-4V	92,300
Ti 3Al-2.5V CWSR	88,750
Ti 3Al-2.5V Annealed	63,900

These values represent the maximum allowable when all stresses are considered using the distortion energy theory of failure and a fatigue requirement of 100,000 cycles full stroke non-reversing stress.

TABLE 1. COILED TUBING MATERIAL PROPERTIES

MATERIAL	E $\times 10^6$ psi	G $\times 10^6$ psi	F_{tu} psi	F_{su} psi	ρ lb/in^3	F_{tu}/G	F_{tu}/E
321 (1/8 H)	28.0	12.5	95000	50000[c]	.286	.0076	.00339
304L (1/8 H)	28.0	11.5	105000	75000	.286	.0091	.00375
6061T6[a]	9.9	3.8	42000	27000	.098	.011	.00424
Ti Commercial[a] Pure Grade B	15.5	6.5	80000	42000	.163	.0123	.00516
21-6-9	28.0	11.0[c]	142000	85200[c]	.290	.0129	.00507
Ti 3AL-2.5V[b] * Annealed	15.0	5.8[c]	90000	58153[c]	.162	.0155	.00600
Ti 6AL-4V[a] *	16.0	6.2	130000	84000	.160	.0209	.00815
Ti 3AL-2.5V[b] * CWSR	15.0	5.8[c]	125000	80769[c]	.162	.0215	.00833

a – MIL-HBK-5
b – Vendor Data
c – Derived Approximate Value
* – Tubing selected for coiled tubing program

TABLE 2. RECOMMENDED WALL THICKNESS FOR COILED TUBING

	TYPE OF SPRING	HELICAL CONFIGURATION			TRI-COIL CONFIGURATION		
MATERIAL	PRESSURE LEVEL	1500 PSI	3000 PSI	8000 PSI	1500 PSI	3000 PSI	8000 PSI
TI 3AL-2.5V Annealed	Wall Thickness in inches	.020	.025	.035	.020	.025	.035
TI 3AL-2.5V CWSR	Wall Thickness in inches	.020	.020	.028	.020	.020	.028
TI 6AL-4V	Wall Thickness in inches	.020	.020	.028	.020	.020	.028

3. HELICAL CONFIGURATION STRESS ANALYSIS

3.1 Definition of Stress. The helical configuration (helical coil spring) had three different applied stresses; 1, the torsional stress caused by mechanical deflection; 2, the ovalization stress caused by the slightly flattened tube trying to regain a round cross section under pressure and; 3, the stress caused by pressurizing a curved tube. The torsional stress was determined by modifying the basic spring equation. The ovalization stress and pressurized curved tube stress were determined by using the techniques given in Reference 1. The stresses obtained from these methods were combined into an equivalent stress using the Distortion Energy Theory of Failure.

All symbols are defined in Appendix A.

3.2 Derivation of Stress Equations

The equation for torsional stress, S_{xz}, for a helical spring made from tubing was derived as follows:

$$S_{xz} = \frac{TX}{J}$$

where:

$$X = \frac{d_o}{2}, \quad J = \pi\,(d_o^4 - d_i^4)/32, \quad T = F \times D_m/2$$

$$S_{xz} = \left[\frac{F\,D_m}{2} \times \frac{d_o}{2}\right] \Big/ \left[\pi\,(d_o^4 - d_i^4)/32\right]$$

See Figure 2 for location of stresses determined by equations 1, 2, and 3.

$$S_{xz} = \frac{8\,FD_m\,d_o}{\pi(d_o^4 - d_i^4)} \tag{1}$$

The torsional stress on the inside (D_m-d_o) of the coil is:

$$S_{xz} = \frac{8FD_m\,d_o}{\pi(d_o^4 - d_i^4)} \times K_i$$

where:

$$K_i = \frac{4c - 1}{4c - 4} + \frac{.615}{c} \text{ and } c = \frac{D_m}{d_o} \tag{2}$$

The torsional stress on the outside ($D_m + d_o$) of the coil is:

$$S_{xz} = \frac{8FD_m d_o}{\pi(d_o^4 - d_i^4)} \times K_o$$

where:

$$K_o = \frac{4c + 1}{4c - 1} - \frac{.615}{c}$$

Since $1/K_i$ is approximately equal to K_o when c varies from c = 4 to c = 50 (2.3 percent error max at c = 4) $1/K_i$ was used in place of K_o when the torsional stress on the outside of the tube spring was calculated, to simplify the program.

So the torsional stress on the outside of the tube is:

$$S_{xz} = \frac{8FD_m d_o}{\pi(d_o^4 - d_i^4)} \times \frac{1}{K_i} \qquad (3)$$

The stroke equation for a helical spring made from tubing was derived as follows:

$$\theta = \frac{Tl}{GJ} \qquad (3A) \qquad \text{but } l = \pi D_m,\ T = \frac{F D_m}{2}$$

From Figure 2, θ can also be expressed as:

$$\theta = \frac{2A_1}{D_m}$$

Substituting terms identified into equation 3A:

$$\frac{2A_1}{D_m} = \frac{FD_m}{2} \times \pi D_m \times \frac{1}{G} \times \frac{32}{\pi(d_o^4 - d_i^4)}$$

$$\frac{2A_1}{D_m} = \frac{16F D_m^2}{G(d_o^4 - d_i^4)}$$

$$A_1 = \frac{8F D_m^3}{G(d_o^4 - d_i^4)} \qquad \text{(Stroke per Coil)}$$

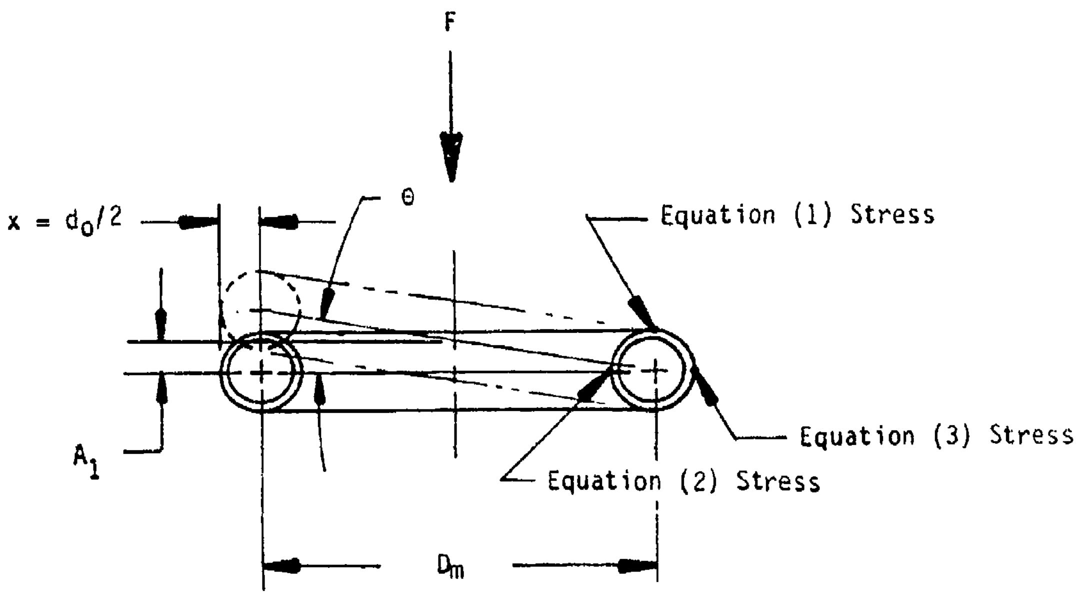


Figure 2. Location of Torsional Stresses on Helical Configuration

$$A_T = A_1 N = \frac{8F\ D_m^3\ N}{G(d_o^4 - d_i^4)}$$

(Total Stroke) The spring pitch must be sufficient to allow this stroke and still allow room for clamps.

Rearranging to solve for force, F:

$$F = \frac{A_T G(d_o^4 - d_i^4)}{8\ D_m^3\ N} \quad \text{(Force)} \qquad (4)$$

Spec MS33611 defines the percent ovalization to be:

$$\phi(^o/o) = \frac{d_o(\text{max}) - d_o(\text{min})}{d_o\ (\text{nominal})} \times 100$$

This equation may be rewritten in specific terms as follows:

$$\phi = \frac{d_a - d_b}{d_o}$$

Since 5 percent ovalization is the maximum allowed, the above equation can be written as follows to obtain the change in diameter at the neutral axis of the tube wall. (See Figure 3)

$$\phi = \frac{2(r + t/2 + \Delta a) - 2(r + t/2 - \Delta b)}{2(r + t/2)}$$

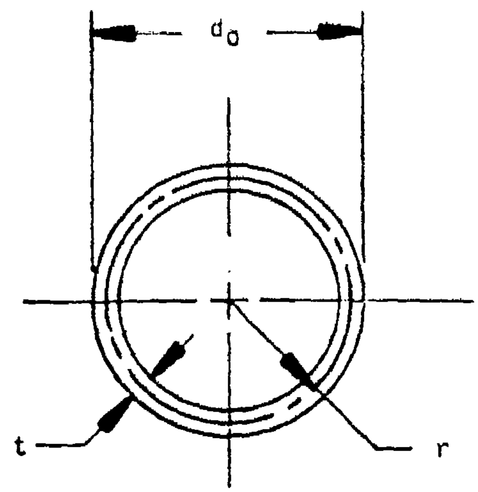
Where:

$$d_a = 2(r + t/2 + \Delta a)$$

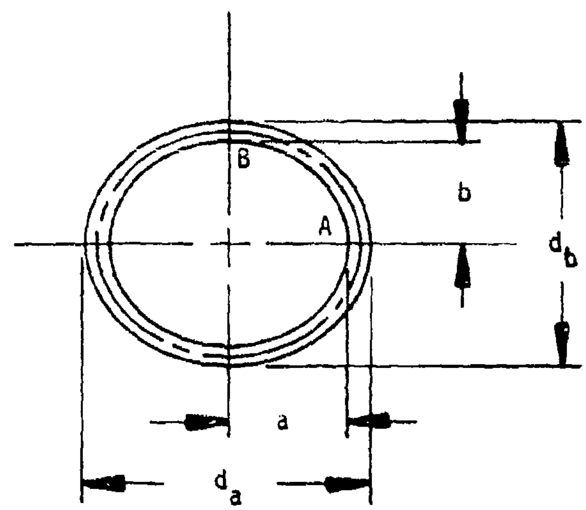
$$d_b = 2(r + t/2 - \Delta b)$$

$$d_o = 2(r + t/2)$$

$$\phi = \frac{2(r + t + \Delta a) - 2(r + t - \Delta b)}{2(r + t)}$$



Before Forming



After Forming

Figure 3. Location of Tube Ovality Dimensions

Simplifying:

$$\Delta a + \Delta b = \phi\ (r + t)$$

The ovalization stress was calculated using the technique of Reference 1. In Reference 1 the measured ovalization of a 3/8 x .020 wall tube showed that the change in max diameter to min diameter was at a ratio of 2.3656/1 as pressure increased. ($\Delta b = 2.3656 \Delta a$). This will not be exact for the 3/16 thick wall tube, but it will be conservative.

$$\text{Since } \Delta b = 2.3656\ \Delta a$$

$$3.3656\ \Delta a = (\phi)\ (r + t)$$

$$\Delta a = .297\ (\phi)(r + t)$$

$$\Delta b = .703\ (\phi)(r + t)$$

$$a = r + \Delta a - \frac{t}{2} = d_i/2 + \Delta a$$

$$b = r - \Delta b - \frac{t}{2} = d_i/2 - \Delta b$$

The quantities a and b are used to calculate the radii of curvature.

From Reference 1, the radii of curvature of the middle surface or the neutral axis of the tube wall, if we call these points A and B are:

$$r_A = \frac{b^2}{a} + \frac{t}{2}$$

$$r_B = \frac{a^2}{b} + \frac{t}{2}$$

These equations are solved for each percent flatness that is needed and the resulting values are inserted in the following equation from Reference 1 to obtain the stress on the inner diameter of the wall at points A and B. The general equation is:

$$f_t = \frac{E\,t\,(r_u - r_p)}{r_p\,(2r_u - t)} \tag{5}$$

r_u = radius at initial ovality
r_p = radius at final ovality after application of pressure.

Specifically the equations for points A and B are:

$$f_{tA} = \frac{Et\,(r_{Au} - r_{Ap})}{r_{Ap}\,(2r_{Au} - t)}$$

$$f_{tB} = \frac{Et(r_{Bu} - r_{Bp})}{r_{Bp}(2r_{Au} - t)}$$

The moment in the tube wall due to elasticity (M_E)is given in Reference 1 as:

$$M_E = M_A + M_B = \frac{2I}{t}\,[(f_{tA}) + (f_{tB})]$$

where:

$I = t^3/12$, moment of inertia of tube wall about the wall neutral axis.

This moment must be equal to the moment due to pressure (M_p) which is given by Reference 1 as:

$$M_p = M_A + M_B = \frac{P}{2}\,(b-a)(a+b+t)$$

The two moments are made equal by decreasing the final percent flatness until the two equations give equal values. This can be accomplished graphically as shown in Reference 1 or by iteration on a computer or programmable calculator. For this program the iteration was performed until the ratio M_E/M_p was greater than or equal to .95. A more accurate number could be attained by reducing the step size of the iteration but a small increase in accuracy would have required a large increase in time.

The circumferential stress due to pressure was calculated using the classic thick wall pressure vessel equation as follows:

$$S_{xi} = \frac{P(d_o^2 + d_i^2)}{(d_o^2 - d_i^2)} \quad \text{(stress at inner surface of tube wall)} \qquad (6)$$

$$S_{xo} = \frac{2P\ d_i^2}{d_o^2 - d_i^2} \quad \text{(stress at outer surface of tube wall)} \qquad (7)$$

The longitudinal stress due to pressure was calculated using the formula in Reference 1 for longitudinal stress in a curved tube.

$$S_L = \frac{P\left\{\dfrac{D_m}{(d_o/(d_o - 2t)^2 - 1} + \dfrac{(d_o - 2t)(d_o - t)\cos\phi}{2t}\right\} + E\,\Delta d_b \cos^3\phi}{D_m + (d_o - t)\cos\phi} \qquad (8)$$

where $\Delta d_b = d_b(\phi_2) - d_b(\phi_1) = 2[b(\phi_2) - b(\phi_1)]$

The appropriate stresses for each case, as shown in Table 3, were calculated and were combined using the Distortion Energy Theory of Failure to reduce all the stresses to a single equivalent stress as shown below.

$$S_{EQ} = [1/2\ ((Sx - Sy)^2 + (Sy - Sz)^2 + (Sz - Sx)^2 + 6(Sxy^2 + Syz^2 + Szx^2))]^{1/2} \qquad (9)$$

See Figure 4 for orientation of stresses used in equation 9.

The tube was checked at three locations as shown in Figure 5 to locate the maximum stress point.

TABLE 3. SUMMARY OF HELICAL CONFIGURATION STRESSES

CASE *	S_x CIRCUNFERENTIAL	S_y LONGITUDINAL	S_z RADIAL	S_{xz} TORSIONAL	S_{xy} S_{zy}
1	EQ7 + (-EQ5 at pt B)	Equation 8 at ρ = 180°	0	Equation 2	0
2	EQ6 + (EQ5 at pt A)	Equation 8 at ρ = 90°	-P	Equation 1	0
3	EQ7 + (-EQ5 at pt B)	Equation 8 at ρ = 0°	0	Equation 3	0

* See Figure 5 for locations.

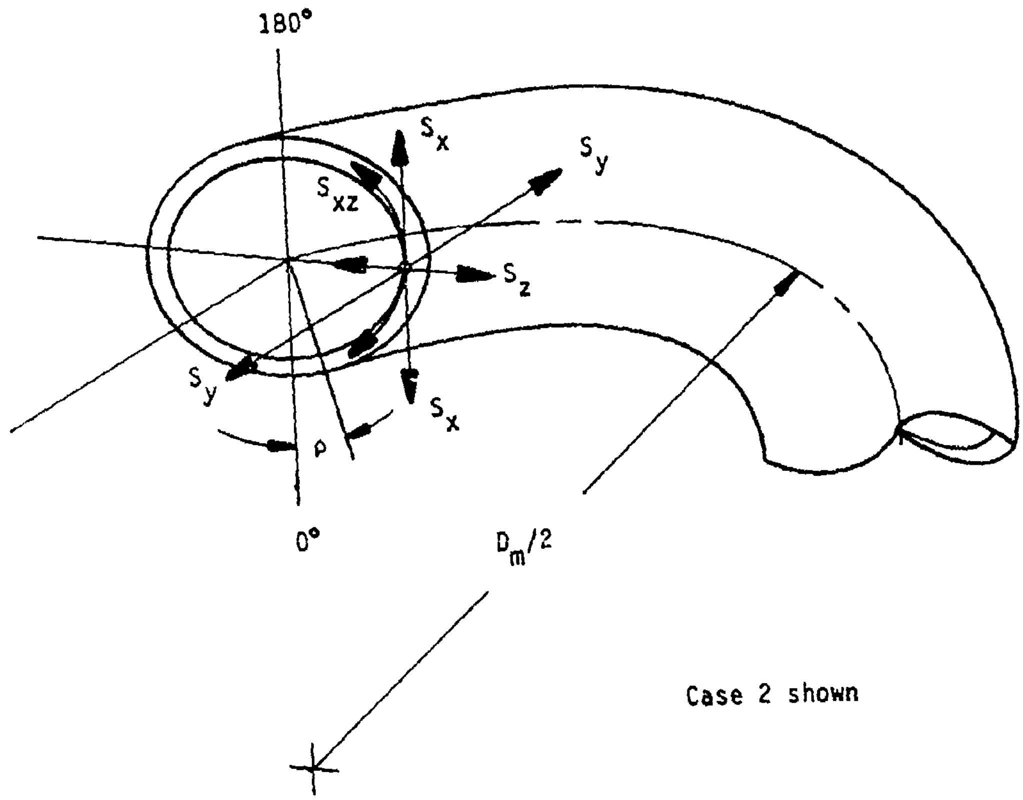


Figure 4. Orientation of Stresses - Helical Configuration

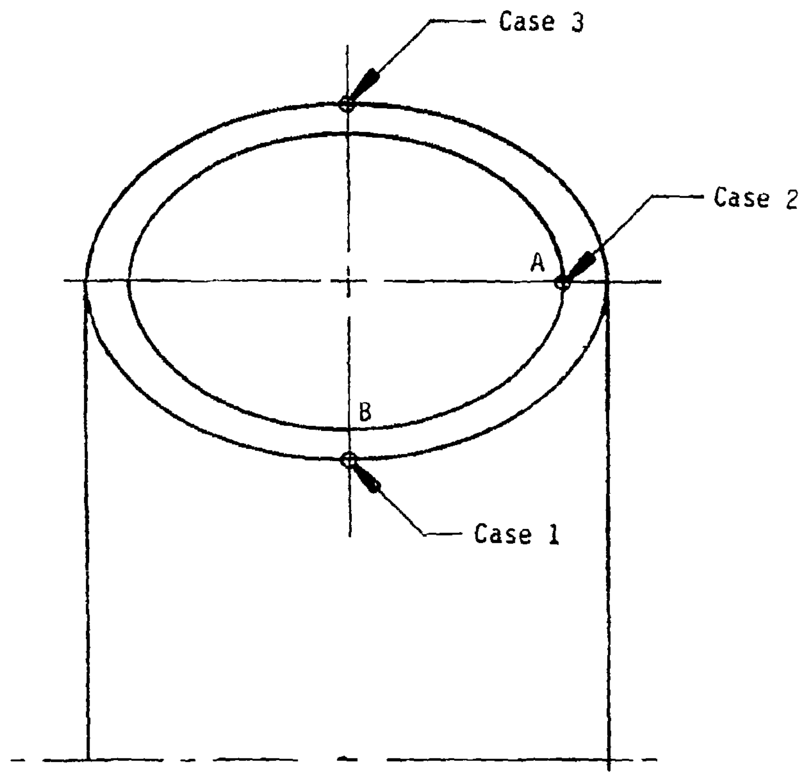


Figure 5. Location of Points Checked for Maximum Stress on Helical Configuration

The allowable stress in the tubing for the helical coils was set at 65,000 psi for the Ti 6AL-4V material, 62,500 for the Ti 3Al - 2.5V CWSR material and 45,000 psi for the Ti 3Al-2.5V Ann material. These stresses represented the maximum allowable for a fully reversing 100,000 cycle life with a scatter factor of 4 for fatigue.

The nomographs in Section 5 for the helical coil tube springs were set up by repetitively solving the equations for each pressure level, tube material, wall thickness and coil mean diameter for a one inch stroke until the stress was within 100 psi of the target allowable stress.

During the initial nomograph set-up it was observed that the stresses due to pressure, and tube ovalization were independent of stroke, and except for some minor bending stresses that were not included in the analysis, the torsional shear stress was the parameter that would size the coil. Examination of Equation 9 shows that the affect of the shear stress on the equivalent stress can be reduced by allowing the spring to have negative shear (extension of the spring). Since each shear stress value is squared, the product is always positive. Since extension springs commonly use allowable stress values that are 80 percent to 85 percent of the allowable stress values for compression springs, it was decided to fix the negative value at 80 percent of the positive value. The amount of stroke, and therefore stress, to be in compression or tension was determined as follows:

Total Stroke $= A$

Compression Stroke $= A_C$

Tension Stroke $= A_T$

$$A = A_C + A_T$$

let: $A_T = .8\ A_C$

therefore: $A = A_C + .8\ A_C$

$$A = 1.8\ A_C$$

$$A_C = \frac{A}{1.8} = .5556A$$

$$A_T = A - A_C = .4444A$$

The affect of the combined use of tension and compression stroke is shown below:

$$F = \frac{A\ G\ (d_o^4 - d_i^4)}{8 D_m^3\ N} \tag{4}$$

$$S_{xz} = \frac{8F\ D_m\ d_o}{\pi(d_o^4 - d_i^4)} \times \frac{1}{K_i} \qquad (3)$$

Substituting Equation 4 for "F" in Equation 3 yields

$$S_{xz} = \frac{8\ [A\ G(d_o^4 - d_i^4)]\quad D_m\ d_o}{8\ D_m^3\ N \quad \pi(d_o^4 - d_i^4)K_i}$$

which reduces to:

$$S_{xz} = \frac{A\ G\ d_o}{\pi\ D_m^2\ NK_i}$$

Solving for N

$$N = \frac{A\ G\ d_o}{\pi\ S_{xz}\ D_m^2\ K_i}$$

Since G, d_o, π, D_m^2 and K_i are constants for this comparison

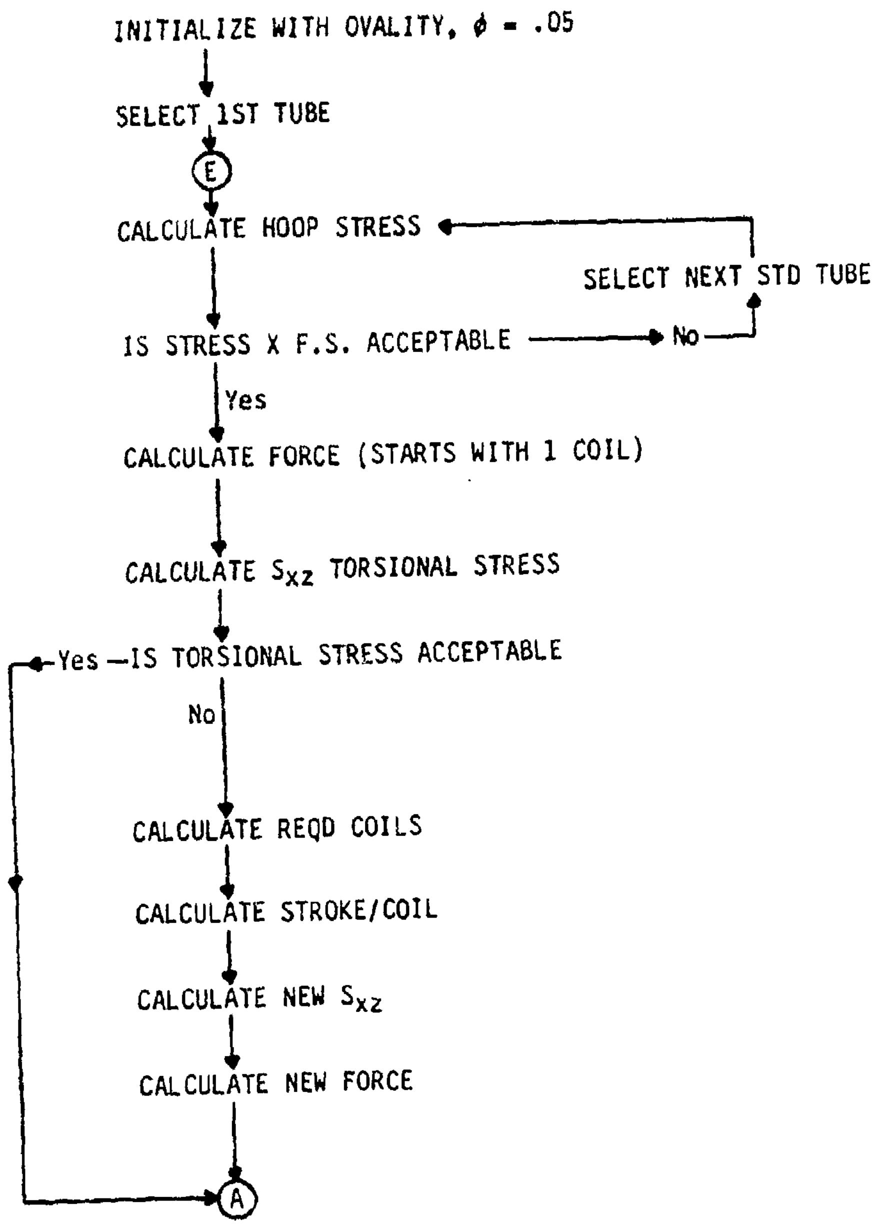
$$\text{let } C = \frac{G\ d_o}{\pi D_m^2\ K_i}$$

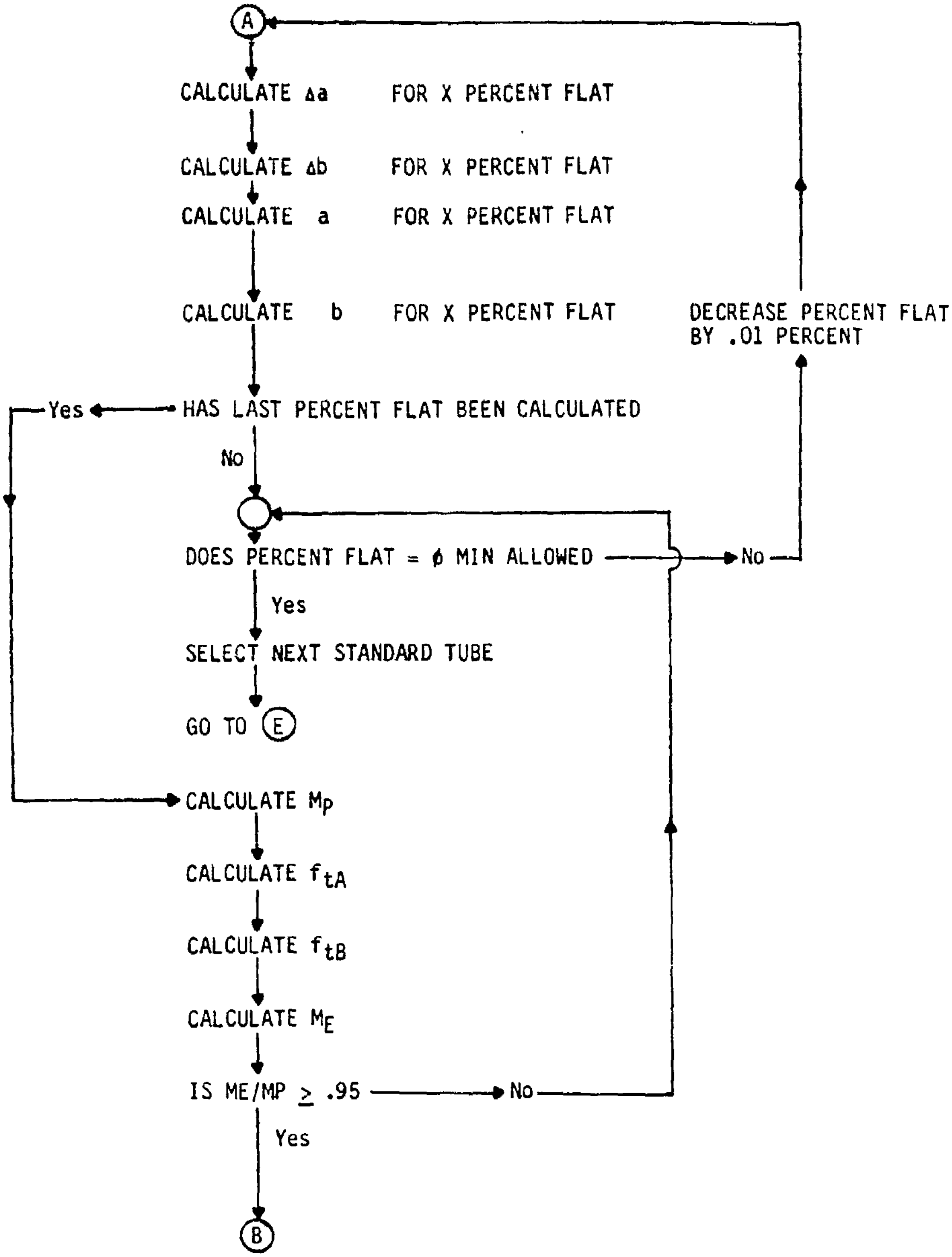
$$N = \frac{AC}{S_{xz}} \qquad (10)$$

The number of coils in a helical spring is directly proportional to the deflection for a given S_{xz}. As shown above, if the spring is extended and compressed to meet the stroke requirements, it requires 44.4 percent less compressive stroke than if it is designed for compressive stroke only. Since the stroke, A, is 44.4 percent less, Equation 10 shows that the number of coils is reduced by 44.4 percent.

3.3 Flow Chart for Helical Configuration Design

Using techniques of Reference 1.




A
CALCULATE Δa FOR X PERCENT FLAT
CALCULATE Δb FOR X PERCENT FLAT
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CALCULATE b FOR X PERCENT FLAT
DECREASE PERCENT FLAT BY .01 PERCENT
HAS LAST PERCENT FLAT BEEN CALCULATED
Yes
No
DOES PERCENT FLAT = ϕ MIN ALLOWED
No
Yes
SELECT NEXT STANDARD TUBE
GO TO E
CALCULATE M_P
CALCULATE f_{tA}
CALCULATE f_{tB}
CALCULATE M_E
IS ME/MP ≥ .95
No
Yes
B

Helical Configuration Design Flow Chart - Continued

(B)

INITIALIZE COUNTERS

CALCULATE Δd_b

PRINT C1 (CASE 1)

CALCULATE S_x ON OUTSIDE WALL AND
ST IN TEMP REGISTER (SBR PROD)

SUM F_{tB} WITH S_x IN TEMP REGISTER

STORE ρ = 180 IN TEMP REGISTER

CALCULATE LONG STRESS SL ARISING
FROM PRESS, PRINT SL (SBR π)
AND STORE SL IN TEMP REGISTER

RCL S_{xz} AND STORE IN TEMP REGISTER

RCL WAHL FACT AND MULTIPLY X S_{xz} IN TR

CALCULATE EQUIV STRESS FOR CASE 1
PRINT S_{EQ}, INITIAL LOAD = 100 PERCENT (SBR EXEC)

Reduce load

No ← HAS EQUIV LOAD AT 2 PERCENT BEEN CALC

Yes

INITIALIZE COUNTERS

PRINT C2 (CASE 2)

RCL S_x ON INSIDE WALL AND STORE
IN TEMP REGISTER

(C)

Helical Configuration Design Flow Chart - Continued

C

STORE ρ = 90 IN TEMP REGISTER

CALCULATE LONG STRESS S_L ARISING
FROM PRESS. PRINT S_L (SBR π)
STORE S_L IN TEMP REGISTER

SUM F_{tA} WITH S_x
RCL S_z AND STORE IN TEMP REGISTER
RCL S_{xz} AND STORE IN TEMP REGISTER

CALCULATE EQUIV STRESS FOR CASE 2
PRINT S_{EQ}, INITIAL LOAD = 100 PERCENT (SBR EXEC)

Reduce Load

No ← HAS EQUIV STRESS AT 2 PERCENT BEEN CALC

Yes

INITIALIZE LOAD COUNTER

PRINT C3 (CASE 3)

CALCULATE S_x ON OUTSIDE WALL AND
STORE IN TEMP REG (SBR PROD)

STORE ρ = 0 IN TEMP REGISTER

CALCULATE LONG STRESS ARISING FROM
PRESS. PRINT S_L (SBR π)

SUM F_{tB} WITH S_x

D

Helical Configuration Design Flow Chart - Continued

D

RCL S_{xz} AND STORE IN TEMP REG
RCL WAHL FACT 1/X AND MULTIPLY BY S_{xz} IN TEMP REG

CALCULATE EQUIV STRESS FOR CASE 3
PRINT S_{EQ}, INITIAL LOAD = 100 PERCENT (SBR EXEC)

Reduce Load

No ← HAS EQUIV STRESS AT 2 PERCENT BEEN CALC

Yes

PRINT DATA (SBR PRT)

R/S

4. TRI-COIL CONFIGURATION STRESS ANALYSIS

4.1 Definition of Stresses. A coil of the tri-coil spring configuration had three different applied stresses; 1, the windup stress caused by mechanical deflection; 2, the ovalization stress caused by the slightly flattened tube trying to regain a round cross section under pressure and; 3, the stress caused by pressurizing a curved tube. All three stresses were determined by the techniques given in Reference 1. The stresses obtained from these techniques were combined into an equivalent stress using the Distortion Energy Theory of Failures. The stresses were highest in the center coil for the tri-coil configuration. The equations derived are for the stresses occurring on the outer diameter of the center coil on the tube outer diameter. This location is identified as point "D" on Figure 6.

4.2 Derivation of Stress Equations. First, the geometric relationships of the tri-coil configuration were analyzed. Referring to Figure 6 the following relationships are given.

H is obtained by iteration to give an equivalent stress which is equal to or less than the maximum allowable tension stress for 100,000 cycles of full stroke with a scatter factor of four.

B is a dimension which gives clearance between coils when the tubing is in the unloaded position. In this case a clearance of .25 inch is used, which with a mean diameter of 2.0 and tube diameter of .1875 sets B to be 2.44 inches.

$$\alpha = 2 \text{ arc tan } (B/2H) \tag{11}$$

$$C = H/\cos(\alpha/2) \tag{12}$$

$$L = 2[(C/2)^2 - (D_m/2)^2]^{1/2} \tag{13}$$

$$\omega = [\text{arc cos } (D_m/C)] - \alpha/2 \tag{14}$$

$$\epsilon = 90 - \omega \tag{15}$$

Reference 2, p 138 shows the angular deflection ($\Delta\alpha$) of a torsion spring made from round wire to be:

$$\Delta\alpha = \frac{3670MN\ D_m}{Ed^4}$$

where d is wire diameter and $\Delta\alpha$ is total angular deflection of coil

For a tube this equation becomes

$$\Delta\alpha = \frac{3670\ MN\ D_m}{E\ (d_o^4 - d_i^4)} \tag{16}$$

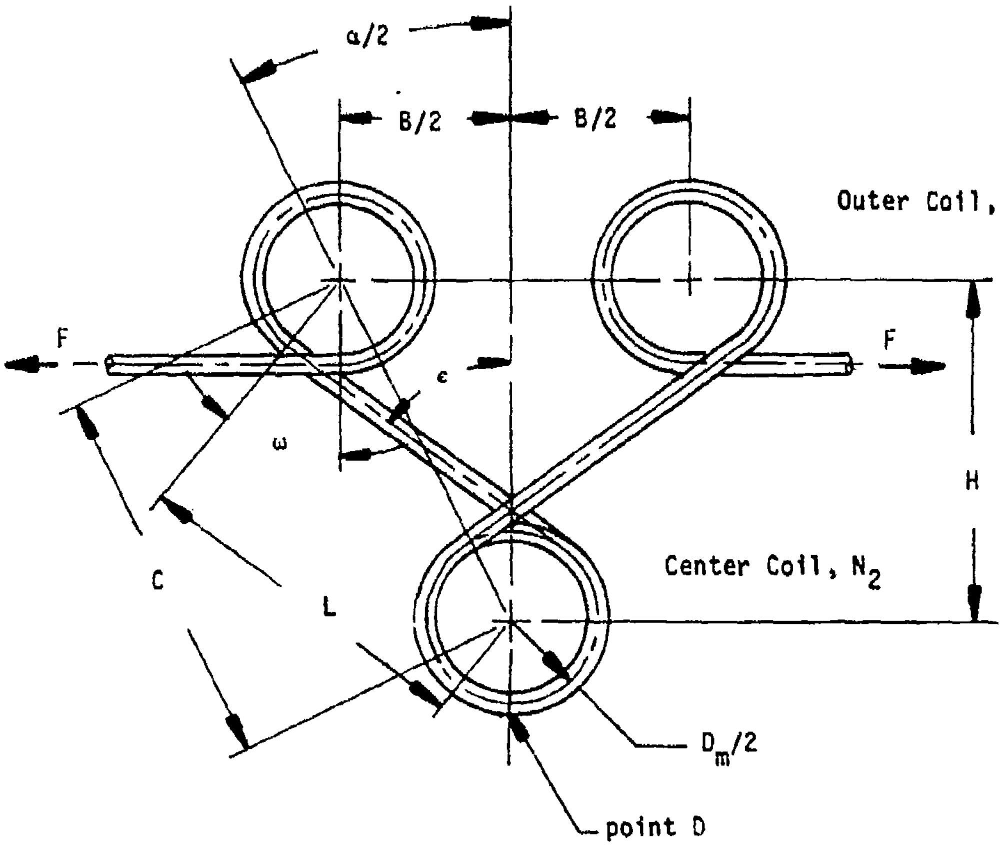


Figure 6. Dimensions Used for Tri-Coil Configuration Stress Analysis

or

$$M = \frac{\Delta\alpha E\,(d_o^4 - d_i^4)}{3670\ D_m\ N} \qquad (16a)$$

Since $M = F \times \frac{Dm}{2}$

$$F = \frac{2\Delta\alpha\ E(d_o^4 - d_i^4)}{3670\ D_m^2\ N_1}$$ where F is force at the outer coil and N_1 is the number of coils in an outer coil

If F is applied at the centerline of the tube as shown in Figure 6, the moment for the center coils is:

$$M = F[H-(Dm/2)]$$

Substituting into Equation (16a) and rearranging:

$$F = \frac{\Delta\alpha E\,(d_o^4 - d_i^4)}{3670\ D_m\ (H-(Dm/2))N_2} \qquad (17)$$ where N_2 is the number of coils in the center coil

$$\Delta\alpha = \alpha_T - \alpha$$

Equation 17 becomes

$$F = \frac{(\alpha_T - \alpha)\quad E\ (d_o^4 - d_i^4\ )}{3670\ D_m\ (H_T - D_m/2)\ N2} \qquad (18)$$

As the coil is stroked through stroke A, angle α becomes α_T and H becomes H_T. To get $\Delta\alpha$, angle α_T and distance H_T between coil centers can be calculated from known information.

$$H_T = \quad [C^2 - ((B + A)/2)^2]^{1/2} \qquad (19)$$

$$\frac{\alpha_T}{2} = \quad \arctan\ [(B + A)/(2\ H_T)] \qquad (20)$$

Once the value for F was known the bending stress was calculated using the technique in Reference 1 as follows:

> NOTE: For calculation of λ, reference one uses the radius r to the neutral axis of the tube wall. However, all calculations were made using the radius $(d_o/2)$ to the outside of the tube wall which made calculated stresses agree with stresses observed on tests of coils with strain gages.

$$\lambda = \frac{t \times D_m/2}{(d_o/2)^2}$$

$$\gamma = \frac{6\lambda^2 - 1}{6\lambda^2 + 5}$$

$$K = \frac{12\lambda^2 + 10}{12\lambda^2 + 1}$$

$$I = \frac{(d_o^4 - d_i^4) \times \pi}{64}$$

$$f_l = \frac{K\gamma\ H_1 F(b + t)}{I} \quad (21)$$

(sign value is a function of position - positive is tension at point "D")

$$\beta = \frac{9\lambda^2}{6\lambda^2 + 5}$$

$$f_t = \frac{K\beta\ FH_1(b+t)}{I} \quad (22)$$

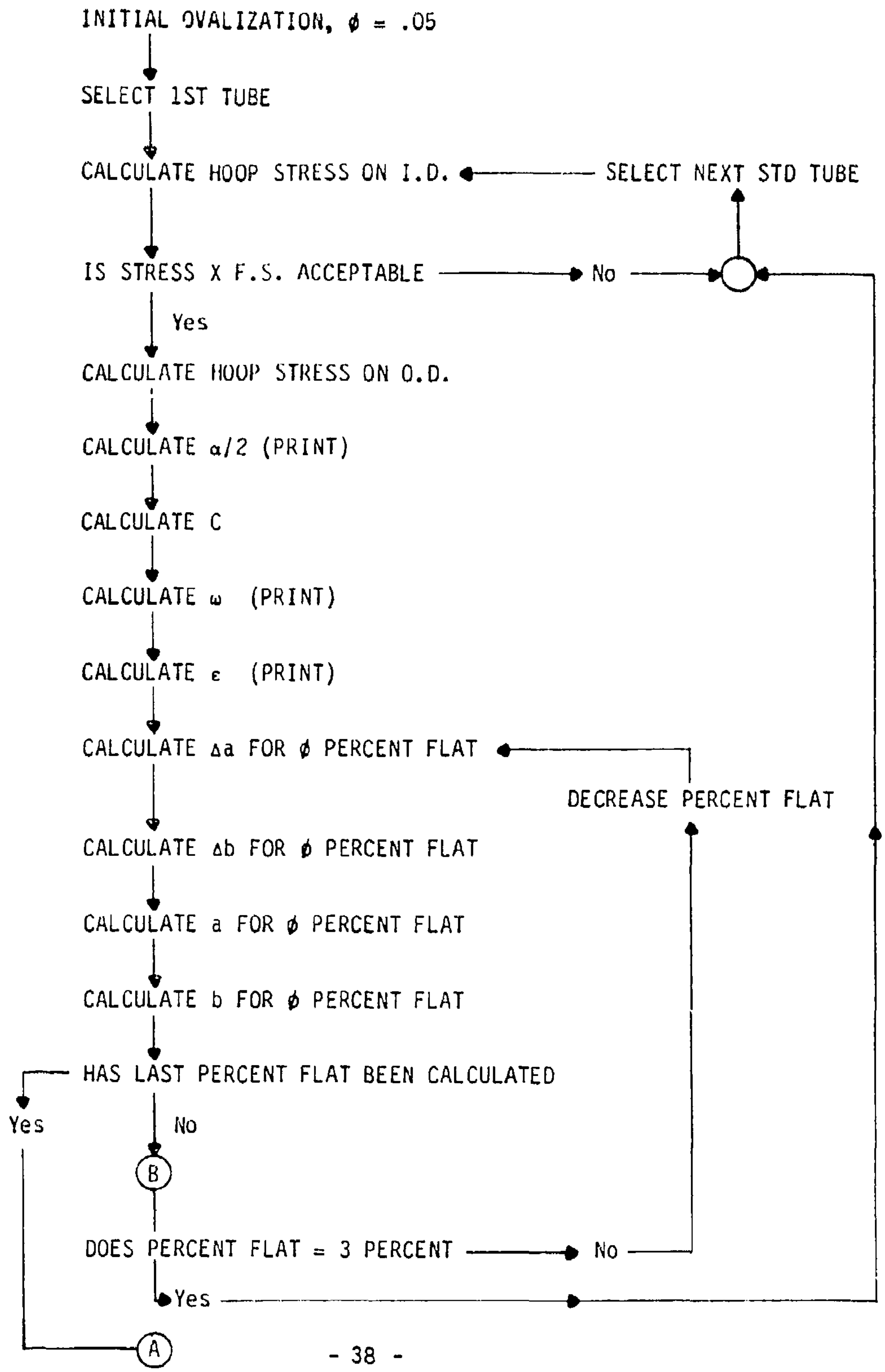
The ovalization stresses and the circumferential and longitudinal stresses were calculated in the same manner shown for the helical coil spring. The stresses were combined using the Distortion Energy Theory of Failure to reduce all the stresses to a single equivalent stress for point D as shown below. S_x, S_y, S_z, S_{xy}, S_{yz}, and S_{zx} are defined on Table 4.

$$S_{EQ} = [1/2((S_x - S_y)^2 + (S_y - S_z)^2 + (S_z - S_x)^2$$
$$+ 6\ (S_{xy}{}^2 + S_{yz}{}^2 + S_{zx}{}^2)]^{1/2}$$

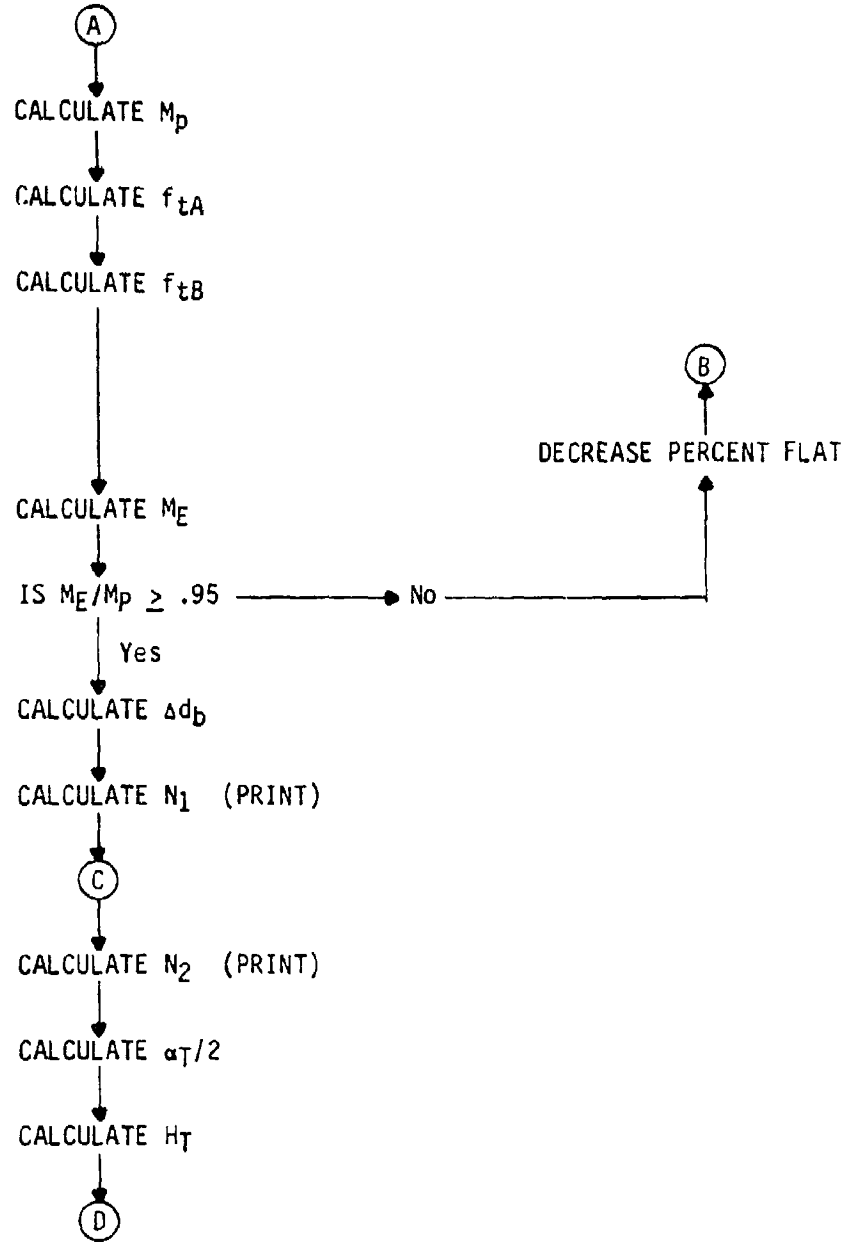
TABLE 4. SUMMARY OF TRI-COIL CONFIGURATION STRESSES

CASE	Sx CIRCUMFERENTIAL	Sy LONGITUDINAL	Sz RADIAL	Sxy, Syz, Szx
D	Equation 7 - (Equation 5 at point B)+ Equation 22 (Tension)	Equation 8 at $\rho = 0°$ Equation 18	0	Assumed = 0

4.3 Flow Chart for Tri-Coil Configuration Design



4.3 Flow Chart for Tri-Coil Configuration Design - Continued



4.3 Flow Chart for Tri-Coil Configuration Design - Continued

D

↓

CALCULATE $\Delta\alpha$ (PRINT $\Delta\alpha$)

↓

CALCULATE FORCE (PRINT F)

↓

CALCULATE λ

↓

CALCULATE γ

↓

CALCULATE K

↓

CALCULATE I

↓

CALCULATE f_1

↓

CALCULATE β

↓

CALCULATE f_t

↓

CALCULATE S_1

↓

CALCULATE S_{EQ} (PRINT S_{EQ})

↓

PRINT DATA

↓

R/S

5. NOMOGRAPHS FOR COIL TUBE DESIGN

5.1 Description of Nomographs

The nomographs in this section were created to allow design of a coiled tube installation without the need for familiarization with the stress analysis. There is a set of nine nomographs for the helical coil configuration and nine nomographs for the tri-coil spring configuration. A set of nine nomographs consists of 3 graphs for each of three materials. The three graphs are each for a different pressure. For example, each set of nine nomographs consists of the following:

Material	Pressure
Ti 3Al-2.5V Annealed	1500
	3000
	8000
Ti 3Al-2.5V CWSR	1500
	3000
	8000
Ti 6Al-4V	1500
	3000
	8000

The nomographs in this section for the helical coil tube springs were set up by repetitively solving the equations for each pressure level, tube material, wall thickness and coil mean diameter for a one inch stroke until the stress was within 100 psi of the target allowable stress.

The nomographs for the tri-coil tube springs were set up by repetitively solving the equations for each pressure level, tube material, wall thickness, coil diameter, and by leg length for a one-inch, two-inch, and three-inch stroke, until the stress was within 100 psi of the target allowable stress.

5.2 Instructions for Use - Helical Configuration. Information needed is: (1) total stroke of installation, (2) operating pressure, (3) maximum mean coil diameter allowed by the installation, and (4) tube material to be used.

Allowable tensile stress in the tubing for the helical coil configuration was set at 65,000 psi for Ti 6Al-4V, 62,500 psi for Ti 3Al-2.5V cold worked stress relieved (CWSR), and 45.000 psi for Ti 3Al-2.5 annealed tubing. These stresses represent .e maximum allowable when all stresses are considered using the distortion energy theory of failure and a fatigue requirement of 100,000 cycles fully reversed stress with a scatter factor of 4 for fatigue.

For example, assume that the total stroke is 3.0 inches. The operating pressure is 8000 psig. The available installation envelope limits mean coil diameter to 2.5 inches. The tubing material to be used is Ti 3Al-2.5V CWSR.

(1) Find Nomograph 6 for Ti 3Al-2.5V CWSR, 8000 psig, Helical Coil.

(2) Place a straight edge on the "+" index at the left of the nomograph thru 2.5 inches mean coil diameter on the right hand scale.

(3) On the intermediate scale read 1.95 active coils per inch of compression stroke and 22 pounds per inch of compression stroke.

(4) Compute total coils using .5556 times the total stroke as the compression stroke.

Total Coils = (1.95)(.5556)(3) = 3.25

Use Total Coils = 3.25

NOTE: This will result in the coils being slightly compressed in the neutral position.

(5) Maximum Force = (22 lb/in)(3 in.)(.5556) = 36.67 lb. The maximum force should be added to the load requirements of the actuation device.

(6) Calculate pitch (p):

$$p = ((.5556A_T)/N) + d_o$$
$$= ((.5556 \times 3.0)/3.25) + .1875$$
$$= .700 \text{ in.}$$

(7) Calculate free length (L_f):

$$L_f = pN$$
$$= (.700)(3.25)$$
$$= 2.276 \text{ in.}$$

where L_f is taken between tubing centerlines.

The resultant helical coil is good for 100,000 cycles of fully reversed stress.

5.3 Instructions for Use - Triangular Coil Configuration. Information needed is: (1) total stroke of installation, (2) operating pressure, (3) maximum allowable centerline distance between coils, and (4) tube material to be used.

The allowable stress in the tubing for the triangular coil was set at 92,300 psi for the Ti 6AL-4V material, 88,750 psi for the Ti 3AL-2.5V CWSR material and at 63,900 for the Ti 3AL-2.5V ANN material. This represented a 42 percent increase in allowable stress over the value used in the helical spring and it was possible because the triangular spring coils were used in "windup" only and had no reversing stress. These stresses represented the maximum allowable for a nonreversing, 100,000 cycle life.

For the example assume that the total stroke is 3.8 inches. The operating pressure is 3000 psig. Tube material is Ti 6Al-4V.

(1) Find Nomograph 17. Ti 6Al-4V, 3000 psig, triangular coil.

(2) Place a straight edge on the "+" index at the left of the nomograph thru 3.8 inches.

(3) On intermediate scales read "H" = 6.69 inches and Force = 7.2 lb/inch of stroke.

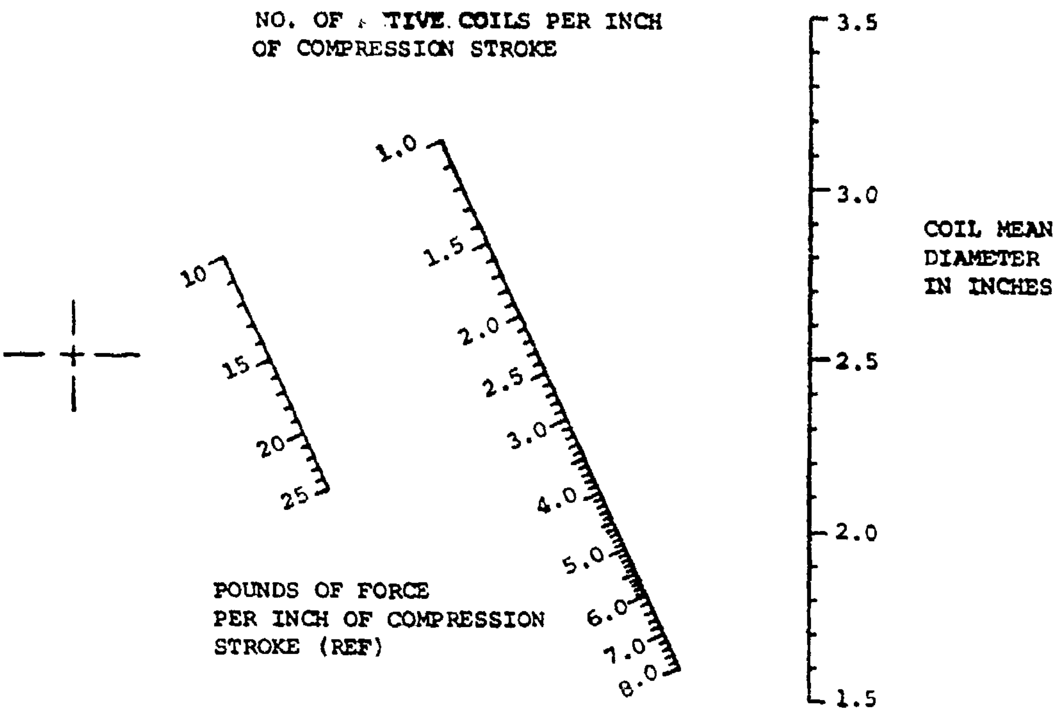
(4) Compute maximum force at full deflection
Force = (7.2 lb/in)(3.8) = 27.36 lb.

(5) The detail dimension for "H" with tolerance must not be less than 6.69 inches.

(6) Calculate B (See Figure 6):

$$B = D_m + d_o + .25$$
$$= 2.0 + .1875 + .25$$
$$= 2.4375 \text{ inches}$$

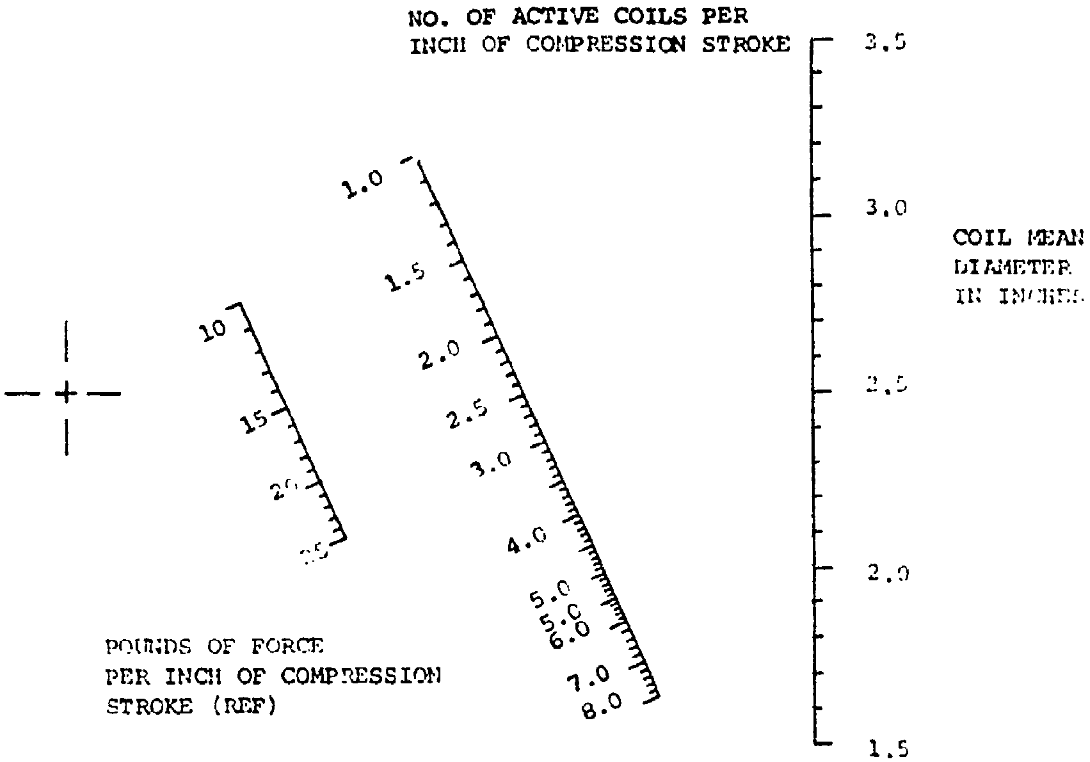
The resultant tri-coil is good for 100,000 cycles of non-reversed stress.



NOTE: Compression Stroke = .5556 x Total Stroke

MATERIAL: T13AL-2.5V ANN
SIZE: .1875 x .020
PRESSURE: 1500 PSIG

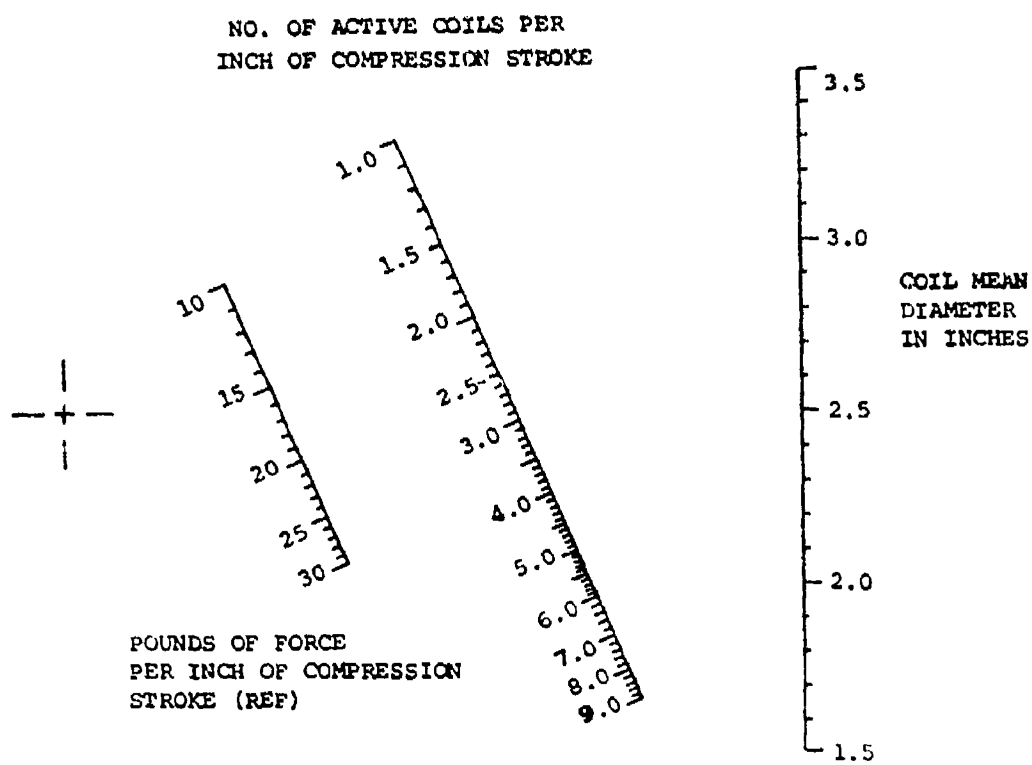
Nomograph 1. Ti 3Al-2.5V Annealed, 1500 psig System, Helical Configuration



NOTE Compression Stroke = .5556 x
Total Stroke

MATERIAL: TI3AL-2.5V ANN
SIZE: .1875 x .020
PRESSURE: 3000 PSIG

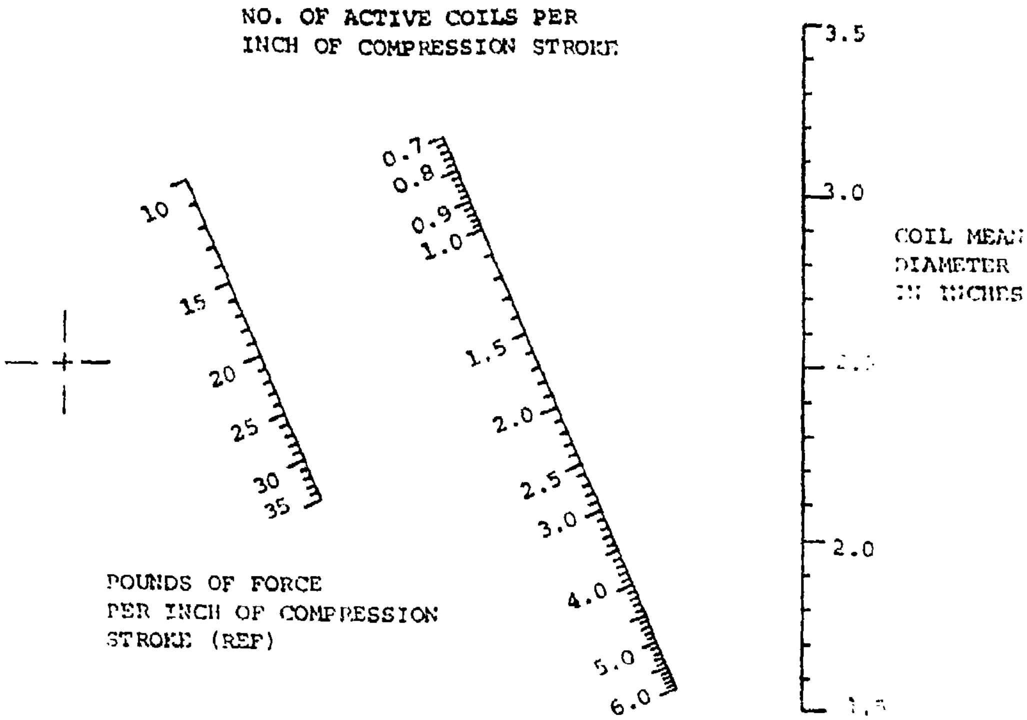
Nomograph 2. Ti 3Al-2.5V Annealed, 3000 psig System, Helical Configuration



NOTE: Compression Stroke = .5556 x
Total Stroke

MATERIAL: T13AL-2.5V ANN
SIZE: .1875 x .035
PRESSURE: 8000 PSIG

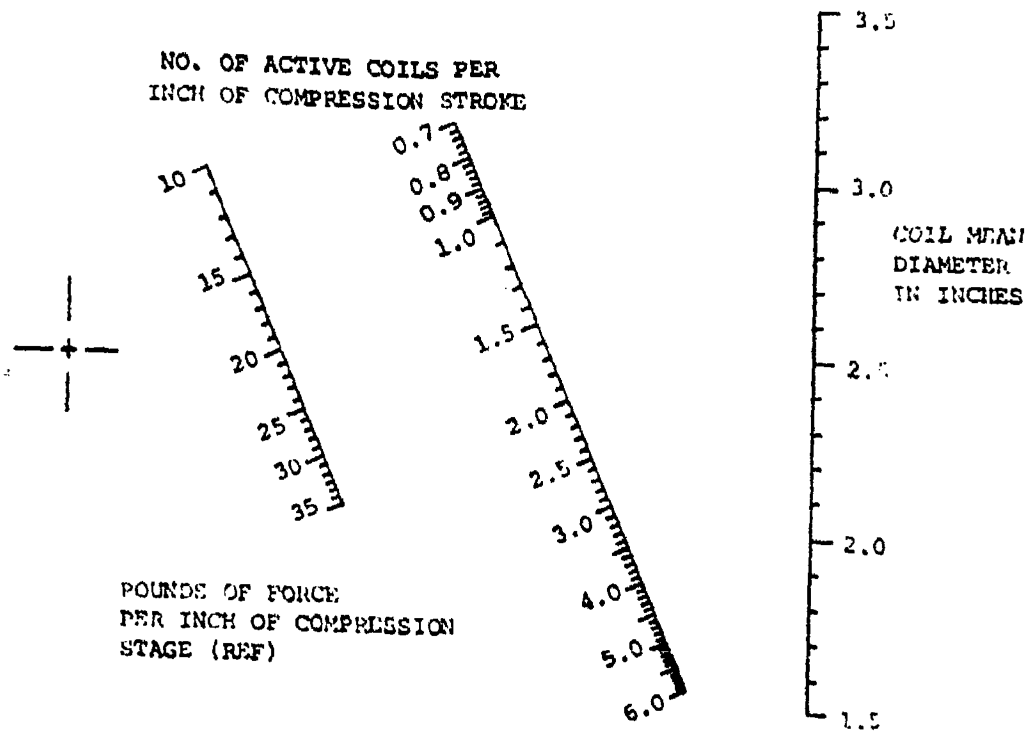
Nomograph 3. Ti 3Al-2.5V Annealed, 8000 psig System, Helical Configuration



NOTE: Compression Stroke = .5556 x
Total Stroke

MATERIAL. T13AL-2.5V CWSR
SIZE: .1875 x .020
PRESSURE: 1500 PSIG

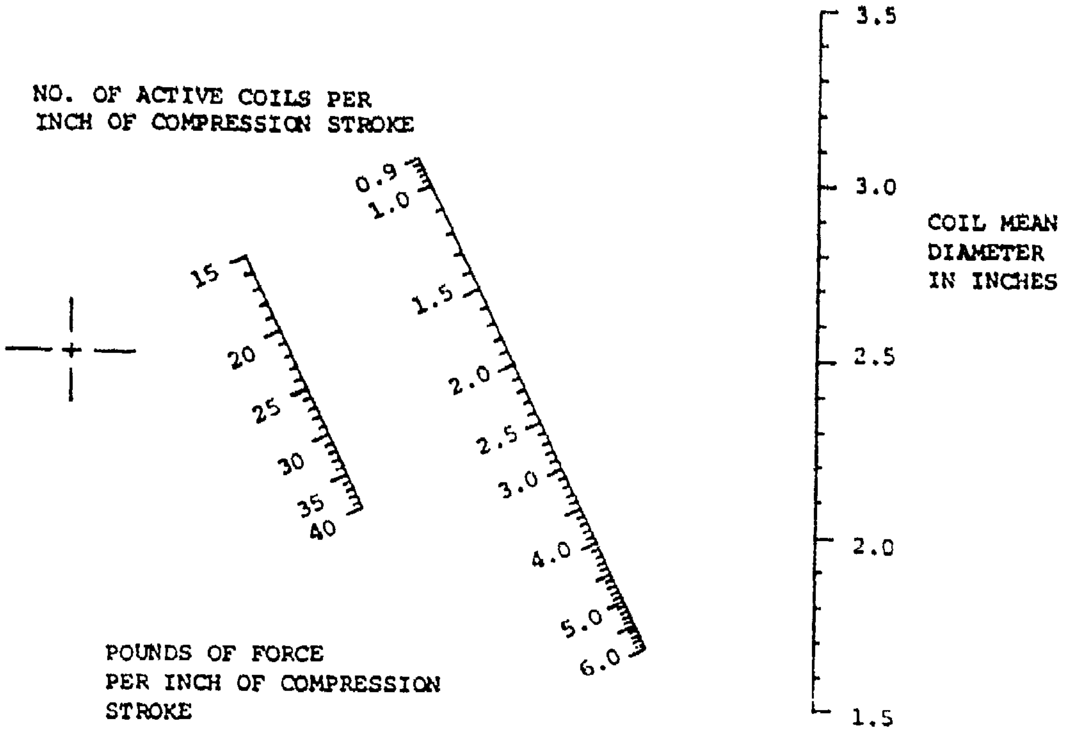
Nomograph 4. Ti 3Al-2.5V CWSR,1500 psig System, Helical Configuration



NOTE. Compression Stroke = .5556 x Total Stroke

MATERIAL. Ti3AL-2.5V CWSR
SIZE: .1875 x .020
PRESSURE: 3000 PSIG

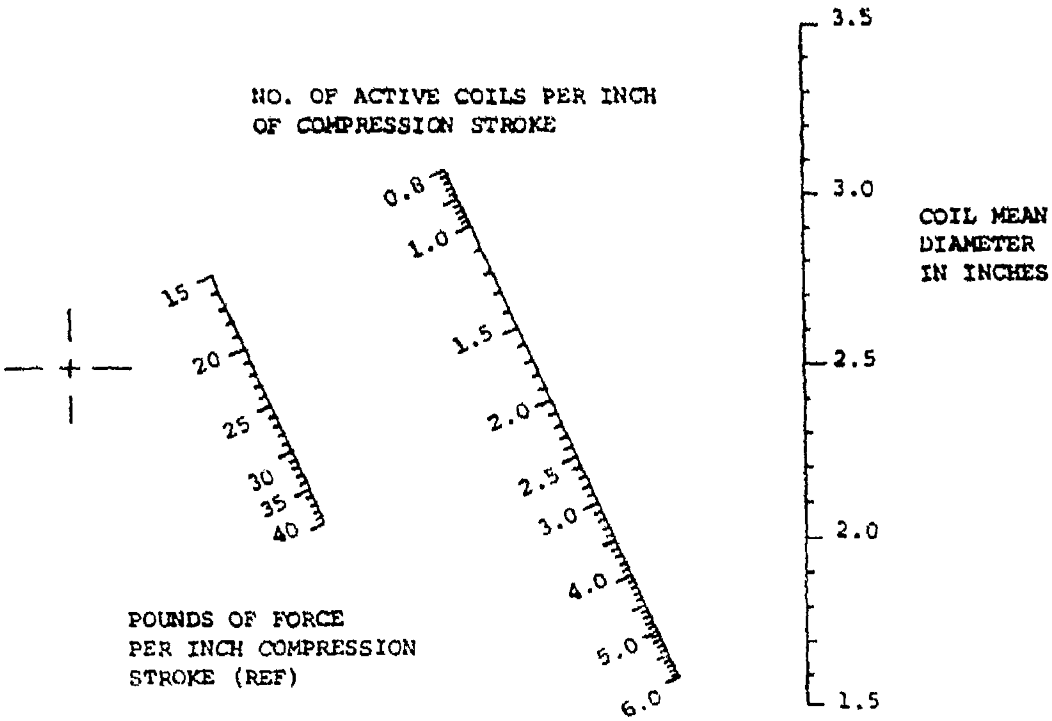
Nomograph 5. Ti 3Al-2.5V CWSR, 3000 psig System, Helical Configuration



NOTE: Compression Stroke = .5556 x Total Stroke

MATERIAL: TI3AL-2.5V CWSR
SIZE: .1875 x .028
PRESSURE: 8000 PSIG

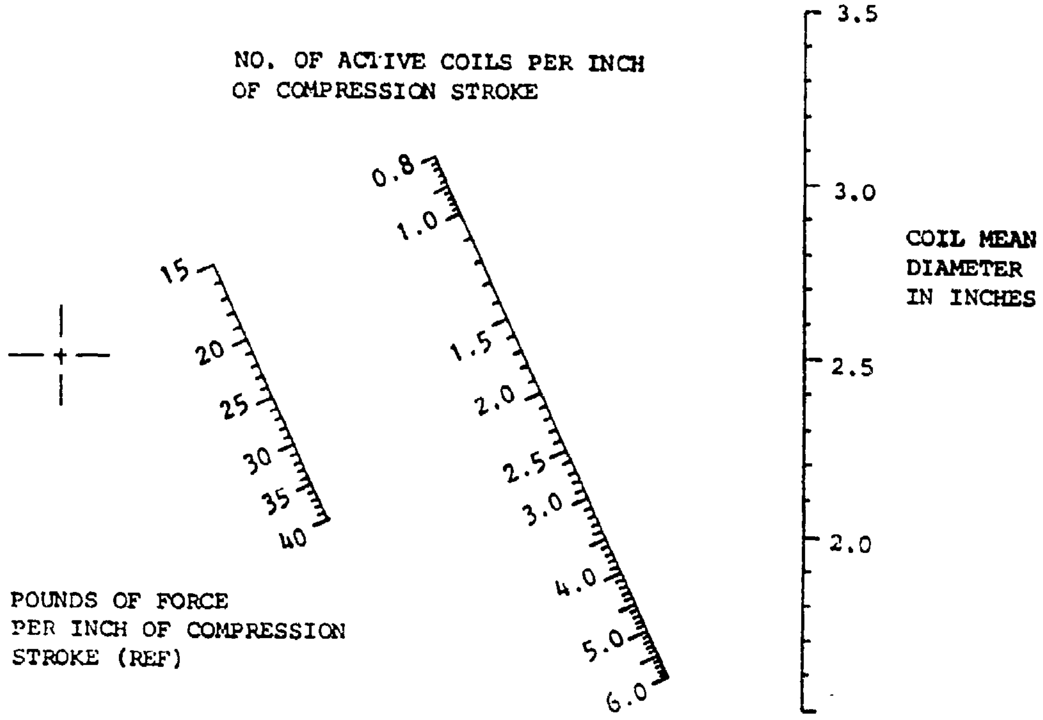
Nomograph 6. Ti 3Al-2.5V CWSR, 8000 psig System, Helical Configuration



NOTE: Compression Stroke = .5556 x Total Stroke

MATERIAL: TI6AL-4V
SIZE: .1875 x .020
PRESSURE: 1500 PSIG

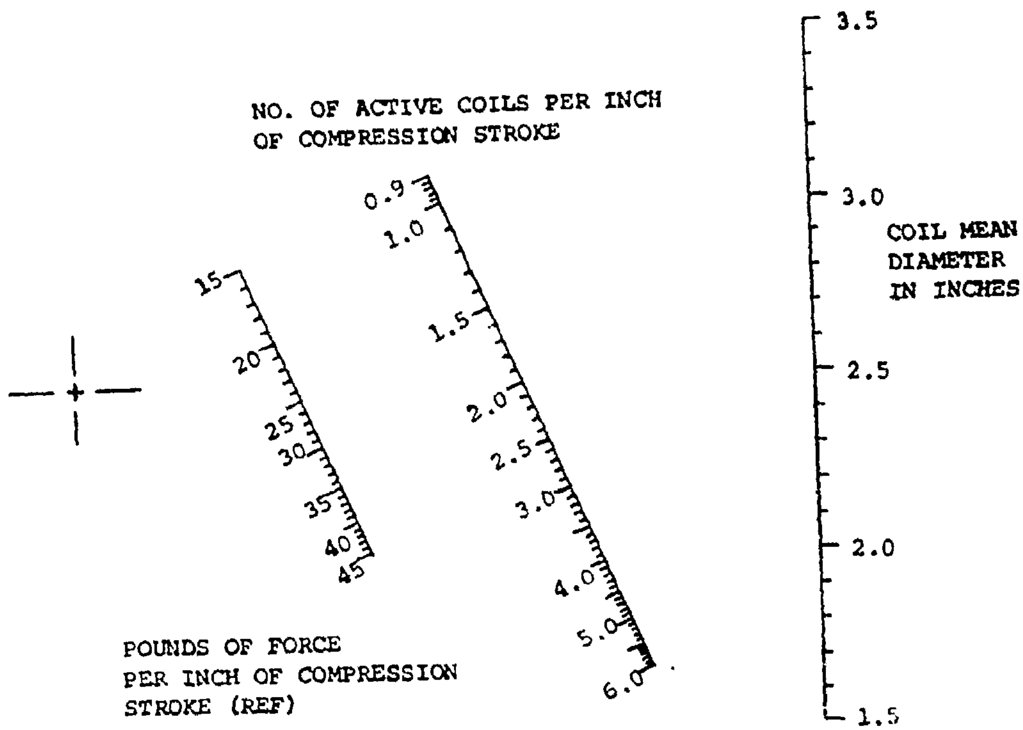
Nomograph 7. Ti 6Al-4V, 1500 psig System, Helical Configuration



NOTE: Compression Stroke = .5556 x
Total Stroke

MATERIAL: TI6AL-4V TUBING
SIZE: .1875 x .020
PRESSURE: 3000 PSIG

Nomograph 8. Ti 6Al-4V, 3000 psig System, Helical Configuration

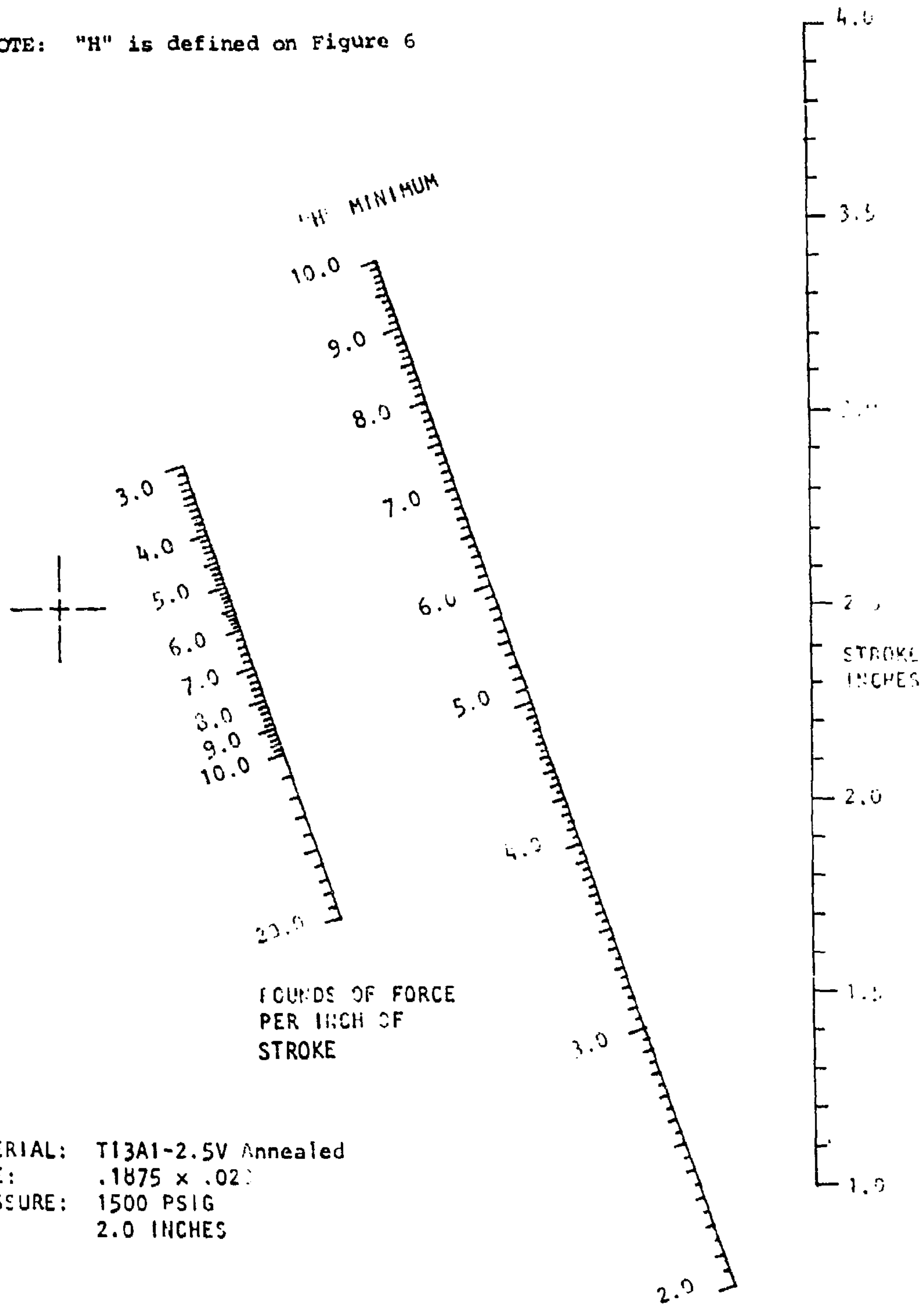


NOTE: Compression Stroke = .5556 x
Total Stroke

MATERIAL: T16AL-4V TUBING
SIZE: .1875 x .028
PRESSURE: 8000 PSIG

Nomograph 9. Ti 6Al-4V, 8000 psig System, Helical Configuration

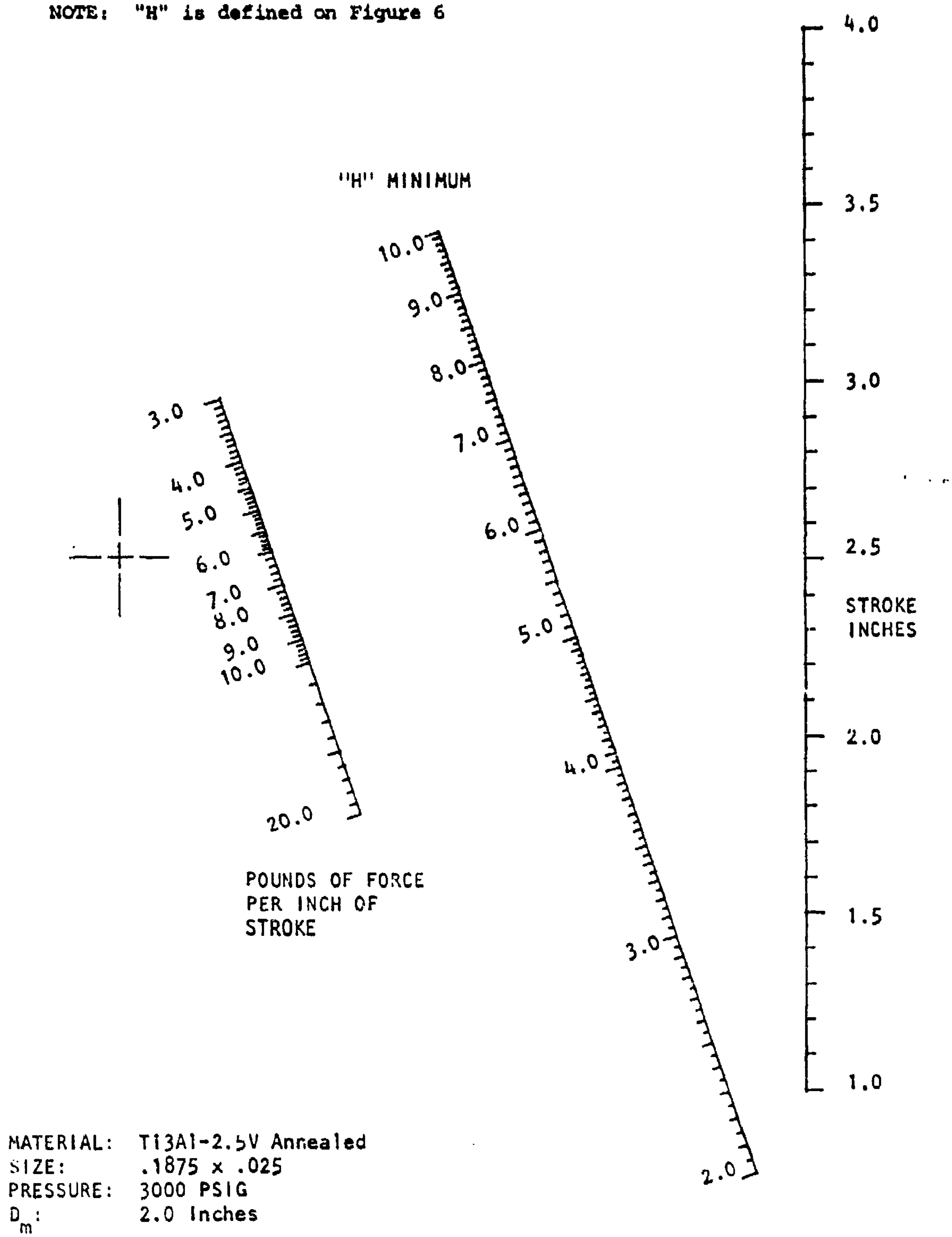
NOTE: "H" is defined on Figure 6



MATERIAL: Ti3Al-2.5V Annealed
SIZE: .1875 x .02
PRESSURE: 1500 PSIG
D_m: 2.0 INCHES

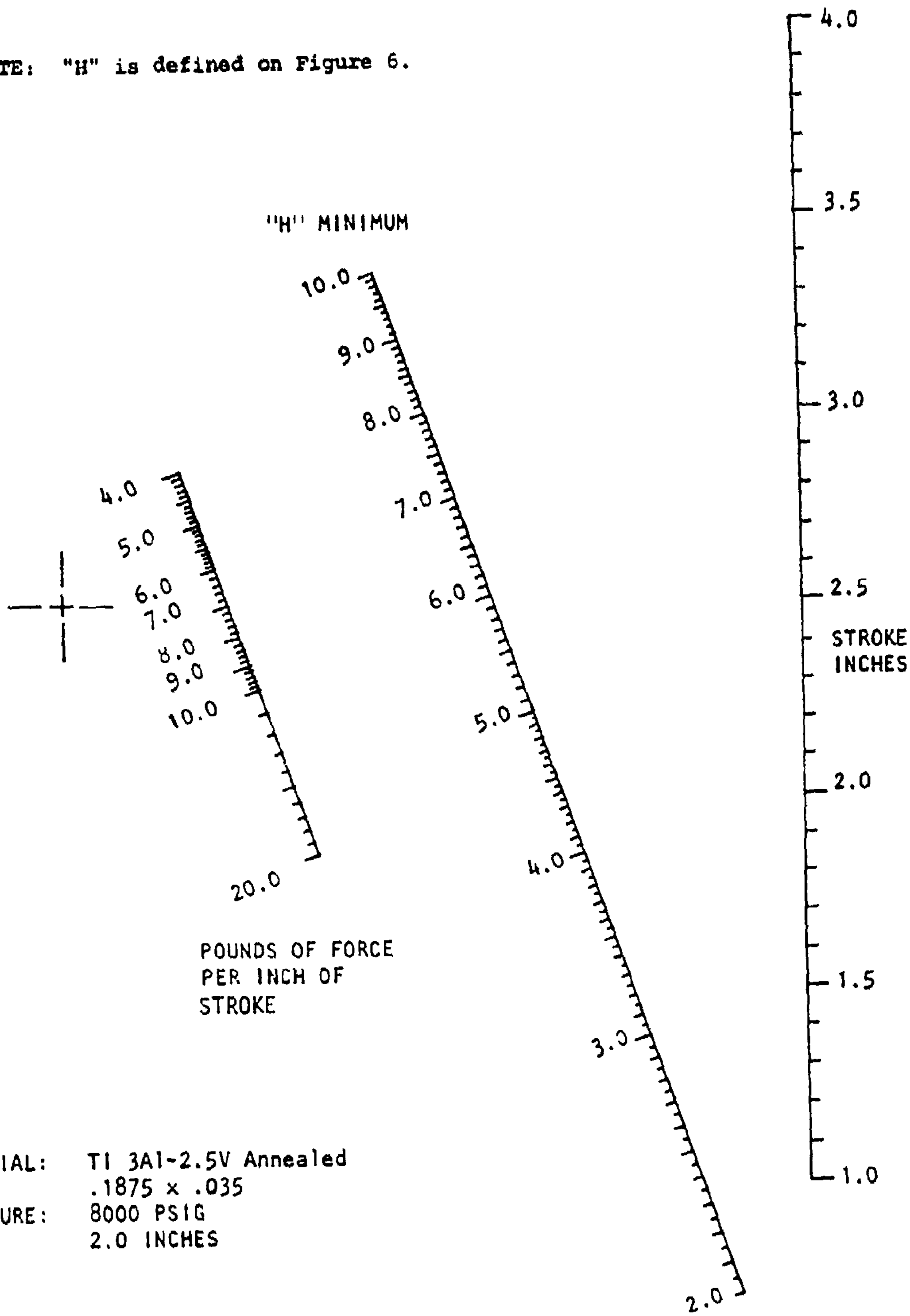
Nomograph 10. Ti 3Al-2.5V Annealed, 1500 psig System, Tri-Coil Configuration

NOTE: "H" is defined on Figure 6



Nomograph 11. Ti 3Al-2.5V Annealed, 3000 psig System, Tri-Coil Configuration

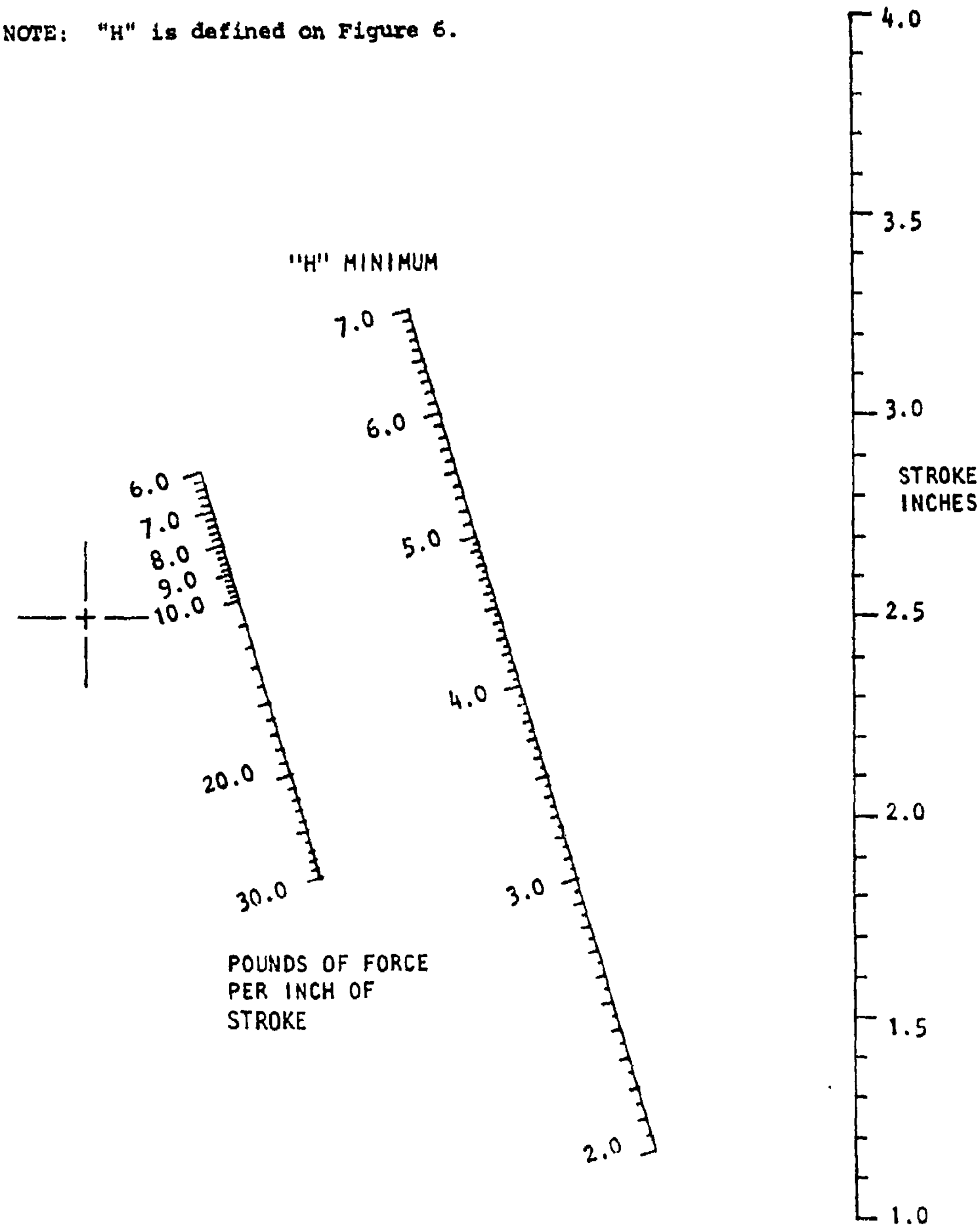
NOTE: "H" is defined on Figure 6.



MATERIAL: TI 3Al-2.5V Annealed
SIZE: .1875 x .035
PRESSURE: 8000 PSIG
D_m: 2.0 INCHES

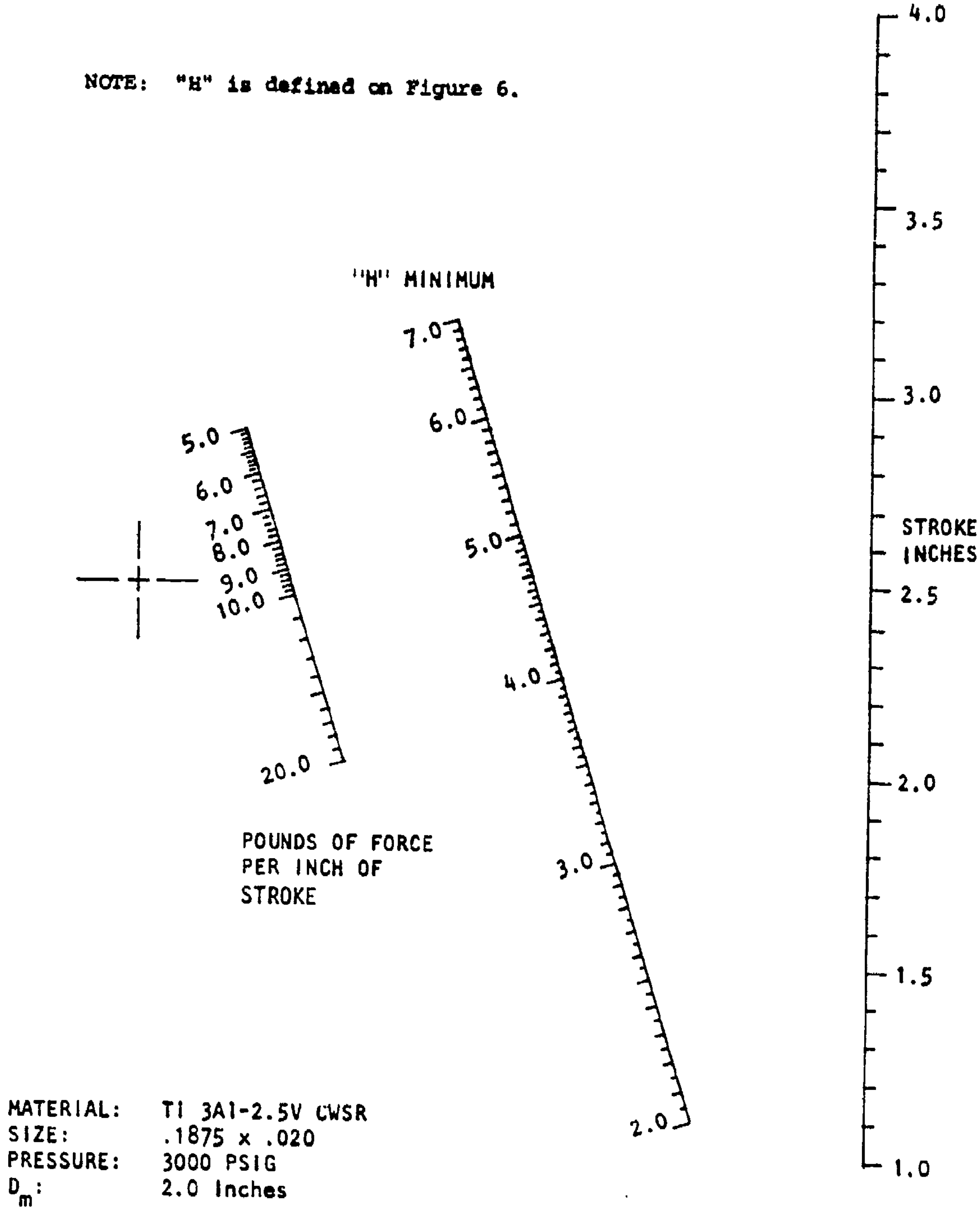
Nomograph 12. Ti 3Al-2.5V Annealed, 8000 psig System, Tri-Coil Configuration

NOTE: "H" is defined on Figure 6.



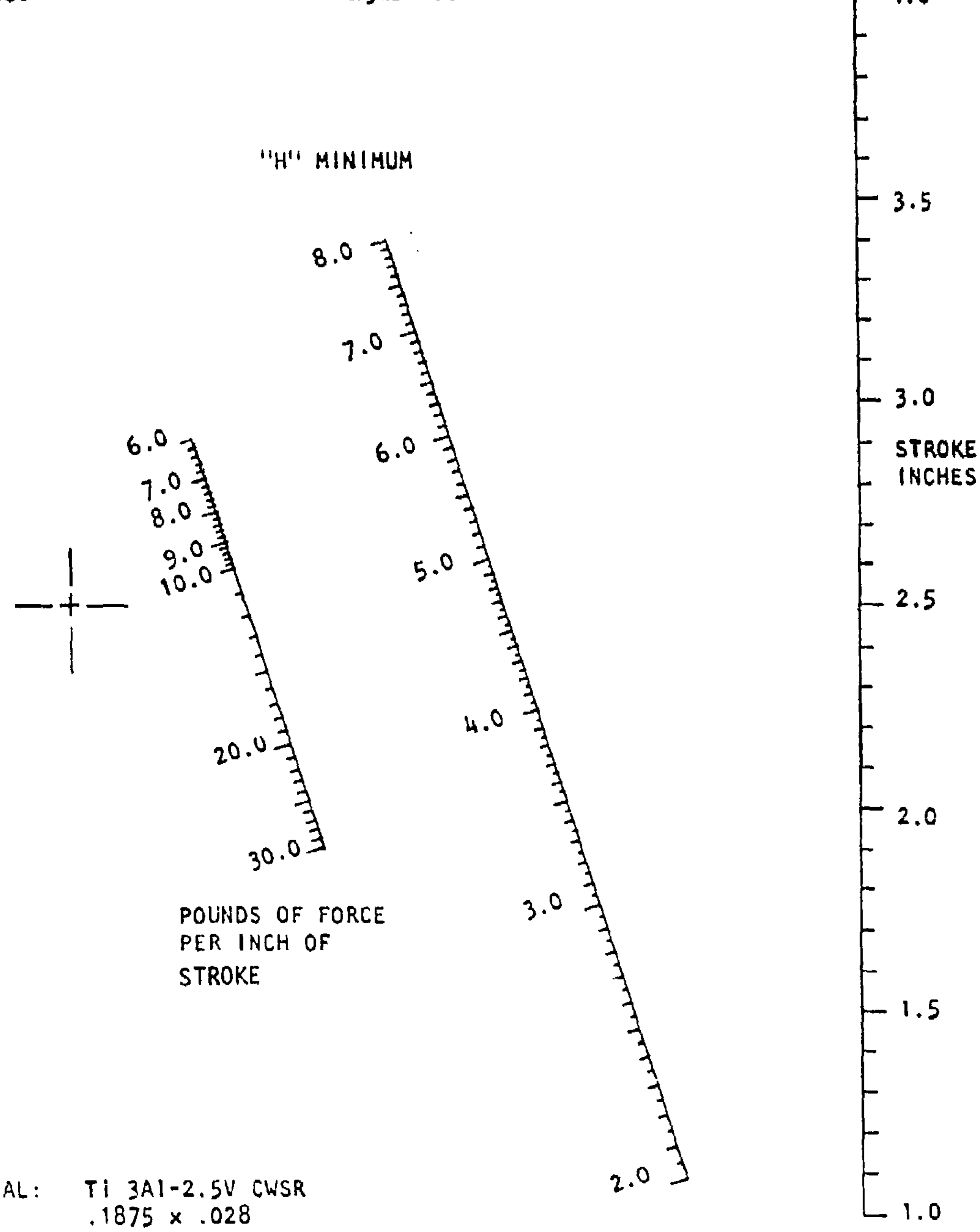
MATERIAL: Ti 3Al-2.5V CWSR
SIZE: .1875 x .020
PRESSURE: 1500 PSIG
D_m: 2.0 INCHES

Nomograph 13. Ti 3Al-2.5V CWSR, 1500 psig System, Tri-Coil Configuration



Nomograph 14. Ti 3Al-2.5V CWSR, 3000 psig System, Tri-Coil Configuration

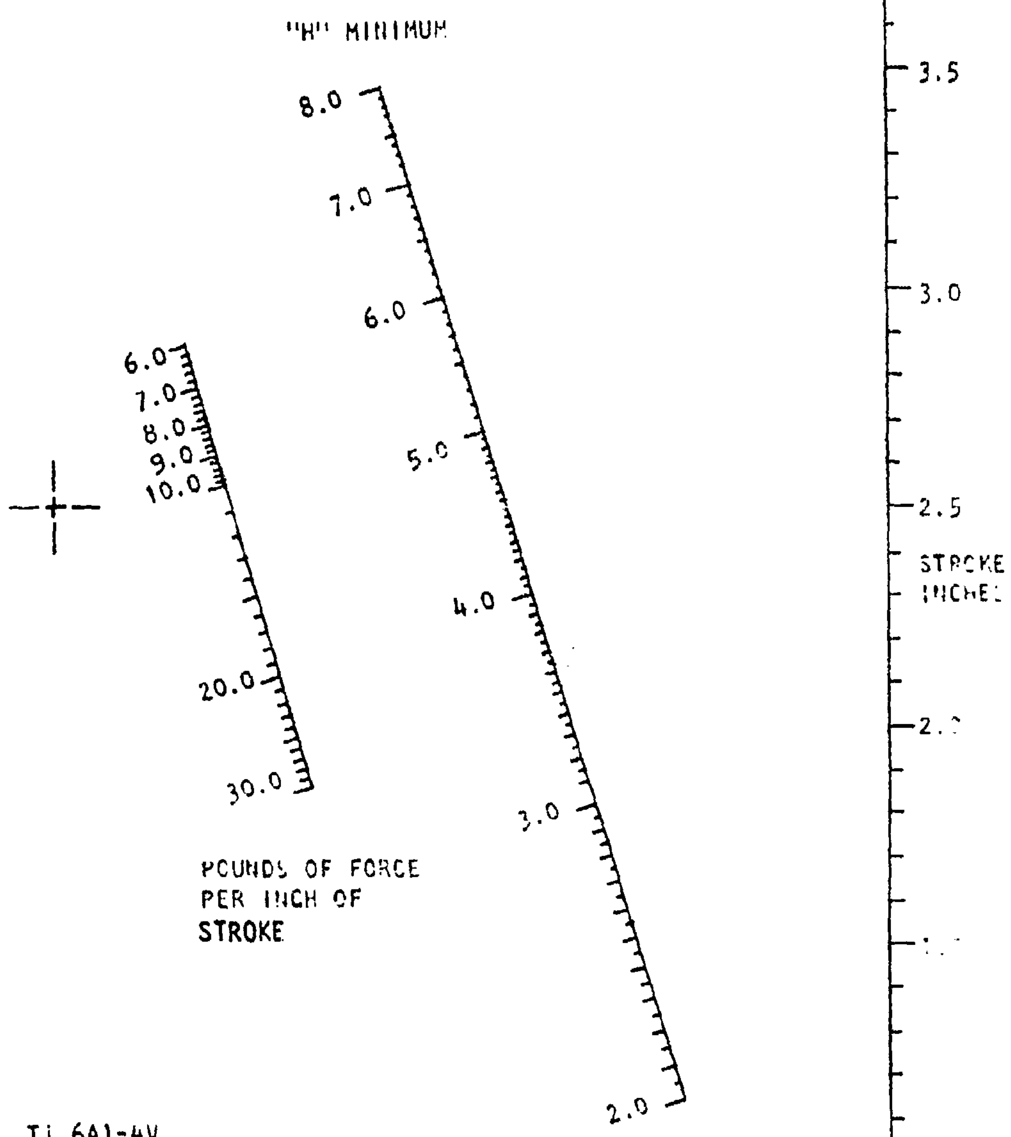
NOTE: "H" is defined on Figure 6.



MATERIAL: Ti 3Al-2.5V CWSR
SIZE: .1875 x .028
PRESSURE: 8000 PSIG
D_m: 2.0 INCHES

Nomograph 15. Ti 3Al-2.5V CWSR, 8000 psig System, Tri-Coil Configuration

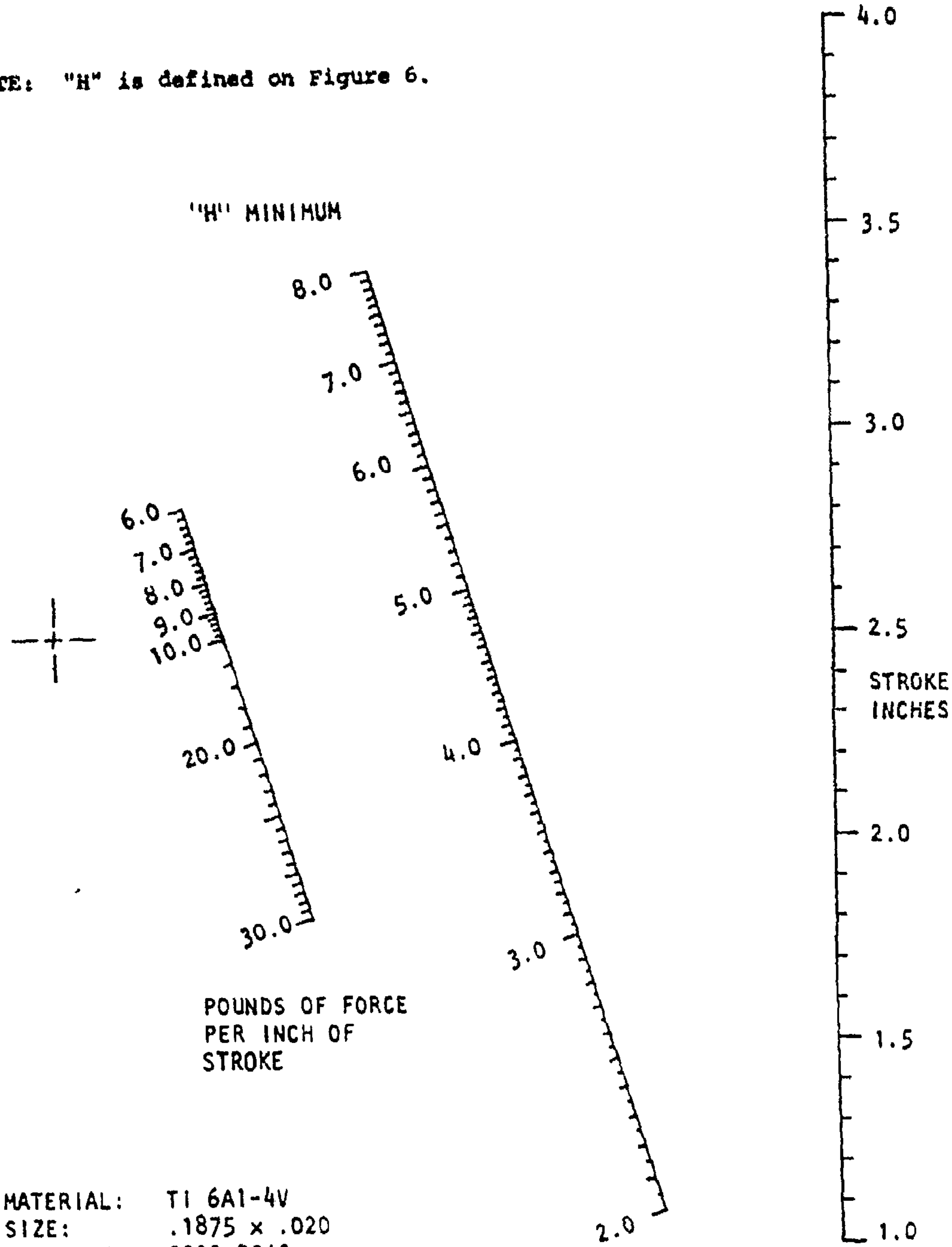
NOTE: "H" is defined on Figure 6.



MATERIAL:	Ti 6Al-4V
SIZE:	.1875 x .020
PRESSURE	1500 PSIG
D_m:	2.0 INCHES

Nomograph 16. Ti 6Al-4V, 1500 psig System, Tri-Coil Configuration

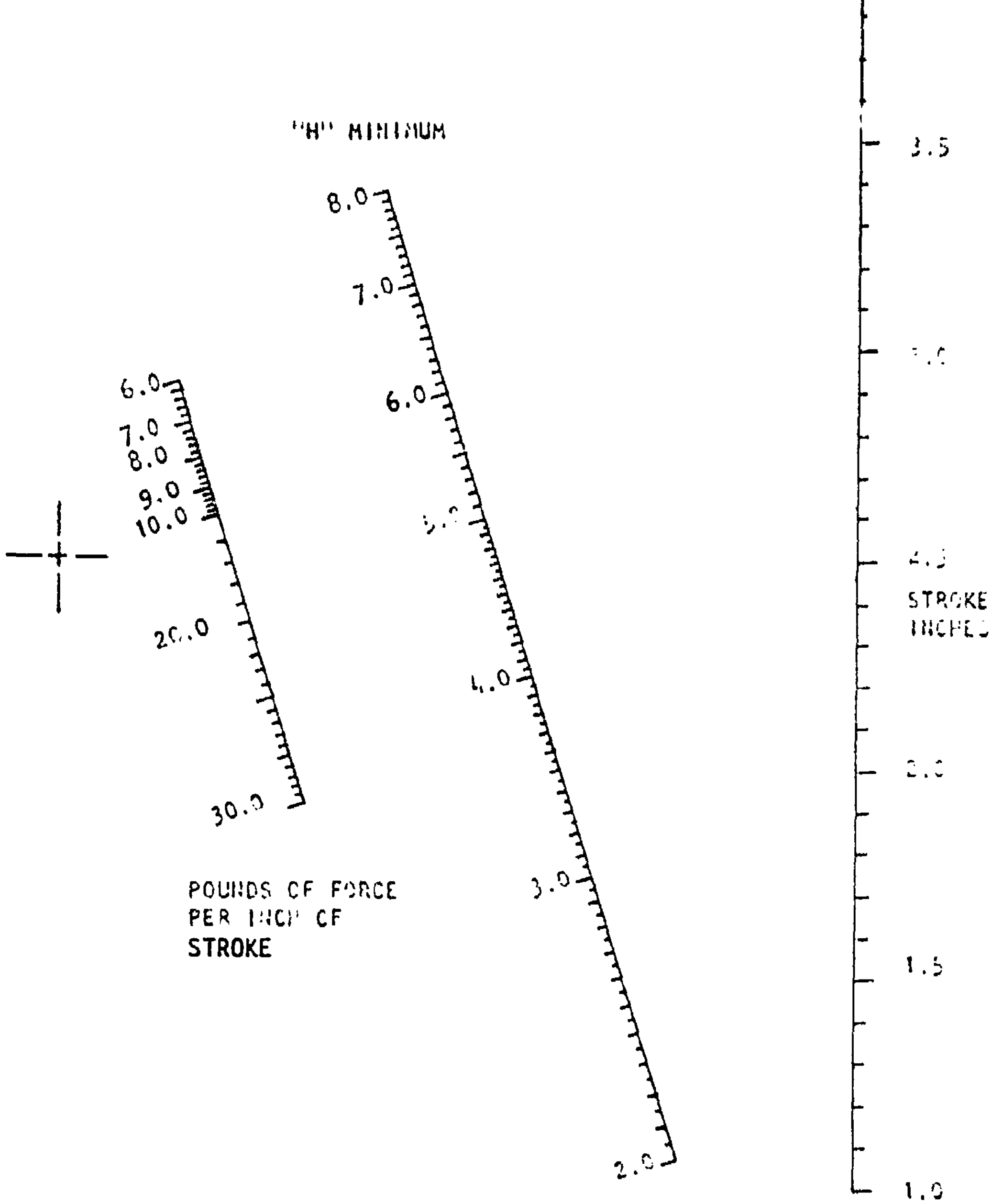
NOTE: "H" is defined on Figure 6.



MATERIAL: TI 6Al-4V
SIZE: .1875 x .020
PRESSURE: 3000 PSIG
D_m: 2.0 INCHES

Nomograph 17. Ti 6Al-4V, 3000 psig System, Tri-Coil Configuration

NOTE: "H" is defined on Figure 6.



MATERIAL: TI6Al-4V
SIZE: .1875 x .028
PRESSURE: 8000 PSIG
S_m: 2.0 INCHES

Nomograph 18. Ti 6Al-4V, 8000 psig System, Tri-Coil Configuration

6. ARBOR DESIGN

6.1 Tube Forming Tools. Two tube forming tools were designed and fabricated. Figure 7 shows drawing 83-00280. This is a tool for .1875 OD tubing. The minor diameter of the helix is 1.012 inches. This tool was used to fabricate the 2.50 inch mean diameter Ti 3Al-2.5V CWSR helical coiled tubes tested.

Figure 8 shows drawing 83-00281. This tool for .1875 diameter tubing has a minor diameter of the helix of 1.327 inches. This results in a 1.625 inch mean diameter helical coil for Ti 3Al-2.5V tubing.

6.2 Tool Design. The following formula was used to determine the minor diameter of the helix for the tube forming tools. Note that the minor diameter D_a is smaller than the coil ID to allow for springback of the material after the coil is formed.

$$D_a = \frac{1.02\ R}{[R/(D_s - R)] + [1.85\ F_{tu}/E]} - d_o$$

$$\text{where } R = .7071\ (d_o^2 + d_i^2)^{1/2}$$

D_a = ARBOR DIAMETER, INCHES

FTU = ULTIMATE TENSILE STRESS ALLOWABLE, PSI

E = MODULUS OF ELASTICITY, PSI

D_1 = TUBING O.D., INCHES

D_2 = TUBING I.D., INCHES

D_s = COIL O.D., INCHES

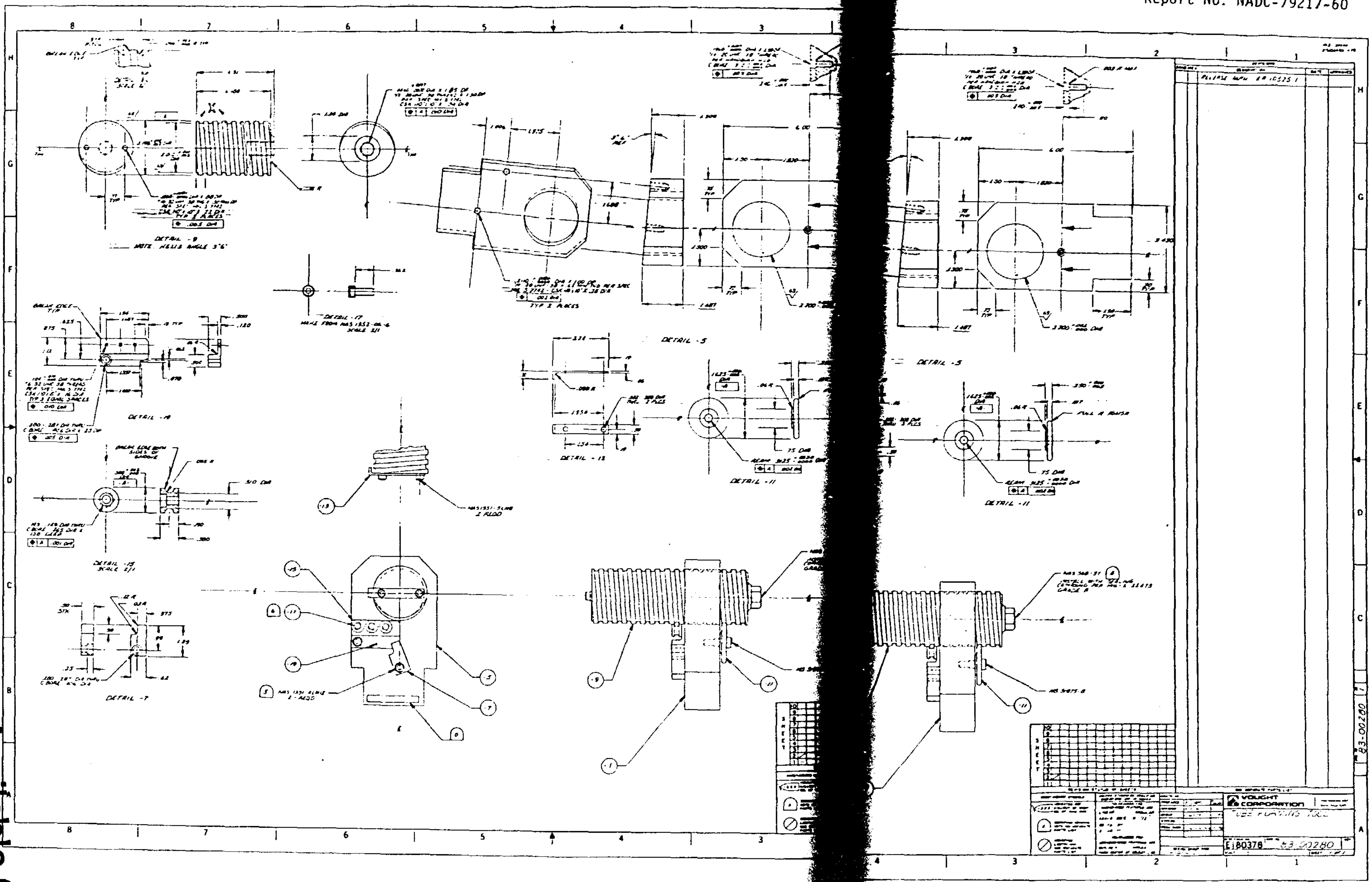


Figure 7. Tube Forming D
2.50 Mean Coil

Figure 7. Tube Forming Drawing 83-00280
2.50 Mean Coil Diameter

- 63 -

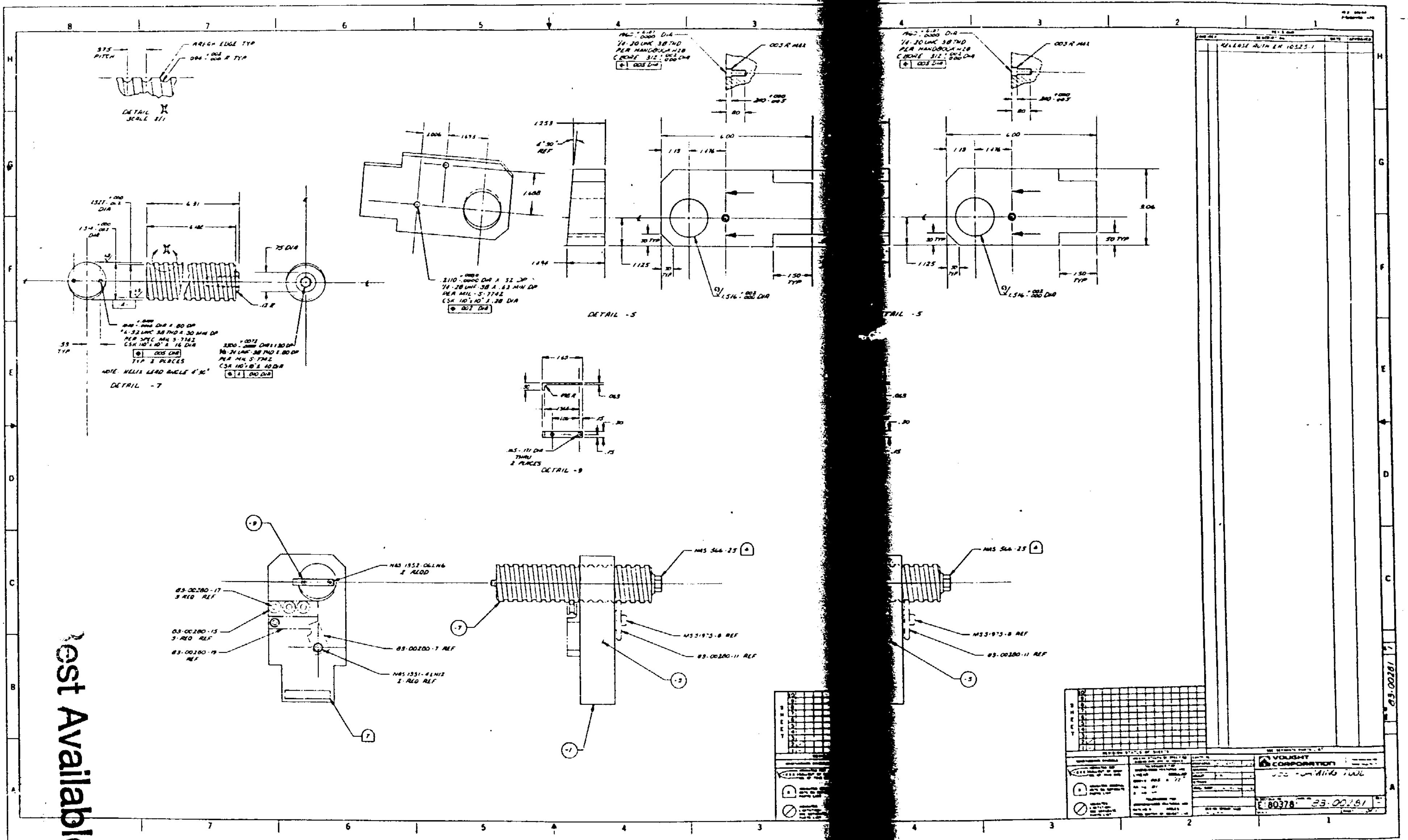


Figure 8. Tube Forming Tool
1.625 Mean Coil D

Figure 8. Tube Forming Tool, Drawing 83-00281
1.625 Mean Coil Diameter

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7. TEST PROGRAM

7.1 Endurance Test. A typical application of flexible plumbing using extension units was selected as the application for endurance tests. The A-7 aircraft roll feel isolation actuator installation was selected as the typical application. The installation is very complex and representative of high density flight control installation on modern fighter aircraft. Figure 9 shows a portion of drawing 215-38020 which is the production installation drawing for the roll feel isolation actuator. The extension units which were replaced by coiled tubing for this program are identified as 215-03003 Ref in zones C5 and C8 of the drawing. This installation was duplicated as to ground points, space available, and motion required, by the fixture shown on Figure 10. The tubes fabricated for the test are shown on Figure 11 which is drawing 83-00283 for the helical configuration and Figure 12 which is drawing 83-00284 for the tri-coil configuration on 83-00283, -1 tube assembly in the PC 1 pressure line; -3 tube assembly is the PC 1 return line. On 83-00284, -1 is the PC 2 pressure line and -5 is the PC 2 return line tested.

After installation the tubes were subjected to 100,000 cycles of $\pm$ 1.14 inches deflection. The tubes were pressurized to 3000 psi in the pressure lines and 100 psig on the return lines throughout the test. No failures or problems occurred.

In both the helical coil and the tri-coil design, the assemblies tested were designed for 3000 psig in the system pressure application and 1500 psig in the system return application. The design analysis for the 3000 psig helical coil required 3.3 coils, but this number of coils would not provide the amount of stroke required when the coils were wound with the pitch provided by the tube forming tools that had been fabricated. The number of coils in the 3000 psig pressure tube was increased to 6.7 coils to provide adequate stroke. The increase in the number of coils decreased the severity of the test below the stress level desired.

The other three tube assemblies survived fatigue damage equal to 5 x 10^6 endurance cycles using the spectrum of specification MIL-C-5503C.

7.2 Vibration Test without Damping. The test fixture was then placed on a vibration test table. Figure 13 shows the assembly oriented for vibration in the vertical axis. Accelerometers were fixed to the coils and fixture at the locations shown on Figures 14 thru 16 and subjected to the random vibration spectrum shown on Figures 35 and 36 of Appendix B, respectively, in the vertical axis and in the lateral axis. The lateral axis test is shown on the shaker in Figure 17. Figure 18 shows the accelerometer location on the helical configuration for the lateral axis test.

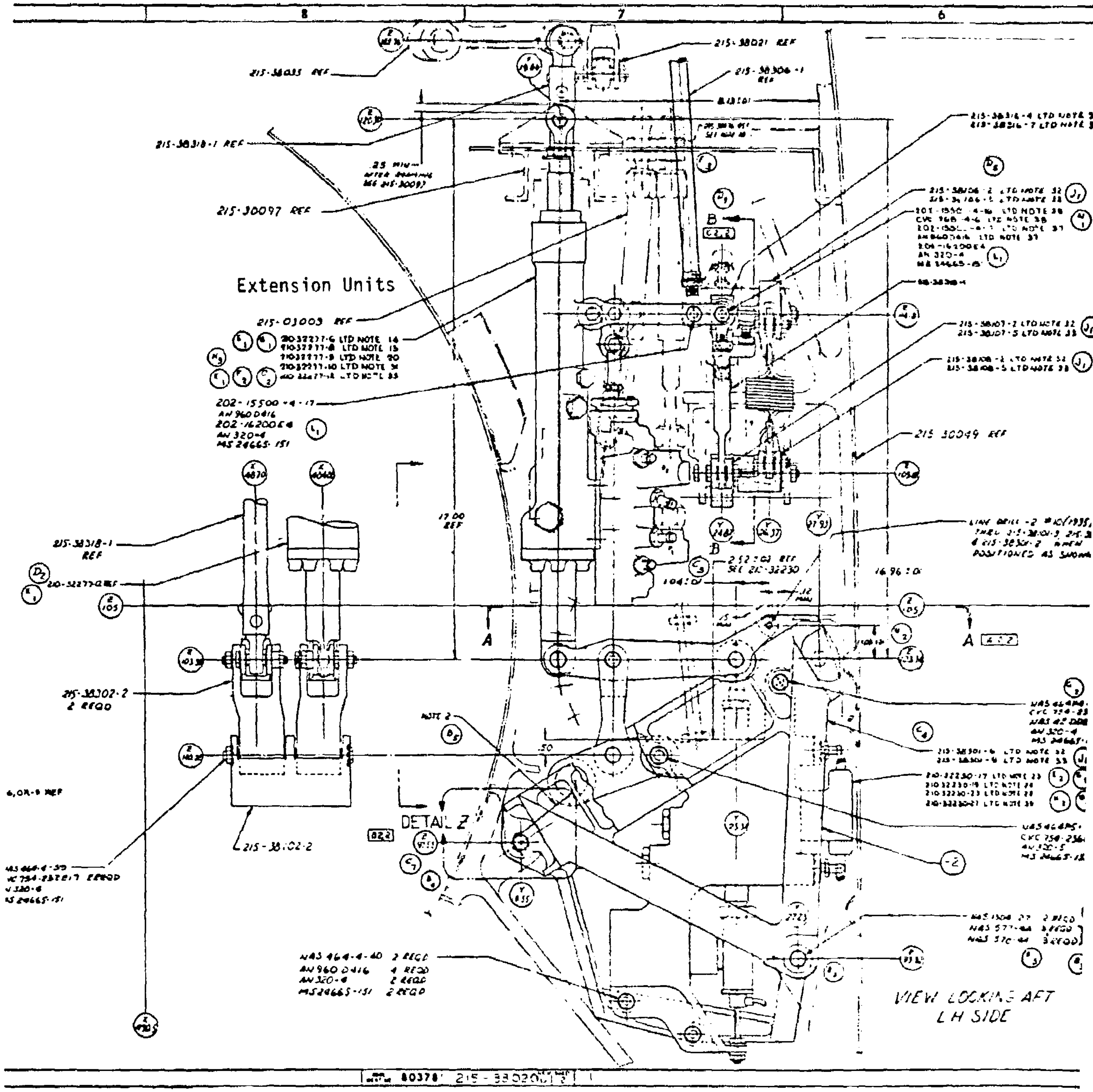
A plot of the transmissibility of the accelerometer on the tubing to the input accelerometer was obtained. This data is shown on Figure 37, Appendix B which shows a transmittal in the vertical axis of 400 at 80 Hz and 55 at 560 Hz for the helical tube. For the tri-coil configuration, Figure 38, Appendix B shows a transmittance of 100 to 150 from 20 Hz to 60 Hz in the vertical axis. Another peak of 100 occurs at 600 Hz. The peak at 1700 Hz is not considered important because of the very low amplitude at this frequency.

The lateral axis vibration test was conducted. Figure 39, Appendix B shows the amplitude ratio of the output to input for the helical configuration. A peak of 800 occurs at 40 Hz and another peak of 190 at 200 Hz. Figure 24 shows a peak of 1000 at 40 Hz for the tri-coil configuration and a peak of 20 at 300 Hz.

Table 5 summarizes the results of all vibration tests.

TABLE 5. SUMMARY OF VIBRATION TEST RESULTS

TUBE CONFIG	AXIS	MAXIMUM TRANSMISSIBILITY - T							
		TEST	FIGURE	T	Hz	T	Hz	T	Hz
Helical	Vertical	Undamped	37	400	80	10	160	55	560
		Damped No.1	43	6	80	3.2	180	0.3	600
		Damped No. 2	49	70	20	70	60	48	600
	Lateral	Undamped	39	800	40	190	200	--	--
		Damped No. 1	45	800	20	760	40	20	1700
		Damped No. 2	51	560	20	420	40	40	1010
Tri-Coil	Vertical	Undamped	38	100	40	160	60	100	600
		Damped No. 1	44	15	20	10	60	30	600
		Damped No. 2	50	96	58	101	160	--	--
	Lateral	Undamped	40	1000	40	20	300	30	600
		Damped No. 1	46	160	40	42	100	5	600
		Damped No. 2	52	200	20	240	40	48	340



Extension Units
215-38021 REF
215-38306-1 REF
215-38035 REF
215-38318-1 REF
215-30097 REF
215-03003 REF
202-15500-4-17
AN960D416
202-16200E4
AN320-4
MS24665-151
215-38318-1 REF
215-38302-2
2 REQD
215-38102-2
215-30049 REF
DETAIL Z
VIEW LOOKING AFT
LH SIDE
NAS 464-4-40 2 REQD
AN960D416 4 REQD
AN320-4 2 REQD
MS24665-151 2 REQD
80378 215-38020

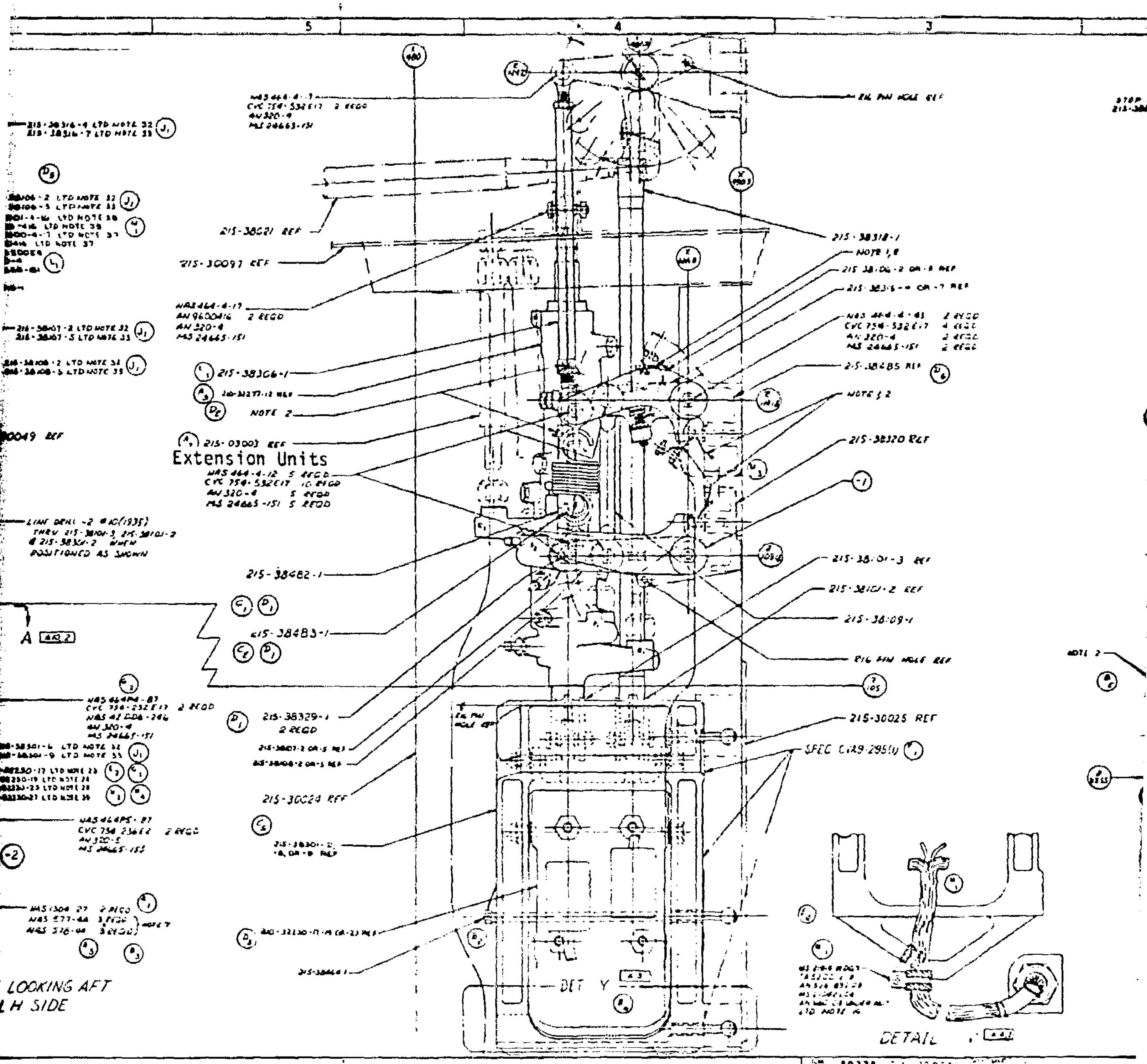


Figure 9. Installati
Roll Feel

12

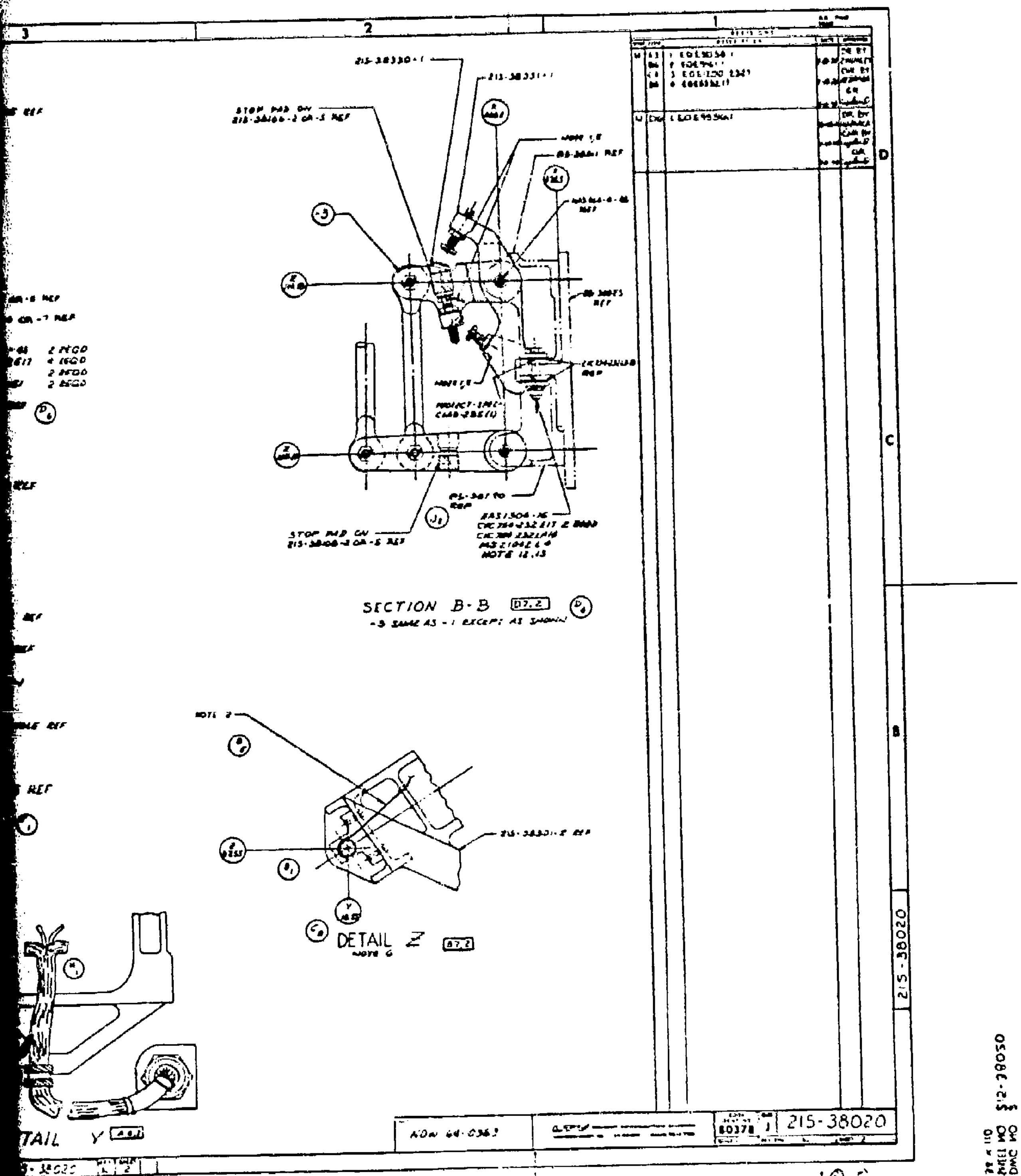
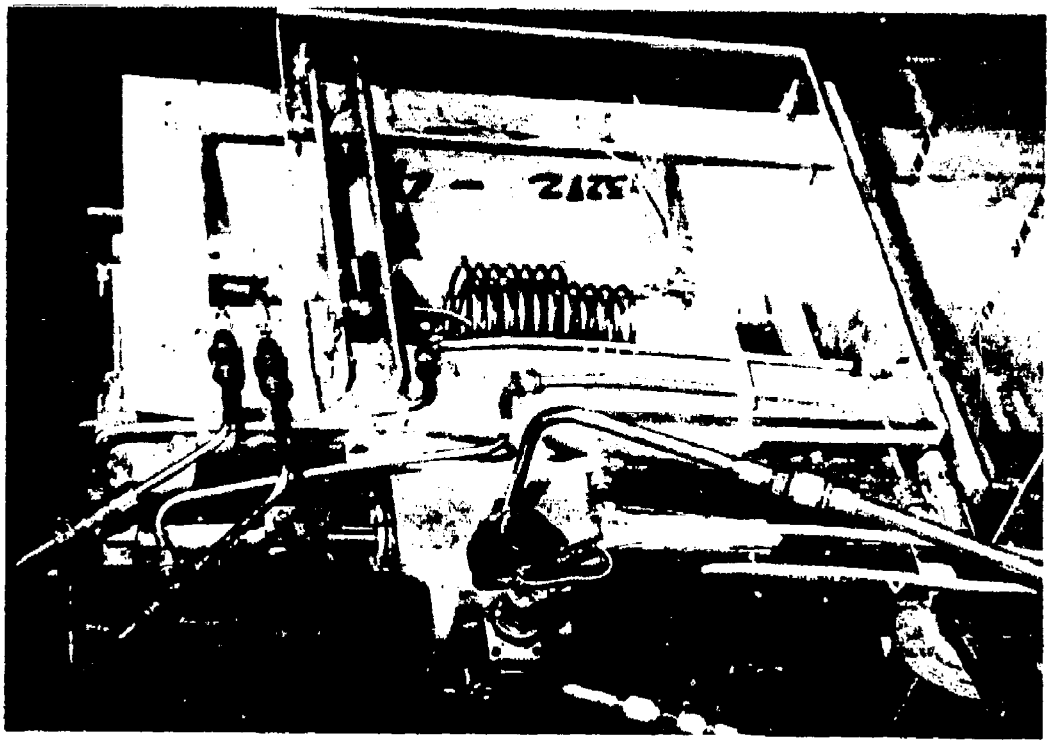
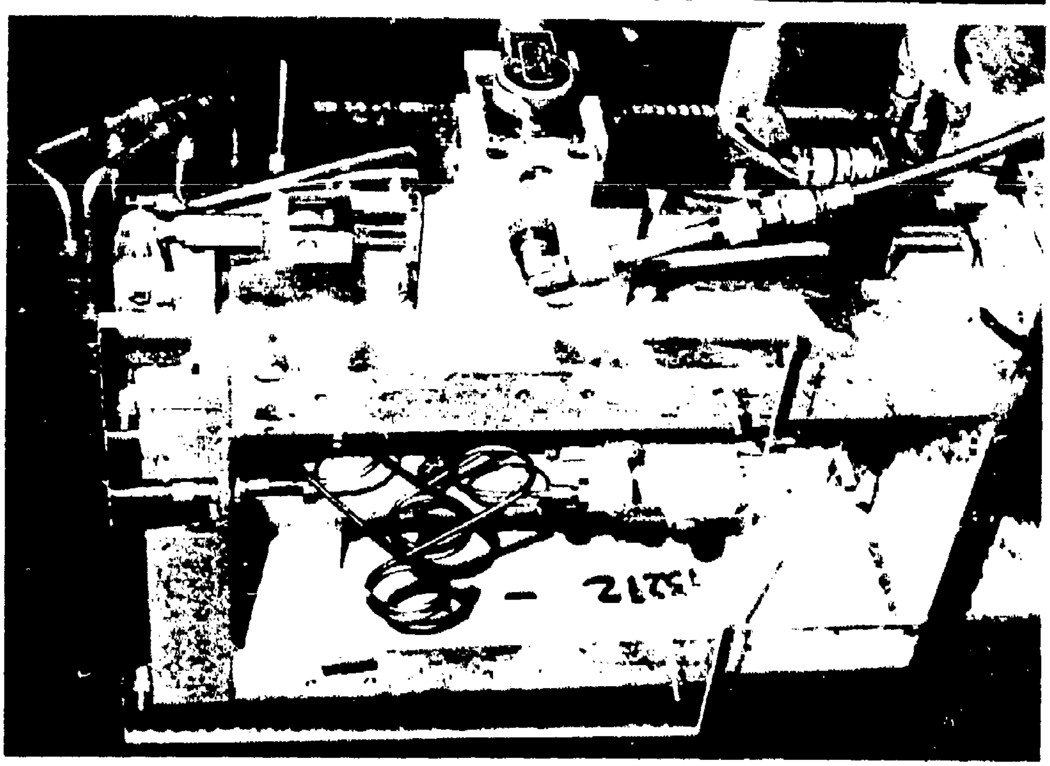


Figure 9. Installation Drawing, 215-38020,
Roll Feel Isolation Package, A-7 Aircraft





DWG NO. PL83-00283 SH 1

M CODE	MATERIAL/MATERIAL SPEC
M1	TITANIUM TUBING, 3AL-2.5V PER SPEC AMS 4943 (ANN)

F CODE	FINISH

REVISIONS

REV	DESCRIPTION	DATE	APPROVED

REVISIONS

REV	DESCRIPTION	DATE	APPROVED
	RLSE AUTH ER10561.1		

SHEET	20	19	18	17	16	15	14	13	12	11	10	9	8	7	6	5	4	3	2	1
REVISION STATUS OF SHEETS																		-	-	-

DASH NO.	USED ON	QUANTITY REQD NA	QUANTITY REQD FA	NEXT ASSEMBLY	FROM	THRU

CONTR NO	N62269-80-C-0205	
PREP	B. JAMES	
CHKD		
GROUP		
STRESS		
PROJ		
DESIGN GROUP NAME	HYDRAULICS	

VOUGHT CORPORATION Post Office Box 5907 Dallas, Texas 75222

COILED TUBE ASSEMBLIES

SIZE B | FSCM NO. 80378 | DWG NO. PL83-00283 | REV

SCALE NONE | SHEET 1 OF 3

Figure 11. Helical Configuration - Endurance and Vibration Test, Drawing 83-00283

DWG NO. PL 83-00283 SH 2

NOTES –

1. THE -5 TUBING COIL WILL BE FABRICATED USING THE 83-00280 TUBE FORMING TOOL. THE -5 TUBING COIL WILL HAVE APPROXIMATELY 5 COILS AS SHOWN.

2. THE -7 TUBING COIL WILL BE FABRICATED USING THE 83-00281 TUBE FORMING TOOL. THE -7 TUBING COIL WILL HAVE APPROXIMATELY 9 COILS AS SHOWN.

3. TUBE ASSEMBLIES ARE SHOWN IN THE RELAXED POSITION. SPRINGBACK ALLOWANCE IS INCLUDED ONLY IN THE COILED AREAS. SOME COMPRESSION WILL BE REQUIRED TO INSTALL TUBES IN THE RETRACTED POSITION.

4. ASSEMBLE AND INSTALL TUBES, SLEEVES AND COUPLINGS PER SPEC CVA 12-182.

5. CLEAN AND PROTECT THE COMPLETED TUBE ASSEMBLIES PER SPEC CVA 12-171.

6. APPLY CVC3016 IDENT TAPE IN ACCORDANCE WITH CVA9-50 IN APPROXIMATE POSITION SHOWN.

7. GENERAL PROTECTIVE SPEC 208-9-220.

8. FOR DEFINITION OF PROTECTIVE FINISH CODE NUMBERS AND APPLICABLE GOVERNMENT SPECIFICATIONS SEE CVA FB-21.

9. LIMITATIONS ON FLATNESS, WRINKLES, SCRATCHES PER MS33611.

10. WORKING PRESSURE IS 3000 PSI FOR -1 AND 1500 PSI FOR -3.

VOUGHT CORPORATION	SIZE B	FSCM NO. 80378	DWG NO. PL 83-00283	REV
	SCALE NONE		SHEET 2	

Figure 11. Helical Configuration - Endurance and Vibration Test (Continued)

DWG NO. PL 83-00283 SH 3

QUANTITY REQUIRED						FIND NO./REF DES	ZONE	FSCM NO	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATL CODE	STOCK SIZE	FIN CODE	NOTE	UNIT WEIGHT
				-3	-1										
					—				-1	ASSEMBLY, COILED TUBE					
				—					-3	ASSEMBLY, COILED TUBE					
					1				-5	TUBING	M1	3/16 x .025 x 60.0			
				1					-7	TUBING	M1	3/16 x .025 x 66.0			
				2	2			96906	MS21921 3J	NUT, SLEEVE COUPLING					
				2	2			96906	MS21922 3	SLEEVE, COUPLING					
				X	X				CVA 12 171	SPECIFICATION					
				X	X				CVA 12 182	SPECIFICATION					
				X	X				CVA9-50	SPECIFICATION					
				2					CVC3016-50	IDENT TAPE				6	
					2				CVC3016-51	IDENT TAPE				6	

VOUGHT CORPORATION

SIZE	FSCM NO.	DWG NO.	REV
B	80378	PL 83-00283	
SCALE NONE		SHEET 3 OF 3	

Figure 11. Helical Configuration - Endurance and Vibration Test (Continued)

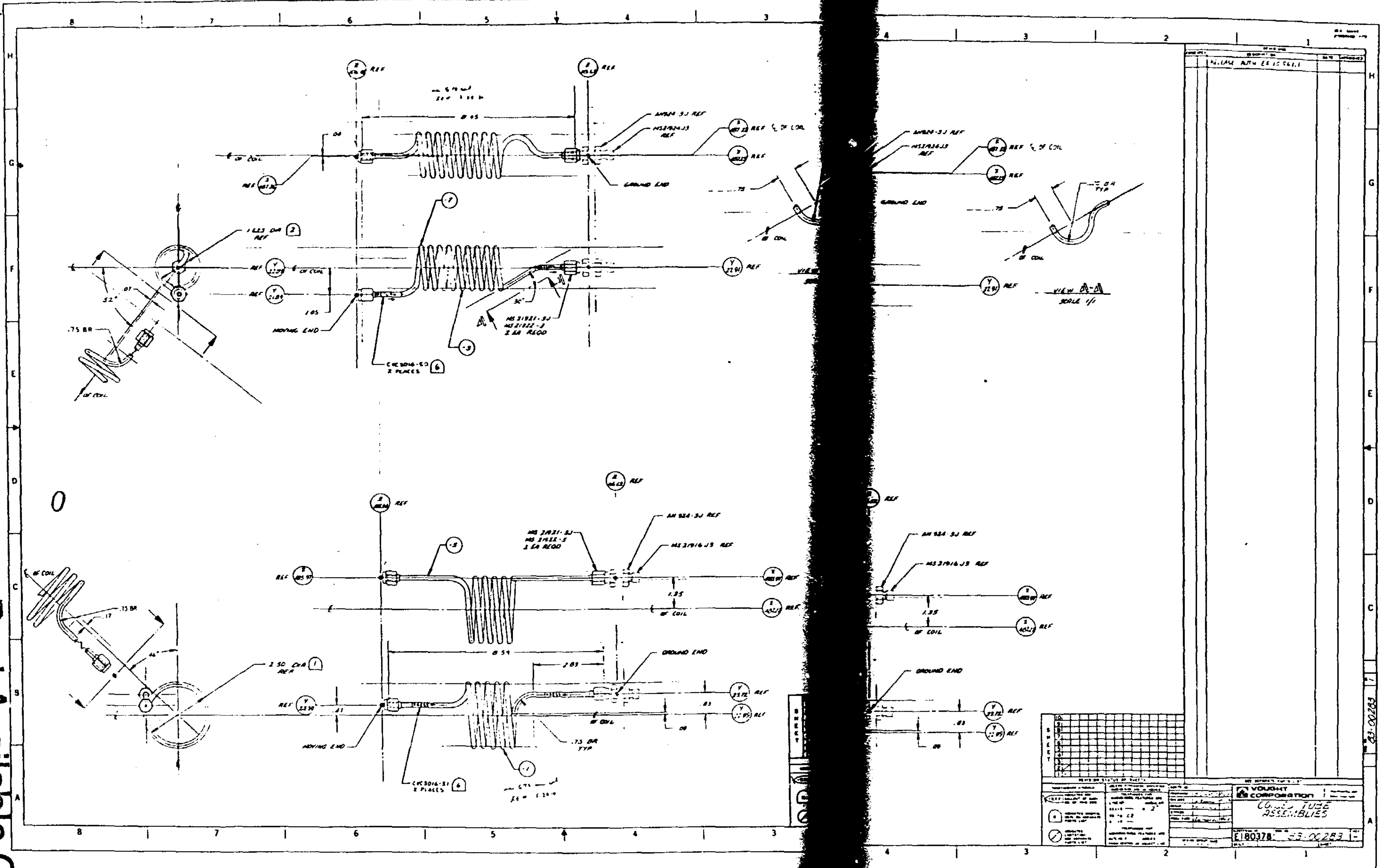


Figure 11. Helical Configuratio

Helical Configuration - Endurance and Vibration Test
(Continued)

- 73 -

DWG NO. PL 83-00284 SH 1

M CODE	MATERIAL/MATERIAL SPEC
M1	TITANIUM TUBING, 3AL-2.5V PER SPEC AMS 4944. (CWSR)

CODE	FINISH

REVISIONS

REV	DESCRIPTION	DATE	APPROVED

REVISIONS

REV	DESCRIPTION	DATE	APPROVED
	RLSE AUTH ER 1056.1		

SHEET							
20							
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2	-						
1	-						

REVISION STATUS OF SHEETS

DASH NO.	USED ON	QUANTITY REQD MA	QUANTITY REQD FA	NEXT ASSEMBLY	FROM	THRU

CONTR NO		
PREP	BOB JAMES	
CHKD		
GROUP		
STRESS		
PROJ		

DESIGN GROUP NAME HYDRAULICS

VOUGHT CORPORATION

COILED TUBE ASSEMBLIES

SIZE B | FSCM NO. 80378 | DWG NO. PL 83-00284 | REV

SCALE NONE | SHEET 1 OF 3

Figure 12. Tri-Coil Configuration - Endurance and Vibration Test, Drawing 83-00284

DWG NO. PL83-00284 SH 2

N

1. THE TUBING ASSEMBLIES SHOWN ON 83-00284 ARE IN THE RELAXED POSITION SPRINGBACK ALLOWANCE IS NOT INCLUDED.

2. THE 1.04 DIMENSION INCLUDES .42 FOR THE MATING BULKHEAD FITTING PLUS .62 FOR STRETCH TO THE NEUTRAL POSITION

3. THE 1.07 DIMENSION INCLUDES .45 FOR THE MATING FITTING PLUS .62 FOR STRETCH TO THE NEUTRAL POSITION.

4. ASSEMBLE AND INSTALL TUBES, SLEEVES AND COUPLINGS PER SPEC CVA 12-182

5. CLEAN AND PROTECT THE COMPLETED TUBE ASSEMBLIES PER SPEC CVA 12-171.

6. APPLY CVL3016 IDENT TAPE IN ACCORDANCE WITH CVA9-50 IN APPROXIMATE POSITION SHOWN.

7. GENERAL PROTECTIVE SPEC 208-9-220.

8. FOR DEFINITION OF PROTECTIVE FINISH CODE NUMBERS AND APPLICABLE GOVERNMENT SPECIFICATIONS SEE CVA TB-21.

9. LIMITATIONS ON FLATNESS, WRINKLES & SCRATCHES PER MS33611.

10. WORKING PRESSURE IS 3000 PSI FOR -1&-3 AND 1500 PSI FOR -5&-7.

VOUGHT CORPORATION P.O. Box 225907 Dallas TX 75265	SIZE B	FSCM NO. 80378	DWG NO. PL83-00284	REV
	SCALE NONE		SHEET 2	

Figure 12. Tri-Coil Configuration - Endurance and Vibration Test (Continued)

DWG NO. 83-00284 SH 3

QUANTITY REQUIRED -7	-5	-3	-1	FIND NO./ REF DES	ZONE	FSCM NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATL CODE	STOCK SIZE	FIN CODE	NOTE	UNIT WEIGHT
			1				-1	COILED TUBE ASSEMBLY					
		1					-3	COILED TUBE ASSEMBLY					
	1						-5	COILED TUBE ASSEMBLY					
1							-7	COILED TUBE ASSEMBLY					
			1				-11	TUBING	M1	3/16 X .035 X 43.0			
		1					-13	TUBING	M1	3/16 X .035 X 46.0			
	1						-15	TUBING	M1	3/16 X .035 X 42.0			
1							-17	TUBING	M1	3/16 X .035 X 44.0			
2	2	2	2			96906	MS21921-3J	NUT, SLEEVE COUPLING					
2	2	2	2			96906	MS21922-3	SLEEVE, COUPLING					
X	X	X	X				CVA9-50	SPECIFICATION					
X	X	X	X				CVA 12-171	SPECIFICATION					
X	X	X	X				CVA 12-182	SPECIFICATION					
2	2						CVC3016-50	IDENT TAPE				6	
		2	2				CVC3016-51	IDENT TAPE				6	

VOUGHT CORPORATION Post Office Box 5907 Dallas Texas 75222 | SIZE B | FSCM NO. 80378 | DWG NO. PL 83-00284 | REV | SCALE NONE | SHEET 3

Figure 12. Tri-Coil Configuration - Endurance and Vibration Test (Continued)

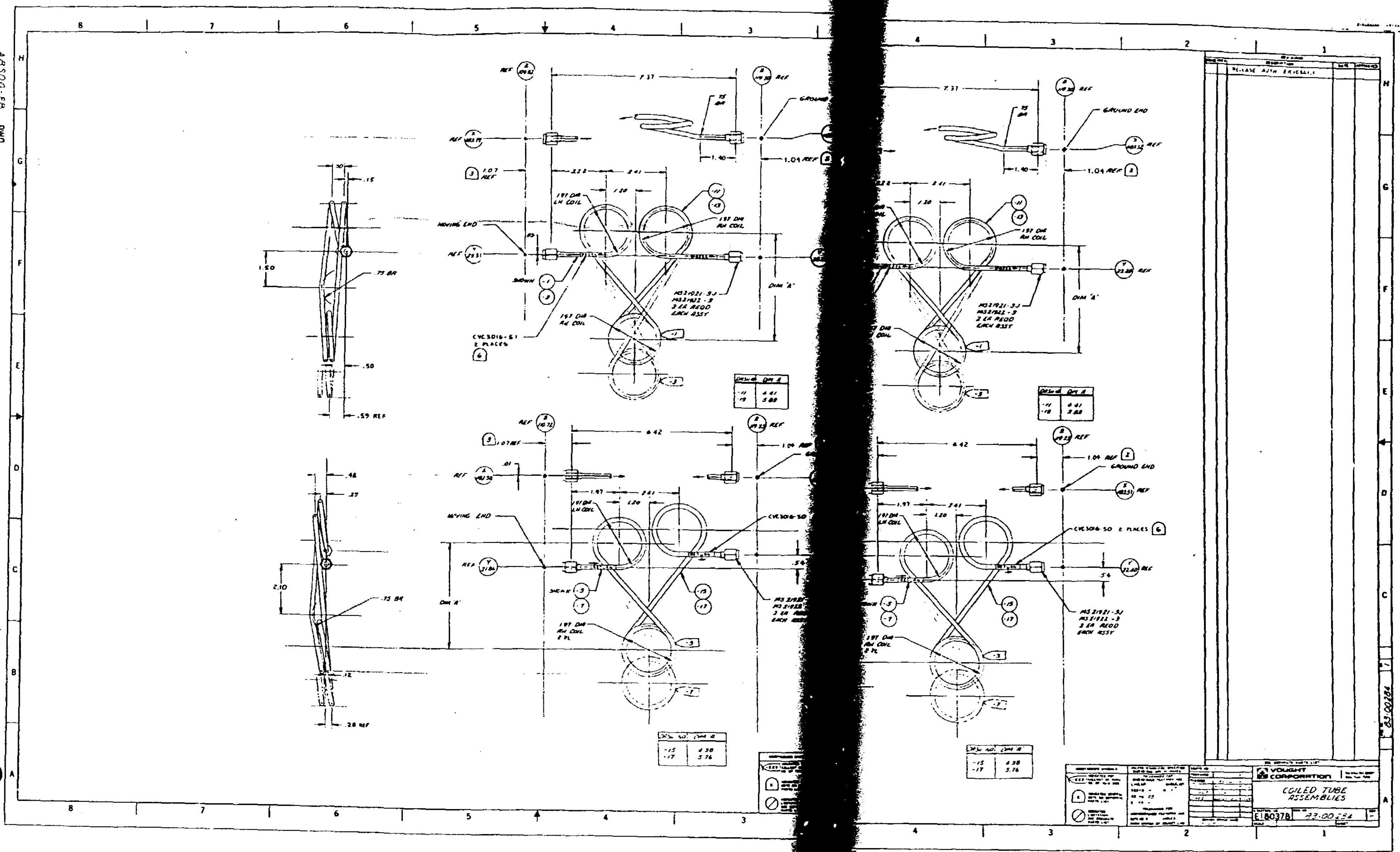

Figure 12. Tri-Coil Configuration - Endurance and Vibration Test (Continued)

Figure 12. Tri-Coil Configuration - Enduranc (Continued)

- 77 -

12

DWG SH 34 X 44 83-00284

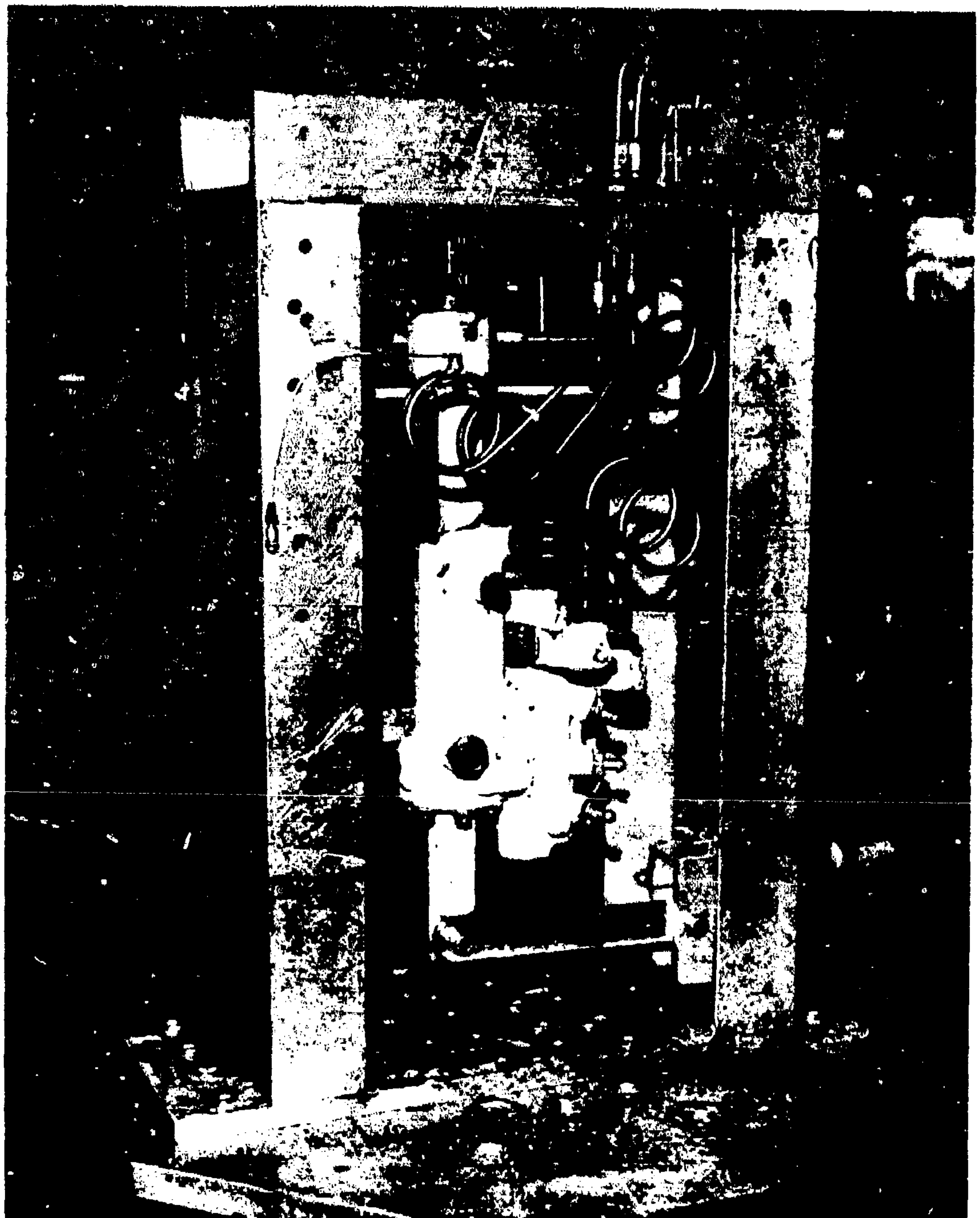

Figure 15. Test Fixture Orientation for Vibration Test in Vertical Axis

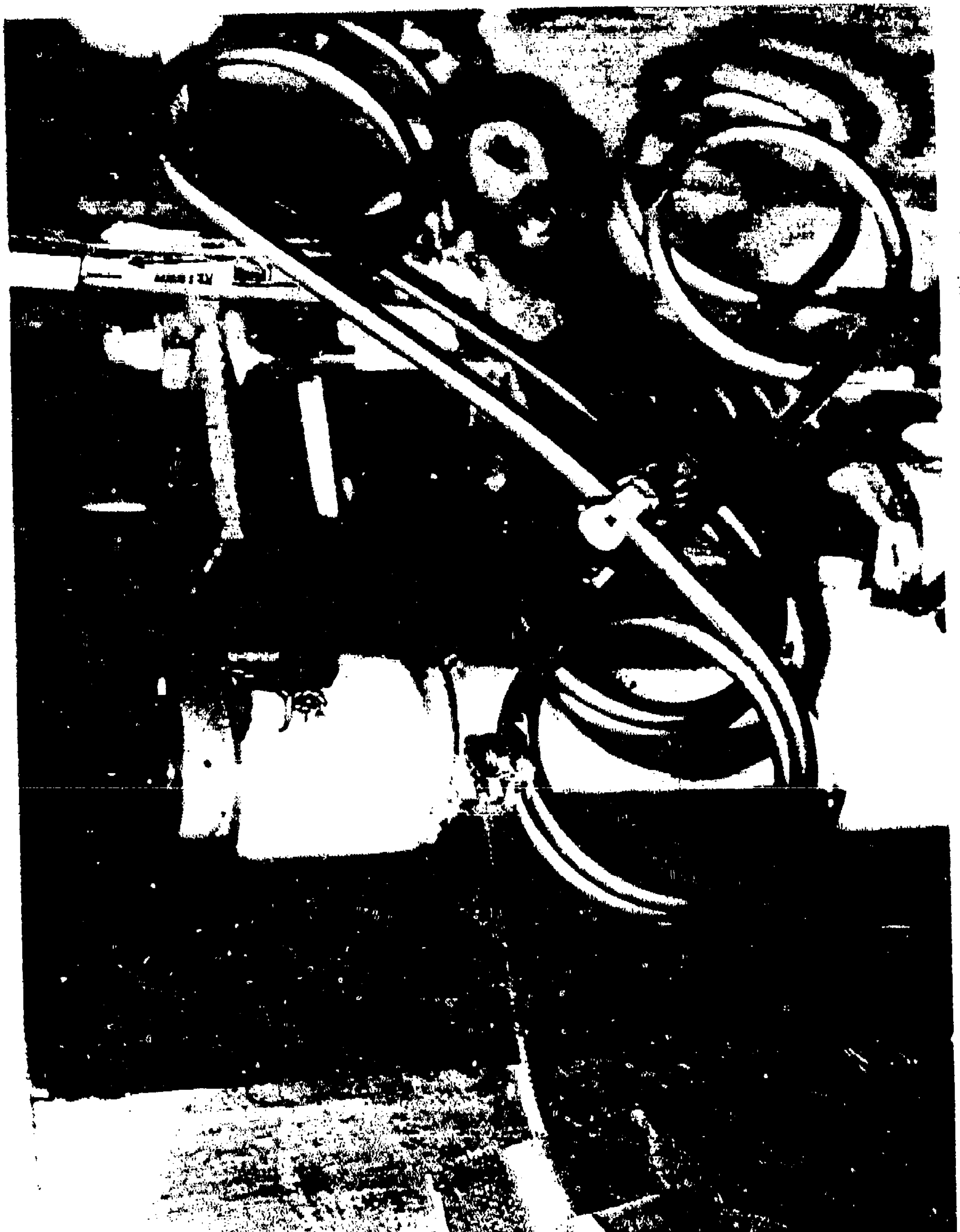

Figure 14. Tri-Coil Configuration with Accelerometer Mounted - Vibration Test, Vertical Axis. Note Tygon tubing and wire used for damped vibration test no. 1

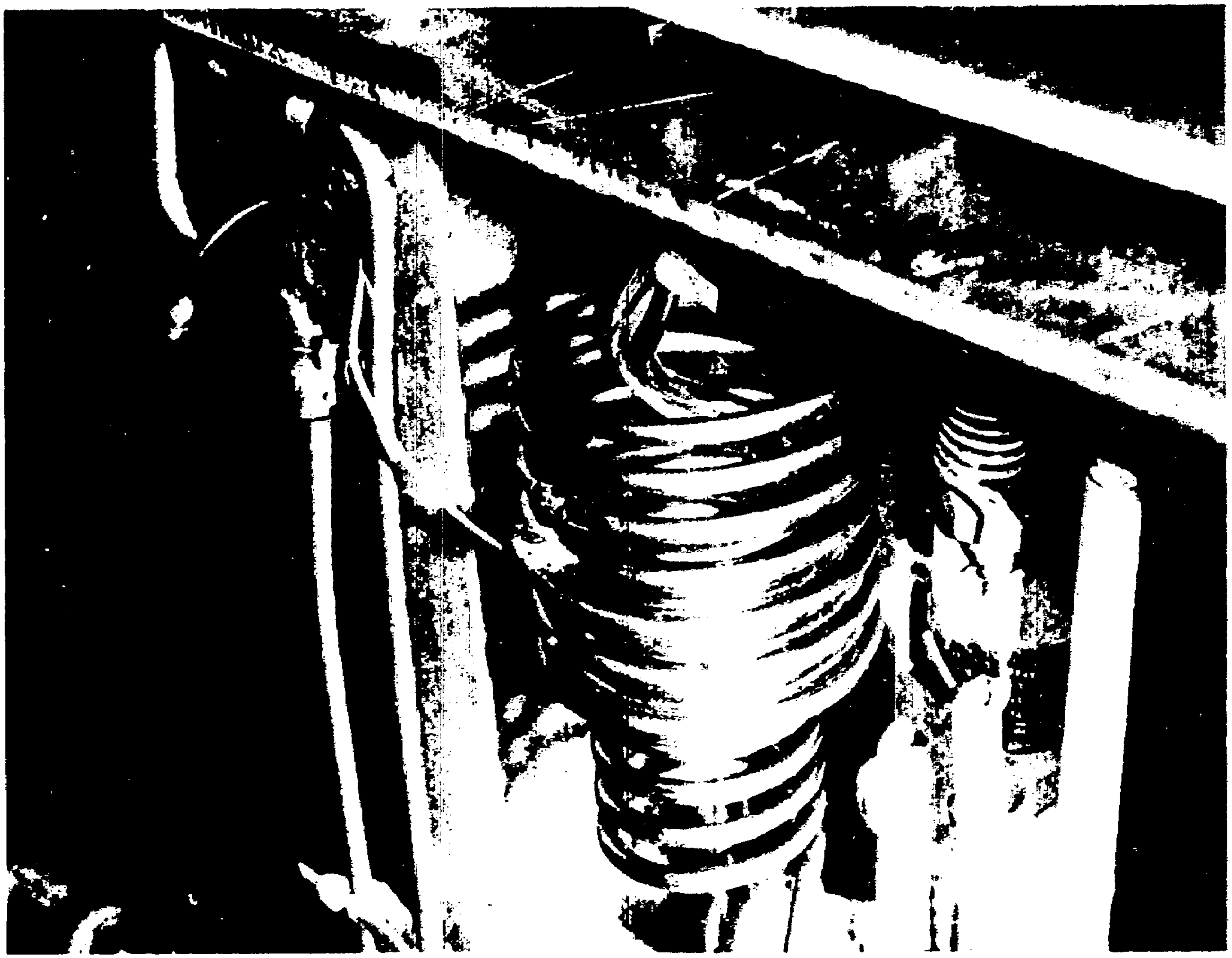

Figure 15. Helical Configuration with accelerometer mounted – [illegible] Test, Vertical [illegible] [illegible] used for [illegible] vibration test [illegible]

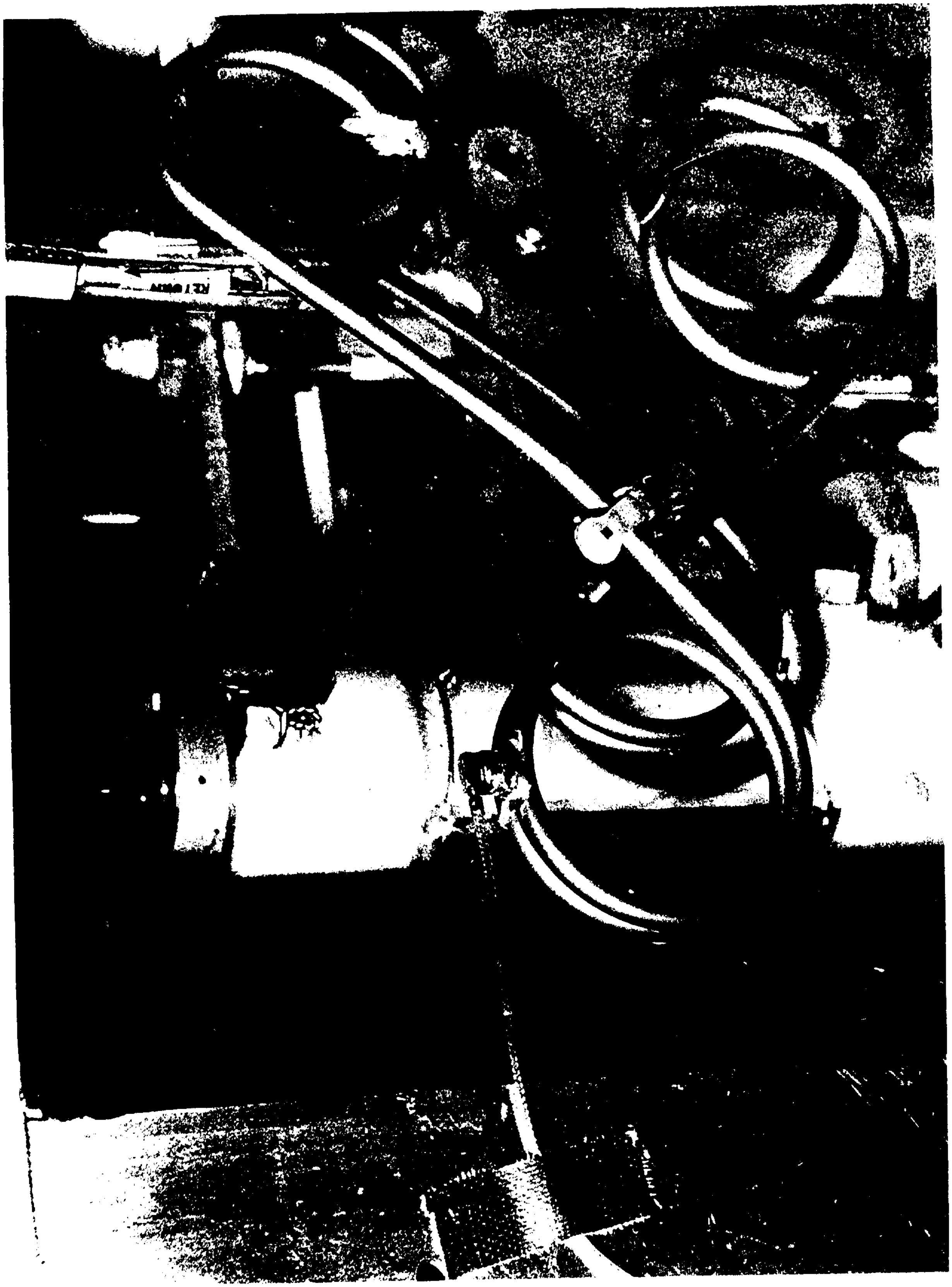

Figure 14. Tri-Coil Configuration with Accelerometer Mounted - Vibration Test, Vertical Axis. Note Tygon tubing and wire used for damped vibration test no. 1

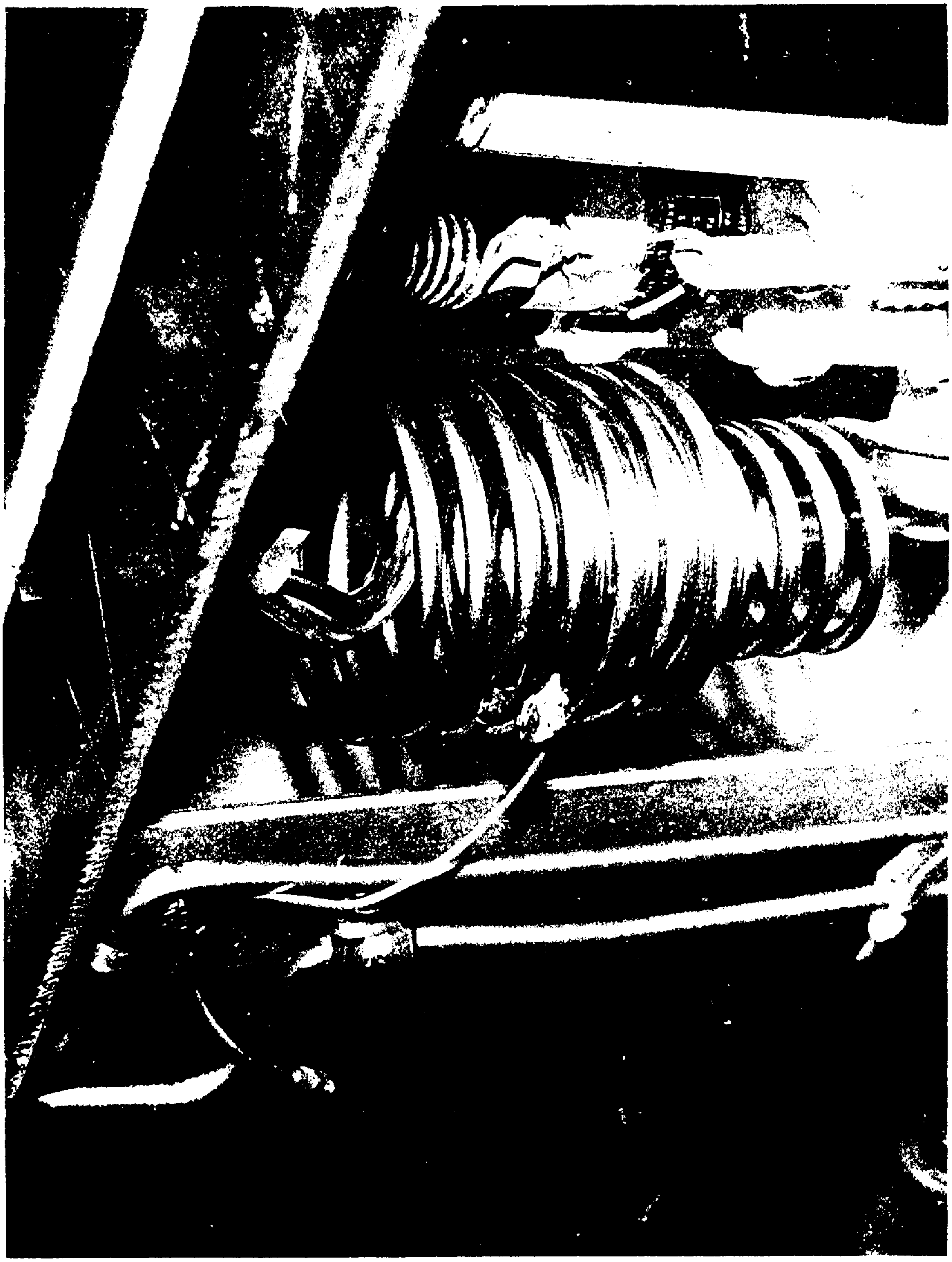

Figure 15. Helical Configuration with Accelerometer Mounted - Vibration Test, Vertical Axis. Note Tygon tubing used for damped vibration test no. 1

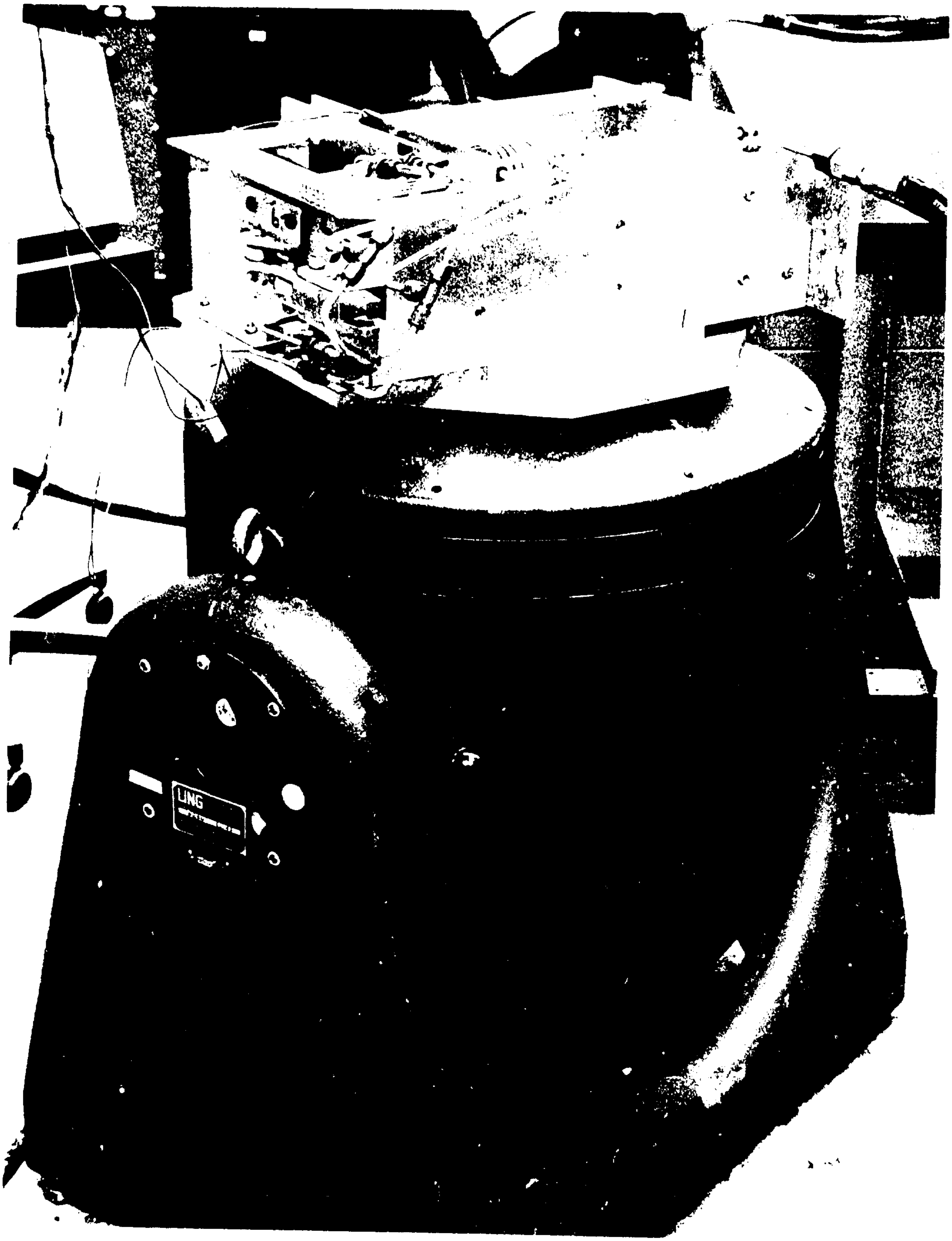

Figure 17. Test Fixture Orientation for Vibration in Lateral Axis

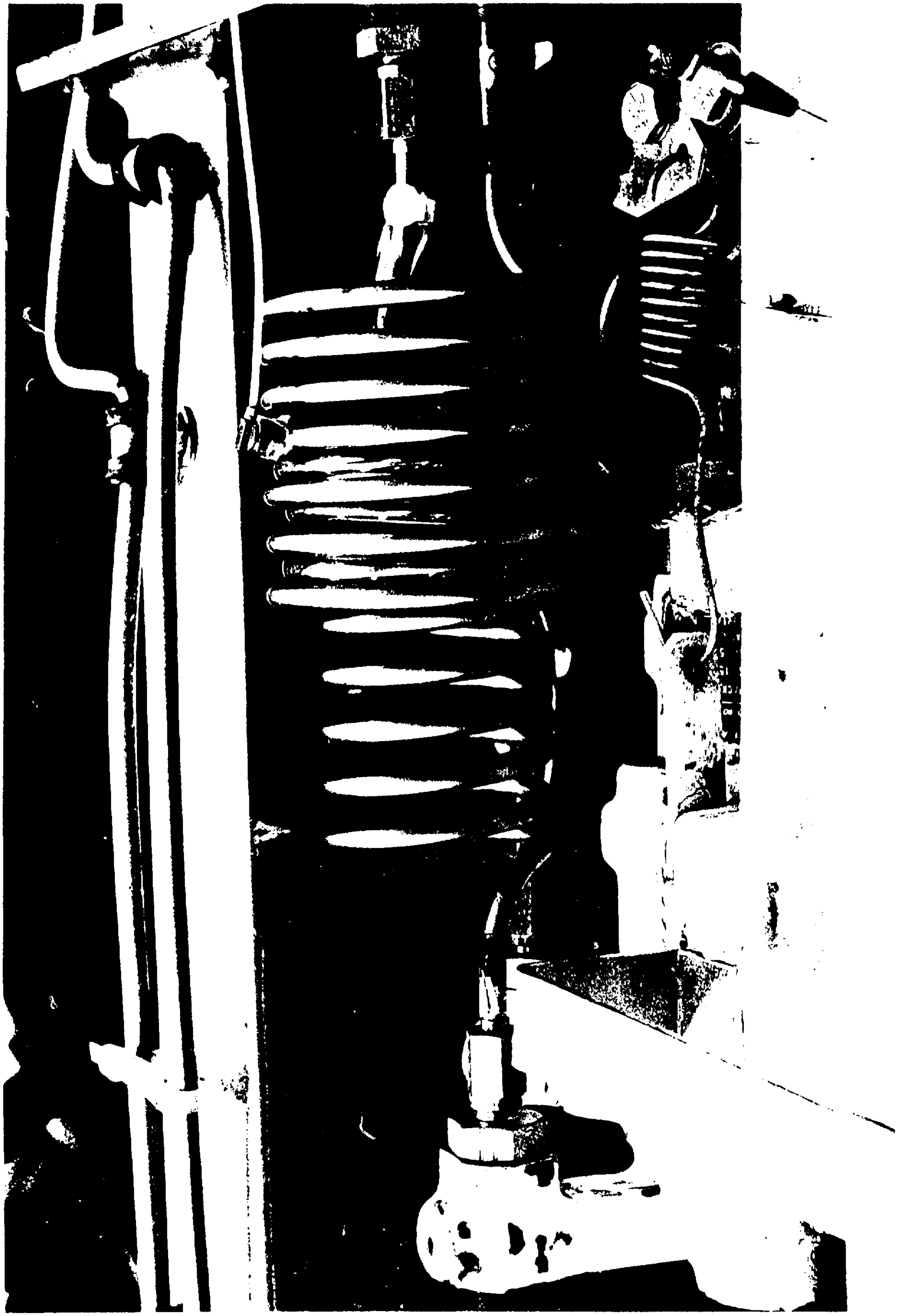

Figure 18. Helical Configuration with Accelerometer Mounted - Vibration Test, Lateral Axis. Note Tygon tubing used for damped test No. 1

7.3 Vibration Tests with Damping

7.3.1 Damped Vibration Test No. 1. As shown on Figure 15 the helical coil was configured with damping method B (Figure 22). The tri-coil configuration was damped with a simple flexible tie between the center coil of one tube assembly and one leg of the other tube assembly as shown on Figure 14. The tie consisted of a length of Tygon tubing which fit exactly between the tube assemblies with a wire bundle tie looped around the tube assemblies and passed through the Tygon tube.

The assembly was subjected to the random vibration input shown on Figure 41 and 42 in Appendix B, respectively for the vertical and lateral axes. The records of transmissibility are shown on Figures 43 and 44 of Appendix B for the vertical axis and Figures 45 and 46 of Appendix B for the lateral axis.

7.3.2 Damped Vibration Test No. 2. The helical coil was fitted with Damping Method A (Figure 21) for the second damped vibration test and is shown on Figure 19. The tri-coil configuration was fitted with a triangular sheet of polyurethane which was secured between the two tri-coil assemblies as shown on Figure 20. In practice this method could probably be reduced to a foam "sandwich" occupying only the immediate space between coils as shown on Figure 25.

The assembly was subjected to the random vibration input shown on Figures 47 and 48 of Appendix B, respectively for the vertical and lateral axes. The records of transmissibility from 20 to 2000 Hz are shown on Figures 49 and 50 of Appendix B for the vertical axis and Figures 51 and 52 of Appendix B for the lateral axis.

7.3.3 Results of Damped Vibration Tests for Helical Configuration. Damping Method B worked well in the vertical axis across the frequency range. Neither method produced a dramatic attenuation in the lateral axis. Damping Method A did work the best of the two methods. Method B would have performed better with more interference between the helical coils and the Tygon tubing. Both methods changed the frequency at which maximum transmissibility occurred in the lateral axis.

7.3.4 Results of Damped Vibration Tests for Tri-coil Configuration. The simple tie between tubes worked well on Damping Test No. 1 in the vertical axis and made a significant reduction in the 40 Hz transmissibility in the lateral axis. On Damped Test No. 2, the foam "sandwich" appears to have lowered the frequencies at which maximum transmissibility occurred in the lateral axis.

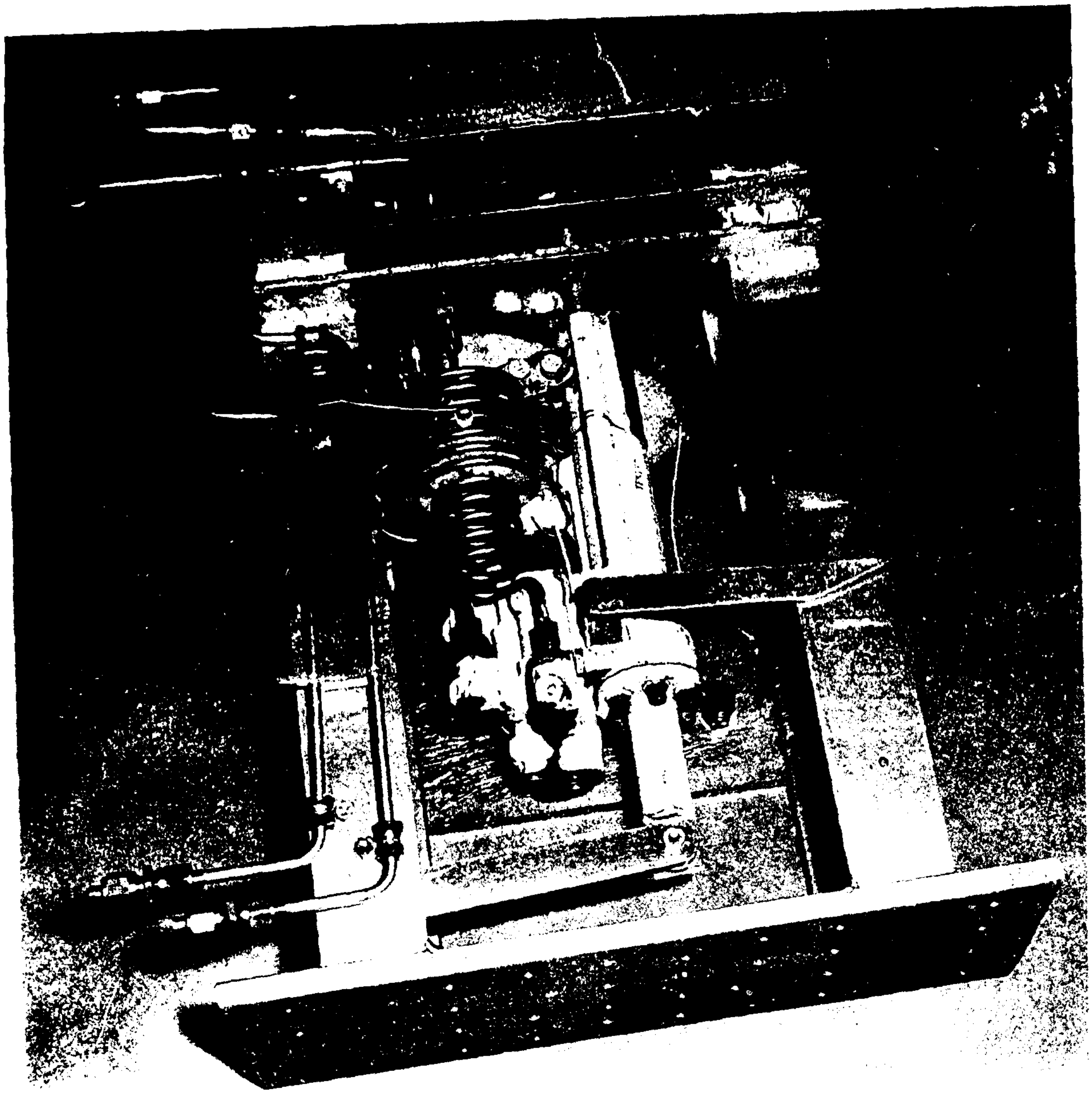

Figure 19. Damping Method A on Helical Configuration - Damped Vibration Test No. 2

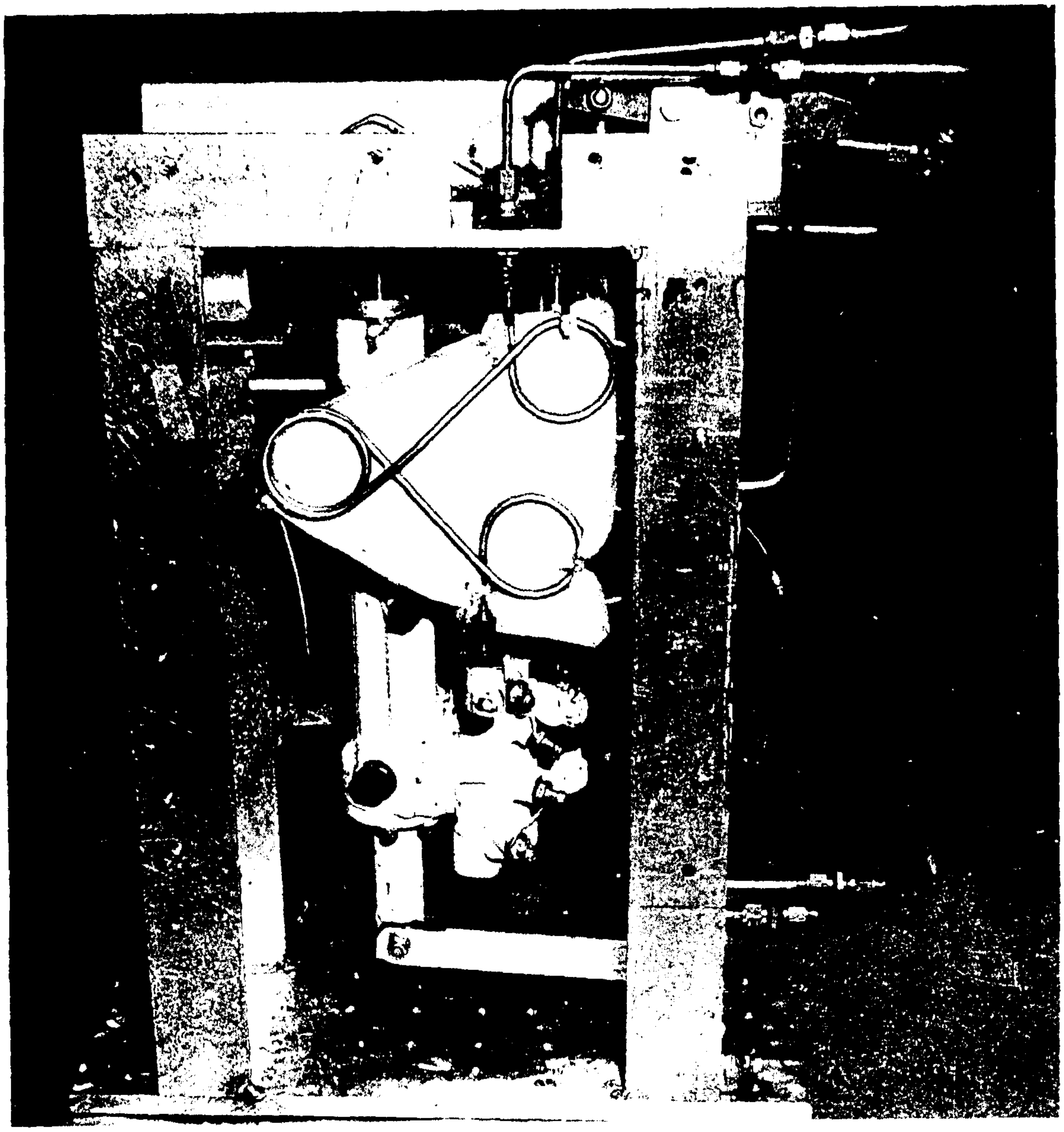

Figure 20. Damping Method E on Tri-coil Configuration - Damped Vibration Test No 2

8. DAMPING STUDY

The damping study resulted in several concepts which should be effective. Some of the concepts could be tried immediately, others would require some experimentation and development before being tested, and evaluated in a future program.

8.1 Damping Method A, Low Density Foam Sheet; Figure 21. This method uses a sheet of low density plastic foam such as latex or polyurethane which is inserted between the inner and outer coils. Damping is achieved by interface to relative motion between coils. The negative aspects of this method are wear of the damping foam due to friction with the coils when the coils are compressed or extended, and loss of properties if the foam has a tendancy to absorb hydraulic fluid.

8.2 Damping Method B, Spiral Plastic Tube Damper; Figure 22. By utilizing a flexible tube which is spiraled between the outer and inner coils, damping may be achieved. The flexible tube is "Tygon" tubing or equivalent. This method requires that the damping tubing be available in diameters which will fit easily between the outer and inner coils. This method is easily incorporated.

8.3 Damping Method C, Low Density Foam Encapsulation; Figure 23. This concept uses silicone foam elastomer, which is a pourable liquid, prior to curing. After curing, the material is capable of extending and retracting with the coil tubes. Some development and experimenation with possible foams is required for this concept.

8.4 Damping Method D, Selective Clamping of Outer Coils to Inner Coils; Figure 24. Some of the inner and outer coils can be clamped to each other. Figure F shows a two spring configuration in which the outer spring has one-half the costs and twice the pitch of the inner spring. With this configuration, the inner and outer coils can be clamped at points A, B, C, D, and E because there is no relative axial motion at those points. The same basic approach can be used with other ratios of coils and pitches, but the clamping locations will differ accordingly. If the outer spring has one-third the coils and three times the pitch of the inner spring, then the inner spring can be clamped at every third coils in a like manner. In either case, both must be wound in the same direction. All clamping must be done at the same clockwise position as the start of the coils to prevent relative motion and tube damage. Other clamp locations are possible if the inner and outer springs start at different clockwise positions. Other clamp locations must be studied carefully to assure that no relative motion exists. Tubing wear may be a problem with use of Method D. The relationship between the pitch and number of coils for Method D is:

$$\frac{p_o}{p_i} = \frac{N_i}{N_o}$$

Where: p = pitch
N = number of coils
i = inner coil
o = outer coil

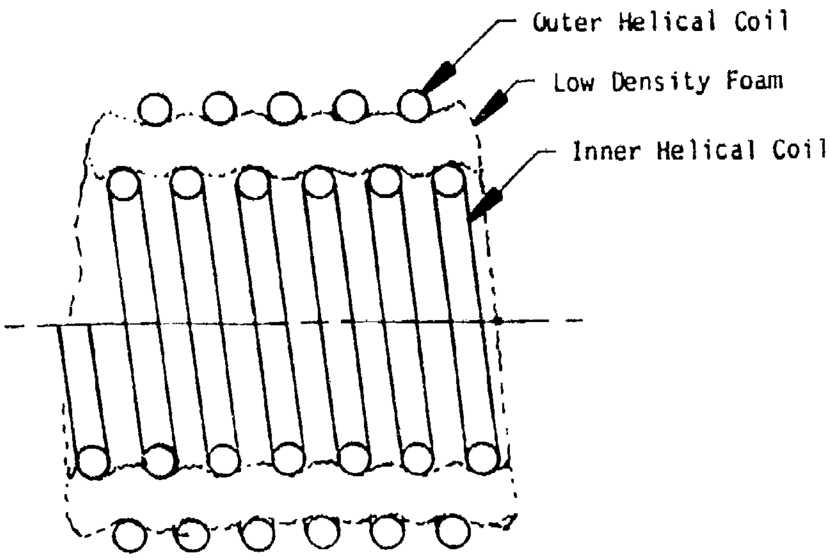

Figure 21. Damping Method A - Low Density Foam

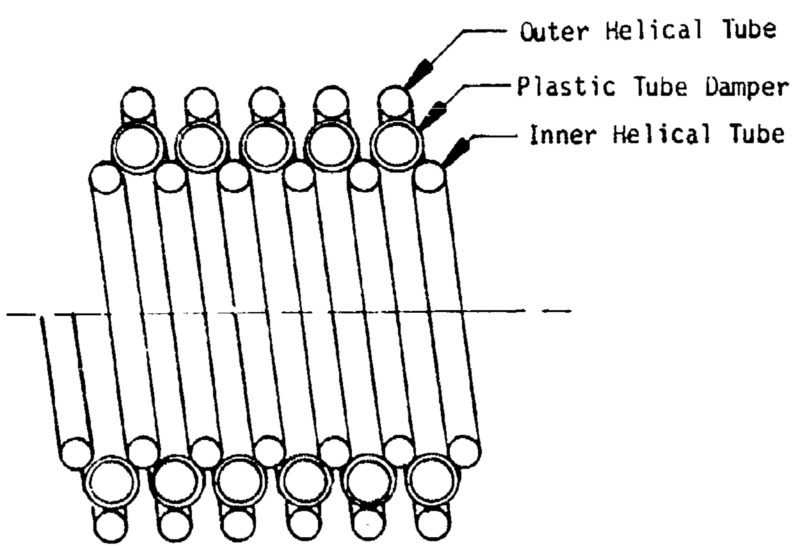

Figure 22. Damping Method B - Spiral Plastic Tube Damper

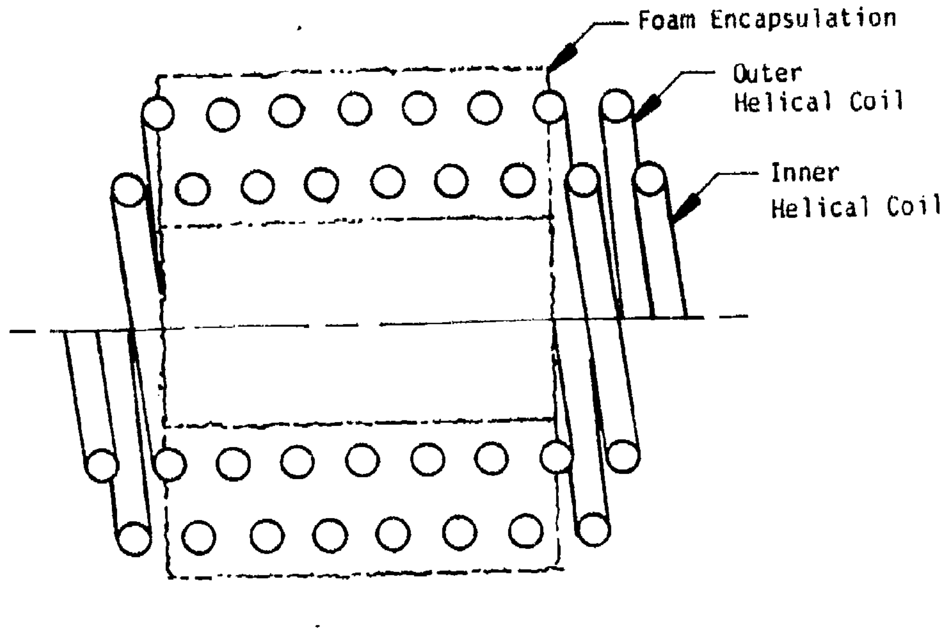

Figure 23. Damping Method C - Low Density Foam Encapsulation

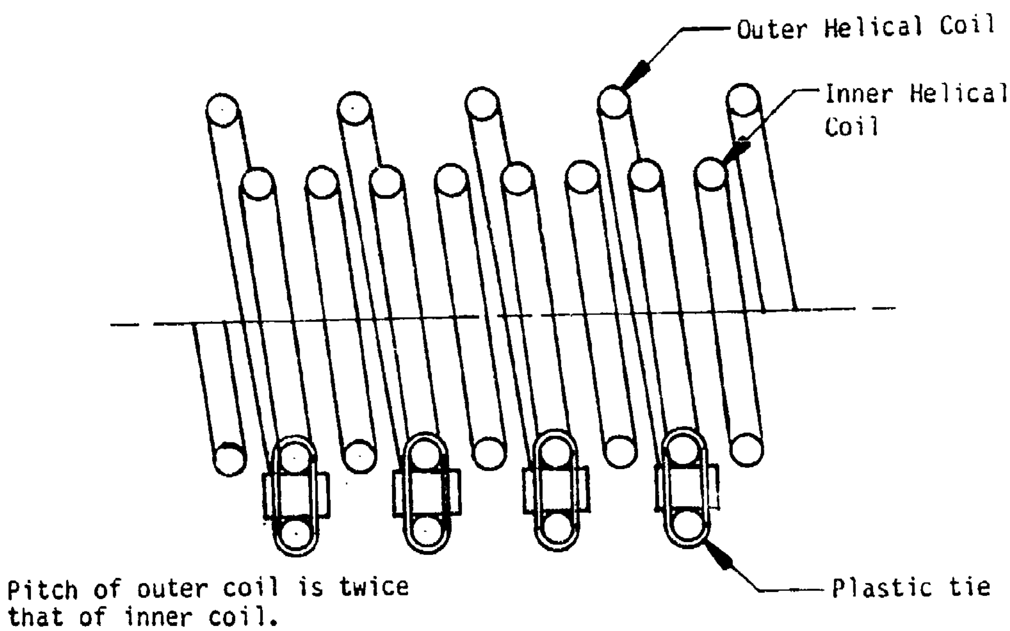

Figure 24. Damping Method D - Selective Clamping of Outer Coils to Inner Coils

8.5 Damping Method E, Foam Sandwich for Tri-coil Configuration; Figure 25. Again, this concept relies on interference of the foam with the coils for damping. Due to the tri-coil configuration and the location of the foam, moderate density foams could be tried for this concept.

8.6 Damping Method F, Tubing Encapsulation of Coils; Figures 26 and 27. Figures 26 and 27 show coils encapsulated in heat shrinkable tubing. The heat shrinkable tubing must be applied to the metal tubing prior to forming. The diameter of the mandrel must accommodate the change in dimensions due to the addition of the heat shrinkable tubing. The coil pitch must be increased also.

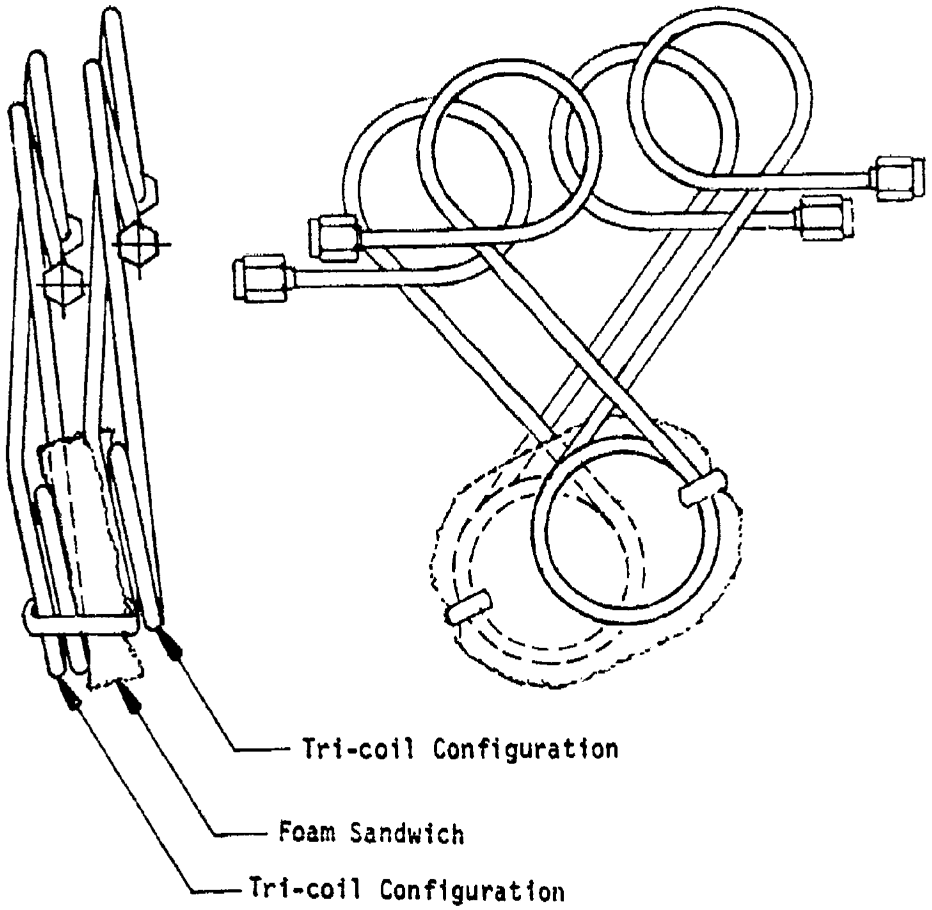

Figure 25. Damping Method E - Foam Sandwich

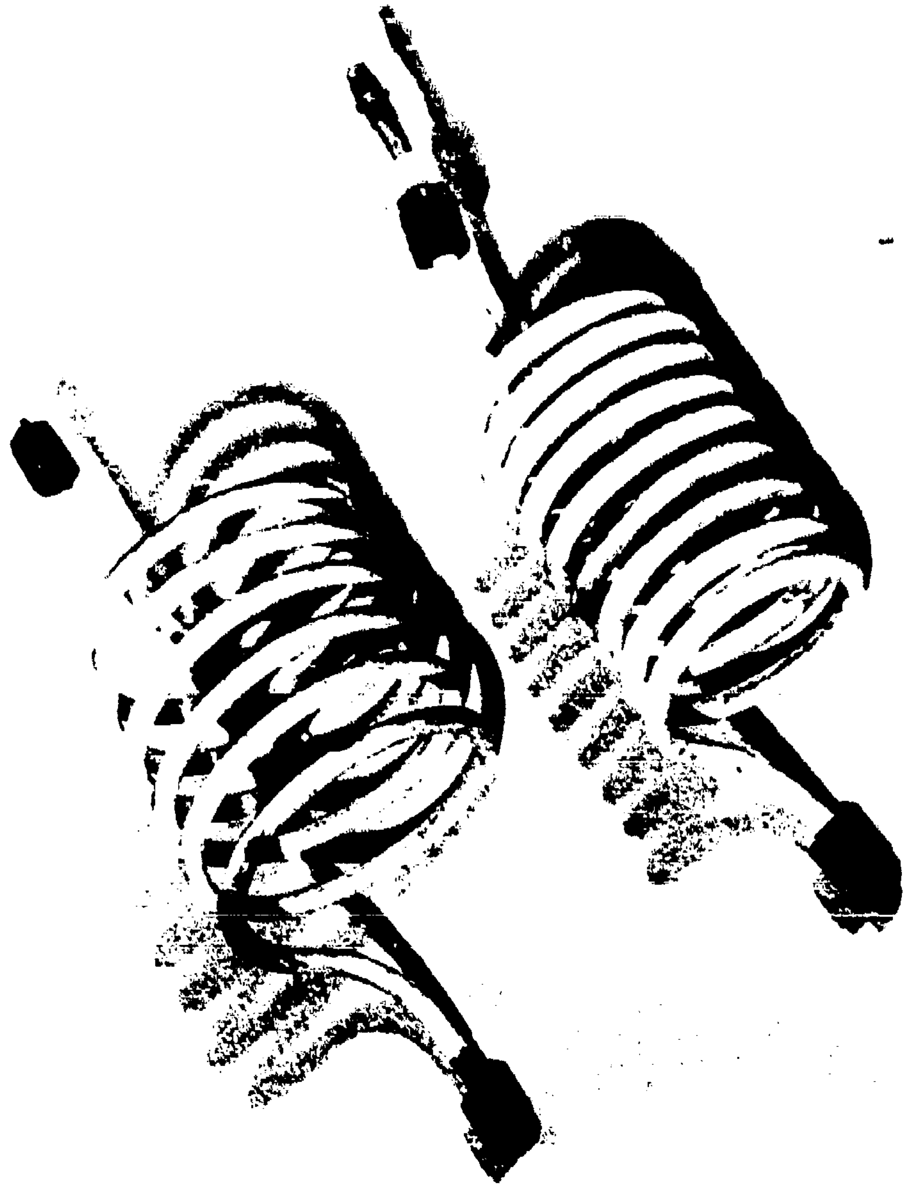

Figure 26. Damping Method F - Tubing Encapsulation of Coils as Applied to Helical Configuration

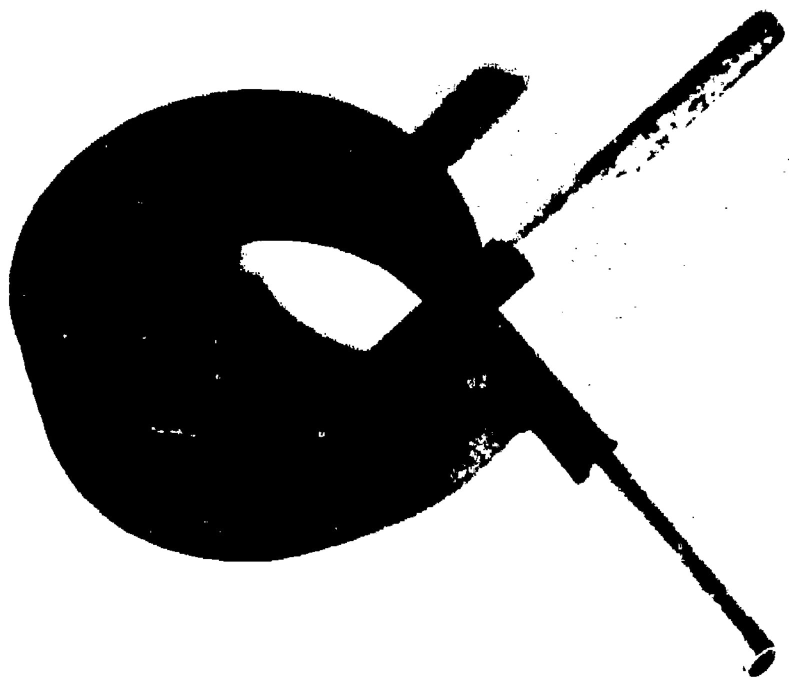

Figure 27. Damping Method F - Tubing Encapsulation of Coils as Applied to Coils Loaded in Torsion

9. TERMINATION AND NESTING STUDY

9.1 Study Description. The task invisioned originally for this study was to deal with special fittings which terminated the coil ends and would eliminate the possibility of fatigue at standard MS flareless fittings. In the course of analysis and test on the configurations in this program, the problem of stress at sleeves on MS flareless fittings was determined not to be significant. A problem with more impact on implementation of coiled tubing was how to form the transition from the helical coils to the end fittings. The type of transition required directly impacts tooling requirements. Therefore, this section is devoted to a description and brief trade study of nesting and termination.

9.2 Simple Tangential Extension; Figure 28. This configuration terminates the coils by a tangential extension of the tube at the end of each coil. This results in simple bends but requires sufficient clearance between coils to allow insertion of the inner coil with fittings.

9.3 Center Termination of Inside Coil; Figure 29. The outer coil still has simple bends but compound bends are required on the inner coil. The inside coil can be easily inserted with this termination.

9.4 Inline Termination of Both Coils; Figure 30. Both coils have compound bends but easy nesting is possible. This configuration would be difficult to produce.

9.5 Reduced Bend Radius Termination; Figure 31. By use of non-standard reduced bend radii, the bends can be made in one plane.

9.6 Double Pitch Coils; Figure 32. By use of a special mandrel, both coils have the same mean diameter but with expanded pitch which allows the coils to be intertwined. Bends are simple. Length of assembly would be greater to accommodate both sets of coils.

Table 6, End Configuration Comparison, summarizes the bend requirements, advantages, and disadvantages of each type of termination.

9.7 Other Possible Configurations. Figures 33 and 34 show two configurations which were made but not tested. These configurations use large diameter coils which would allow the tube to be installed around the outside of the actuator. Figure 33 is an oval concept. Figure 34 shows large diameter circular coils as they might appear installed around an actuator. These configurations have the advantage of being able to be shaped to conform to the wing contour or compartment in which the actuator is installed.

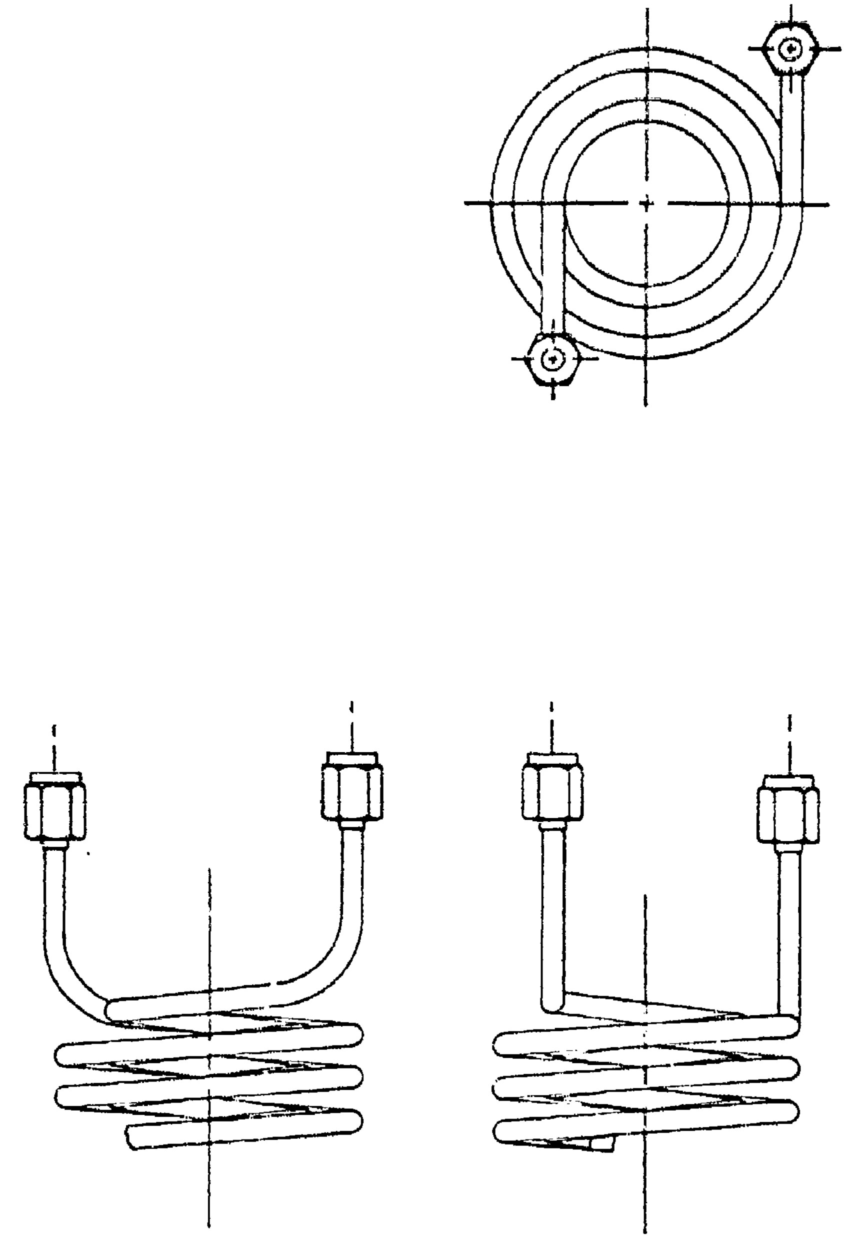

Figure 28. Simple Tangential Extension

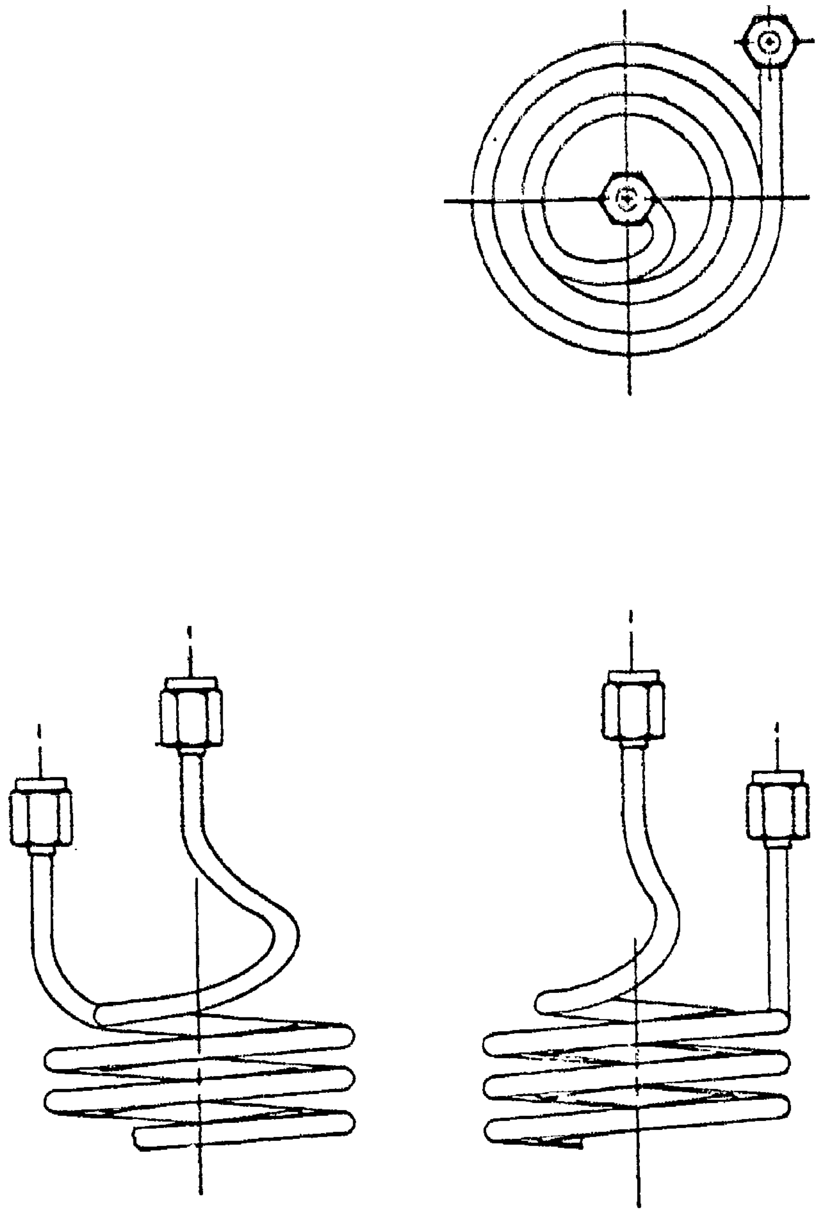

Figure 29. Center Termination of Inside Coil

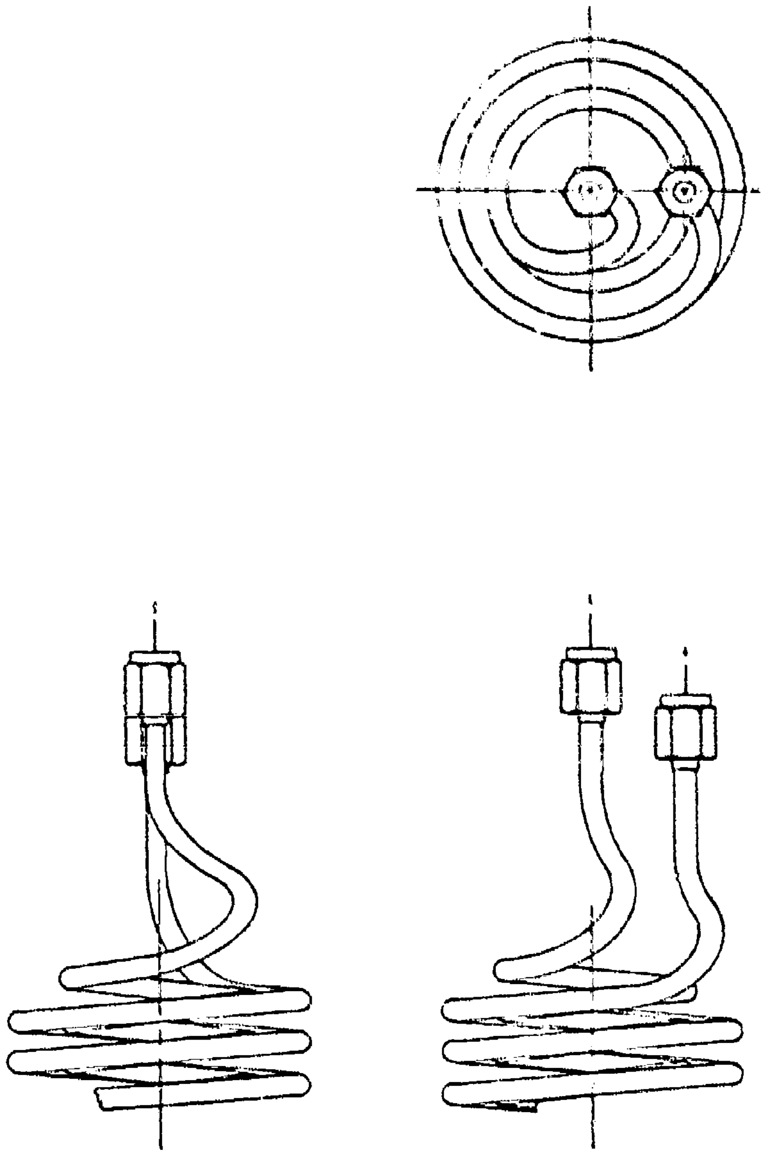

Figure 30. Inline Termination of Both Coils

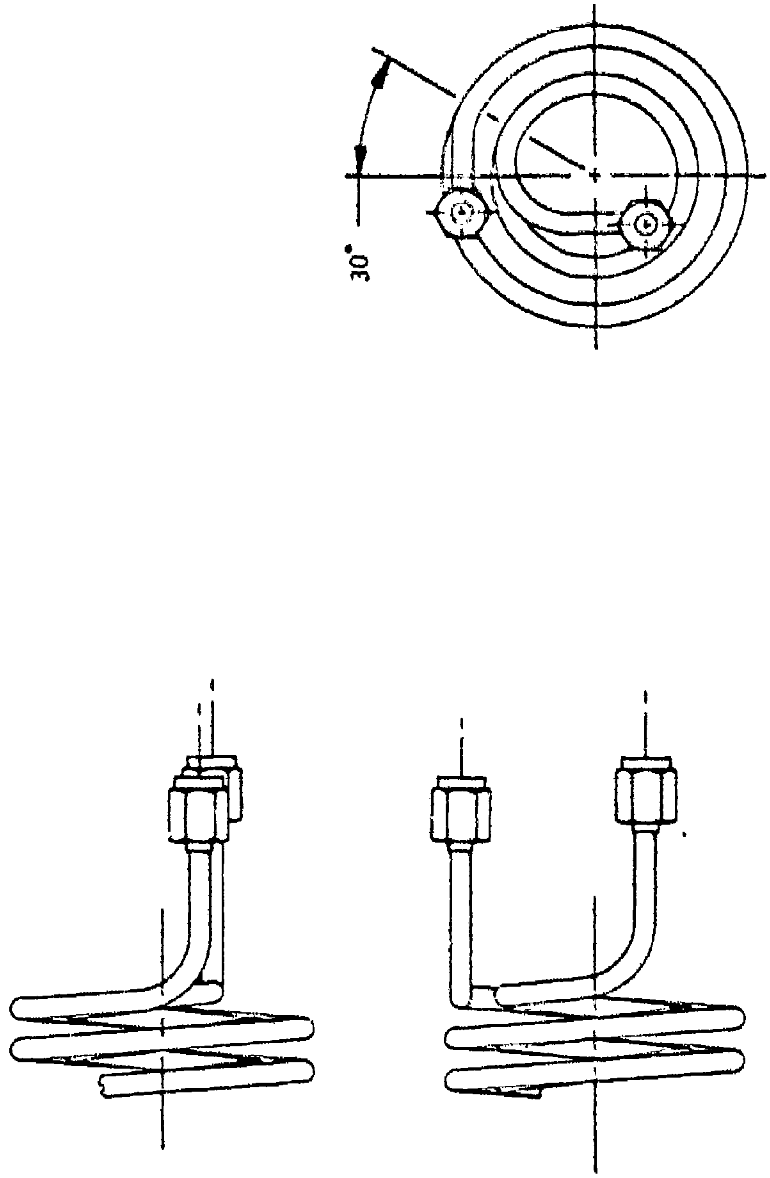

Figure 31. Reduced Bend Radius Termination

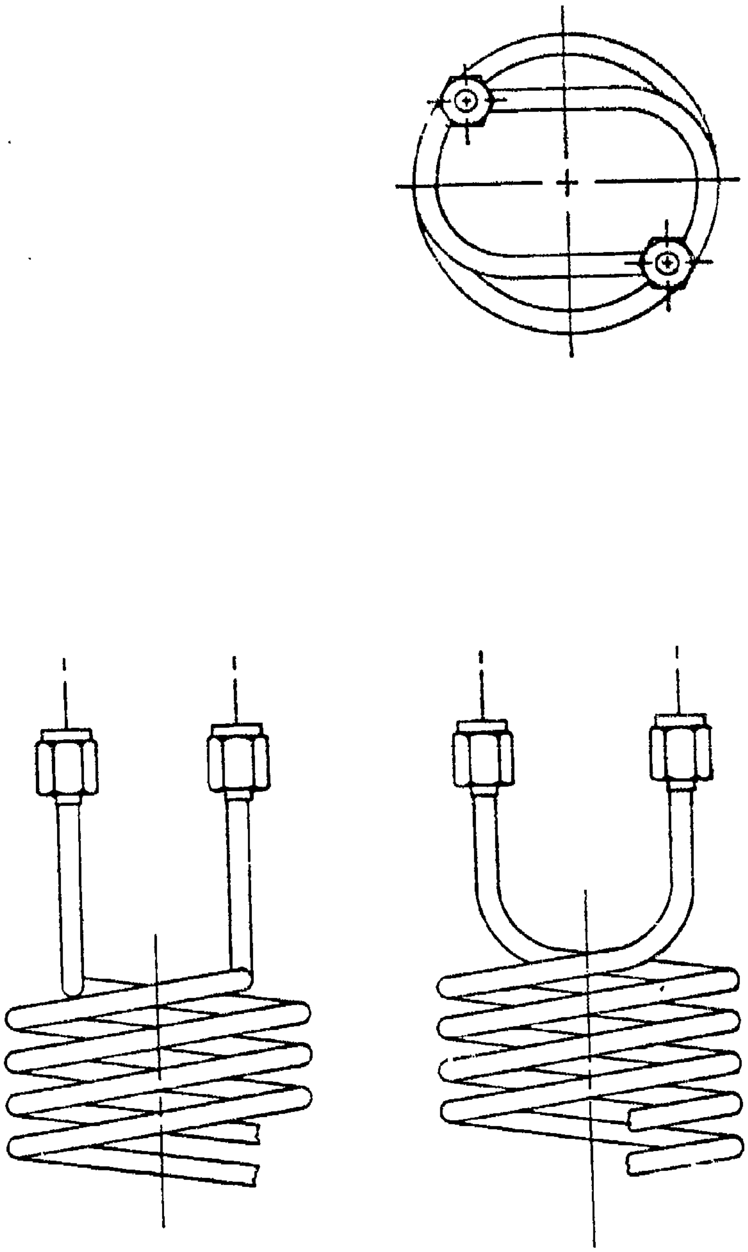

Figure 32. Double Pitch Coils

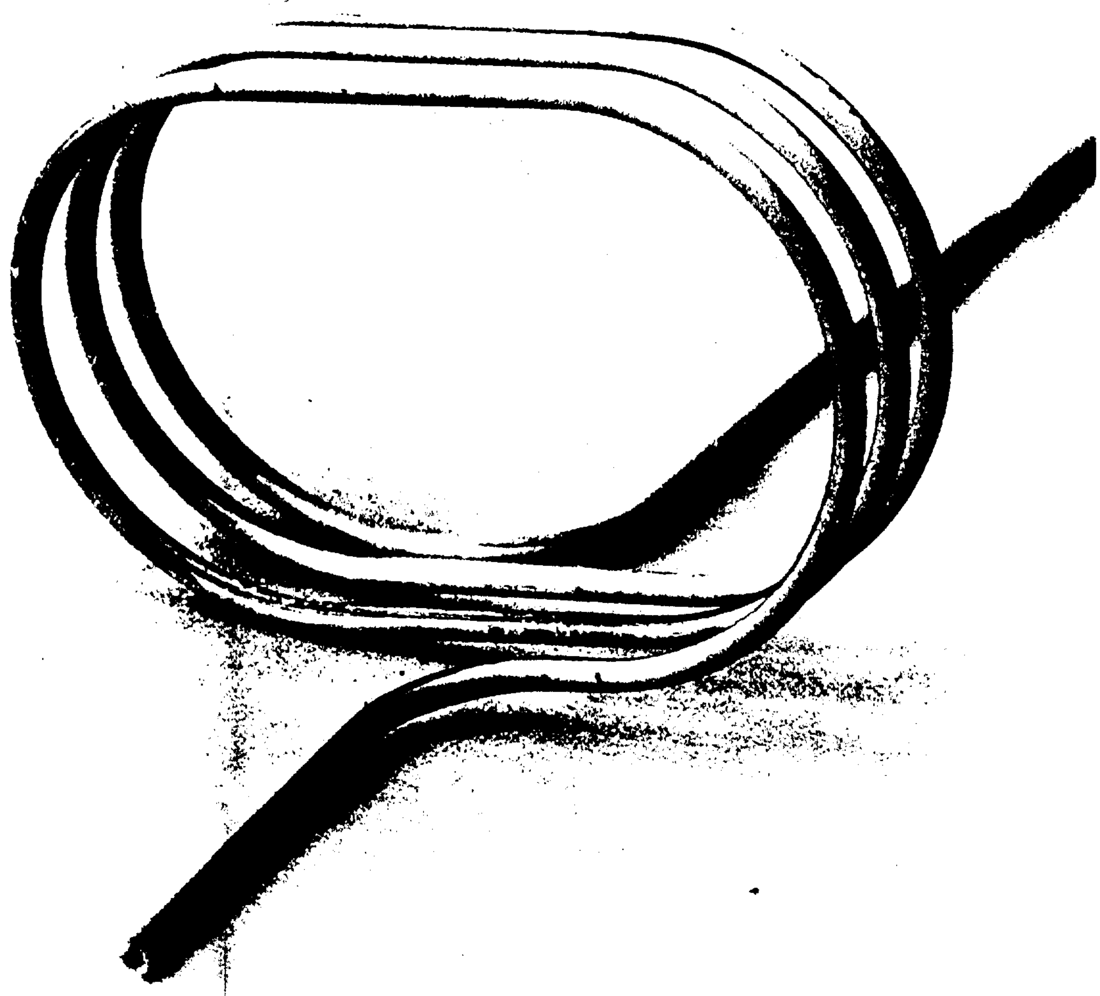

Figure 33. Oval Configuration

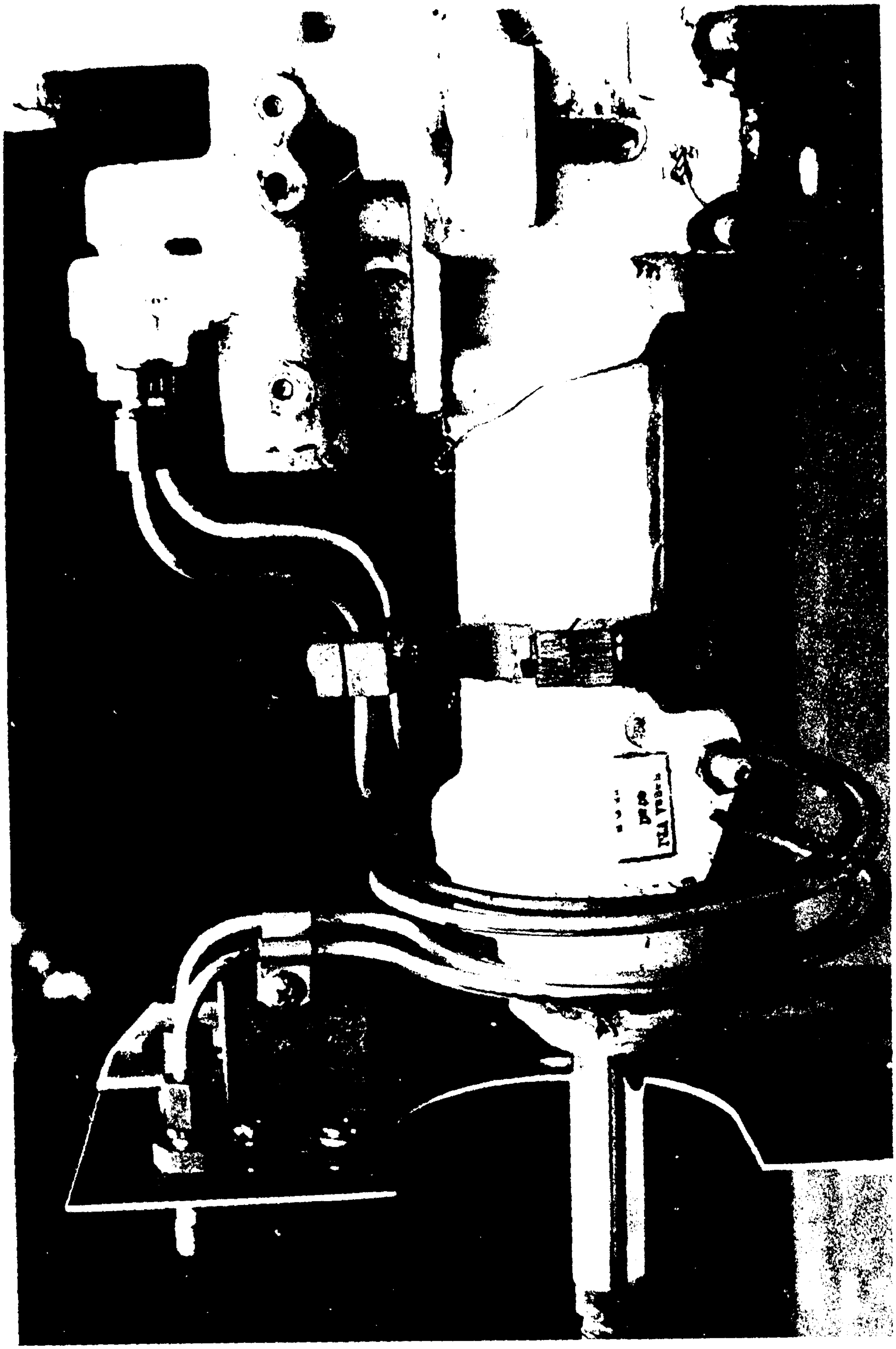

Figure 34. Large Diameter Circular Coils

TABLE 6. END CONFIGURATION COMPARISON

CONFIGURATION DESCRIPTION	TUBE BEND TYPE SIMPLE CURVATURE	TUBE BEND TYPE COMPOUND CURVATURE	SPECIAL REQUIREMENTS	ADVANTAGES	DISADVANTAGES
Figure 28			Clearance between coils must be large so coils can be nested.	Lowest cost	Bulky
Inside coil	X				
Outside coil	X				
Figure 29			Compound curves require special tools	Ease of nesting coils	Difficult to duplicate
Inside coil		X			
Outside coil	X				
Figure 30			Compound curves require special tools	Ease of nesting coils	Difficult to duplicate
Inside coil		X			
Outside coil		X			
Figure 31			Tight bend radius must be in the same plane as the helical coil	Ease of nesting coils Low cost	---
Inside coil	X				
Outside coil	X				
Figure 32	X		Special tooling required to wind coils	Allows Min No. of coils	Extra length

10. CONCLUSIONS

10.1 Materials. The materials selection study used the ratio of tensile strength to torsional modulus as the function to be maximized. A material with the highest ratio is considered a good spring material. The three materials selected were cold worked stress relieved Ti 3Al-2.5V, annealed Ti 3Al-2.5V and Ti 6Al-4V. Material properties such as ultimate shear strength and torsional modulus for titanium alloys are not readily available.

10.2 Endurance Test. The successful completion of the 100,000 cycle endurance test indicates that the design equations derived were adequate and that helical or tri-coil configurations can be designed to pass any endurance spectrum.

10.3 Vibration Tests. Undamped configurations showed amplification of input acceleration. Relatively simple damping methods attenuated the amplification of random vibration. In the brief tests conducted, the use of low density foams and plastic ties was moderately succcessful but further work is required.

10.4 Fabrication of Coils. Using the tools described in this report, fabrication of coils was very straight-forward. Difficulty was encountered in the forming of the bends from the helical coil to the straight lengths going to the end fittings. The hand forming techniques used would be acceptable for the small quantities involved in a test and development program, but would not be satisfactory for production of tubing in greater quantities.

10.5 Configuration. The configurations evaluated are satisfactory for replacement of hose or extension units on moving barrel actuators. The variety of shapes will allow installation of coiled tubing in very compact envelopes which are cylindrical or planar with finite thickness. The design equations and nomographs derived allow tailoring of the coiled tube design for any stroke.

11. RECOMMENDATIONS

Having established a good technical foundation for the design of the helical and tri-coil configurations, additional work is required to provide the complete technical base to allow design and fabrication of these configurations in confidence and also to support reliability predictions.

11.1 Recommendations on Additional Tests

- Test recommended wall thicknesses to failure for both the helical and tri-coil configurations.
- Determine the torsional modulus (G) for Ti 3Al-2.5V CWSR alloy.
- Evaluate vibration damping methods to establish the most effective techniques.
- Fabricate and test configurations such as the diamond and oval shapes which wrap around the actuator.
- Set up a test demonstration to establish that auto-frettage of tubing is not required.

11.2 Recommendations on Additional Analysis and Design

- Conduct a trade study of end termination to establish production tooling requirements.
- Write a Fortran language computer program of the design algorithm used.
- Create nomographs for larger tube sizes such as 1/4 and 3/8 inch diameter.

11.3 Recommendations on Demonstration and Education

- Make static displays to illustrate the various configurations possible.
- Prepare a movie which examines the tooling required, illustrates the configurations possible, and details damping methods.
- Make presentations on the coiled tube program to major aircraft manufacturers and to cognizant Naval technical personnel.
- Investigate retrofit of coiled tube installations on fleet aircraft in specific applications.

REFERENCES

1. Cooke, Conrad H.; Stouffer, Ronald D., "Study of Coiled Tubing for Aircraft Hydraulic Systems", Final Report WADC T.R. 55-121 to Wright Air Development Center, Glenn L. Martin Company, February, 1955.

2. Wahl, A. M., "Mechanical Springs", 2nd Edition, McGraw-Hill Book Company, New York, 1963.

APPENDIX A
SYMBOLS

The symbols defined below are used throughout this report.

α = Included angle of extensions passing through coil centerlines on tri-coil configuration, with coils unloaded, degrees.

α_T = New angle α in loaded position, degrees

ϕ = Ovality of tube, decimal percent - subscript 1 denotes unpressurized condition; subscript 2 denotes pressurized condition

Δa = Change in length of semi-major axis of ellipse at inner tube wall, inches

Δb = Change in length of semi-minor axis of ellipse at inner tube wall, inches

Δd_b = Change in ovality of tube across d_b from unpressurized to pressurized condition

A = Total stroke; inches

A_1 = Stroke per coil measured at center line of coil, inches

B = Distance between outer coil centers in tri-coil configuration in unloaded position, inches (see Figure 6)

C = Distance between outer and center coil centers on tri-coil configuration, inches (see Figure 6)

D_m = Coil mean diameter, inches

E = Modulus of elasticity, psi

F = Load on spring, lbf

F_{su} = Ultimate shear strength allowable, psi

F_{tu} = Ultimate tensile strength allowable, psi

G = Modulus of rigidity, psi

H = Vertical height of tri-coil in free state, inches

H_T = Vertical height of tri-coil in loaded state, inches

I = Moment of inertia, in^4

J = Polar moment of inertia, in^4

K = Parameter used in Reference 1

K_i = Wahl correction factor, inside of coil

K_o = Wahl correction factor, outside of coil

APPENDIX A
SYMBOLS - CONTINUED

L	=	Straight length of tubing tangent to coils on tri-coil configuration, inches (see Figure 6)
M_A	=	Moment taken about point A of oval tube, in-lb
M_B	=	Moment taken about point B of oval tube, in-lb
M_E	=	Moment in tube wall due to elasticity, in-lb
M_P	=	Moment due to pressure, in-lb
N	=	Number of active coils
P	=	Internal pressure, psi
S_{EQ}	=	Equivalent uniaxial stress, psi
S_x	=	Total stress in circumferential direction of tube, psi
S_{xz}	=	Total torsional stress in tube, psi
S_y	=	Total stress in longitudinal direction of tube, psi
S_z	=	Total stress in radial direction of tube, psi
T	=	Torque, inch-pounds
γ	=	parameter used in reference 1
λ	=	Parameter used in reference 1
a	=	Length of semi-major axis of ellipse, inches
θ	=	Angular displacement of a single coil, radians
b	=	Length of semi minor axis of ellipse, inches
d_a	=	Major axis of oval tube at outer wall, inches
d_b	=	Minor axis of oval tube at outer wall, inches
d_i	=	Tube inside diameter, inches
d_o	=	Tube outside diameter, inches
f_{tA}	=	Stress due to ovality at semi-major axis of elliptical tube cross-section, psi.
f_{tB}	=	Stress due to ovality at semi-minor axis of elliptical tube cross-section, psi
r	=	Radius at wall neutral axis of perfect round tube, inches
r_A	=	Radius of curvature to tube wall neutral axis at semi-major axis of ellipse, inches
r_B	=	Radius of curvature to tube wall neutral axis at semi-minor axis of ellipse, inches
β	=	parameter used in reference 1
t	=	Tube wall thickness, inches

APPENDIX B
DATA FROM VIBRATION TESTS

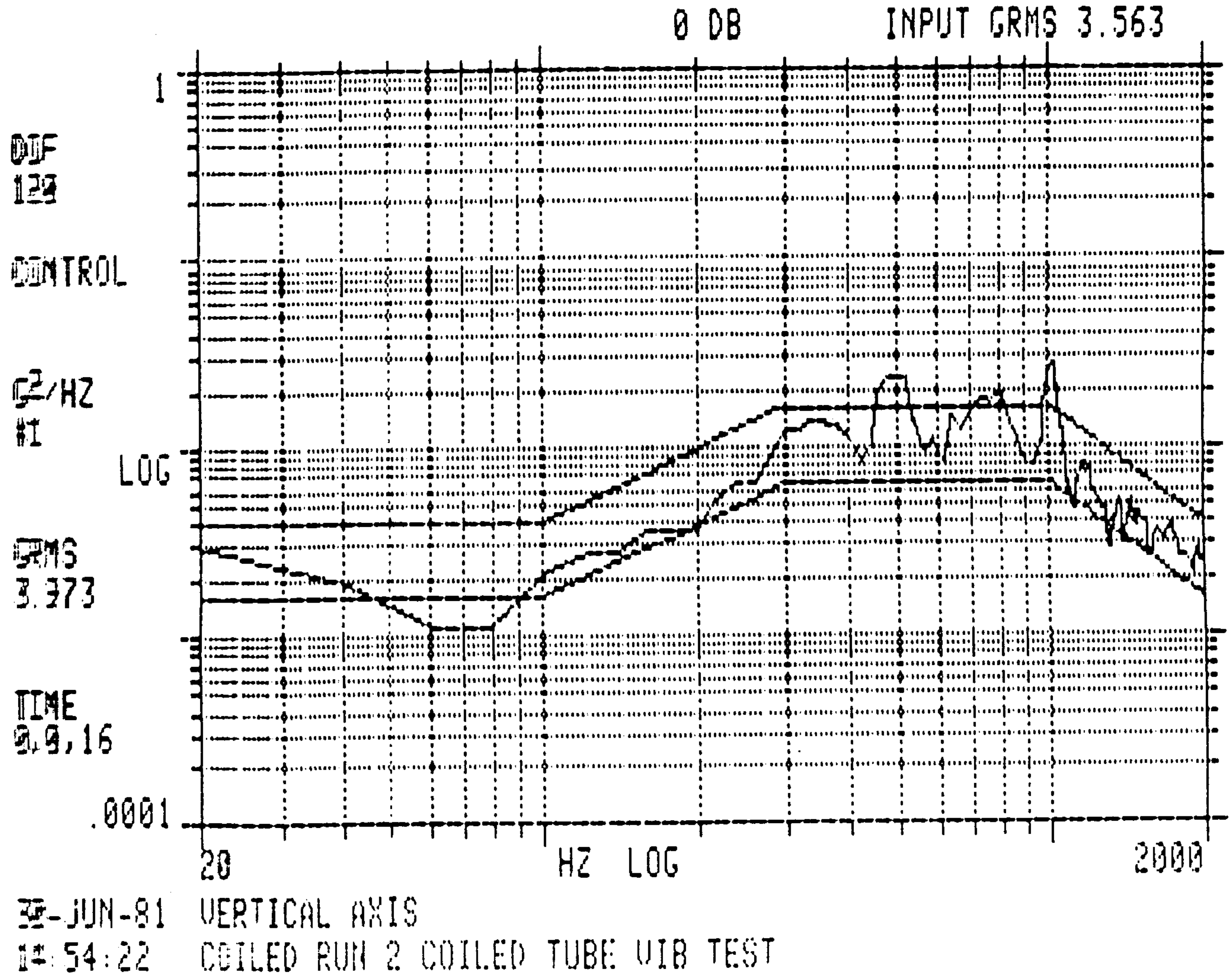

Figure 35. Undamped Vibration Test, Input for Vertical Axis

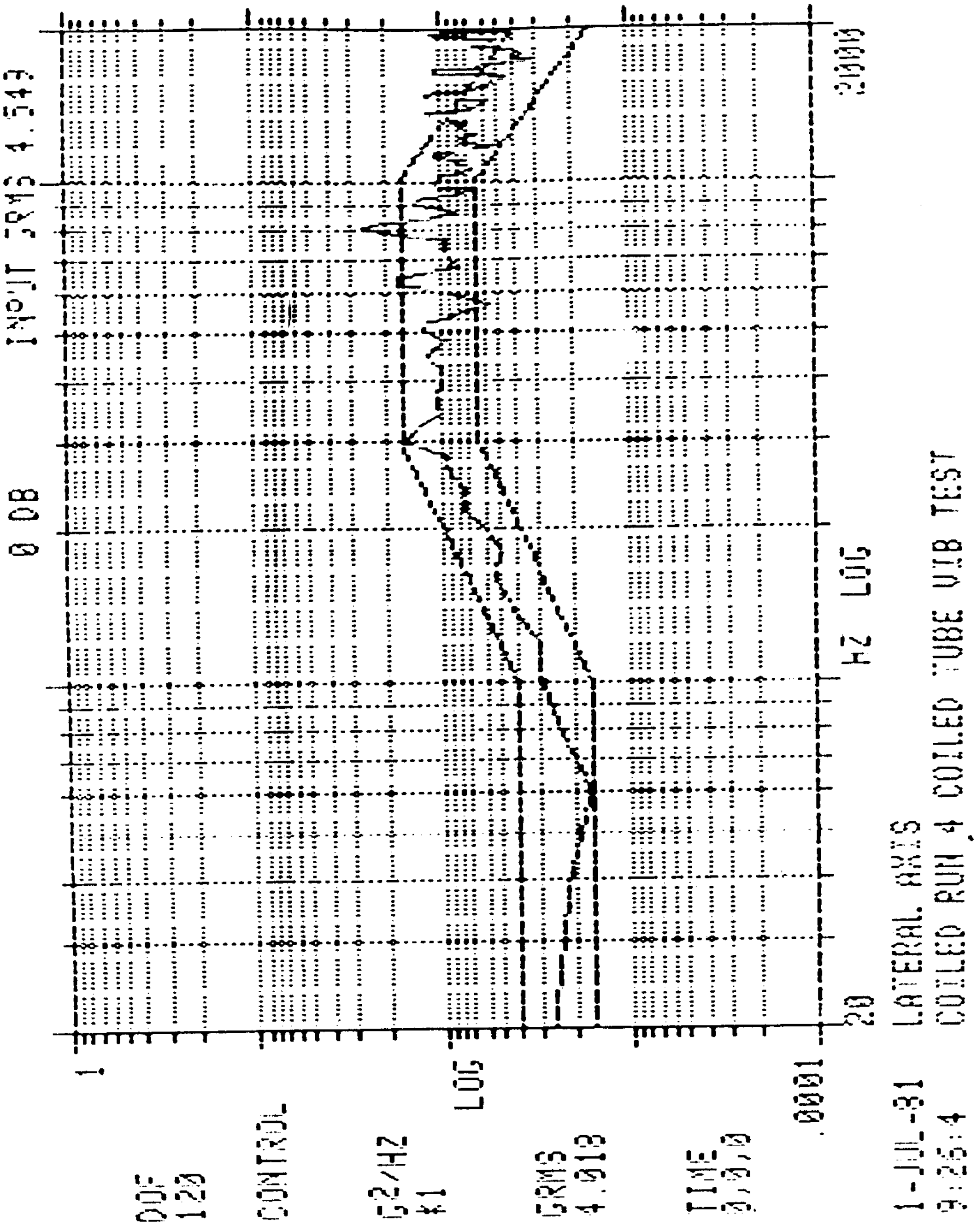

Figure 36. Undamped Vibration Test, Input for Lateral Axis

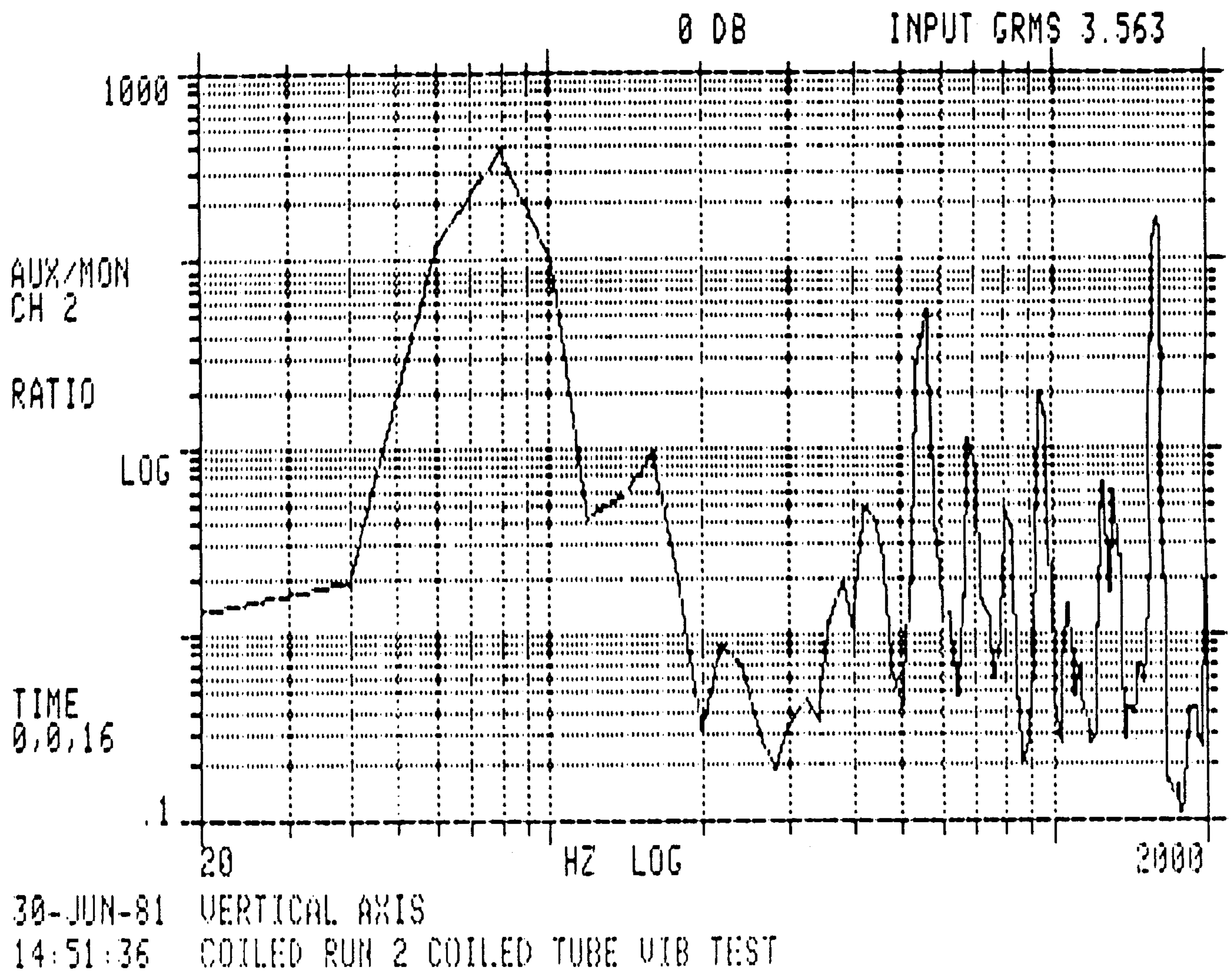

Figure 37. Undamped Vibration Test, Transmissibility of Helical Configuration in Vertical Axis

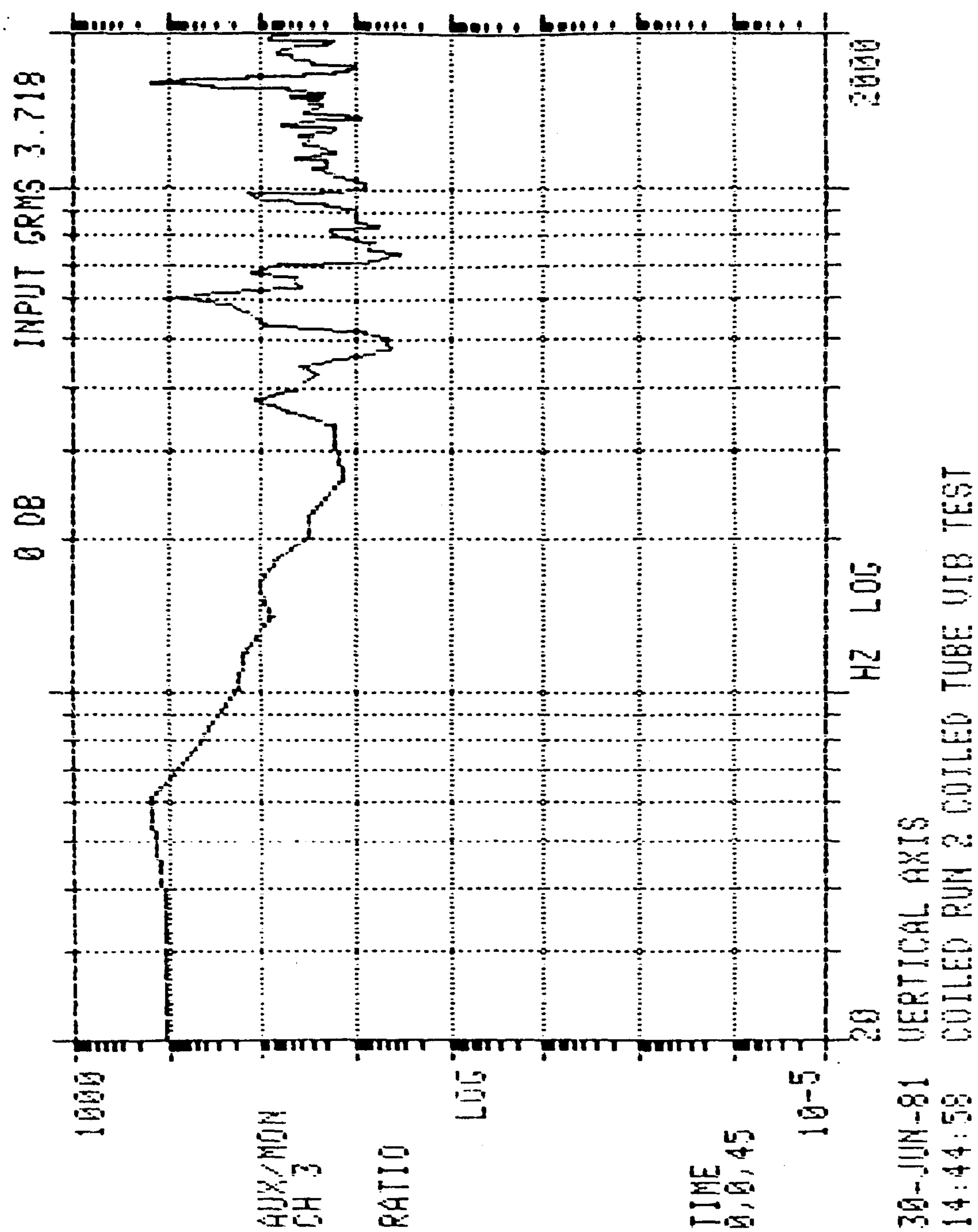

Figure 38. Undamped Vibration Test, Transmissibility of Tri-Coil Configuration in Vertical Axis

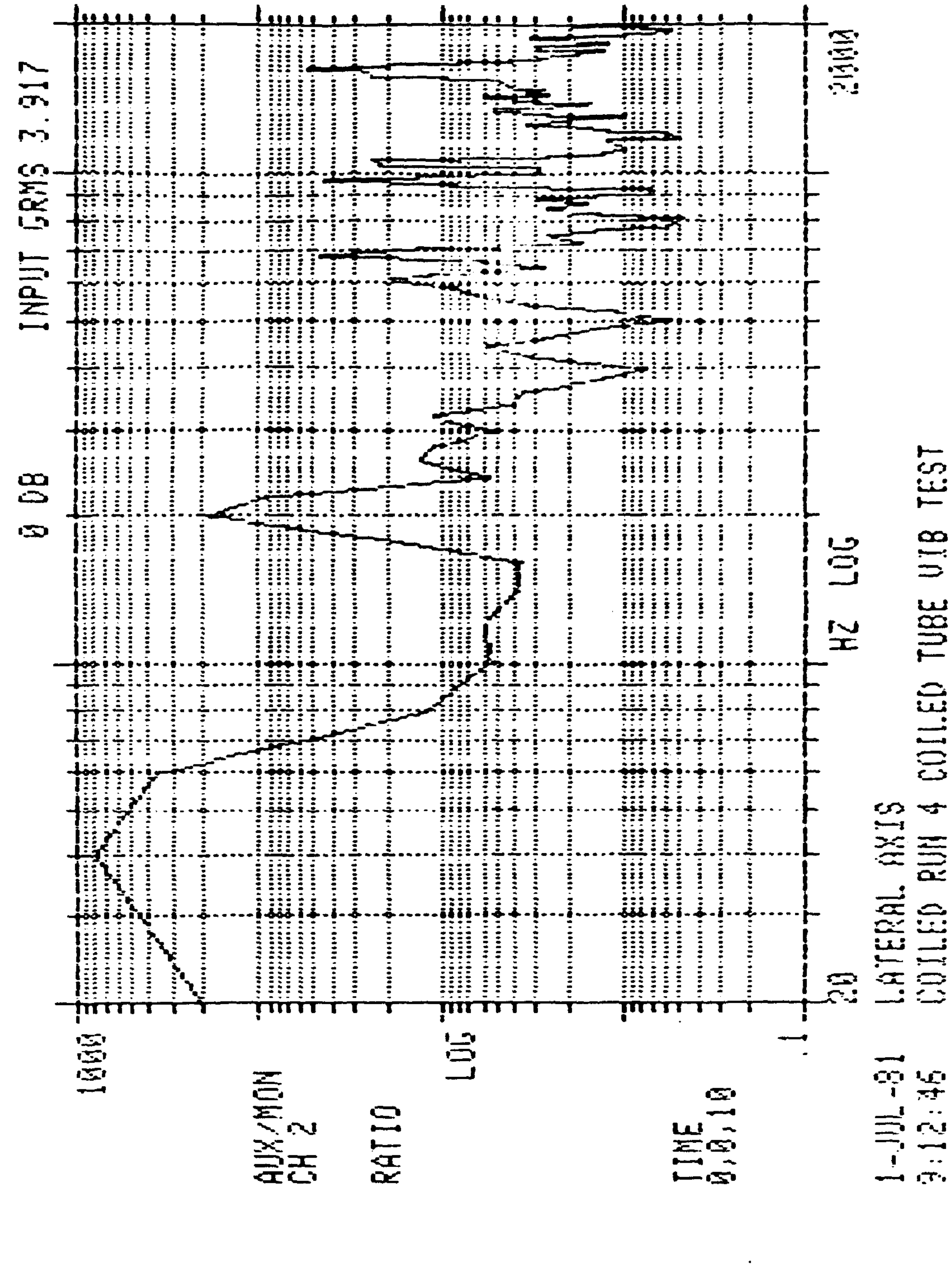

Figure 39. Undamped Vibration Test, Transmissibility of Helical Configuration in Lateral Axis

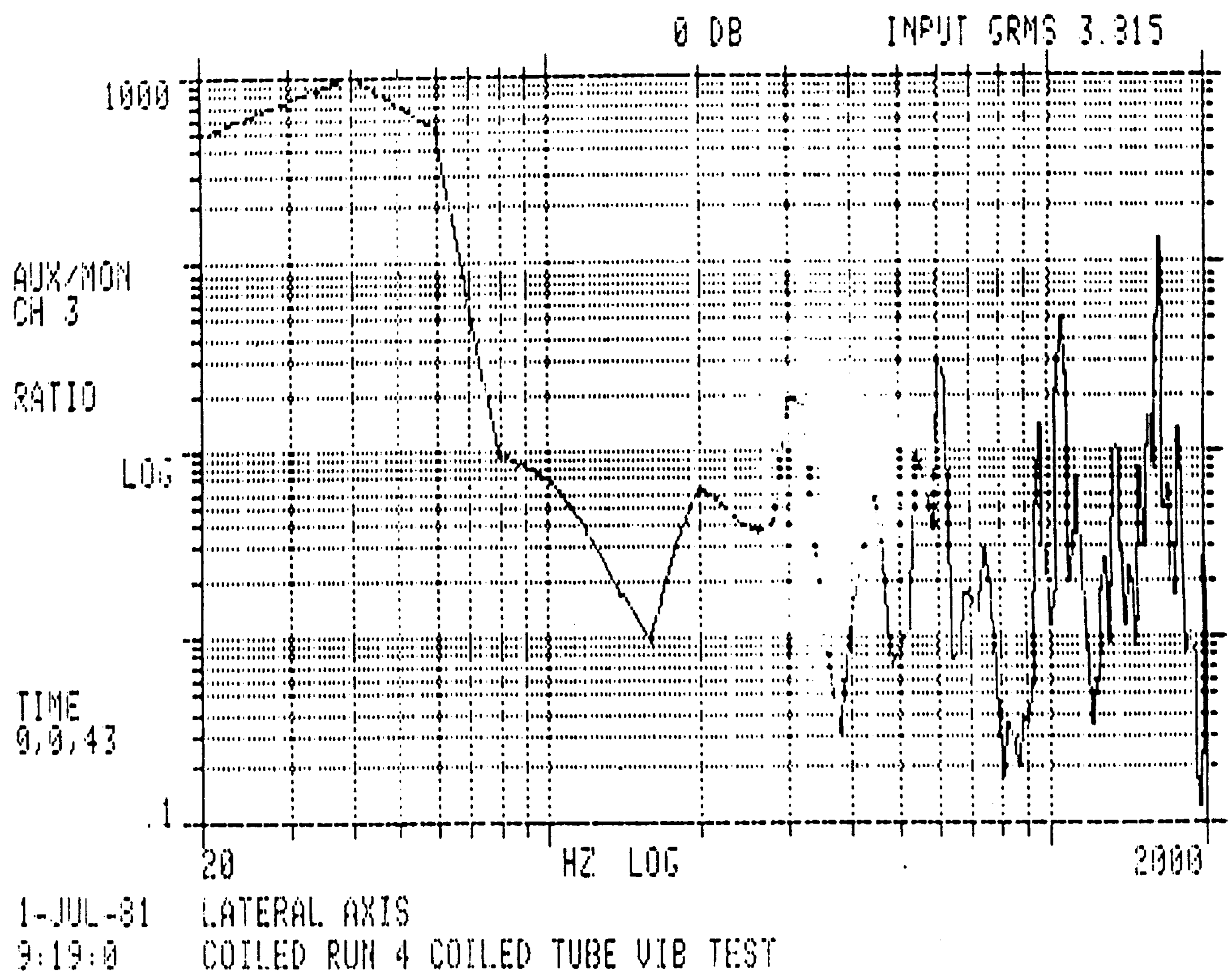

Figure 40. Undamped Vibration Test, Transmissibility of Tri-Coil Configuration in Lateral Axis

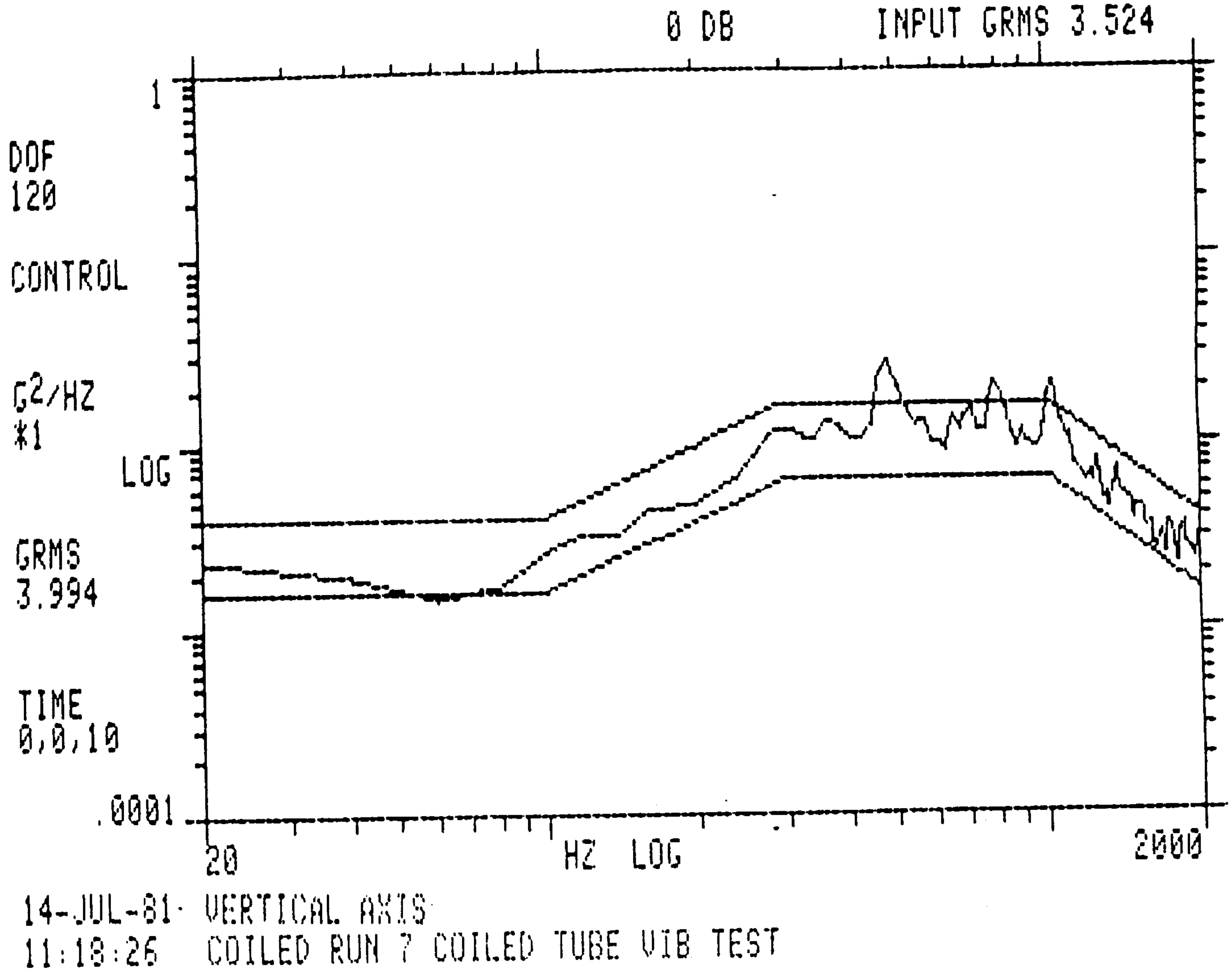

Figure 41. Damped Vibration Test No. 1, Input for Vertical Axis

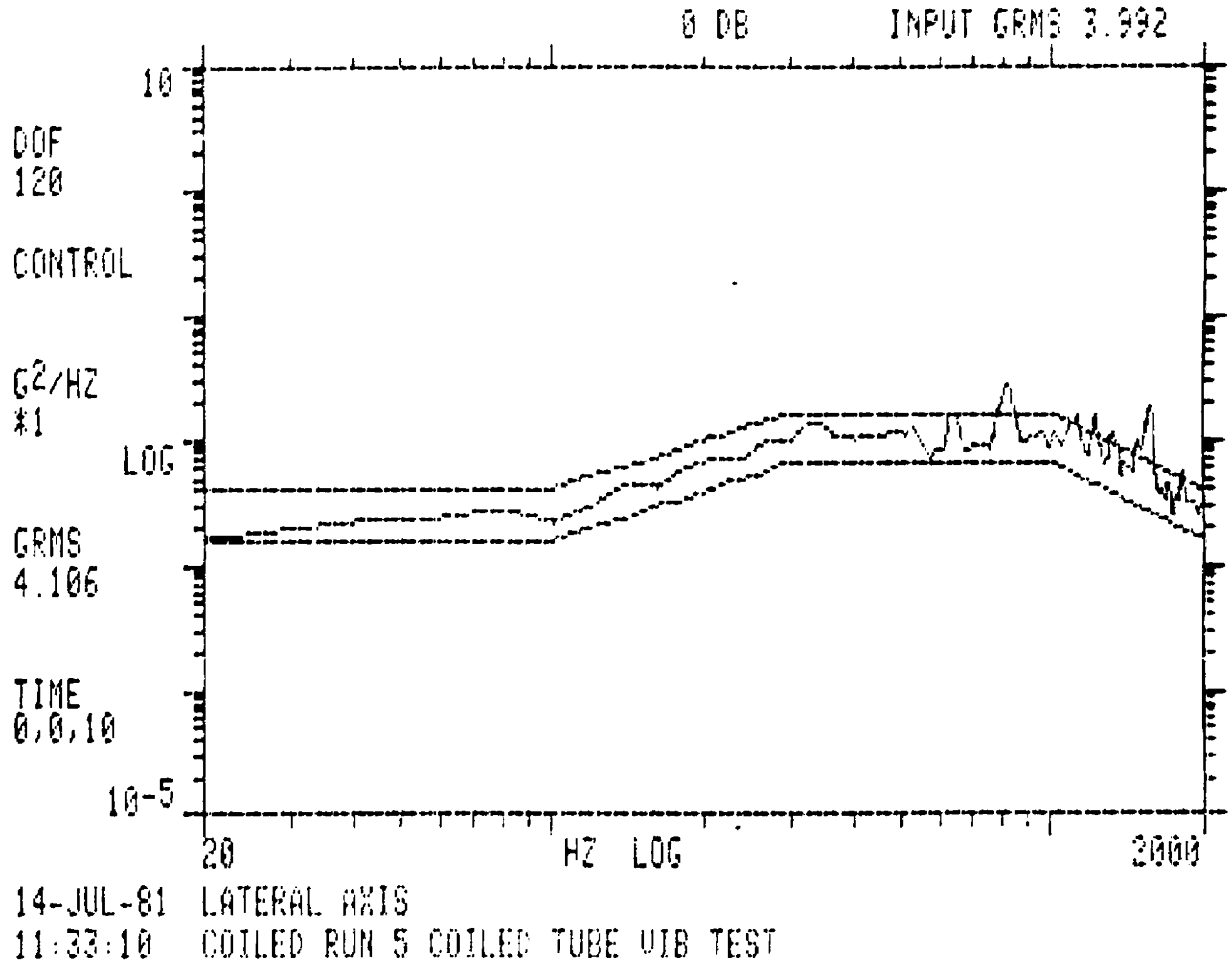

Figure 42. Damped Vibration Test No. 1, Input for Lateral Axis

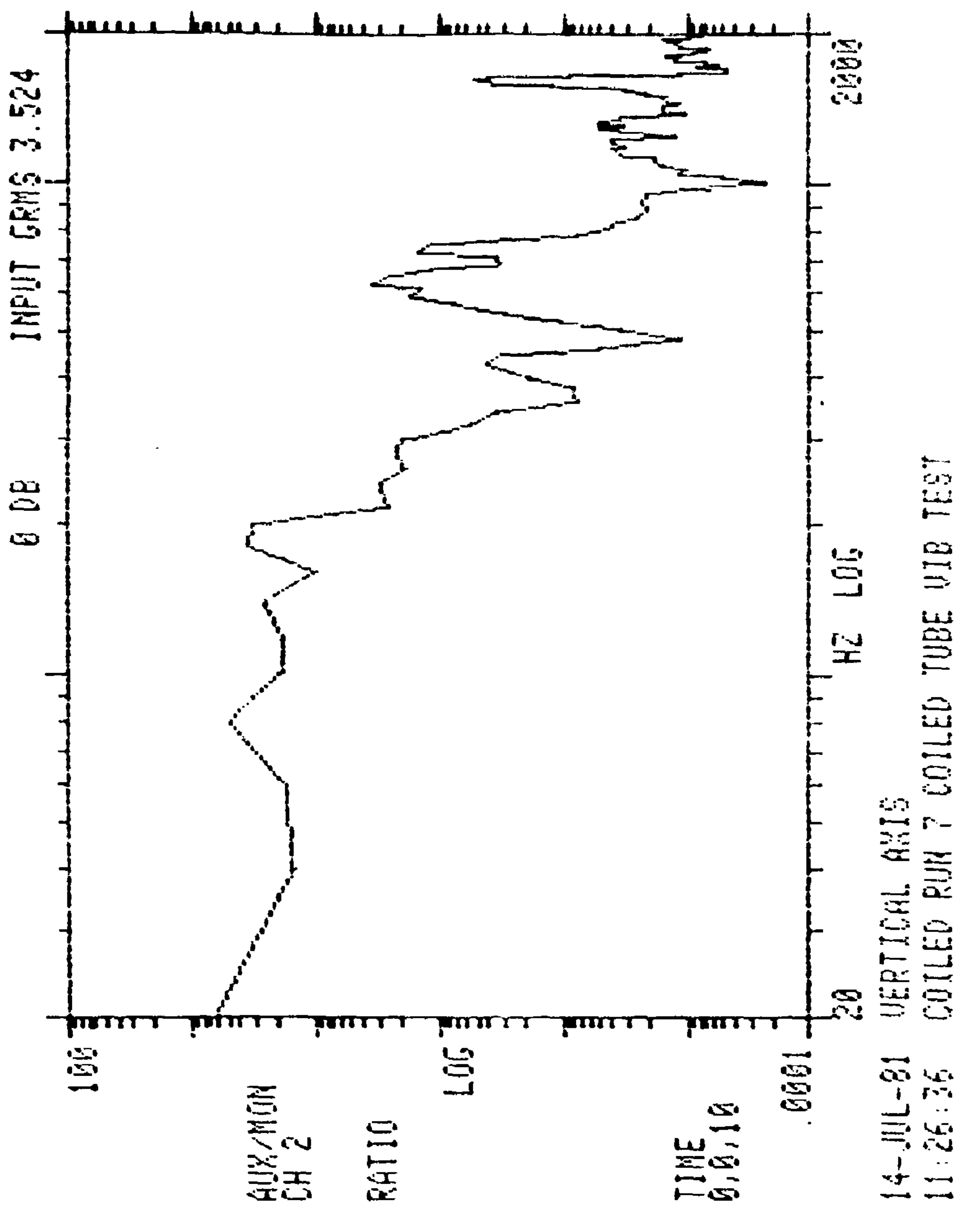

Figure 43. Damped Vibration Test No. 1, Transmissibility of Helical Configuration in Vertical Axis

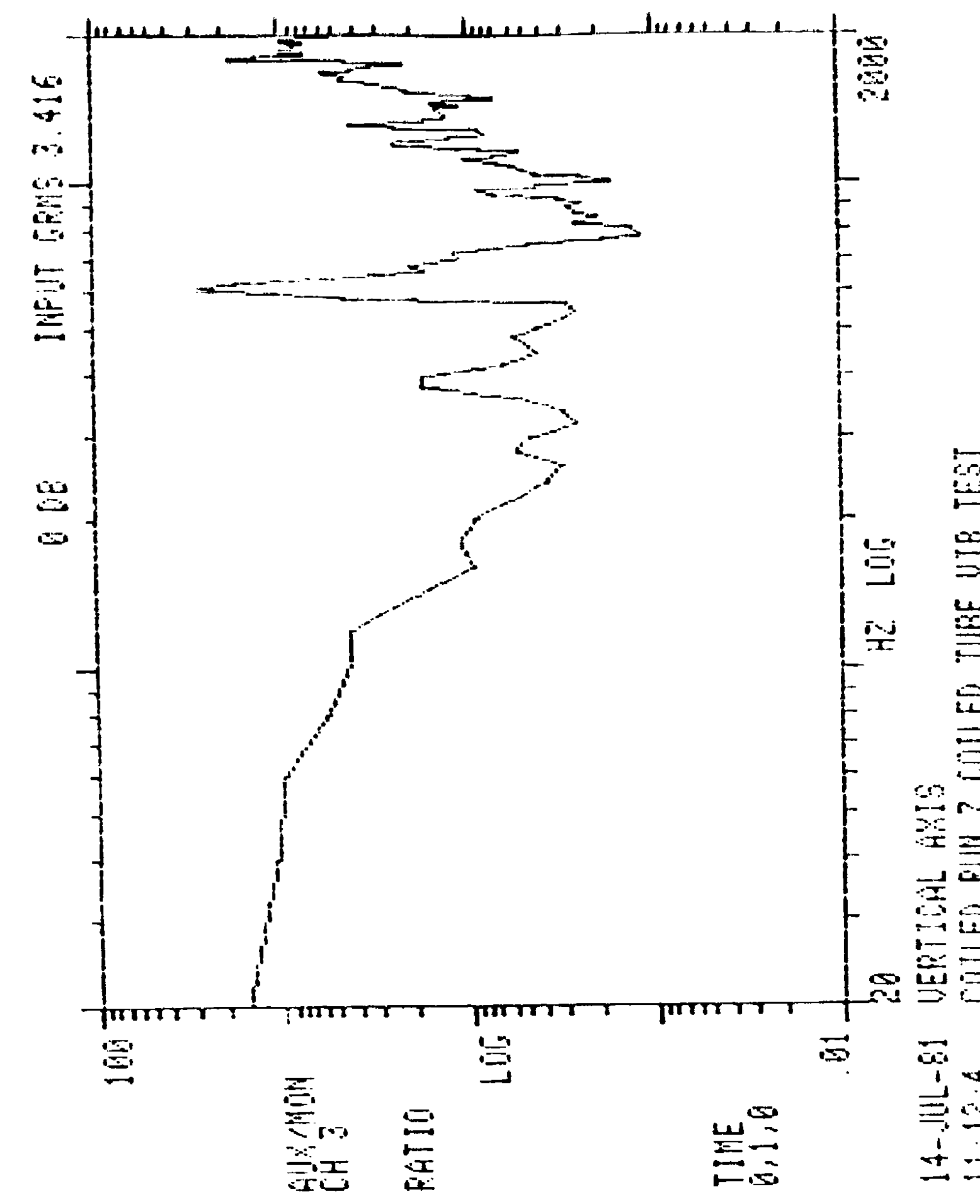

Figure 44. Damped Vibration Test No. 1, Transmissibility of Tri-Coil Configuration in Vertical Axis

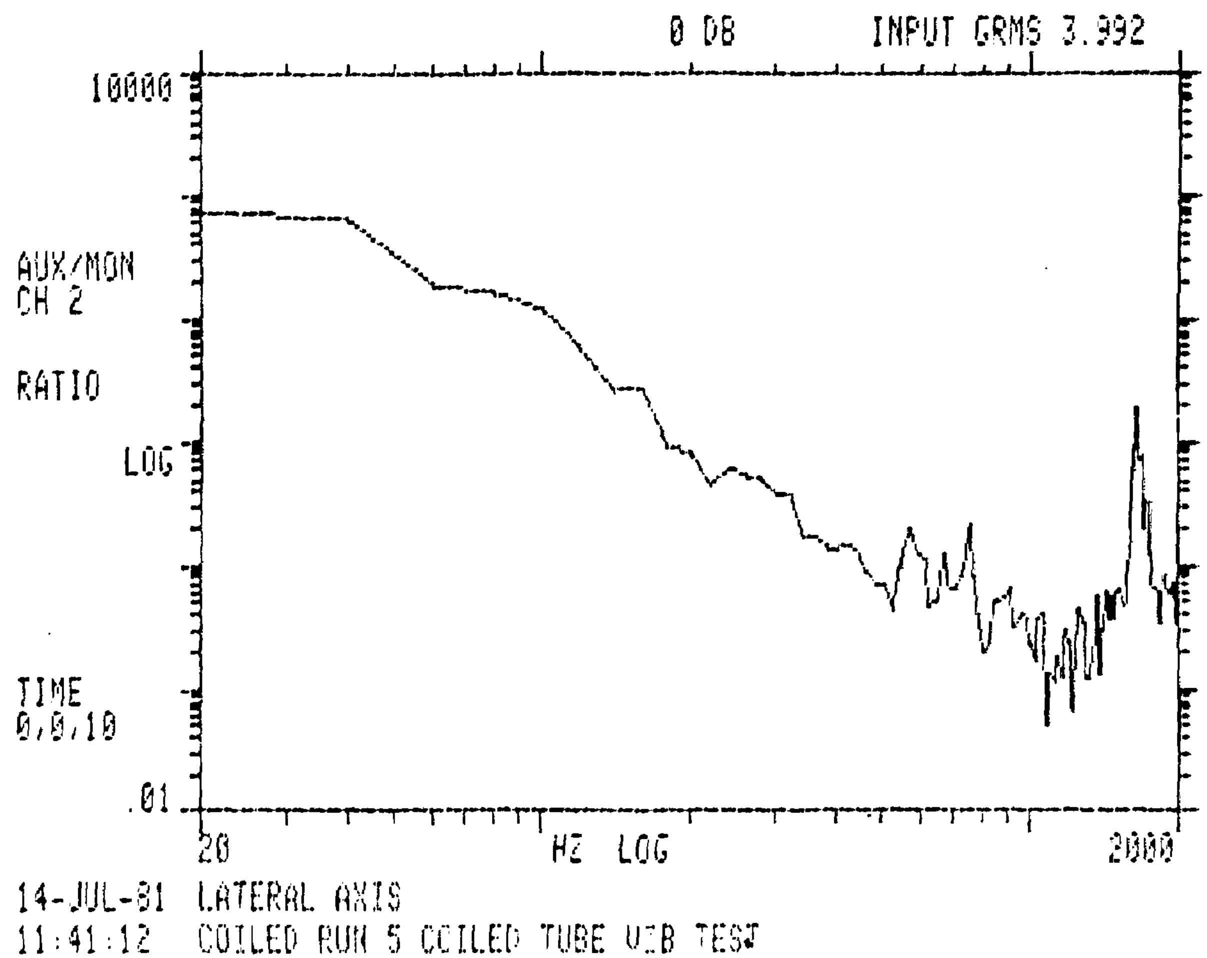

Figure 45. Damped Vibration Test No. 1, Transmissibility of Helical Configuration in Lateral Axis

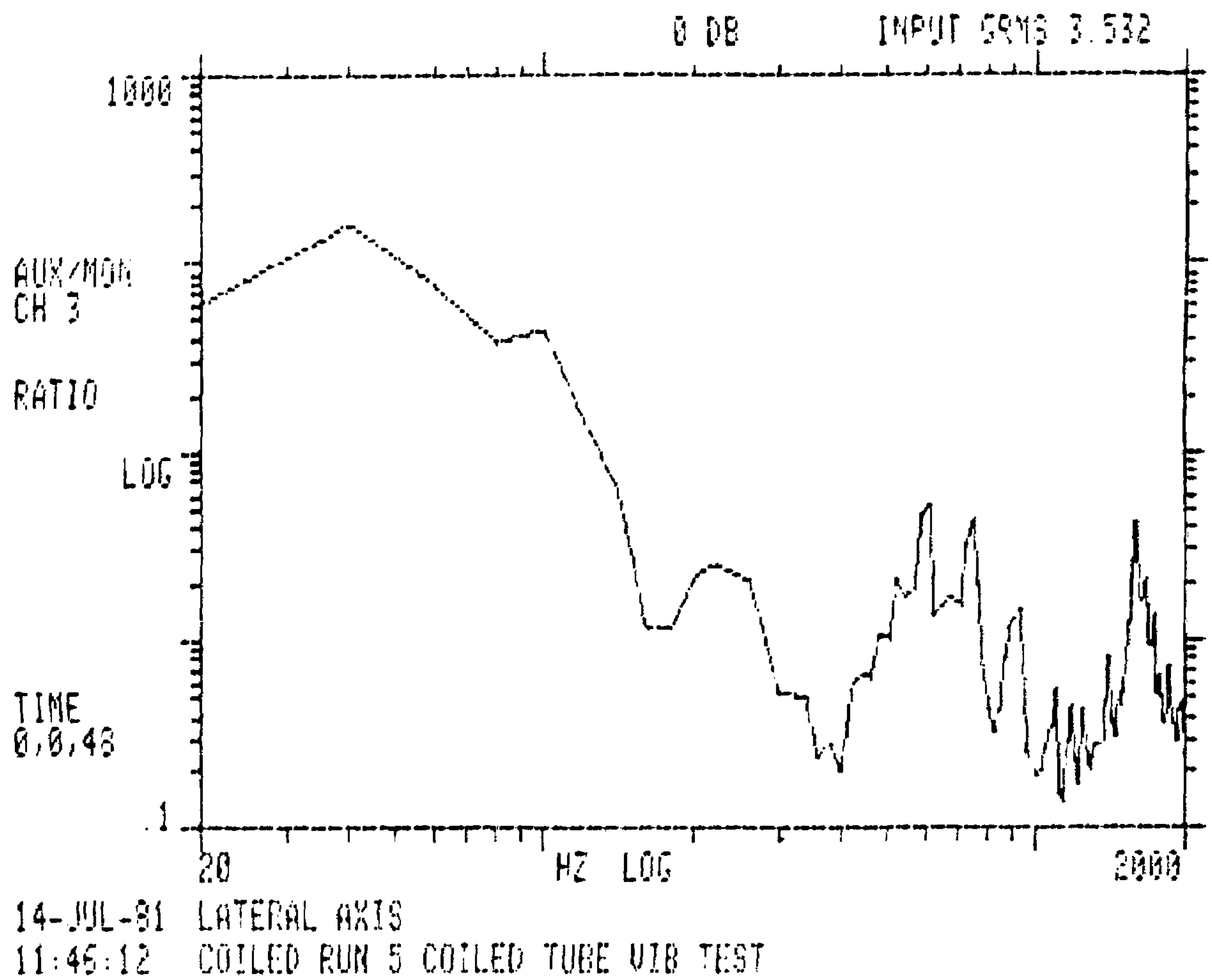

Figure 46. Damped Vibration Test No. 1, Transmissibility of Tri-Coil Configuration in Lateral Axis

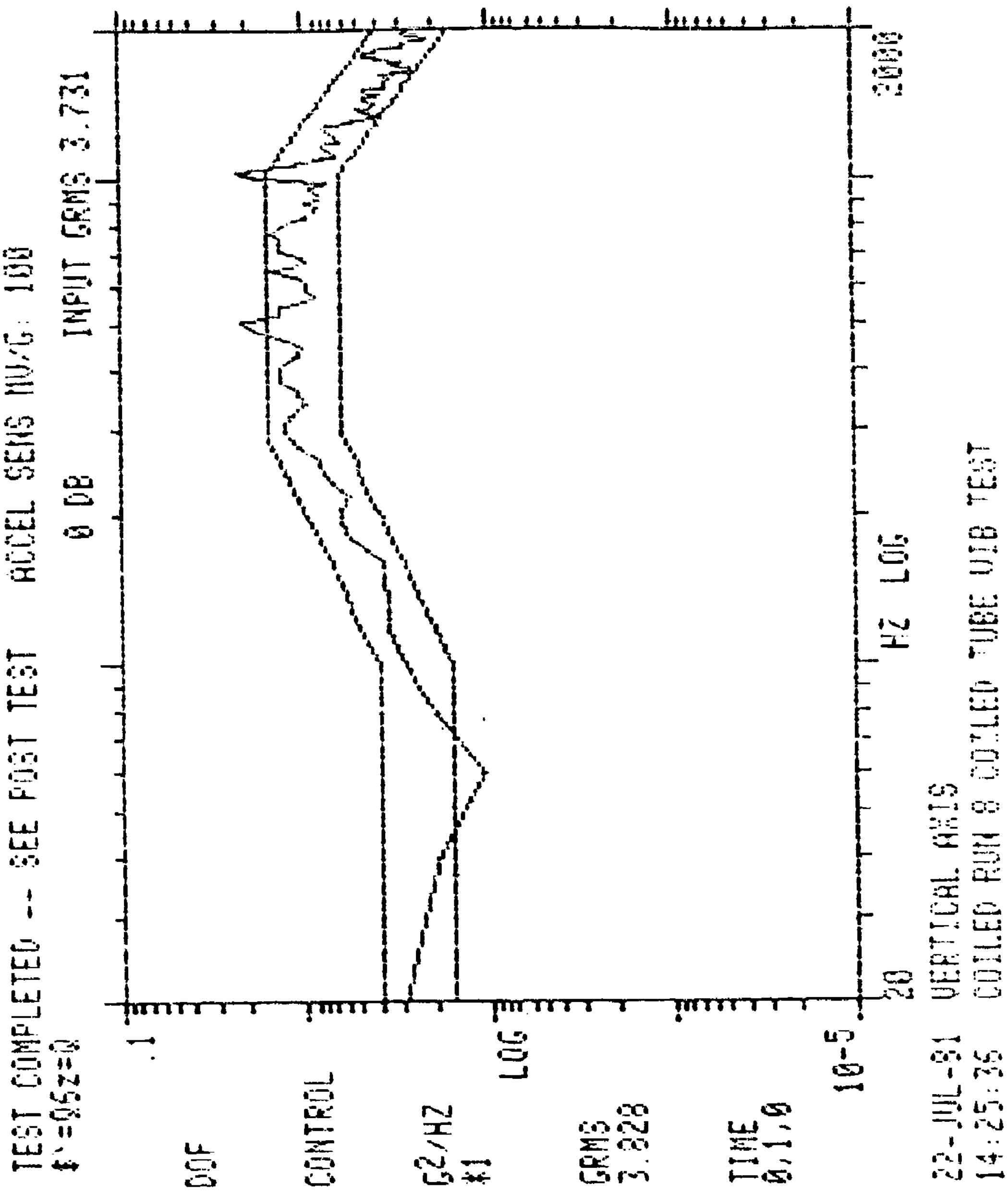

Figure 47. Damped Vibration Test No. 2, Input for Vertical Axis

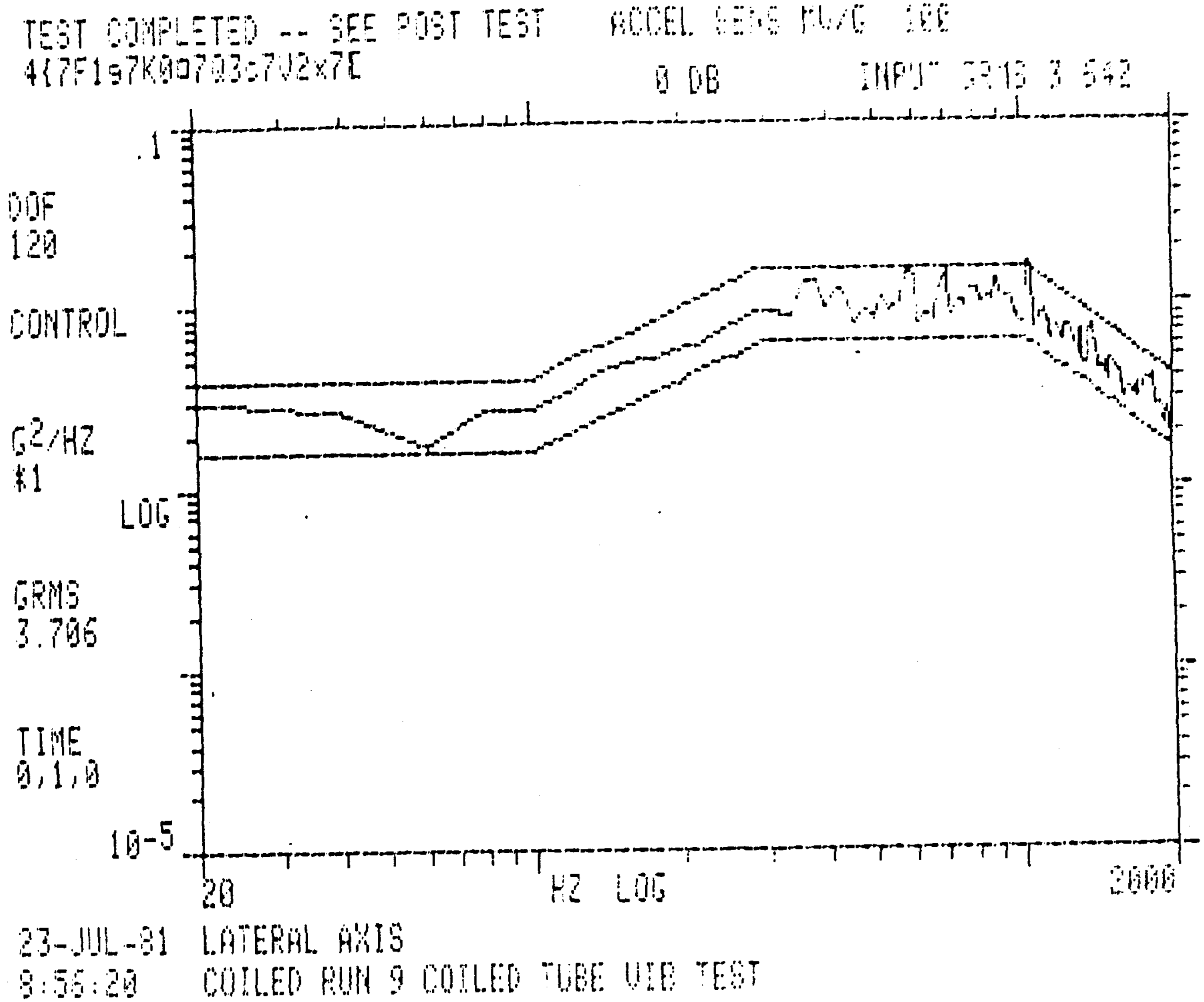

Figure 48. Damped Vibration Test No. 2, Input for Lateral Axis

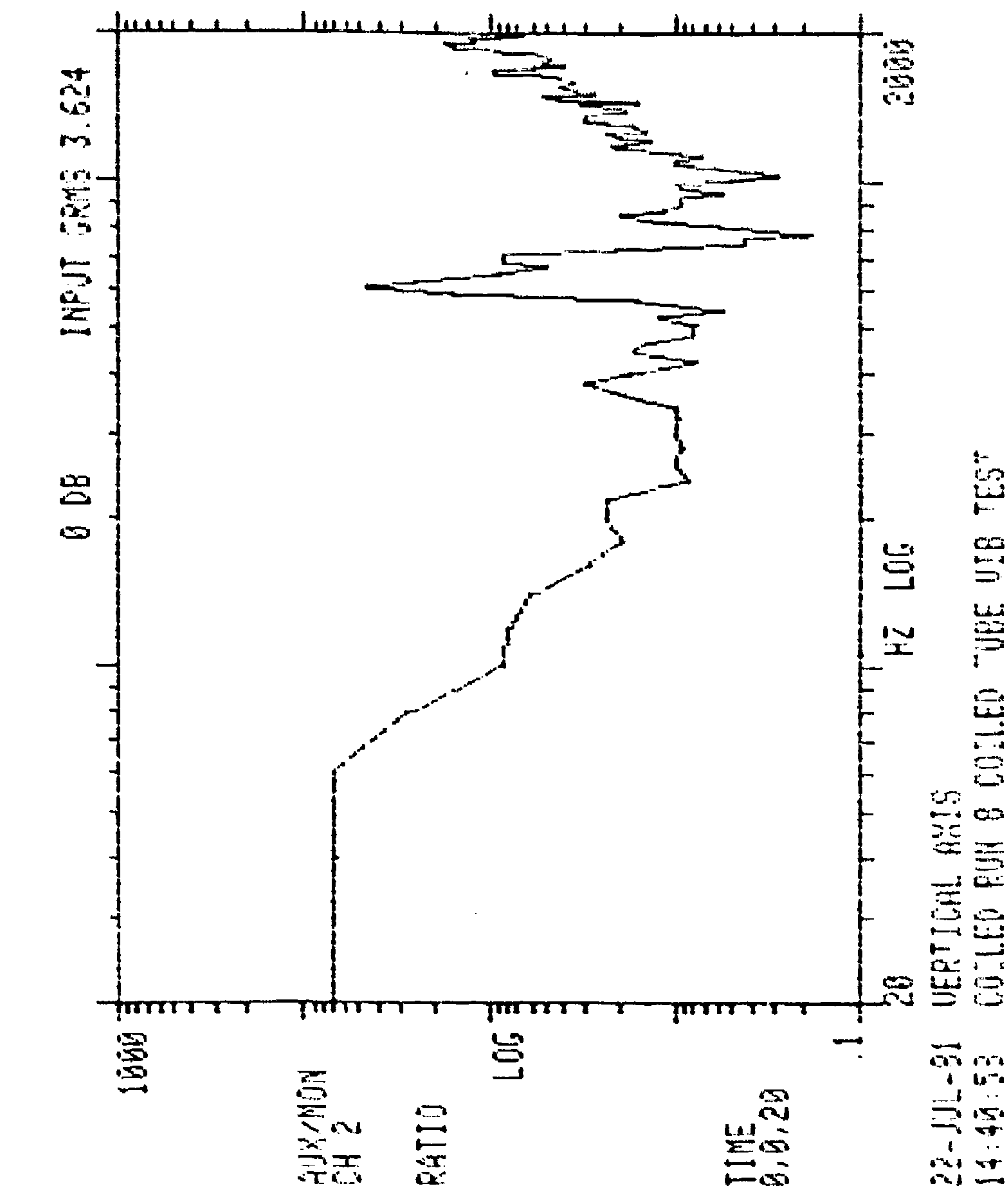

Figure 49. Damped Vibration Test No. 2, Transmissibility of Helical Configuration in Vertical Axis

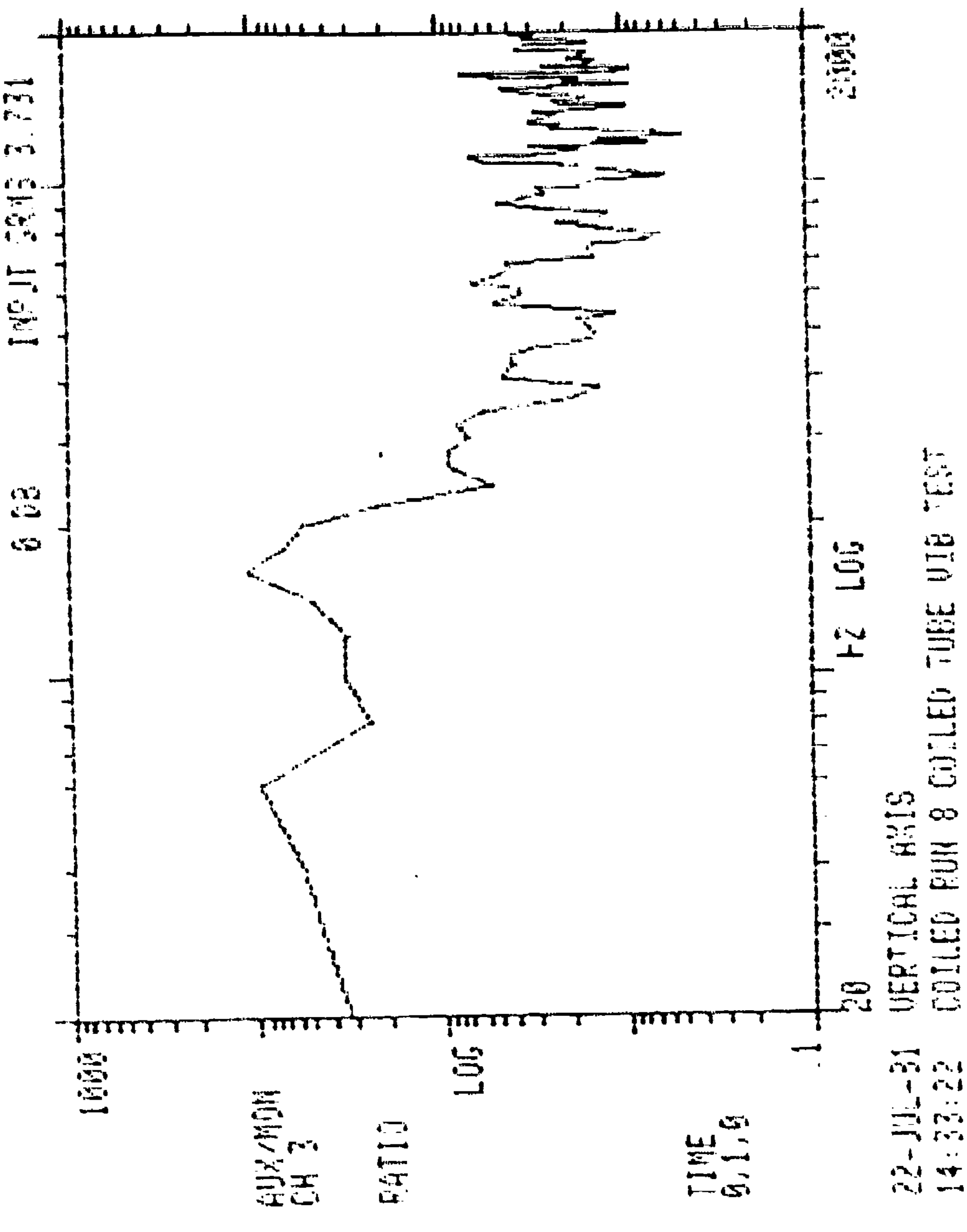

Figure 50. Damped Vibration Test No. 2, Transmissibility of Tri-Coil Configuration in Vertical Axis

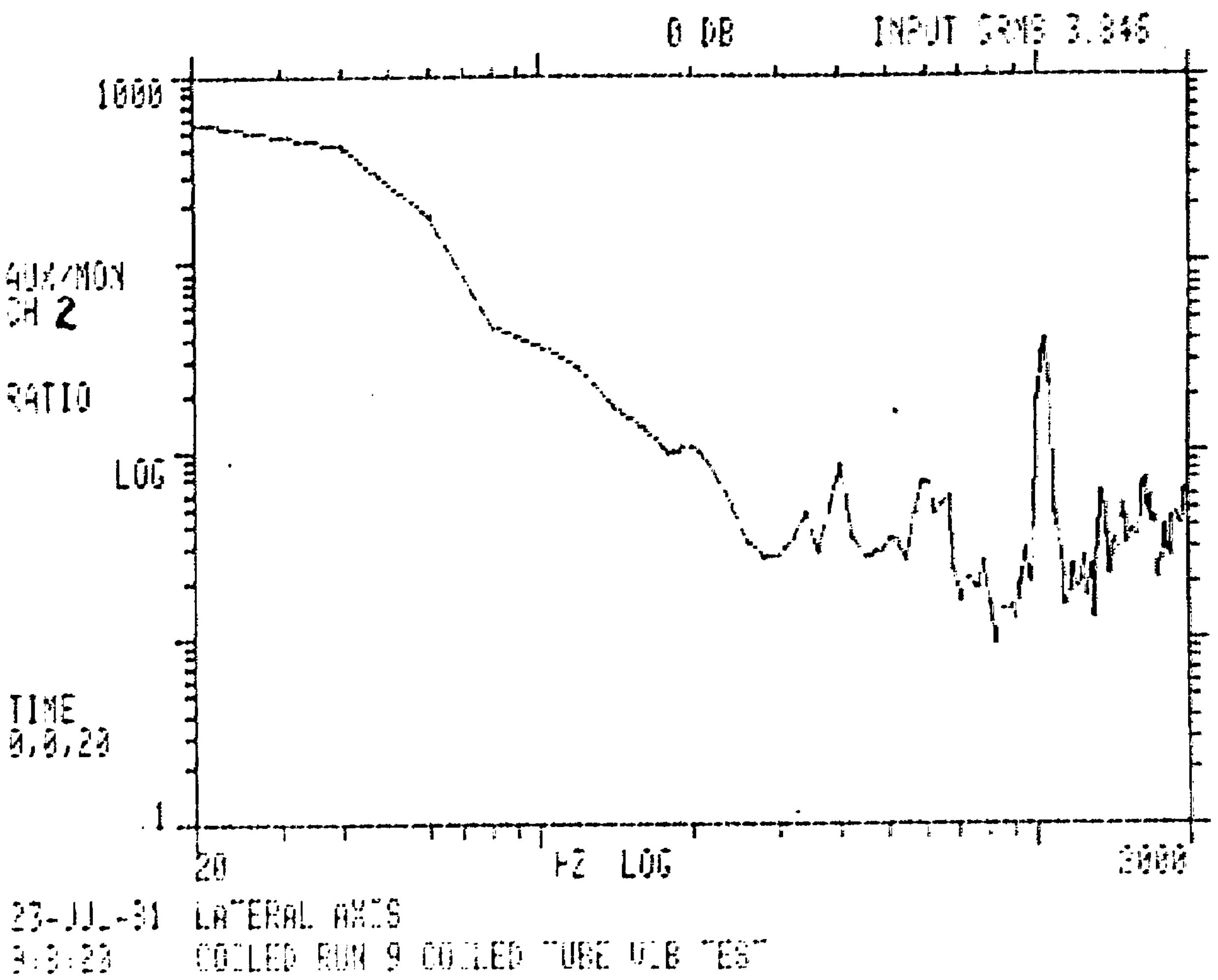

Figure 51. Damped Vibration Test No. 2, Transmissibility of Helical Configuration in Lateral Axis

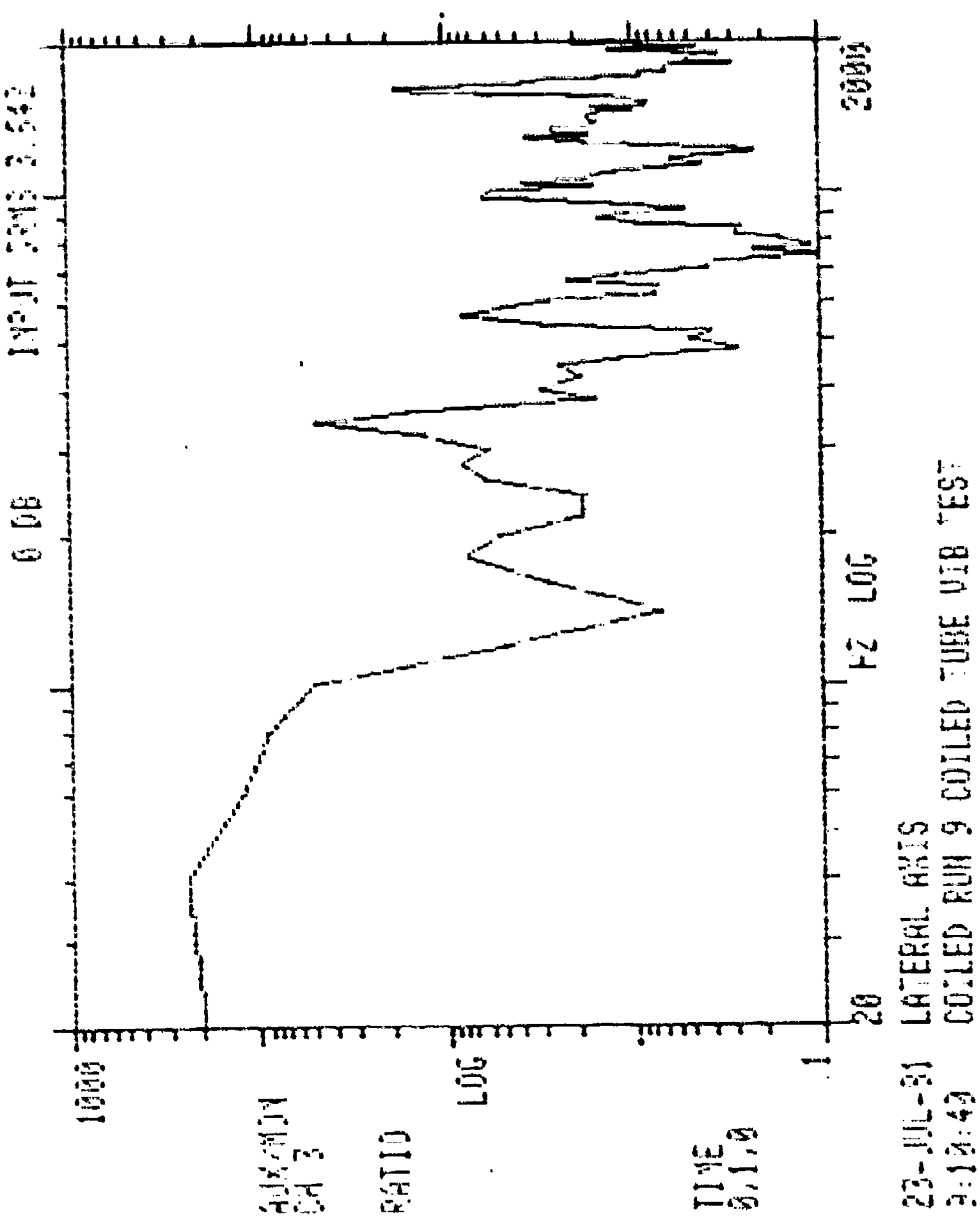

Figure 52. Damped Vibration Test No. 2, Transmissibility of Tri-Coil Configuration in Lateral Axis

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